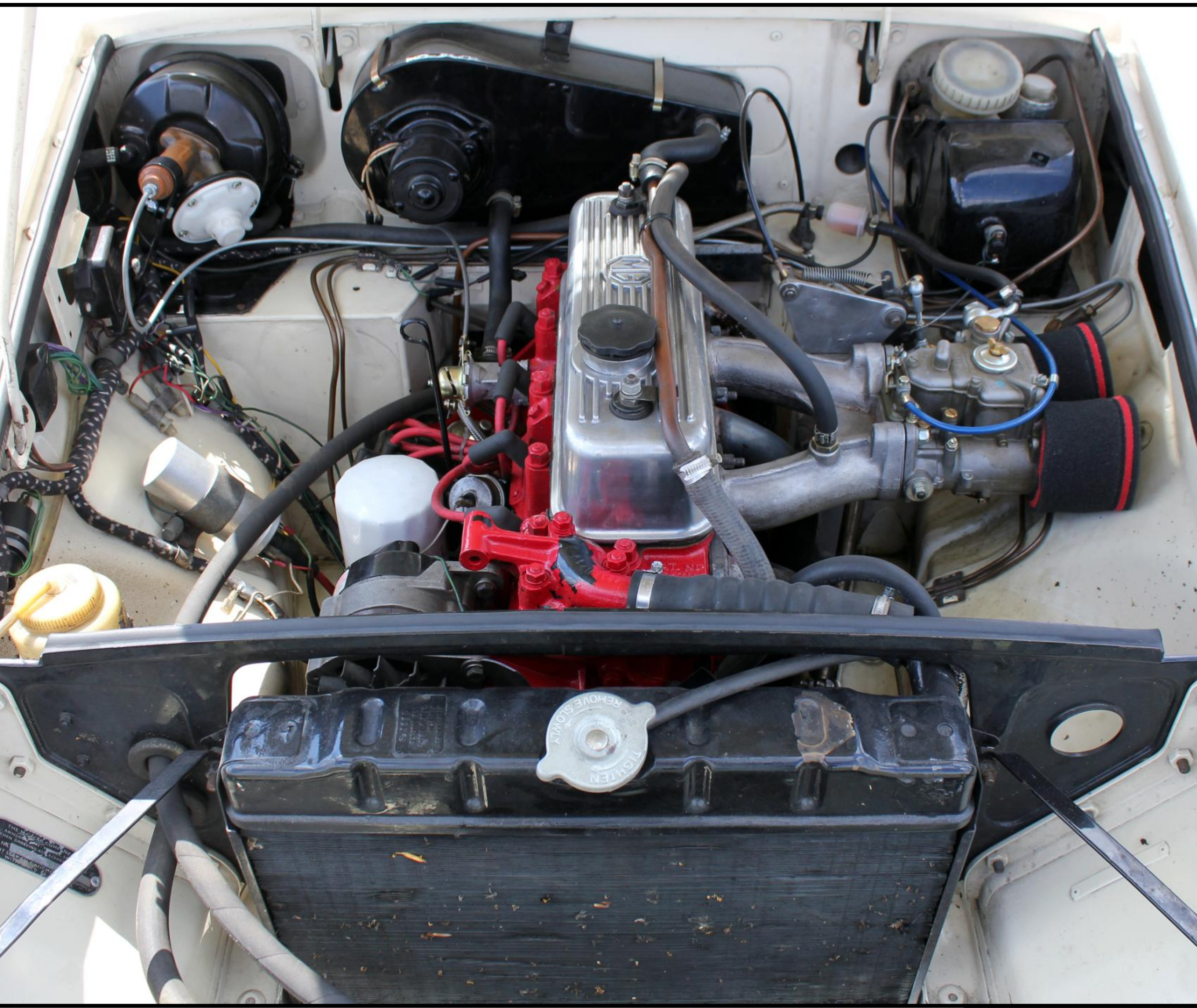
TUNING THE MGB 4-CYLINDER ENGINE



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About this Publishing

This article has been sent to me from a well-known Motor Journalist from Germany – himself owner of a tuned MGB – while I am still carrying out a MGB GT restoration. I would never be able to write about engine tuning in such a professional manner by myself.

Although I do not know the author of this work and have never received any permission to publish this extraordinary and helpful work, I did not want to miss the chance to share all the knowledge of these 1.000 plus pages with MG enthusiast all over the world. Trying to find the author or details has never been a success but it appears that it has been written somewhat about 2005/2006.

So whenever there will be any well-founded interests of not publishing these information please let me know. However, I hope you can enjoy this detailed tuning guide to live what MG always has been – Safety Fast!

Sven Wedemeyer – **mgbgt.wordpress.com**

Berlin, July 19th 2011

...so this is where the original article starts:

About The Author And His Car

MG's are a thing of the heart, not of the wallet. I do not use my B as a Daily Driver, although I do drive it to work every once in a while just to make people jealous. Mind you, it's not a "Trailer Queen," but I do go to some lengths to keep it as good looking as I can. You know, washed & towel-dried, waxed, covered while parked in the garage, etc.

It was in horrible condition when I bought it. One day, while out for a pleasure drive in my Honda Del Sol VTEC on the mountain roads of nearby West Virginia, I stopped for a cup of coffee at a small roadside Mom & Pop style restaurant. While sitting there reading a copy of the local newspaper that someone had left in the booth, in my thoughts I bemoaned the appliance-like lack of "personality" in today's cars. In the "Cars For Sale" section I spotted an ad for an "MGB Convertible". I asked the waitress if the phone number was local and she confirmed that it was. Investing a quarter in the payphone outside, I called and inquired if the car was still for sale. It was. The elderly gentleman who was selling it said that it had belonged to his brother who had recently passed on after a long battle with cancer and that the family was selling it to raise money to help pay off his remaining debts.

Following his directions, I soon arrived at the address. He showed me to an ancient wooden garage with a sagging green roof where we swung the doors open and pushed it out into the light of day where I could examine it. The only good thing about it was that the body was straight, had no significant rust, the body panels, with the sole exception of the severely-dented front valance, were undamaged and original, and it still wore its original Glacier White paint. It also had a nonfunctioning overdrive unit.

Beyond that, it needed everything. Like for example, a dashboard. The instruments and switches were in a cardboard box on the floor. The top was tattered and its windows had turned a nice opaque amber from age. The windshield was cracked. The seats were split and sagging, plus the carpet reeked of mildew. The dry rotted, nearly bald tires mounted on corroded wire wheels were hardly confidence-inspiring. If the springs were to have been sagging any more than they were, then the car would have been resting on its bump stops. The only parts of the electrical system that worked were the alternator and the ignition. Using a set of jumper cables, the engine took some coaxing to start, smoked, and quit on me twice as it wheezed its way home from West Virginia. This was probably merciful as the delays gave my ears time to stop ringing from the roar of the rusted-out exhaust system. The brakes were so bad that I didn't dare take it beyond third gear, which was also a blessing as this reduced my shifting and thus I didn't have to grit my teeth to the screaming grind of the clutch's bad throw-out bearing. To put it mildly, it was in very poor shape. For a paltry $600, I had become the proud owner of a "project car." My wife took one look at it and promptly dubbed it "the mouse trap," which was a pretty good name for it as mice had in fact been living in it for a while (found a wondrous stash of nuts behind the splash panel inside the left front fender).

What followed in the coming months after my proud acquisition could only be described as an increasingly psychotic experience. My bank account was devastated, and my mechanical skills, not to mention my patience with parts suppliers, plus my general sanity, were tested to their limits. But it was worth it.

In order to give you an idea of what it looks like now, I'll have to describe it to you as I don't have a scanner. It's a Chrome Bumper 1972 MKII Roadster, painted Mineral Blue. It has a MKI grill and MKI British Market taillights (amber top lens). I gave it 15" chrome 72 wire spoke wheels by Dayton (not Dunlop). All of the trim, including the bumpers, was replated with marine chrome. The aluminum of the windshield frame was re-polished and re-anodized in the original silver color and a new Triplex windshield installed. The interior is all black leather in early MKI piping pattern from Mike Satur over in the UK. Seat, door and carpet piping is in Mineral Blue in order to match the paint. The information provided by instrumentation is important to me, so from left to right they are: Fuel Gauge, Tachometer, Combination Oil Pressure/Water Temperature, Speedometer, Oil Temperature, Voltmeter, Ammeter, Clock. Walnut door cappings, dashboard, radio speaker console panel, door pull handles, emergency brake handle, and gearshift knob lend a certain ambiance of British quality to the detailing. The steering wheel is a leather-covered early MKII with wire spokes, mounted on a 1969 steering column modified with the original 1972 cowl and column-mounted switches. A period-correct OE AM/FM Radiomobile radio resides above a tunnel-mounted octagonal speaker grill and surround wired to a power antenna and a modern single speaker produces surprisingly good sound. Not wishing to create inconsistency in the amenities department, I chose to mount on the dashboard a MKI map/courtesy light on the right for the passenger and a center-mounted MKII courtesy light. The dashboard vents were deleted as being unnecessary in a Roadster. A Chrome MKI ashtray forward of shift lever, a tunnel-mounted early MKII (separate) leather console with a forward external change pocket (actually an ashtray without a cigarette holder) and a pair of footwell-mounted MGB-GT mapholders completes the vintage feel of the interior. The Securon inertia seat belts were a bolt-in affair. The Cabriolet-style top is of red Mohair with an insulated liner.

So much for the cosmetic side of the car. Looks mean nothing if it does not run, so let's get mechanical. The suspension uses new stock rate springs with a 7/8" front stabilizer bar and a 5/8" trailing rear stabilizer bar, plus a Panhard Rod to control rear axle sway. The lever-arm dampers are rebuilds from World Wide Auto (better than new or Apple Hydraulics rebuilds) with uprated damping on the front units. The brakes are stock, except for the use of RV8 Carbon Kevlar brake pads and shoes to decrease problems with fade when driving hard on mountain roads. To aid in this pursuit, the tires are Michelin P195/60R15 Exaltos. The hydraulic systems use Teflon-lined stainless steel braided hoses. The rear axle is a Hardy-Spicer Banjo-type with a torque-biasing limited-slip differential made by Quaife Engineering in the UK. I kept the clutch, driveshaft, and halfshafts stock as they are more than capable of handling the power of any streetable B Series engine. The transmission is equipped with a standard Laycock de Normanville overdrive and has been rebuilt with the ratios from the 1975-1976 models (taller first gear).

The engine has a compression ratio of 10.5:1, is balanced to within one gram, and has a Peter Burgess-modified Derrington crossflow cylinder head with Mangoletsi intake manifolds and Weber DCOE 45 carburetors. The Peco Big Bores exhaust manifold is Jet-Hot coated and is joined to a Peco free-flow exhaust system. The camshaft is a Piper 285 operating thru modified 18V tappets, tubular chrome-moly pushrods, three-angle stainless steel valves and lead-free seats. The ignition is provided by a Crane/Allison pointless conversion Lucas 45 distributor with an Aldon-modified spark curve and working thru an uprated 30Kv coil with silicone leads. The cooling system uses an early cast iron coolant pump (die cast impeller, so no cavitation at high RPM), a seven-bladed fan, a custom aluminum shroud, and a Morris downflow four-row L-type radiator recored in aluminum. The oil cooler is a 10 row model mounted behind a louvered front valance connected to the engine by braided stainless steel hoses with an inline oil thermostat.

Disliking the original "black lump" paint scheme of the 18V engine, I decided to do it as a piece of industrial art. The block, water outlet elbow, and carburetor bodies were painted MG Red. The head, all exposed nuts, engine front and back plates, blanking plate, and motor mount plates were all done in Aluminum. The rocker arm cover and the sump are Ford Blue. The coolant pump is Orange. The timing cover, fan shroud, and all studs are Gloss Black. The tappet chest covers are done in British Racing Green. The fan was yellow right out of the box. The carburetor float bowl covers, alternator castings, heater valve, and oil filter head are polished. The heat shield is black. All in all, it looks like it belongs on a stand in an industrial museum, right next to a locomotive. It's a different approach, to say the least, and some people find it fascinating to look at, asking what the different items are.

The electrical system is improved as well. Modern blade-style fuses are used for each circuit and extra relays make for greater efficiency. The headlights are French Cibies, which produce more light over a much better pattern. All of the turn signal and brake light bulbs are uprated versions for greater brightness. To handle these extra loads the looms use the heaviest gauge automotive wire possible for each circuit with soldered connectors. A 43 Amp Lucas alternator powers the whole system.

All in all, I'd say that it's a pretty competent car for something that was designed well over fifty years ago. However, that's not to imply that I'm done with it yet. Some say that I've never fully recovered from what is commonly called "The MGB Experience". Since the original rebuilding of the car, I've installed a TR7 clutch driven plate to go along with the new bonnet and engine backplate, both in aluminum. I also switched to the 60 lb lighter Hardy-Spicer banjo axle with another Quaife Engineering torque-biasing limited-slip differential and MGA Twin Cam disk rear brakes. The beat goes on.

How did I get the inspiration for such a car? Well, like every "Project Car", there's a story behind that: Many years ago the Directors of MG wanted to produce a special version of their MGB to be issued in limited numbers to their top executives as a company perk. It was not to be sold to the public, so very few people outside of MG ever knew about the project. Vanden Plas, then owned by BMC, which was the parent corporation of MG, was to do the paint job and the interior. They were also to handle assembly. The bodies were delivered rough-primed from the Pressed Steel subsidiary's body assembly plant in Swindon directly to the Vanden Plas factory in London. All of the chrome plating was done specially by an outside firm that specialized in show-quality finishes. The engine and suspension were both specially modified and fine tuned to a formula established by the factory race department. Because these cars never appeared on the assembly line at the Abingdon plant, they did not appear on the production records. To avoid resentment in the ranks, if anyone "outside the loop" asked about the cars, they were simply told that they were being created for the Sales Department for car show purposes.

One day back in the early `70's while on leave from the service I went to see my best friend, Rick. He was at home for the holidays and was soon to return to college. He needed a part for his 1967 MGB Roadster. The dealership did not have it in stock and said that the part could not be obtained until after the holidays. Having to return to college prior to that, Rick decided that the only thing to do would be to go to the warehouse of the US Distributor which was in nearby Leonia, NJ. Although I knew that the distributor almost certainly would never sell directly to the public, I decided to go along with him for the ride.

When we arrived, being a pair of Jersey Wise Guys, we parked in the Employee's parking lot. As I got out of the car I noticed a rather nice Roadster parked nearby, aside from all the other cars in the lot. I noticed that it had chrome wire wheels and a luggage rack. I went over to it for a closer look. Then I noticed the paint job. I put my face close to it and could see the pores in my face. I looked into the interior of the cockpit and was astounded. I called Rick over, yelling "Hey, Rick! You gotta see this!"

The dashboard was done in walnut, as were the matching door cappings, door handles, shift knob, and the grip of the brake lever. The carpet was obviously a Wilton wool carpet. The seats and door panels were all done in leather with piping that matched the paint. The red top was an odd cloth-like material (I later found out that it was mohair) and had a headliner. Everywhere we looked there was nothing but quality. Underneath I spotted a Panhard rod and a trailing rear stabilizer bar, which I knew was not put on MGBs at that time. This car was obviously very special. I now believe that this car was the one issued to the President of MG's US operations.

To make a long story short, the Distributor wouldn't sell us the part, but I never forgot that car. When my second wife decided that she wanted a used 1969 MGB, we got it. She loved that car. She washed it, cared for it, and even learned how to do her own oil changes. One of my fondest memories from those days is the one of her laughing with a smudge of dirty oil on her nose. I always wanted to build hers into a reproduction of that special MGB just for her, just to show her my love for her. Sadly, at age twenty-five, she died before I could do it.

So, today I finally own it. I built it with my own two hands, just as I would've for her. When I part with it, it'll go to our son. She would have wanted it that way. I know she'd be proud. Like I said, "MG's are a thing of the heart, not of the wallet."

Steve S. – Virginia, USA

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Forward

There are certain rules that you should go by in considering the purchase of any particular MGB. Try to remember that this is a classic car which has been out of production for 30 years. The oldest ones (1962 models) are 48 years old. When you buy one you will have no real idea of how it has been maintained over the course of its life. Perhaps the current owner loves it, coddles it, and keeps it in a garage, but who can say that it did not go through a period of neglect and abuse at some time? If you're a realist, you must automatically presume that this is the case. If the car has not had a documented new-from-the-ground-up restoration within the last 50,000 miles, then you would be a fool to think that it would be as reliable as a garage-kept three-year-old Honda. If you rely on *any* unrestored classic car to be your daily driver, you will ultimately be disappointed and will be glad when its' new owner drives it away.

Fully restored B's are a rarity, and that's why they are so admired and sought-after. A *fully* restored B, just like any fully restored classic car, is not cheap to create. It could well have $20-25K in cash invested in it, with nothing included to this figure for the countless number of hours during which its owner performed a labor of love on it. If the restoration was done by a commercial outfit, then it is only realistic to expect that corners will have been cut and quality will have been compromised, especially where the eye cannot see. A cheaply overhauled engine and a rebuilt clutch do not a new car make. Any MGB selling for $8,000 will need to be repeatedly worked on during your period of ownership.

However, quality parts are still relatively easy to get via the Internet, although you may have to do some searching for certain unusual parts, and their prices are surprisingly competitive, but you will have to wait for them to be delivered. Your local auto parts store will have little or nothing for your car and probably will not even know what it is. Service will probably need to be done by you personally as garages that are versed in this car are rare. If you don't *enjoy* the idea of performing mechanical work on your own car as a hobby, any classic car is not for you unless you are rich, period. You will need to have tools, and I do not mean a claw hammer, an adjustable wrench, and a screwdriver that is kept in a drawer in the kitchen. You will need, *just to get started*: full sets of combination wrenches, sockets, extensions, screwdrivers, a torque wrench, a grease gun, a spark plug socket and Champion gauge, a strobe light, a dwellmeter and a feeler gauge, a multimeter, side cutters, a wire stripper, a soldering iron, a full set of good pliers, a hydraulic floor jack, jack stands, a slide hammer set, and maintenance manuals. The car is basically simple, easy to understand and work on, a plus that cannot be said for the cars of today.

MGB owners are a fraternity, initiated by and bonded together by the trials of ownership, as well as by an enthusiasm for what is truly the ordinary man's Aston Martin. If you have perfectionist tendencies, love tackling a problem, and get a real feeling of accomplishment from setting something right, then you'll find out something about why the B is one of the most beloved of all the classic cars ever built and why their owners have so much pride in them.

So there you have it. If you just want an open roadster to use as a daily driver, buy a Mazda Miata. My daily driver is a Del Sol VTEC. Although it will positively crucify the MGB on any comparison criteria scale, and has the reliability of a kitchen appliance, it also has all the soul and character of one. Like, for example, a refrigerator. One does not *love* an appliance. An MGB actually has an individual personality, and that's why we sometimes find ourselves talking to them as we work on them. Improperly maintained, it will be like a recalcitrant child. Properly maintained, it's like having an English gentleman as a best friend: friendly, forgiving, and good, *civilized* fun.

A Little Background About The B Series Engine

There are few mysteries about the engine employed in the MGB. This is not a state-of-the-art, fuel-injected, dual-overhead-camshaft, four-valves-per-cylinder, variable-valve-timing, microchip-controlled technological wonder, festooned with interconnected sensors and switches, all linked to mysterious black boxes, and guaranteed to both intimidate and befuddle NASA engineers. Instead, this engine is something more along the order of an archeological relic from a bygone age of motoring, something that was intended be maintained by its owners with simple hand tools and to also be produced in alternate versions that were to be installed in farm tractors and diesel-engined taxis. In today's world of laser weapons, it seems as anachronistic as a sword. Crude, yet still highly effective in a very intimate way.

Keep in mind that the genesis of the B Series engine originated in August of 1944 when it had become obvious that the defeat of Germany was close at hand. Leonard Lord, Chairman of the Board and General Manager, gathered together his three top engine designers from the engineering staff at the Austin Design Office, Eric Barham, Jimmy Rix, and Bill Appleby, and gave them the assignment of designing a pair of all-new engines that would enable the company to get a jump on the competition in the soon-to-come postwar market. Their initial undertaking was an overhead valve 1200cc development of a pre-war side-valve Austin engine. However, this engine could not have its displacement increased, and was regarded as nothing more than a stop-gap development that would buy time for the new generation of engines to be properly developed. After a series of design studies, in January of 1952 the team started work on an entirely new engine. The ultimate evolutionary results were the legendary A Series and B Series engines. Such was their development potential that they both remained in production well into the 1990s.

The block of the engine was designed to the British Standard (BS) 1452-17 in which the coolant jacket extended down to just below the level of the piston rings when the piston was at Bottom Dead Center (BDC). Flow-cast of grey iron, the engine block was allowed to slowly cool so that graphite crystals would form within its matrix, assuring reasonable machinability. The B Series engine blocks were originally cast at the old Nuffield foundry at Cowley, displaying the now-cryptic acronym of MOWOG in raised letters. MOWOG stood for <u>MO</u>rris, <u>WO</u>lseley, and M<u>G</u>, the original car companies that comprised the Nuffield Group, which owned the foundry. Later, due to growing production demands, this output was supplemented by the Austin foundry at Longbridge, the casting molds still being solely produced by the Cowley facility with the traditional MOWOG acronym proudly included. In the early 1970's Qualcast took over the work of pouring the castings, but as before, the casting molds were still produced at the Cowley facility, complete with the traditional MOWOG name.

During the era in which the B Series engine was designed, hydraulic tappets for automotive applications were still in their technological infancy; therefore, the engine was designed to use mechanical chilled iron (white iron) tappets, the production of which was entrusted to Hepolite. Setting of valve lash clearances was accomplished by means of a simple, manually adjustable ball end and jam nut mechanism on the lever end of the rocker arm. The rocker arms pivoted on bronze bushings, much like those found in a steam engine. Oil from the rocker arm assembly was allowed to drain down the pushrod passages in order to lubricate the upper ends of the tappets, and then through two holes respectively positioned in the bottom of the tappet chest between cylinders #1 and #2 and between cylinders #3 and #4, thus bypassing the lobes of the camshaft. This simple approach offered the designers the opportunity to wisely leave the camshaft exposed to the crankcase so that both its drive gear for the oil pump and its lobes could be lubricated by a pressurized spray of oil emitting from the lower ends of the connecting rods. This system of lubrication was made all the more effective by having the camshaft rotate in a clockwise direction, thus ensuring that its nose as well as the ascending and descending ramps of the lobes received a spray of lubricating oil upon their surfaces prior to the tappet exerting any pressure upon them. This desire to lubricate the lobes of the camshaft, as well as the nose and the lower sections of the tappets, dictated the narrow .996" (25.3mm) thickness of the big end of the connecting rods. Adequate bearing support was then achieved by using a large-diameter (2.0212" / 51.34mm) big end design.

The engineers at the factory prudently decided to employ a rather stout Renold duplex camshaft drive chain that had an even number (52) of 3/8" (.375" / 9.53mm) pitches (spaces between links). The sprockets were both given an even number of teeth, 20 teeth on the camshaft drive sprocket on the crankshaft drive sprocket and 40 teeth on the camshaft driven sprocket. This combination precluded any single roller of the camshaft drive chain from contacting the same sprocket tooth each time it made a consecutive circuit, thus preventing uneven wear and consequent vibration and inconsistency in both valve and ignition timing, as well as prolonging the lifespan of the drive system.

Its Heron-type cylinder head design integrated the pistons into the overall combustion chamber design by featuring concavities in their crowns. It also made use of Weslake-patented combustion chambers, which were a marked advance beyond previous technology, allowing for superior airflow characteristics and fuel / air charge distribution while permitting excellent flame propagation. The incoming fuel / air charge was directed toward the spark plug and away from the hot exhaust valve, minimizing the possibility of preignition, as well as allowing less ignition advance to be used. The siamesed intake ports, like some other features of the engine, were largely the result of production economics. By using siamesed intake ports the intake manifold could be of efficient, yet simple design and thus still be relatively inexpensive to produce. In addition, the pushrod passages could be neatly situated between the ports, thus keeping the cylinder head as well as the engine block as compact and light as possible. The placement of both the intake and the exhaust manifolds together on the same side of the cylinder head meant that only one mating surface needed to be machined, and fewer manifold mounting studs and their attendant threaded bores were required. It also allowed the distributor, oil filter, and generator (dynamo) to be placed on the opposite side of the engine for easier accessibility, thus greatly simplifying maintenance.

There are also some distinct engineering advantages to this approach. By placing the intake ports with their cool incoming fuel / air charge next to the hotter exhaust ports, this area of the cylinder head is better cooled than it would be in a crossflow design, hindering warpage by enhancing heat transfer from the exhaust valves and thus extending their lives, although this configuration does allow more heat to accumulate in the walls of the intake ports. This condition of radiant heat being detrimental to fuel / air charge density, it consequently reduces power output potential. However, the engineers wisely chose to adopt an exhaust port design of rectangular cross section in order to permit the use of wider adjacent coolant passages in order to maximize cooling of this critical area without a compromise of port flow capacity as a circular cross section exhaust port of equal cross-sectional area would have otherwise entailed.

Due to the relatively small surface area of the roof of the combustion chamber, the undersquare (small-bore, long-stroke) configuration gives better thermal efficiency and thus better fuel economy, as well as providing a greater surface area on the exterior of the cylinder walls in order to minimize the heat transference problems that are inherent to the cast grey iron material that was chosen for the engine block to be made of. It also gives better scavenging effect, thus extending the powerband. By requiring an inherently larger volume crankcase in order to accommodate the long stroke of the crankshaft, power-robbing "Pumping Losses" could thus be minimized.

The cylinders were of the Wet Liner type, in the original design being directly exposed to coolant flow over their entire exterior surface area inside of a large capacity coolant jacket. The bore centers of the later larger-displacement versions of the engine could be located the same distance apart as those of the earlier, smaller displacement versions of the engine so that the later engine could take advantage of the designer's intent that the engine have an inherent "developmental stretch" in order to give later larger-displacement versions the potential to be produced on the same tooling, thus keeping both Research and Development costs, as well as production costs, within reasonable limits.

A high capacity Holbourne-Eaton positive displacement eccentric rotor oil pump was provided in order to supply oil to the crankshaft bearings. These were 1.125" (28.58mm) wide for the front, center, and rear crankshaft bearings, and .875" (22.23mm) wide for the intermediate crankshaft bearings of the five-main bearing version of the engine. They all were given diameters of 2.125" (53.98mm), a full .125" (3.18mm) larger than that of the previous 1622cc three main bearing version of the engine. This produced an almost unbreakable crankshaft with lots of overlap between its journals and counterweights. The main bearing mounts were provided with exceptionally heavily gusseting as a diesel version of the B Series engine was to also be produced. This imparted exceptional rigidity to the engine block. The oil pump was driven directly from the camshaft by helically cut gears, thus minimizing noise output.

Although the B Series engine design is truly a compromise, it is a brilliant one that modern mechanics recognize as being one that was far ahead of its time when first introduced. It was further improved with the introduction of its five-main bearing version. Certainly, there were other new engine designs that were even more advanced in the mid-to-late 1940s, but this one was intended to be available in cars that ordinary people could afford to own and operate. In those days, that made it special, and its designers had every reason to be proud. During an era when full race engines struggled to reliably produce 1 BHP per cubic inch, when the 18G engine arrived in 1962 it boasted 95 BHP from a mere 110 cubic inches, giving it a specific output of .864 BHP per cubic inch, and this was an engine that could reliably be used as a daily driver! In its heyday, it was impressive indeed. Pretty fantastic for a relic whose basic design is almost two-thirds of a century old! A true classic engine design for a true classic car!

Getting Started Right

Everybody who is about to rebuild the tired engine of their MGB entertains the thought of improving upon the power output of this classic engine design. However, nobody wants to end up with a temperamental beast. Properly built with quality components and knowledgeably modified, an enhanced-performance version of this engine should last as long as an engine rebuilt to Original Equipment specifications. It should also be reasonably reliable enough to be used as an everyday car.

Since you are rebuilding the engine, this is a good opportunity to do it the Peter Burgess way. As a former professional mechanic who has built custom engines, I can assure you that I have thoroughly read both of Mr. Burgess' books "How to Power Tune MGB 4-Cylinder Engines", as well as its companion volume "How To Build, Modify, And Power Tune Cylinder Heads", and that his theories are both sound and logical. His reputation as the top MGB engine tuner is both earned and well deserved. His books should be in every MGB owner's library. His website can be found at http://www.mgcars.org.uk/peterburgess/ . If you have not studied his books, they are available from Veloce Publishing through their website at http://www.veloce.co.uk/newtitle.htm . I wholeheartedly agree with his statement "The entire engine system needs to be considered as a whole; otherwise the gains from component changes may not be fully realized."

Before you begin, you will need to have a proper Service Manual. I would recommend that you purchase a reprint of the original factory service manual that the MG dealers had for their mechanics to consult. To my knowledge, there is nothing that can compare with it for completeness. Its actual title is "The Complete Official MGB," although it is often called "The Bentley Manual" as it is printed by Bentley Publishers. Bentley also sells a reprint of the Factory Parts Catalogue, which is very useful for discerning just which part is appropriate for your particular MGB. Their website can be found at http://www.bentleypublisher.com/ where you can eliminate the middleman and order it / them direct.

During the MGB's eighteen-year production run, its engine was repeatedly modified in order to meet the requirements of the marketplace. It is usually helpful to know just what you are working with:

Engines Used in the Production MGBs				
Prefix	**Engine Numbers**	**Car Numbers**	**Production Dates**	**Notes**
18G-(R)U-H/L	101-31,121	101-31,793	May 62-Feb 64	1
18GA-(R)U-H/L	101-17,500	31,794-48,766	Feb 64-Oct 64	2
18GB-(R)U-H/L	101-91,175	48,767-138,400	Oct 64-Nov67	3
18GD-(R)We-H/L	101-6,945	138,401-158,230	Nov 67-Oct 68	4
18GD-Rc-H/L	101-240	138,401-158,230	Nov 67-Oct 68	5
18GF-We-H	101-13,647	138,401-158,230	Nov 67-Oct 68	6
18GG(R)We-H/L	101-29,642	158,231-256,646	Oct 68-Aug 71	7
18GG-Rc-H/L	101-955	158,231-256,646	Oct 68-Aug 71	8
18GH-(R)We-H	101-43,548	158,231-218,651	Oct 68-Aug 70	9
18GJ-(R)We-H	22647-43,548	187,701-218,651	Oct 69-Aug 70	10
18GK-(R)We-H	101-26,226	219,001-256,646	Aug 70-Aug 71	11
18V-581-F-H	101-5,302	258,001-366,400	Aug 71-Nov 73	12

18V-581-Y-H/L	101-5,302	258,001-366,400	Aug 71-Nov 73	13
18V-582-F-H	101-22,341	258,001-366,400	Aug 71-Nov 73	14
18V-582- Y-H/L	101-22,341	258,001-366,400	Aug 71-Nov 73	15
18V583-F-H	101-870	258,001-327,990	Aug 71-Aug 73	16
18V583-Y-H	101-870	258,001-327,990	Aug 71-Aug 73	17
18V-584-Z-L	101-19,491	258001-294,250	Aug 71-Aug 72	18
18V-585-Z-L	101-2,751	258,001-294,250	Aug 71-Aug 72	19
18V-672-Z-L	101-38,094	295,251-360,069	Aug 72-Sept 74	20
18V-673-Z-L	101-6,550	295,251-360,069	Aug 72-Sept 74	21
18V-779-F-H	101-535	336,500-360,069	Nov 73-Sept 74	22
18V-780-F-H	101-7,224	332,000-360,069	Nov 73-Sept 74	23
18V-836-Z-L	101-5,401	360,301-367,818	Sept 74-Dec 74	24
18V-837-Z-L	101-1,504	360,301-367,818	Sept 74-Dec 74	25
18V-846-F-H	101-917	360,301-409,400	Sept 74-June 76	26

18V-847-F-H	101-40,188	360,301-523,002	Sept 74-Oct 80	27
18V-797-AE-L	101-9,361	367,901-386,267	Dec 74-Aug75	28
18V-797-AE-L	101-2,007	386,601-409,400	Aug75-June76	29
18V-798-AE-L	101-1,694	367,901-386,267	Dec 74-Aug75	30
18V-798-AE-L	101-2,007	386,601-409,400	Aug75-June76	31
18V-801-AE-L	101-14,801	382,135-409,400	June75-June76	32
18V-802-AE-L	101-3,509	382,135-409,400	June75-June76	33
18V-883-AE-L	101-50,984	410,001-523,002	June76-Oct 80	34
18V-884-AE-L	101-10,425	410,001-523,002	June76-Oct 80	35
18V-890-AE-L	101-12,059	410,001-500,904	June76-Dec 79	36
18V-891-AE-L	101-4,584	410,001-500,904	June80-Dec 79	37
18V-891-AE-L	101-5,389	517,591-523,002	June 80-Oct 80	38
18V-892-AE-L	101-3,559	414,629-523,002	Aug 76-Oct 80	39
18V-893-AE-L	101-1,277	414,629-523,002	Aug 76-Oct 80	40

Engine Prefix Codes:

U Manual transmission w/center gearchange, first gear not synchronized.

WE Manual transmission w/center gearchange, first gear fitted w/ synchromesh.

R Laycock de Normanville Overdrive (optional), R found in front of either U or We.

Rc Borg Warner Model 35 automatic transmission (optional from 1967/68).

H High Compression, 8.8:1.

L Low Compression, 8.0:1.

F Two carburetors, Home and Export markets except for North America.

Z Two carburetors and emission control, North America only.

AE Single carburetor and emission control, North America only.

Notes:

1) Three-main-bearing crankshaft, Overdrive optional, Low compression optional

2) Three-main-bearing crankshaft, Closed circuit breathing, Overdrive optional, Low compression optional.

3) Five-main-bearing crankshaft, Closed circuit breathing, Overdrive optional, Low compression optional.

4) Home/Export (not USA, Canada to Aug 1968), Five-main-bearing crankshaft, Closed circuit breathing, Fully synchronized transmission, Alternator, Overdrive optional, Low compression optional.

5) As 18GD, Home/Export (not USA, Canada to Aug 1968), Closed circuit breathing, Automatic transmission, Alternator, Low compression optional.

6) As 18GD, USA (1968 models) and Canada (from Aug 1968), Air pump and injectors, Closed circuit breathing, Fully synchronized manual transmission, Alternator, Overdrive optional, Low compression optional.

7) As 18GD, Home/Export (not USA), Carburetor ventilation instead of closed circuit breathing, Overdrive optional, Low compression optional.

8) As 18GG, Home/Export (not USA), Carburetor ventilation instead of closed circuit breathing, Automatic Transmission, Low compression optional.

9) As 18GF, North American export, Carburetor ventilation instead of closed circuit breathing, Overdrive optional.

10) As 18GH, Evaporative loss control system, For cars for California.

11) North American export, 1971 model year, Evaporative loss control system, Overdrive optional.

12) Home Market, two SU HS4 carburetors, without Overdrive.

13) Export (not North America), Two HIF4 Carburetors, Low compression optional, without Overdrive.

14) Home Market, Two SU HS4 carburetors, with Overdrive.

15) Export (not North America), Two HIF4 Carburetors, Low compression optional, with Overdrive.

16) North American export, Two HS4 Carburetors, Automatic Transmission.

17) Export (not North America), Two HIF4 Carburetors, Automatic Transmission.

18) North American Export, 1972 Model year, Two HIF4 Carburetors, without Overdrive (to car number 294,987 on 1972 GT models).

19) North American Export, 1972 Model year, Two HIF4 Carburetors, with Overdrive (to car number 294,987 on 1972 GT models).

20) North American Export, 1973 & 1974 model years, Two HIF4 Carburetors, without Overdrive.

21) North American Export, 1973 & 1974 model years, Two HIF4 Carburetors, with Overdrive.

22) Home/Export (not North America), Two HIF4 Carburetors, without Overdrive.

23) Home/Export (not North America), Two HIF4 Carburetors, with Overdrive.

24) North American Export, "1974 ½" model year, Two HIF4 Carburetors, without Overdrive.

25) North American Export, "1974 ½" model year, Two HIF4 Carburetors, with Overdrive.

26) Home/Export (not North America), with Overdrive.

27) Home/Export (not North America), without Overdrive.

28) North American export, 1975 model year, single Stromberg carburetor, without Overdrive.

29) 1976 model year, Canada only.

30) North American export, 1975 model year, single Stromberg carburetor, with Overdrive.

31) 1976 model year, Canada only.

32) USA, single Stromberg carburetor, with catalyst, without Overdrive, at first for California only, from start of 1976 model year (386601) on all USA cars.

33) USA, single Stromberg carburetor, with catalyst, with Overdrive at first for California only, from start of 1976 model year (386601) on all USA cars.

34) USA (not California), w/ catalyst, without Overdrive.

35) USA not (California), w/ catalyst, with Overdrive.

36) California, w/ catalyst, without Overdrive.

37) California, w/ catalyst, with Overdrive.

38) As above, Continued for Japan.

39) Canada, no catalyst, without Overdrive.

40) Canada, no catalyst, with Overdrive.

Removing The Engine

Pulling the engine out of the car is considered to be something of a Rite Of Passage for MGB owners, and need not be an exercise in fear. Get at least one friend to help out, as there are moments when it is not an easy job on your own. Although it may seem that the removal would be easier if the engine and transmission were separated while still in the car, the easiest way for the amateur mechanic to do it is to pull the engine and transmission out as one unit with the engine hoist located directly in front of the car. It is possible to pull the engine separately, but to do so incurs the risk of damaging the first motion shaft of the transmission. As for those who feel that the first motion shaft of the transmission is essentially incapable of being damaged, I dare them to put their beliefs to the test by sliding their engine part-way onto the end of the shaft, and then releasing the hoist to simulate an accidental release!!! In addition, realigning the engine with the transmission still in place can be maddening for the unpracticed. To do so requires that the transmission be jacked up until it touches the transmission tunnel in order to clear the height of the front crossmember. Then comes the headache of getting the gap between the backplate and the bellhousing parallel all round while the engine is hanging from a hoist. For those with repeated experience, it's easy to forget how difficult it is for a first-timer, especially when he is working alone.

First, disconnect the ground (earth) on the battery. If you may be facing a rebuild of the engine, you will then find that it is best to remove the retaining bolt of the crankshaft's pulley wheel on the harmonic balancer (harmonic damper) prior to removing the engine from the car. In order to loosen the retaining bolt of the crankshaft's pulley wheel on the harmonic balancer (harmonic damper) for removal, the most professional method is to use an impact wrench after bending back the securing tab of the lockwasher. This method has the advantage that you do not have to worry about blocking the flywheel (if engine is separated from transmission, or blocking the wheels with transmission in gear). Another, and much more commonly used method for loosening the crankshaft's pulley wheel on the harmonic balancer (harmonic damper) retaining bolt, is to place a 1 5/16" wrench / breaker bar against the left chassis rail (as you are facing the engine from the front). Put some wood or something thick and soft against the left chassis rail in order to prevent it from being scratched. Your wife's pillow is good for this (she won't mind you borrowing it for such a worthy cause). Take care to disconnect the High Tension (HT) lead from either the ignition coil or the distributor cap so that the engine cannot start. Blip the starter briefly, and the retaining bolt will be jarred loose. When reinstalling, note that there is a flat on the middle round section of the harmonic balancer pulley that you must hammer the tab of the lock washer down onto. This holds the tab onto the harmonic balancer pulley wheel, and you can then tab up the head of the retaining bolt as normal. Be aware that the retaining bolt of the crankshaft's pulley wheel on the harmonic balancer (harmonic damper) in the end of the crankshaft is not standard Whitworth, but is of Whitworth form: Diameter 1 5/8"-16 TPI, 1 1/16 Full Thread. If it is found necessary to clean up the threads, the operation must be confined to cleaning up only. These threads are highly stressed and must always be up to full size. Thus, it is not correct to use an American SAE-UNF form tap or die in order to clean up these threads, though you might get by in the end of the crankshaft, but definitely not on the bolt. To my knowledge, apart from the 1/4-inch BSPP (British Standard Pipe Parallel) threads in the drain port on the side of the engine block, this is the only other British form threaded fastener on any of the B Series engines. Note that BSF is 5/8"-14, rather than 16 TPI. UNF is 5/8"-18. CEI starts with 20 TPI at ½" and larger.

Remove the gear change lever (gear shift lever) surround, raise the gearlever boot, then unscrew the gear change lever (gear shift lever) retaining bolts and lift out the gear change lever (gear shift lever). Drain the oil from the oil sump , and then disconnect the oil cooler hoses (flexible pipes) from both the engine block and the oil filter stand, and then disconnect as the case may be, either the oil pressure gauge hose (flexible pipe) or the oil pressure senor wiring from the engine. Disconnect both the throttle and choke (fuel / air control) cables, and then disconnect the fuel lines from the carburetors. Remove the carburetors as a single unit, the heat shield, and the intake manifold, along with the exhaust manifold, fan, distributor, alternator, heater valve, hot water pipe, hot water hoses (flexible pipes) and oil filter stand in order to both lessen the total amount of weight to be moved about and to also protect these components from being damaged. If your engine is equipped with antipollution equipment, it should also be removed prior to attempting to remove the engine from the car. This should be done not only in order to reduce both the weight and the bulk to be lifted and maneuvered, but to protect the antipollution equipment. Even if you have no intention of reinstalling it, you can always sell it on Ebay to an owner who lives in a location where emissions testing is a prerequisite for obtaining an inspection sticker! Drain the coolant from the radiator and, if you are fortunate enough to have a functioning petcock valve installed onto the side of your engine, drain the coolant jacket of the engine block as well. Next, disconnect the thermal transmitter for the coolant temperature gauge, and then disconnect the coolant hoses (flexible pipes) from the coolant pump and the coolant outlet elbow housing. Now, crawl under the car. Do not forget to disconnect the front mounting bracket for the exhaust system located on the bellhousing of the transmission and to remove the grounding (earthing) strap. Chrome Bumper models have the ground (earth) strap on one of the front rubber engine mounts where it connects the engine front plate to the chassis. The Rubber Bumper cars have their ground (earth) strap on the right transmission mount where it connects from the forward machine bolt that holds the transmission mount to the transmission and the transmission support crossmember. While you are under the car, remove both the electric starter and its solenoid, disconnect the clutch slave cylinder hydraulic hose (flexible pipe) and the clevis pin of its pushrod, and then remove the clutch slave cylinder from the bellhousing, as well as the speedometer drive cable from the main transmission casing. Next, disconnect the driveshaft (propeller shaft), and then disconnect the electrical connections of the solenoid that is located on the Overdrive unit. Crawl out from under the car, and then loosen the 3/8”-18 UNC machine bolts of the front rubber engine mounts, and then remove the gearshift knob and the shift boot retainer plate. Be aware that ¼ x 28 (fine thread) x ½” PoziDriv countersunk round head machine screws are used to attach the transmission tunnel cover to the transmission tunnel, and they screw into welded-on nuts on the reverse side of the transmission tunnel, except for the front one which has a similar nut on a removable plate that covers the extended hole forward of the shift tower. They are unaccountably long and usually of two lengths, and stick down into the tunnel, where their lower ends rust. The length (and pointed ends on Original Equipment ones) is apparently so that you can get all the various layers aligned on assembly. The original screws are not Phillips head screws, although commonly mistaken for such. Be warned that if you use a Phillips head screwdriver, you will chew up the heads of these PoziDriv machine screws. If this mistake has already been made, replacements can be found at these firms: McMaster-Carr at: http://www.mcmaster.com/ , MSC at: http://www.mscdirect.com/ , or Metric Multistandard Components Corp at: http://www.metricmcc.com .

Remind yourself of how much fun you are having, then crawl back under the car, and remove the machine bolts that secure the rear transmission mount to the underside of the car. Now, crawl back out from under the car and whistle a happy tune as you proceed to remove the 4 machine bolts that secure the oil cooler, and then remove the machine bolts that secure the radiator diaphragm to the body of the car. Remove both the radiator and the radiator diaphragm, along with the oil cooler as well as its hoses (flexible pipes) in order to both give more room in which to maneuver the engine / transmission package and to decrease the angle to which the engine / transmission package must be tilted, making its extraction from the engine compartment much easier. This will also avoid damage to the radiator. Raising the rear axle of the car up about 8 inches to 12 inches on jackstands will allow the tail end of the transmission to drop down lower and give you a better relative angle. Beg, borrow, or buy a load leveler mechanism so that you can alter the angle of the engine in order to allow maximum maneuverability as you lift it in cramped quarters and make the extraction much, much easier. You might feel that such a device is an unnecessary luxury, but it is worth every cent not to scratch up your paint or dent and / or crease the sheetmetal inside of the engine compartment. This is why professional shops always have a load leveler for removing engines!

Removing the fan from the engine is a good idea if you are using a mobile engine hoist, as on some types of mobile engine hoist, the fan can catch on it and be damaged. Place the base of the mobile engine hoist as close as possible to the engine bay and do not extend the arm of the mobile engine hoist any further than is necessary. Use the rocker arm studs as lift points only if you are certain that they are Original Equipment items as some of the replacement studs nowadays are of dubious quality. Most failures will occur as a load is applied at an angle to an attachment point, so make those attachments strong, or, better yet, make them nonexistent by using a sling. Although some use a length of chain enclosed in a bicycle inner tube, I prefer to lift the engine with a strap of heavy nylon webbing. Not only is it strong and easy to undo knots from, but its greater surface area in contact with the engine block makes slippage less probable to occur and it is less likely to damage paint. Pass the strap between the engine and its backplate, cross it over above the rocker arm cover and loop it under the coolant pump, and then tie the ends off with a simple square knot above the engine. With the hook placed behind the knot, it will not slip backwards, plus the square knot is self-tightening and will not slip either. Always remember the cardinal rule to never, ever, put any part of your body anywhere below a suspended engine.

Transmission Support Crossmember

When you prepare to reinstall the engine, jack up the rear of the car and set the rear axle onto axle stands so that you will have adequate clearance for the tail end of the transmission. Leave the engine tilted with the transmission at a lower level in order to make it easier for your fingers to install the machine bolts of the front mounts. Do not make the classic Beginner's Mistake of tightening down the front rubber engine mounts and then trying to install the transmission support crossmember onto the end of the transmission package. Instead, before attempting to install the engine, attach the transmission support crossmember onto the transmission and leave its 5/16"-18 X ¾" long mounting bolts loose. It is much easier to start the transmission bolts by hand, and then tighten the front rubber engine mounts with the engine hanging on the hoist before tightening the rear transmission mount.

You should note that there are two mounting holes with captive nuts on each side of the car for the machine bolts that secure the transmission support crossmember to the transmission. The reason for the two sets of holes is so that you can alter the position of the transmission support crossmember as appropriate. This feature is necessary because in the case of the three-synchro transmissions there is a difference between the distances between the front engine mount and the mounting point on the rear of the transmission of the standard transmission and the one that is equipped with an Overdrive unit. The transmission support crossmember needs to attach about an inch further forward for the Overdrive unit-equipped version of the transmission. Tighten the transmission support crossmember machine bolts using a half-height swiveling socket, with a four or five-inch extension. With this tool, you can get to those rear machine bolts a lot easier. However, if you are not fortunate enough to have a half-height swiveling socket available, the transmission support crossmember can be modified for easier installation. Slot the holes on the metal that the mounts sit on, and then drill two 24mm holes in positions that will allow a long extension bar with a socket to reach each of the machine bolts on the transmission mount. Next, bolt the transmission mounts onto the transmission, and then attach the "top hat" middle part. Bolt the transmission support crossmember and it to the top hat part. Using a long 3/8" drive extension you will be able to tighten the mounting bolts easily, before attaching the four transmission support crossmember-to-body machine bolts.

Front Engine Mounts

When new front rubber engine mounts and their brackets are installed, inspection usually reveals that the assembly is already bending toward the engine block. That means it is prestressed in compression, and as the engine rocks, the stress cycles from compression to tension and back again, ultimately leading to fatigue failure. This condition is at its most severe on the Left Hand engine mount bracket, since torque effect causes that side of the engine to lift under acceleration, whereas the Right Hand bracket tends to remain in compression, except during hard engine braking. If you fit a spacer of approximately 1/8" (.125" / 3.18mm) thickness between the bracket and the engine block at the large machine bolt, you will prestress the bracket in such a way as to prevent the cycling through zero, which reduces or eliminates fatigue failure. This compressive preload also keeps the rubber mount plates parallel, greatly increasing the life of the mount itself. If the mounts are correctly shimmed, then the force on the rubber mounts will be at right angles and they should not sag, even over a long period of time. The need for these shims is determined by the dimension across the mountings in the chassis, which varies due to build tolerances. You can determine if they are needed by examining the mounting rubbers. The sides should be at 90^{o} to the ends when under the weight of the engine. If they slope towards the engine at the top, then you need to add shims. If they slope away from the engine, then you need to remove shims.

Be aware that the later front engine plate of the 18V engines from the 1975 through 1980 Rubber Bumper models have a repositioned engine mounting boss on its camshaft side and thus will not accept the rubber engine mounts of earlier cars. The engine blocks that are fitted with this front engine plate can be readily identified by their engine numbers: 18V-836-Z-L, 18V-837-AE-L, 18V797-AE-L, 18V-798-AE-L, 18V-801-AE-L, 18V-802-AE-L, 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L, 18V-891-AE-L, 18V-892-AE-L, and 18V-893-AE-L.

In the case of the rubber engine mounts used in Rubber Bumper cars, this round type of engine mount (also used on the GTV8) theoretically does not need shims in order to correct the alignment as the chassis rail brackets have slots in them so that the studs can take up whatever position that they require. Those, as well as the angled faces of both parts, theoretically take up any dimensional differences that are likely to be encountered between the chassis rails. However, the stud on the mounting rubber will not drop lower than the point where the steel disc of the rubber engine mount hits the ledge at the bottom of the chassis rail. Consequently, most Rubber Bumper MGBs have two spacers on each side.

When replacing the late model rubber engine mounts that are used in the Rubber Bumper models, it is easiest to undo the nuts that secure them to the chassis brackets, and then jack the engine up so that the studs of the engine mounts clear the chassis brackets. The rubber engine mounts have a top and a bottom, of course. The bottom is a single stud that goes through a slotted chassis bracket and is accessed from underneath. These are real pigs, needing a 16-point combination wrench that has to be turned over twice for each flat! Of course, that is the easy side, the difficult side has to have the steering rack completely removed first!!

The procedure for replacing old front rubber engine mounts on a 1975-1980 Rubber Bumper model is a bit more involved than it is for the earlier Chrome Bumper models. The steering rack has to be moved all of the way forward in order to permit you to get at the nut holding the rubber engine mount to the chassis. Installation of new rubber engine mounts is impossible with the steering rack in place. Mark the orientation of the universal joint of the steering column so that it can be reassembled correctly. Next, lift the front wheels off the ground. Remove the four bolts that secure the steering rack in place, being careful to locate and keep any shims that are underneath it for later replacement. These are of critical importance as they effect the alignment of the two halves of the steering column. Now, gently move the steering rack forward. The front wheels will assume a massive toe-in position, but the steering rack will be out of the way so that the rubber engine mounts can be installed. Doing it in this manner does not disturb any of the steering geometry.

If the nuts are free on the bottom studs, it is easier to remove the upper nuts and bolts of the mount, then you can spin the mount off the nut. Once the nuts are removed, you will have to first tilt the engine in order to extract the first one, and then tilt the engine in order to extract the other. Next, remove the rubber engine mounts from the front plate of the engine. Note that the rubber engine mounts have their chassis plate studs offset from the center. When attaching the rubber mount to the brackets on the block, the stud must go into the lower of the two possible positions. Remember to reinstall any spacers on the appropriate side of the rubber engine mount. Many cars will have had a second spacer fitted on the driver's side in order to prevent the exhaust manifold from contacting the steering column as the engine mounts age and compress. If you find that you need to add a spacer, it is more easily fitted between the rubber engine mount and the chassis bracket. Because the rubber engine mounts are set at an angle, it is not possible to drop the engine straight onto the chassis plates with its rubber engine mounts attached, even though the chassis brackets are slotted. Tilt the engine in order to get one stud in, and then tilt the engine the other way so that the first stud is at the top of its slot and you should then be able to install the other stud. Before the stud goes all the way through, fit the lock washer and start the nut. When you lower the engine all the way, aim to get both of the studs at the same positions in their respective slots in order to ensure that the engine is correctly aligned. Expect the driver's side to be tricky because access is severely limited by the steering column passing through the chassis bracket. Wedge the nut into an open-ended spanner and stick the lock washer onto the nut with some chilled grease, and then place the nut/washer into the slot in the chassis bracket. When the engine is raised, you can carefully hold the nut and then spin the rubber engine mount in order to get it together, along with its spacer, into the nut a few turns, and then secure the mount to the engine plate. Under the bolt head you will need to fit a thick washer that has been contoured to fit inside of the bracket. If this machine bolt is bottomed in the hole, then the bracket will break, and the threads of the machine bolt will be damaged when you remove it. Rather than having to undo the nut completely and raise the engine enough for the stud to clear the chassis bracket, cut a slot in the rubber engine mount instead of a hole for the stud of the engine mount. You will then be able to slacken the nut, raise the engine slightly on that side, and then slide in the spacer. Bolting up the stud nuts is a long, slow job, as you have to turn the open-ended wrench (spanner) over twice for each flat, so be patient. These extra spacers will require a slightly longer machine bolt. The use of Loctite will ensure that the large machine bolt does not work loose. If it does, it will cause the bracket to fracture across its bolthole, in addition to the usual crack at the bend of the bracket. Do not omit the shim plates, and be sure that the mounts are driven to the bottom of the slots in the frame (if the original square offset spacers are fitted, they will only fit with the mount all the way down at the bottom). Note that the square spacer has an offset hole. The mounting stud goes all the way at the bottom of the slot in the frame (tap it down with some weight on the mount), and the offset of the square spacer serves to keep it there so it cannot move upward should it became loose. Because the threads are usually damaged, run a sizing die over the threads before installation so that the nut will be free-running. Be sure to use anti-seize compound on the threads so that any future disassembly will be a more straightforward affair. Install the square spacer under the bracket so that its widest part is uppermost, it just fits up against the top edge of the cavity. It helps if you can get the underside of the frame bracket clean and use adhesive to hold the square washer up while you install both the nut and the lockwasher.

Determining if you need to shim the rubber engine mounts is a simple matter because the rubber blocks deform if the engine is too low - the top and bottom faces will not be at right angles to the plate. Simply add shims equally to both sides until both of the rubber blocks sit square. If there are clearance problems with the bellhousing or the exhaust manifold / steering column, simply changing a shim from one side to the other will move the engine in the opposite lateral direction while leaving the engine at nominally the same height.

At first appearances, the installation of the rubber bushings into the transmission mount seems to many to be a formidable task. The smaller of the bushing's two flanges is 1¼" in diameter and about ¼" thick, while the hole through which it must pass is only about ¾" in diameter. It appears to be a job that requires a man with at least three hands. However, installation of the rubber bushings into the transmission mount is not as difficult as it initially seems to be. The hole through which it must pass is only about ¾" in diameter. First, this task can be made considerably easier by heating the rubber in hot water in order to soften it up, and then using a lubricant will make the procedure yet easier, as well as protect the rubber from chafing. Secure the yoke in a vise. Tie off one end of a thin cord, in the direction of one end of the yoke. Loop the cord, and then pull it up through the yoke hole. Pass the loop around the bushing, and then place the edge of the bushing flange into the yoke hole. As you do this, it helps to use your free hand in order to oblongate the bushing. Initially, pull the cords almost parallel to the bushing groove. As more of the flange begins to enter the hole, change your direction of pull downward, until eventually you are pulling straight down. In this manner, you will gradually peel the circumference of the flange through the hole. Tying off the cord leaves one hand free to manipulate the bushing, and also to change the pull angle of the cords. Obviously, the wiser you are at choosing your tie-down point, the better this procedure will work.

The Antipollution System

If your engine is a post-1967 North American Market MKII model, then it is equipped with an antipollution system. In order to get better performance out of the engine, it will be necessary to remove some of the components of this system. Prior to doing this, check with your State Officials to find out if this is illegal. Be advised that in some states where it is illegal to tamper with a vehicle's antipollution system, it is not required to be maintained once a car has reached a certain age, so specifically inquire about this issue as well.

However, you may find that you are required to retain a functioning air injection system. If this proves to be the case, then you will need to do some extra maintenance in order to get the system to perform properly.

Cleaning Air Injection Ports

When you remove and inspect the cylinder head, be aware that the air injection ports are commonly all plugged with carbon deposits. On the cylinder heads of engines that were originally equipped with an air injection system, the air injection ports are drilled into the spark plug side of the cylinder head. The bores are threaded and steel double flared tubing is inserted into these ports in order to seal against the air injection manifold. These tubes are about 1.5" in length and may become stuck inside of the cylinder head, however. In trying to clean the ports out, they may get stuck to the tool that you are using to clean them with. Hot tanking doesn't seem to damage them, but it may loosen them up enough to enable them to be removed.

Using an electric drill and the smallest drill bit that you have, setting the drill bit to go not much deeper than 1.5" plus the depth of the threaded portion of the port. Now, carefully drill out the crud. It is messy, so, if you happen to have a shop vacuum handy, you can use it to suck up the dust as you remove it. After the first pass with the smallest drill bit, switch to the next larger size drill bit. The internal bore of the tubing is roughly the same size as that of the air injection manifold, but always remember that the goal is to only bore out that deposit of crud, and not any metal.

You can test to see how successful you have been by reinstalling the cylinder head and starting the engine. If the ports are clear, then some exhaust gases will blow out of the injection port. With the ports clear, there will be a louder exhaust note, with a loud pop or bang for each cylinder along with a moderate puff of gas coming from each injection port,. It is not fiery hot when the engine is cold, but I do not know what it would be like when the engine is hot.

Once you have confirmed that the ports have been cleared, do your best to clean up the threads, the flares, and everything else before reinstalling the air injection manifold. Because the flare fittings on the air injection manifold commonly seize to the manifold tubing, making removal impossible without destroying the air injection manifold, I recommend coating the fittings and lubricating the tubing with Anti-Seize compound so that it moves freely during installation. Most anti-seize compounds are rated to 1,500° Fahrenheit (815.6° Celsius). That means the air injection manifold and the cylinder head would burn off their paint before the anti-seize compound burns away. Since engine temperatures rarely exceed 230° Fahrenheit (110° Celsius), and never get beyond 500° Fahrenheit (260° Celsius) without damage, I would not worry about that.

Removal of the Antipollution System

If your state's regulations do not forbid your removal of the antipollution system, be aware that it is desirable to retain certain items of the antipollution system, so do not start by simply stripping everything off. Instead, proceed with the same knowledgeable, methodical approach that you would use toward any other part of the car.

It is important to retain the crankcase ventilation system. Properly maintained, crankcase gases are drawn into the combustion chambers of the engine by the vacuum created by the fuel induction system, either through the intake manifold as in the 18GB, 18GD, and 18GF engines, or through the carburetors as in the later engines. This permits the crankcase to function in a partial vacuum of about -2 PSI which not only reduces power loss due to the pistons, connecting rods, and crankshaft forcing the atmosphere inside of the crankcase to move about (a condition that is technically termed either "Windage Loss" or "Pumping Loss"), it also causes the oil mist inside of the crankcase to condense more rapidly while being drawn upwards towards the camshaft and tappets. Because the oil mist becomes more quickly and more highly condensed in the partial vacuum, more of it tends to fall into the oil sump rather than remaining in suspension as a fine mist and being drawn into the fuel induction system. An oil separator that works on the turbulence principle is incorporated into the design of the front cover of the tappet chest in order to assist in preventing this. The mesh inside of the oil separator is designed to catch the oil mist in the air that travels from the crankcase to the carburetors. If it is in good condition, then the oil returns to the sump in liquid form and only air & fumes go into the carbs. On the other hand, if the oil separator is choked with carbon deposits, then the oil travels up the mesh because of the reduced availability of air passages as well as at a faster air speed. Consequently, gravity does not get a chance to return the condensed oil to the sump. Due to the fact that the effectiveness of the system is dependent upon vacuum, all connections between the fuel tank, the vapor separator, the adsorption canister, the rocker arm cover and its oil filler cap, the dipstick, the distributor, as well as between the oil separator and the fuel induction system, must be well sealed in order to maintain the optimum state of vacuum. Should you attempt to run the engine without the oil filler cap in place, you will find that because the airflow into the fuel induction system lacks restriction, so much air will be drawn into the fuel induction system through the intake manifold and thus bypassing the carburetors that the engine will not run.

In addition, without the partial vacuum induced by this system, the pressurized gases inside of the crankcase of the B Series engine would cause oil on the cylinder walls to be blown past the piston rings into the combustion chambers, leading to carbon buildup on the roofs of the combustion chambers as well as on the piston crowns, with consequent preignition problems. The carbon can also collect inside of the groove provided for the compression ring, causing the compression ring to seize (Bet'cha can't guess how I know this!). In addition, an excess of these pressurized gases and oil mist would also be vented partially through the breather tube of the rocker arm cover, and resulting in an oily film inside of the engine compartment of engines equipped with a vented oil filler cap (BMC Part # 12H 1836) of the 18GA, 18GB, 18GD, and 18GH engines. In the case of 18GJ, 18GK, and 18V engines equipped with a nonvented oil filler cap (BMC Part # 13H 2296), pressurization of the fuel tank as well as of the adsorption canister will occur, interfering with its function. Without some sort of pressure relief, normal engine operation will allow blow-by to create pressure inside of the crankcase that can force oil out through the path of least resistance. In such cases, a seemingly incurable seeping leak at the base of the rocker arm cover is the usually the first external symptom. Oil inside of the hose (flexible pipe) that connects the rocker arm cover to the adsorption canister would be an internal symptom of the occurrence of this event. In order for the excess pressurized gases inside of the crankcase to arrive at the rocker arm cover, they would also have to travel up the past the pushrods and the oil drainback holes that are located in the floor of the tappet chest. This means that the excess pressure of the gases would be forced upward around the tappets, decreasing the additional lubrication of both the balled lower ends of the pushrods and the upper sections of the tappets that is supplied by both the oil mist from the crankshaft and the oil that runs down the pushrods from the rocker arm assembly onto the upper sections of the tappets. The pistons would also have to work against the pressure trapped inside of the crankcase, retarding their downward movement (i.e., "Pumping Loss"), thus causing more combustion heat to be transferred to both the cylinder walls as well as to the roof of the combustion chamber, making the engine run hotter and reducing power output. Thus it must be understood that all of this is prevented by drawing all of the pressurized gases inside of the engine out through the front cover of the tappet chest and into the fuel induction system under the effect of an induced vacuum, and as such, the system contributes to long-term reliability, as well as to a prolonged lifespan of the engine.

If yours is an 18GA, 18GB, 18GD, or 18GF engine equipped with a Smiths PCV Valve (BMC Part # 13H 5191, Moss Motors Part # 360-630), it should be retained in order to reduce atmospheric pressure inside of the engine, thus reducing oil consumption and consequent accumulation of carbon inside of the combustion chambers, as well as reducing power-robbing windage loss. The intake port on the PCV valve is connected to a tube on the front cover of the tappet chest, which contains a wire gauze mesh that acts as both an oil-trap and as a flame-trap. The oil filler cap is of the vented type, which admits fresh air into the crankcase via a restriction orifice and a wire gauze filter. The restriction orifice limits the flow of air through the engine and the PCV valve into the intake manifold to a low level. This low level of airflow has two purposes: the first being the prevention of the fuel / air mixture from becoming too lean (weak), and the second being to provide a partial vacuum inside of the crankcase in order to ensure that crankcase fumes do not escape into the atmosphere, but instead are burnt inside of the engine. The PCV valve itself is a simple mechanism, consisting of a rubber diaphragm that has a coil spring-loaded valve beneath it. When the engine is not running, the valve is fully opened by spring pressure and the diaphragm is retracted. When the engine is running, airflow creates a vacuum within the crankcase and under the lower surface of the diaphragm, which with atmospheric pressure bearing down upon its upper surface, pushes the diaphragm down, overcoming the resistance of the coil spring and closing the valve. This reduces the airflow, which decreases the vacuum inside of the crankcase and below the inner surface of the diaphragm, which allows the diaphragm to move up again, opening the valve to give more airflow, which results in an increase in vacuum, and so on, and so on, ad infinitum, ad nauseum. In practice, the sprung diaphragm continually maintains a balance between crankcase pressure and atmospheric pressure, resulting in the relatively constant rate of airflow through the valve, and thus the engine, under conditions of varying levels of intake manifold vacuum.

However, the PCV valve is known to be prone to problems. The condition of its rubber diaphragm should be regularly checked. Should it rupture, considerable quantities of oil mist will be transferred into the combustion chambers through the fuel induction system. This will be evidenced by a telltale bluish exhaust smoke and a seemingly lean running condition. In addition, should the compression rings begin to fail, the resulting overpressurization of the crankcase will cause oil mist from the engine to saturate the oil separator tube of the early version of the front cover of the tappet chest and be transferred onward into the combustion chambers through the fuel induction system, the consequent reduction of the octane level of the fuel / air mixture and carbon buildup eventually resulting in problems such as preignition, sometimes called "pinging". If the oil filler cap on the rocker arm cover does not seal properly, you will lose the crankcase vacuum. As a result of the increase in pressure inside of the crankcase, the PCV valve will open further. When the PCV valve cannot increase flow enough to restore crankcase vacuum, it will be open as much as it can be, allowing an excessive amount of air into the intake manifold. The fuel / air mixture will become consequently lean, resulting in a fast and slightly rough idle. In addition, should the intake orifice of the rocker arm cover happen to become clogged, it will stop all ventilation airflow through the crankcase. This will cause a higher vacuum inside of the crankcase that will cause the PCV valve to close, making the engine idle slightly slower, and the fuel / air mixture will be a little rich, producing an abnormally slow, rough idle. This can foul spark plugs and cause carbon accumulation inside of the combustion chambers, as well as accumulating both water and contaminants inside of the crankcase. The resultant damage could lead to expensive repairs. That is why service instructions call for changing the vented type oil filler cap periodically. Third, if the piston rings or cylinder walls are worn beyond the normal operating tolerances, or if the piston rings are stuck in their piston grooves, the volume of blow-by gases produced by may be more than the PCV valve can process. In this case, the PCV valve will be completely open and there may also be an excess of pressure inside of the crankcase. This results in reverse flow through the intake vent of the rocker arm cover and possible expulsion of oil past the oil seals. In the interests of reliability, you may wish to replace the PCV valve with a more efficient front cover of the tappet chest that incorporates an external oil reservoir / return chamber from a later version of the engine.

The Tappet Chest Covers

The later rear cover of the tappet chest (BMC Part # 12A 1366) has the advantage of being less prone to distortion and leakage, partly as a result of its having a groove for retaining the later synthetic rubber oil seal that is less prone to heat-induced failure. The front cover of the tappet chest for the later 18V engines (BMC Part # 12H 4395), found on 18V-797-AE, 18V-798-AE, 18V-801-AE, 18V-802-AE, 18V-846-H, 18V-847-H, 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L, 18V-891-AE-L engines) is preferable to preceding designs due to its more open design that permits better breathing characteristics and for having incorporated into its design a superior external oil reservoir / return chamber which minimizes the transfer of oil mist into the fuel induction system. Because the external oil reservoir / return chamber is often partially clogged with carbon deposits as well as dried oil residue, it should be soaked in a strong solvent such as carburetor cleaner, and then thoroughly blown out with compressed air prior to reinstalling it. If it is reinstalled in a partially clogged state, it will inhibit the outward flow of gases from within the crankcase, sometimes to the point of creating overpressurization and attendant oil leakage from whatever weak sealing may be present anywhere on the engine. In addition, it will not be able to properly condense oil mist. The oil mist will then be drawn into the fuel induction system and reduce the octane level of the fuel / air mixture, which in turn will lead to preignition. It will also result in carbon buildup on the roof of the combustion chamber as well as on the heads of the valves and the piston crowns, either of which can also result in preignition. You may find that it will be necessary to use the 18V rocker arm cover (BMC Part# 12H 1836) with its built-in restrictor in the elbow. That system was designed to work with the later, more open, form of front cover of the tappet chest.

When installing the gaskets onto the covers of the tappet chest, remember that the rubber O-rings on the bolts tend to take a set when left in place, so always replace them with new ones in order to obtain an effective seal. Use either Permatex Ultra Black RTV Gasket Maker, Permatex Aviation Form-A-Gasket sealant, or Loctite Hi-Tac to glue the gaskets to the covers and then allow it to harden overnight so that they will not move during installation. Do not use silicone-based Permatex Blue RTV or Permatex Red Ultra RTV sealant on any of the engine gaskets as they are both prone to failure and contamination of the engine oil under hot operating conditions. In addition, they can also substantially contribute to swelling of the seals. Excessive swelling of a shaft seal's elastomeric lip is a good indicator that the lip material and the lubricant in use are not compatible. Swelling of the seal material can be particularly problematic if some materials, such as silicone, come into contact with oil at high temperatures. Under such conditions, softening, swelling, and reversion of the seal material can occur. If material swell becomes an issue, check to be sure that the elastomer in use is compatible with the lubricant and any other fluids coming into contact with the seal, either during cleaning, installation, or operation. This includes any solvents that may be used during teardown. You should also check to be sure that system fluids are not being contaminated in some way; contamination could cause an otherwise acceptable seal material to swell or degrade.

Do not be mislead by any product with a term such as “gasket maker” included in its name. This is merely a marketing ploy. There is no reason to feel confident without the structural element of the actual gasket, so never attempt to use any sealant as a substitute for an actual gasket. The 5/16”-24 UNF nut for the shallow rear cover of the tappet chest should be torqued to 2 Ft-lbs, while the 5/16”-24 UNF nut for the deeper front cover of the tappet chest should be torqued to 5 Ft-lbs. Exceeding these torque values may result in distortion of the tappet chest covers, as well as crushing of the gaskets, leakage being the result.

If you choose to not remove the hose (flexible pipe) that leads from the fitting on the center of the intake manifold to the gulp valve, it can be simply blocked with a plug, or, after removing the intake manifold, threads can be tapped into the intake manifold with a ¼” (.250”) NPT tap and a nipple installed to function as a plug.

At this point, you may remove both the hoses and the check valve that connect the air pump to the Air Injectors atop the cylinder head. Next, remove the air pump, its air cleaner, and the attendant mounting brackets. When the engine is equipped with the air pump, the gulp valve is necessary in order to prevent backfiring when closing the throttle at high engine speeds, so remove the gulp valve along with its hoses and its attendant hardware as well. At idle speed the intake manifold vacuum with the gulp valve is in the order of 18” Hg to 20” Hg, while on the overrun it rises to 23” Hg to 25” Hg without the gulp valve. This is not enough to make a significant difference in terms of the amount of fuel pulled out of the fuel jet; thus the gulp valve is unnecessary once the air pump is removed.

If the idea of removing your air pump causes you to experience pangs of conscience, consider the fact that the hot exhaust gases pass through the exhaust system in pulses, not as a steady flow. When a high-pressure pulse of exhaust gases is passing the air injector port inside of an exhaust port, an anti-back-flow valve prevents the hot gases from entering the air injector system to any meaningful degree. The air injected into the exhaust port can enter only when pressure inside of the exhaust port has decreased, which is after the inertia of the pulse of exhaust gases has carried it well past the air injector port, leaving a low-pressure area in its wake that permits the anti-back-flow valve to open. This being the case, the air that is injected is always sandwiched in between the pulses of exhaust gases, mixing with it only after entering the turbulence inside of the muffler (silencer), at which point temperatures have dropped to the point that combustion has ceased, thus accomplishing little other than the dilution of the exhaust gases exiting the exhaust pipe. Way back when the system was introduced, the EPA measured for pollution in only terms of a standard of "Parts Per Million" (PPM, as they called it back then), thus the diluting system helped to satisfy the EPA standards, along with such tricks as lowering the compression ratio, leaning out the fuel / air ratio, and changing the ignition timing curve to initiate combustion earlier during the compression cycle. However, several ecologically-minded scientists and liberal politicians, emboldened at the time by having recently succeeded in forcing Congress to legislate an eventual ban on leaded fuels, loudly protested that the technology being used by the shameful capitalist auto industry was a sham perpetrated by its bourgeois lackeys at the expense of the poor, suffering, downtrodden masses in order to prevent themselves from being forced into spending millions of dollars of precious plutocratic profits on the development and use of "meaningful" technologies. When the EPA responded to the mounting political pressure by changing its test standards to reflect the actual total amount of pollution emitted, manufacturers quickly switched to catalytic converters and quietly discontinued their use of air pumps.

Next, remove the Air Injectors from the cylinder head and replace them with 7/16"-20 UNF fine-threaded iron machine bolts ¾" in length, making sure to coat heir threads with antisieze compound. These somewhat rare items can be obtained from any supplier to boiler repair shops. Do not be tempted to use steel Allen head set screws because they will have to be bottomed out into the cylinder head in order for their threads to create an effective seal. Should a casting defect be present, the resulting stress stemming from the different coefficients of expansion between that of the steel of the Allen-headed plug and that of the grey cast iron of the cylinder head can result in cracks forming between the walls of the exhaust ports and those of the coolant passages adjacent to where the plug is seated. This does not occur when the steel air injector plugs are seated in place because, being hollow, the steel of which they are fabricated can expand inwards and thus not place any stress upon the material of the cylinder head. A rare practice, in the event that the air pipes need to be replaced in future, is to put a ¼" ball bearing under the plug in order to prevent the plug from damaging the seat. However, jamming a ball bearing between the bottom of the plug and the seat would both concentrate and increase the thrusting stress on the thin port wall. Ba-a-a-d practice. Should cracking occur, the coolant system will be pressurized by the venting exhaust gases when the engine is running, leading to leaks at the hose (flexible pipe) junctures, vapor lock inside of the coolant system, and, in some cases, a blown cylinder head gasket. When the engine is not running and the exhaust valve is closed, coolant will puddle atop the exhaust valve as well as leak into the exhaust system. If the exhaust valve is open, the coolant will enter the combustion chamber and trickle down into the crankcase, polluting the oil. One might reason, “Are we not to also fret about the same thing because of steel studs securing the manifolds to the cylinder head?” The steel cylinder head studs and their mounting threads in the cylinder head are engineered to work together, while a solid steel Allen-headed plug and the threads in the air injector ports are not, so such reasoning in such a case is fallacious. Of every ten cylinder head castings that Peter Burgess examines for their potential for rebuilding purposes, he has to reject nine due to cracks having already developed, most commonly in the vicinity of the exhaust valve seats for #2 and #3 cylinders. In our case, we are not dealing with brand new grey iron cylinder head castings, but old, tired ones. Having your cylinder head reworked by an expert such as Peter Burgess is not cheap, but is a highly worthwhile investment. However, the prerequisite removal of material from the interior of the passages and combustion chamber further weakens the casting. Putting it at risk by using Allen-headed steel plugs thrusting against steel balls when you could use something more appropriate is just plain foolish, no matter how theoretically minimal you may think the risk might be. In our reality, it is usually worse than you might think at first guess.

If your car is equipped with an adsorption canister BMC Part # 13H 8511), do not discard it. This system uses absolutely no energy to operate, so it should be retained in functional condition. Two of the three ports on the top are connected to the fuel tank and float chamber vent ports of the carburetors. Any fumes from expansion of fuel or the filling of the fuel tank or carburetor float chambers migrate into the adsorption canister, there to be adsorbed by the activated charcoal granules, the resultant fume-free air being vented out of the port at the bottom of the adsorption. As the level inside of the fuel tank drops while driving, fresh air travels in the opposite direction through the adsorption in order to replace it. The third port on top of the adsorption is the important one as far as crankcase ventilation is concerned. On engines that are equipped with this system, the oil filler cap is a non-vented type, and a tube with a restrictor is provide on the back of the rocker arm cover.

This port is connected to the third port on top of the adsorption canister. The restriction in the rocker arm cover tube provides the same function as before, i.e., preventing excessive weakening of the fuel / air mixture as well as ensuring a small negative pressure inside of the crankcase. The vacuum created by the fuel induction system pulls fresh air in through the bottom port at of the adsorption canister, through the granules which purges them of any adsorbed fumes as well as filtering particles out of the drawn-in air, through the engine picking up any oil fumes, any fumes from either source being burned in the engine.

However, over time both the filters and their enclosed activated charcoal degrades into fine particles to the point that fine particles of it can be drawn into the crankcase. One of the things that you do not want inside of your crankcase is fine particles of charcoal. Rebuilding the adsorption canister is not difficult. The bottom unscrews in order to expose the retainer ring. The retainer ring is the piece that looks like a wheel that has three spokes. Notice that the retainer ring has six tabs in it that fit into matching slots in the bottom of the adsorption canister. These have to be very carefully depressed in order for the retainer ring to be released. After the retainer ring has been removed, you will find a filter element that is made of plastic gauze. Remove it, and then carefully examine it. If it is in decent condition, then set it aside for re-use. Underneath of the plastic gauze filter element you will find the old activated charcoal. It has the appearance of black pellets. These should be discarded. Under the activated charcoal, you will find another plastic gauze filter element like the first one, and a steel mesh screen. If the steel mesh screen is intact, save it for re-use. Next, you will find a special washer and a spring. These should be cleaned for re-use. Should the two plastic gauze filter elements appear to not be serviceable, you can make replacements from filter paper by cutting it into 3 ½" (89mm) circles. Lacking this, you can use plain old coffee filters. Look around for some large enough to cut the discs from without leaving any holes. For replacement activated charcoal, use common aquarium filter activated charcoal that is available from any store that sells aquarium supplies. You will need about eight ounces of activated charcoal. Rinse it with pain water in order to remove any stray fine particles, and then dry it inside of an oven. When you are ready to reassemble your adsorption canister, place the spring and the special washer into position inside of the bottom of the adsorption canister. Next, install the steel mesh, and then place either your old plastic gauze filter element or your new, homemade paper filter on its top. After that, put in about eight ounces of the new activated charcoal and then gently shake it down in order to settle it. It should fill the adsorption canister up to the point where the other filter and retaining ring will just fit in place. It you overfill by a small amount, it will not matter as the spring will compress enough to keep pressure on the activated charcoal. Note that it is necessary for the entire packed bed of activated charcoal to be held firmly in place by the spring so that friction will not produce fine particles that can foul the filter. Place the second filter element on top of the activated charcoal, and then place the retainer ring on top of it. Be sure that the retainer ring locks in place with its six tabs in their matching slots. Next, screw the bottom of the adsorption canister back into place, reinstall the adsorption canister into your car, and then reinstall the hoses. The center port on the top of the adsorption canister is for the purge line hose (flexible pipe) that goes to the rocker arm cover, while the port on the front of the top of the adsorption canister is for the vapor hose (flexible pipe) that attaches to a Y connection to the two vapor hoses (flexible pipes) which are attached to the overflow vents on the float bowls of the carburetors, and the port on the rear of the top of the adsorption canister is for the vapor hose (flexible pipe) that comes from the expansion chamber for the fuel tank. The port on the bottom of the adsorption canister is for the air vent line hose (flexible pipe). Note that heater hose (flexible pipe) material does not work well in these applications. In addition, avoid plastic hose (flexible pipe) as it has a tendency to harden and crack. Instead, go to a good auto parts store, such as NAPA rather than a discount parts stores, and tell them that you need an emissions-quality hose (flexible pipe) for the hose that connects the center port of the adsorption canister to the rocker arm cover and also for the hose from the front cover of the tappet chest to the fuel induction system. This hose (flexible pipe) is also known to some vendors as “oil resistant” hose (flexible pipe) and, like fuel line hose (flexible pipe), is engineered to withstand the presence of both oil and fuel, as well as their vapors. The smaller hoses (pipes), both on the adsorption canister and the vapor separator can me made from Original Equipment fuel line hose (flexible pipe) with no problems.

Avoid the steel-braided hoses (pipes), as when they perish you cannot tell until fuel leaks from them, and their high pressure capacity is irrelevant in a fuel system that operates at a pressure of a mere 2.7 to 3.8 PSI. A word of warning about the type of clamps used to secure the fuel hoses (flexible pipes): Over time, the rubber of the fuel hoses (flexible pipes) will suffer distortion under the clamp and thus reduces the pressure between fuel hose (flexible pipe) and the mounting tube. Hoses (flexible pipes) that have been secured with worm-type threaded type clamps will tend to weep, whereas fuel hoses (flexible pipes) that are secured with clamps of the “constant pressure” spring clip type do not suffer from this problem.

One seldom-thought-of method of reducing the chances of preignition and detonation while running under heavy load conditions at high engine speeds is to install an electronically controlled Exhaust Gas Recirculation (EGR) system and a knock sensor in order to trigger its operation. Originally conceived of as a method to reduce emissions of nitrogen oxides (NOX) pollution in the exhaust gases, it recirculates minute amounts of exhaust gases into the intake manifold through the EGR valve, the volume of exhaust gases being determined by the Inside Diameter (I.D.) of the orifice mounted atop the crossover balance tube of the intake manifold. The greater the volume of exhaust gases being recirculated, the less sensitive to heavy load conditions the engine will be. By carefully tuning the recirculation system, it is possible to extract the highest level of performance under normal load conditions without endangering the engine under conditions of extremely heavy loadings. Because the exhaust gases contain almost no oxygen, the metering of the fuel / air ratio need not be altered. Although these exhaust gases are hot, they actually have a cooling effect on compression / temperature ratio by diluting the fuel / air mixture slightly and thus reducing charge density. This decreases the Effective Compression Ratio (ECR), consequently reducing both the octane requirements of the engine and reducing its compression / temperature ratio to a point that preignition and detonation does not occur, as well as having the side benefit of reducing the formation of NOX. Because some of the intake charge is entering through the crossover balance tube of the intake manifold, velocity at the fuel jet bridges of the carburetors is also decreased, resulting in reduced atomization of the fuel. The larger droplets of fuel thus produced take longer to combust, which also decreases the likelihood of preignition or detonation. If your intention is to build a very high compression engine (Geometric Compression Ratio (GCR) above 9.5:1) using cast iron cylinder heads and you are unwilling to compromise on what would normally be the optimum valve timing or ignition spark curve for your desired performance characteristics, this is an option that you may want to have available.

You should retain the Anti-Run-On Valve (BMC Part # 12H 4295, Moss Motors Part # 367-110) fitted on the 1973 and later models as its purpose is to induce such a strong vacuum to the chamber above the fuel inside of the float bowls that the fuel cannot exit the fuel jets when the ignition is switched off, thus preventing the car from running on. When the ignition is turned off, the ignition switch energizes this solenoid-actuated Anti-Run-On valve in order to close it, and then the oil pressure switch (BMC Part # BHA 5197, Moss Motors Part # 141-715) releases it after the engine has stopped and oil pressure has fallen. The Anti-Run-On valve is open when the engine is running, allowing fresh air to be pulled through the adsorption canister, clearing it of the vapors that have expanded into it from both the fuel tank and the carburetor float bowl chambers, then through the rocker arm cover and the tappet chest into the fuel induction system to be consumed inside of the combustion chambers. The rocker arm cover (BMC Part # 12H 3252) of the North American Market 18GJ, 18GK, and 18V engines is equipped with a 5/64" restrictor tube in order to prevent the fresh air that is being drawn in through the rocker arm cover from overly diluting the fuel / air mixture and causing lean running. The size of the hole in the restrictor tube of the rocker arm cover has been restricted to 5/64" in order to create a significant -2 PSI vacuum both inside of the crankcase and to the fuel induction system. If you enlarge the size of this hole, then you will adversely effect the tuning at idle speed. This may result in the loss of smooth idle, or an inability to idle at the correct low engine speed. It will also result in an overly-rich fuel / air mixture off-idle, the vacuum leakage remaining relatively static and not increasing proportionally as the throttle is opened. Much the same happens with a genuine vacuum leak, i.e., at the intake manifold gasket.

This Anti-Run-On system can be readily retrofitted onto 1970 through 1971 18GJ and 18GK engines as well as onto the 1972 18V-584-Z-L and 18V-585-Z-L engines, all of which have the necessarily modified fuel tank (BMC Part # NRP 4), adsorption canister (BMC Part # 13H 5994, Moss Motors Part # 367-100), nonvented oil filler cap (BMC Part # 13H 2296, Moss Motors Part # 460-125), nonvented fuel tank cap (BMC Part # BHH 1663, Moss Motors Part # 202-755), and restrictor tube equipped rocker arm cover (BMC Part # 12H 3252) as Original Equipment. Conversions of earlier engines will need all of these items, as well as the oil pressure switch (BMC Part # BHA 5197, Moss Motors Part # 141-715). Do not remove or disconnect the Vapor Separator that connects the fuel tank to the Adsorption Canister. These procedures having been performed, you can now set out on a quest for more power.

Finally, if your engine is from a post-1974 model, remove the EGR Valve and its hose (flexible pipe) and control pipe, the fuel shutoff valve, and the vacuum advance valve. Be aware that the 1975 and later Rubber Bumper cars made for the North American Market had a vacuum advance valve mounted atop the cover of the pedal box. Its purpose was to allow the vacuum advance control feature on the distributor to function only in fourth gear. This was an emissions-related feature that actually damages performance in the lower three gears. As such, it can be deleted with no fear of failing a state-mandated emissions test, as its removal will have no effect upon an idling engine.

Preparing A Foundation For More Power

You must accept the fact that more power will increase both wear and stress on your engine's components. Hence, it is important that the basic components and systems of the engine provide a sound foundation. Remember: if anything is worth doing, it is worth doing right.

Never reuse old gaskets, seals, oil gallery plugs, frieze plugs, core plugs, flywheel bolts, camshaft bearings, bushings, main bearings, tappets, valve springs, thrust washers, piston rings, circlips, wrist (gudgeon) pins, rocker pedestal studs, rocker shafts, cylinder head mounting studs, manifold mounting studs, connecting rod bearings, rocker arm bushings, connecting rod bolts, or the crankshaft main bearing cap studs or bolts. None of these items are expensive, and recycling them into your engine is not only false economy, but also an open invitation to future mechanical failure.

Have all of your components, including the oil sump , rocker arm cover, crankshaft, engine block, heads, connecting rods, and rocker arms hot tanked in caustic cleaning solution in order to remove the years of accumulated crud that is to be found inside of all old engines. Prior to this being done, insist that all of the gallery/core/frieze plugs be removed from the engine block so that the chemical solution can get into all of the spaces inside of the engine block. However, be aware that all too often hot tanking alone is insufficient; it is very common to have serious blockages that do not completely dissolve out. Consequently, it is necessary to remove all of the plugs of the all of the oiling passages inside of the engine block and then clean them out with brushes. Unthreaded plugs can be removed by drilling them and threading them with ¼-20 UNF tap, and then using a socket head cap screw and a socket in order to remove them. Use rifle bore brushes to thoroughly clean all of the passages. Once you think that you have them clean, do not be surprised if you see more crud remaining inside of the nooks and crannies. Dig it out, and then clean the engine block again. If you do not, it will come loose and destroy your engine, more often sooner than later.

In order to be aware of the location of the various plugs and fittings that will need to be removed so that the passages within the engine block may be adequately cleaned, it is best to understand the engine's oiling system itself.

Just above the oil pump, a passage runs horizontally from the pump outlet port to the back of the engine block where it is blocked with a press-fitted plug. There it intersects a lateral passage, which is located a few inches toward the centerline of the engine block. This lateral passage is plugged on the outside with a 3/8” BSP threaded hex-head plug that is sealed by means of a copper washer and serves to duct oil to a point just inboard of the position of the seat of the oil pressure relief valve. There it intersects with a descending passage which serves to conduct oil flow to the input end of the oil pressure relief valve. This descending passage is stopped at its bottom with a press-fitted plug. Another passage passes laterally and parallel beneath the upper passage in order to intersect the descending passage. Its inner end is of a relatively smaller diameter, while the outer end is counter-drilled to a larger diameter and machined in order to form the tapered seat for the oil pressure relief valve, allowing for a slip-fit of the relief valve. The outer end of the passage is plugged with a ½” BSP threaded domed spring cap, which retains the compression spring for the oil pressure relief valve. This plug is factory-sealed with two fiber washers, but one sealing washer will usually work just as well. In order to allow for fine adjustment of relief pressure, building up thickness of sealing washers will reduce tension of the spring of the oil pressure relief valve oil and thus slightly relieve pressure on the body of the relief valve if so desired. In a corresponding manner, installing shims under the spring inside of the oil pressure relief valve will increase the tension of the spring of the oil pressure relief valve and thus increase the pressure of the oil relief valve. However, a less time-consuming method of adjusting the relief pressure is to install an adjustable oil pressure relief valve from Advance Performance Technology (APT Part # OPRV-ADJ). This clever, yet simple device permits the tension of the pressure relief spring to be adjusted to any desired level while the engine is running.

A descending passage intersects the oil pressure relief valve passage immediately outboard of the seat of the oil pressure relief valve. This passage is left open at the bottom in order to provide an open circuit for the oil from the oil pressure relief valve bypass to drain into the oil sump . Another descending passage passes parallel to it in order to intersect the oil pressure relief valve passage farther outboard, well behind the oil pressure relief valve, and is plugged at the bottom with a press-fitted plug. A lateral passage connects both of these descending passages and is stopped at the outside with a press-fitted plug. In combination, the lateral passage and the inner descending passage make available a free-flowing vent from the rear of the oil pressure relief valve to the oil sump . This allows the oil pressure relief valve to move freely with no pressure interference from behind. The shallow hole in the side of the engine block near the plug is a tooling hole that was used for alignment purposes during the original machining of the engine block.

An ascending passage passes parallel to the back of the engine block and at an angle from the right rear corner of the engine block in order to intersect the upper lateral passage. This sends oil to the high-pressure gallery, which runs the full length of the right side of the engine block to intersect the ascending passage, and is blocked at both ends with press-fitted plugs. This ascending passage terminates on the right with a special threaded fitting that is sealed with a copper washer. This fitting has a long nose that extends into the engine block with a very close fit inside of the passage so that it will be sealed around its nose. It then ducts oil flow from the passage to the outside of the engine block while blocking any cross-flow between the ascending passage and the high-pressure gallery. With the fitting properly installed, oil exiting the engine block passes through external plumbing for the oil cooler and / or through the oil filter assembly to reenter the engine block on its right side, flowing into the high pressure gallery which feeds the main bearings of the crankshaft by means of descending passages that pass obliquely upwards from the gusseted main bearing saddles. Oil then flows thenceward to both the big end bearings of the connecting rods and to the camshaft bearings. Should an improper fitting without this extended internal nose be employed, the oil will pass freely from the rear lateral passage into the high-pressure gallery, bypassing the oil cooler and / or the oil filter.

On top of the oil filter mounting area on the right side of the engine block is a downward-angled passage that is fitted at its outer end with a press-fitted plug. This passage marginally intersects the tapped passage for the oil filter mounting bolt and exits inside of the crankcase just aft of the center web of the engine block casting and just ahead of the #3 cylinder bore. This passage serves as a drain in order to eliminate any possible hydraulic lock when installing the center bolt for the oil filter canisters used on the 18G, 18GA, 18GB, 18GD, 18GF, and 18GH engines so that the for the center bolt oil filter canister can be screwed all the way in without resistance, and then accurately torqued to 15 Ft-lb.

The low-pressure gallery runs the full length of the engine block above the camshaft, and is fitted at both ends with press-fitted plugs. Oil is fed into the low-pressure gallery from the center camshaft bearing through an extension of the passage from the crankshaft's center main bearing. The low-pressure gallery intersects the journal passage for the top spigot of the oil pump driven gear, supplying oil to both it and its drive gear on the camshaft.

An ascending passage intersects the rear camshaft bearing in order to feed oil from the rearmost camshaft bearing upward into the cylinder head. The oil for lubricating the rear camshaft bearing is fed into the bottom of the bearing. The rear journal of the camshaft has a circumferential groove and two opposite grooves running part of the way across the length of the journal so that when the grooves align with the passage inside of the bearing as the camshaft rotates, oil pulses through to the upper oil passage and onward to the rocker shaft. Corresponding with this passage is another ascending passage. A horizontal passage runs from the back of the cylinder head forward about 2” below the rear exhaust port, intersecting the ascending passage and is stopped at the back with a press-fitted plug.

Another ascending passage intersects the horizontal passage in order to feed oil into the rear rocker shaft pedestal and lubricate the rocker arm assemblies. It should be noted that, with the exception of the 12H 906 cylinder head casting that was used on the 18G, 18 GA, and 18GB engines, drainage channels are cast into the top surface of the cylinder head in order to duct oil from the vicinity each cylinder’s set of valves to the pushrod passages for additional lubrication of the intake tappet. This is due to the fact that the 19% heavier weight of the intake valve produces greater pressure upon the interface of the lobe and the tappet, plus the lobe for the intake valve has a steeper ramp profile than that of the exhaust lobe, and thus benefits from a greater degree of lubrication.

Within a visible depression immediately behind the front engine plate on the lower right hand side of the engine block is a small flush-fit press-fitted plug. This plug closes a cross-drilled passage that supplies oil from the front camshaft bearing to the camshaft drive chain tensioner. Above the sump flange nearby there is also a slotted screw plug that blocks its unused port. This unused port is for the dipstick tube that is used on other versions of the B Series engine.

Hot Tanking

Be sure to remove the aluminum Engine Number Tag from the engine block prior to hot tanking, as the caustic chemicals will dissolve it. The engine number plate on MGB engine blocks is held in place by two rivets that are driven into holes in the side of the engine block. Be aware that these rivets have steep wedging threads on their shanks. Simply file a notch on either side of each rivet so that it can be securely gripped, clamp a set of vice-grip pliers onto the rivet, then twist the rivet counterclockwise (anticlockwise). You will find that using this method allows the rivets to come out quite easily. Always discard them and replace them with new ones. If the engine identification plate is missing, there is a way to date the age of the engine block. On the right hand side of the engine block, in the area between distributor and oil filter, there are three numbers that form a circle and that are slightly raised, e.g., 30 12 71, which tells the day, the month, and the year during which it was cast (in this case 30th day of December, 1971). The top one is the day, the left one is the month and the right one is the year. In the early 1970s the month code changed to a letter, thus 12 G 3 would be 12th day of July 1973.

After hot tanking, all of the internal passages should be chased out thoroughly with brushes and flushed. Be sure to tell your machinist that the area around the rear cylinder inside of the coolant jacket of the engine block is commonly a trap for sediment and to be sure that all of it is removed.

On bare metal, rust forms quickly. It is important to clean off any surface rust and be sure the surface is chemically clean. Some facilities have a second hot tank with an acid-based solution for removing rust. However, should this not be available, you can remove the rust yourself. Under no circumstances should hydrochloric (muriatic) acid be used to remove rust from any of the engine components. It will chemically interact with the rust and impregnate the remaining iron surface with hydrogen, resulting in hydrogen embrittlement of the metal that will lead to cracking. Hydrochloric acid reacts with the iron oxide (rust) to form soluble ferric chloride, thus leaving a clean exposed metal surface. However, it then also reacts vigorously with the iron to form ferric chloride and hydrogen. If you leave an iron engine block in a bath of hydrochloric acid long enough, you will eventually have a bath of ferric chloride (and enough hydrogen for a Zeppelin!). Instead, use Naval Jelly, which contains phosphoric acid. Being a thick gel, it will cling to the surface being treated instead of running everywhere as would happen with an acid that is in liquid form. Phosphoric acid reacts with the iron to form ferric phosphate, which adheres to the surface. This protects the surface and also slows down the reaction, so it is more or less self-limiting. Phosphoric acid reacts very slowly with iron, thus hydrogen emission is much lower and hydrogen embrittlement of the metal is insufficient to present a significant structural problem that can result in the formation of cracks.

After removing the rust, rinse the naval jelly off thoroughly, then blow the metal dry with compressed air or your wife's hairdryer (she will not mind you borrowing it for such a noble purpose, of course). Do not be surprised when afterwards you notice a dark-purple-hued, thin passivated layer of ferric chloride where the phosphoric acid has removed the rust. This is a natural result of chemical interaction. In order to prevent rust, apply a coat of WD-40 to the coolant passages inside of the cylinder head, the coolant jacket, and to all of the oil passages. Once this has been done, take care to prevent machining chips and machining dust from getting into the ports and the passages by blocking them off either with rubber plugs (available at most better hardware stores) or with short lengths of tapered wooden dowels.

As an aside to the subject to rust removal, there is a safe, simple non-destructive process of rust removal for smaller parts that will have no effect on the physical dimensions of the article that is to be cleaned of corrosion. Upon completion of the process, there will be no damage other than that which has been caused by the original corrosion itself. All that is required is a solution of 100 grams of sodium carbonate to every liter of distilled water that will be needed in order to completely cover the item that is to be cleaned of corrosion. A suitable-size plastic or glass container and a 6 Volt battery charger capable of supplying 4 to 6 Amps on a continuous basis without overheating is also required. Fill the container with the solution of sodium carbonate and distilled water, and then suspend a lead strip into it in order to function as an Anode. Fold the lead strip so that it overhangs the side of the container and projects into the solution, keeping it separate from the items that are to be cleaned. Next, suspend the items to be cleaned from an electroconductive metal rod or tube that is firmly fixed to the top of the container. This metal rod or tube can be made of copper, likewise the wires that are to be used for suspending the items. WARNING: It is important that the items being cleaned do not come into contact with each other as they will certainly be damaged should arcing between them happen to occur. With both the anode and the items to be cleaned suspended in the solution, connect the positive (+) red lead from the battery charger to the lead anode. Next, connect the negative (-) black lead from the battery charger to the copper rod/tube supporting the items to be cleaned. Finally, connect the battery charger to a power supply and turn on the power. A slight effervescence will be noticed rising in the solution from the items being cleaned, signifying that the process has begun. You will soon notice small flakes of rust falling off from the items as it is loosened. A few hours will remove light to medium rust, but severe rust will take considerably longer. Leave the items in the solution for as long as it takes, because as I have already said, no damage will occur to them at all. For stubborn rust deposits, the process may be hastened by lifting the offending items from the solution, scrubbing them with a soft nylon brush, and then returning them to the solution. When clean, remove the parts from the solution, then rinse them in clean warm to hot water. Afterwards, dry and then lightly coat them with WD-40 or oil in order to prevent rust. The result will be a metallic luster on the items cleaned, and a light shot of penetrating oil while they are still warm should see all screws removed easily. If any of the screws are still resistant to turning, simply be patient and return the item to the solution.

Machinework

All threads in the engine block should be chamfered so that the uppermost section of the threads will not pull out above their deck surfaces. In each case, the chamfered recess need not be of greater diameter than that of the threads. Be aware that chamfering will most likely deform the threads that are immediately beneath the chamfer, so reshaping them with a rethreading tap is essential. Deformed or dirty threads in the engine block can reduce clamping force on a gasket in the same manner as dirty or damaged threads on the machine bolts do, so they should be chased with a rethreading tap after chamfering. A rethreading tap and die are designed to clean and straighten threads. Using a regular cutting tap or die can cause a weakness in the threads by causing stress fractures at the point of cutting. This cutting action will damage rolled threads and reduce the maximum amount of torque that the machine bolts can sustain. In addition, all of the untapped machined passageways should be reamed smooth to the diameter recommended by the manufacturer of the plugs that are to be installed in them. The ports of the coolant passages should be lightly chamfered in order to prevent them from shearing the cylinder head gasket, thus preventing gasket material from breaking off and making cylinder head gasket removal into a future headache. Finally, clean all of the passageways in order to remove any debris. Note that no plugs of any kind should be installed until after all machinework on the engine block has been performed and the engine block thoroughly cleaned out in order to ensure removal of all grit and metal swarf as these passages and chambers can become a repository of such materials. Insist that new oversize bronze plugs be shrink-fitted and pressed in to a depth that is slightly beneath the surface of the engine block so that they will not interfere with proper gasket sealing of the oil sump and the front and back plates. If you wish, you can do this yourself. In order to shrink-fit them into the engine block, the plugs should be sprayed with WD-40 in order to displace any moisture on them, placed into a well-sealed Ziploc bag in order to prevent ice from forming on them, and with the thermostat on the deep freeze set as low as it will go, they should be left in there to chill overnight. That shrinks them to a smaller diameter. When they are ready to be installed, they should be taken out, then immediately seated into the engine block with a flat-nosed punch. When they warm to room temperature, they will be in there good and tight because they will have expanded in place!

The only way to get them out will be to drill and tap threads into them and use a puller! Bronze, being an alloy of tin and lead, has a higher coefficient of expansion and contraction than iron. It thus expands more than iron when it gets hot, so there is no way that they will ever come out while driving down the road.

Stainless steel Frieze / core plugs should be used for the same reason. Their high chromium content also means lots of expansion when hot, so once they are properly seated, they will not pop out. Make sure that they have a good concentric seating surface by specifying that an end mill bit be used to clean up their seating surfaces in the engine block. It is not the cheap way to do it, but it always works. If this cannot be performed due to insufficient engine block material thickness, the existing seating surfaces should be power cleaned with a rotary wire brush. When you install the Frieze / core plugs, be advised that their surfaces should be dished inward with a plannishing hammer so that they will be deformed properly into the bore and provide an effective seal. Should the dish be too deep, the plug will shrink and it will come out easily. However, even with the most meticulous preparation, Frieze / core plugs have been known to pop out when the engine experiences detonation. In order to prevent this unhappy experience, thin aluminum flat bar strips can be fabricated to hold in place the three Frieze / core plugs on the side of the engine. Simply use a small cap screw on each end of the strap and a combination of washers to take up the gap behind the bar. If anybody notices this modification, point out to them that the MG Factory Racing Team did this on their works MGAs. The Frieze / core plug at the rear of the engine block can also be secured by using one of the thick machine washers that is used to clamp the intake and exhaust manifolds to the cylinder head and tapping in 3/8" UNF threads or its suitable 9mm metric equivalent. You will also need a matching machine bolt with threads all the way to the cylinder head about 1 inch long. Install the washer with its stepped side facing the rear engine plate, sliding it in until it locates over the hole in the engine backplate, and then insert the machine bolt. Tightening the machine bolt will force the washer against the engine backplate and thus secure the core plug in its recess. This simple design can be further improved by turning a flange into the washer so that it will positively locate in the hole in the engine backplate.

While the engine block is at the machine shop, you may wish to consider having it modified in order to allow the installation of an Original Equipment petcock-style drain valve (BMC Part # 3H 576, Moss Motors Part # 470-240). The drain port thread is 1/4-inch BSPP (British Standard Pipe Parallel thread). The standard thread designation is G-1/4. This is a Whitworth form parallel fastening thread, 19 threads per inch, and major diameter is 0.518” (13.157-mm). Removal is most easily accomplished with a 3/8” Whitworth wrench (spanner). This was a common feature of the earlier, smaller displacement B Series engines that was continued into the production of the 18G engines that were destined for use in the MGB. Unfortunately, the coolants of that era did a rather poor job of protecting the coolant passages inside of the cast iron engine block from corrosion. As the engine expanded and contracted during heating and cooling, small particles of rust would flake off from the walls of the coolant passages and chest. The primary reason for the provision of a petcock-style drain valve was that the liquid coolant system was designed in such a configuration that circulating sediment could accumulate in certain ‘dead’ areas of flow. This presented both a potential trouble point, as well as an opportunity to make use of such an area as a collection point so that the sediment so it could be removed. Note that coolant drained from this location will also reveal evidence of an impending disaster, i.e., metal flakes from an eroded impeller of the coolant pump. The sediment tends to settle into the area around the base of the #4 cylinder and into the adjacent drain tap, clogging it if not drained on a regular basis. This drain tap is located at just such a catch-point in a pocket recessed into the floor of the coolant jacket where the sediment can settle. If that recessed pocket is kept clean, the circulating sediment will keep settling there, and can be gotten out by frequently draining off a bit of coolant. Once the recessed pocket fills, sediment will then settle in other low velocity areas, becoming very hard to remove. That is what coolant system flushing chemicals are supposed to do. Sometimes they work, but sometimes they do not. If the system is periodically drained here at the petcock, the sediment will be removed and proper circulation can be maintained. If this task is not attended to, then the sediment can build up until it interferes with coolant circulation, in which case you will begin to get either local and / or general overheating problems. Its secondary purpose was to allow coolant to be drained from the cylinder head and the upper sections of the engine block without going to the trouble of draining the coolant system from beneath the car, thus permitting the easy removal of the cylinder head without having coolant run down the sides of the engine block. This permitted a new cylinder head gasket to be installed alongside the road without losing too much coolant. As such, it is a quaint holdover from a bygone age of motoring, much like the micro-adjuster on the Lucas 24D4 distributor. Ultimately, it continued to be fitted until the introduction of the Mark II models in November of 1967. It was fitted to the 5-main-bearing engines “as required” from that point onwards. Specifically by what criteria the factory determined when it was “required” as yet remains obscure. However, while modern formula coolants have largely reduced this silting problem, the drain tap remains a viable option. Just be sure to get all of the rust out of the inside of the engine and you should be fine from there on.

Be sure that all of the main bearing support surfaces are line-reamed and then line-honed afterwards, plus all of their oiling holes carefully deburred. If possible, it would be wise to have the rocker arms, heads, engine block, crankshaft, and connecting rods either magnafluxed or, better yet, x-rayed in order to be certain that there are no cracks present. If the rocker arms are not to be replaced by new ones, then all of the rocker arm faces should be resurfaced on a contour grinder and rehardened to 54-56 ROC. Because the depth of hardening on the external layer of the original working surfaces of the rocker arms is only a few thousandths of an inch thick, this rehardening after resurfacing is an absolute necessity in order to attain a durable working surface.

Be sure that both the mating surface of the cylinder head and the deck of the engine block have been skimmed flat and that all of the stud mounting holes and coolant passages are chamfered, or at best you will ultimately experience a blown cylinder head gasket or at worst a cracked cylinder head. Flycutting lacks precision and should be used only as a cost-cutting measure for removing metal prior to the final precision cut. The cylinder head gasket may fail due to shearing effects. An end mill produces a superior finish for the cylinder head gaskets of street machines because the grooves left behind by the end mill provides a surface that the cylinder head gaskets can bite into and thus produce a better seal. Note that the deck of the engine block must be parallel to the axis of the crankshaft. After end milling, the ridges at the edges of the grooves that remain from the end milling process on the surface of the deck of the engine block and on the edges of the combustion chamber and the cylinder bores should be removed. The valve seat counterbores must be carefully deburred and smoothed to preclude the possibility of "hot spots" forming and consequently triggering preignition. A surface ground finish is acceptable only for racing engines that use only copper cylinder head gaskets and face frequent disassembly. The Original Equipment specification for the piston crown to deck depth dimension is .040" (1.016mm), which is rather standard for achieving acceptable levels of squish (quench) in a standard engine. When calculating piston crown-to-deck depth dimension, always allow .002" to .003" (.0508mm to .0762mm) for the connecting rod stretch that is produced by inertial forces at high engine speeds. That is, an actual .040" (1.016mm) running clearance would be the result of a measured cold clearance of .042" to .043" (1.0668mm to 1.0922mm) when measuring on the workbench.

Establishing piston crown-to-deck clearance is a relatively straightforward procedure. After oiling the crankshaft main bearings and crankshaft journals in order to protect them from scuffing, install them and torque the crankshaft main bearing caps and both front and rear crankshaft main bearing plates to their Original Equipment recommended torque values of 70 Ft-lbs. Next, after oiling the cylinder bores, install each of the connecting rod and piston assemblies using one old compression ring in order to stabilize and center each of the pistons inside of their cylinder bores. Be sure that they are properly vertically aligned. Using a dial indicator mounted on a magnetic stand affixed to the deck of the engine block, rotate the crankshaft back and forth until the front piston is indicated as being at Top Dead Center. At this point, a depth micrometer or the extending end of a vernier caliper can be used in order to establish the piston crown to deck clearance (i.e., piston height) by measuring directly above the axis of the wrist (gudgeon) pin. This is also the perfect opportunity to check the accuracy of the Top Dead Center timing mark on the harmonic balancer (harmonic damper) pulley wheel. If the mark does not properly align, use an offset key in order to set it to its proper position.

At this point, a word about machining tolerances is in order. When a machinist checks the specifications of a part that he is about to work with, he always takes note of the Machining Tolerances to which it is to be machined. The term “Machining Tolerance” refers to the allowable amount of variation for a specified dimension. For example, the specified bore diameter of the Original Equipment engine is 3.1600” (80.264mm), with a machining tolerance of +.0005” (+.0127mm) oversize or -.0005” (-.0127mm) undersize. This means that a bore diameter that is anywhere between 3.1595” (80.251mm) and 3.1605” (80.277mm) is acceptable. In the case of a five-main bearing engine, the Original Equipment specified clearance gap between the cylinder bore and the piston skirt is .0021” (.05334mm) to .0033” (.08382mm) at the top of the piston and .0006” (.01524mm) to .0012” (.3048mm) at the bottom of the piston. In practice, this implies that a 3.1589” (80.9219mm) skirt diameter piston in a 3.1595” (80.2513mm) diameter cylinder bore or a 3.1599” (80.2615mm) skirt diameter piston in a 3.6005” (91.4527mm) diameter cylinder bore would be technically acceptable. However, either would be far from ideal as in either case friction would be greater than the engineering ideal. In an Original Equipment specification five-main bearing engine the engineering ideal would be to have a skirt clearance of .0027” (.069mm) at the top of the piston and a .0009” (.023mm) skirt clearance at the bottom of the piston, thus minimizing friction at operating temperatures. The average general-purpose engine machine shop has equipment that can hold a machining tolerance of

.001” (.0254mm) and a good engine machine shop can hold a machining tolerance of .0005” (.0127mm), while a really top-notch engine machine shop can hold a machining tolerance of .0001” (.00254mm)”. Obviously, the tighter the machining tolerances to which the engine components can be held to the theoretical engineering ideal, the more powerful and longer-lived the engine can be. However, the services of an engine machine shop that can hold a .0001” (.00254mm) machining tolerance are never cheap to hire because of the higher cost of both its precision equipment and its connected maintenance, not to mention the superior skill level of those entrusted to work with such high-precision equipment. In engineering, as with all other things in life, you only get what you are willing to pay for. In terms of power output, how significant can trying to hold as close to the engineering ideal be? The original prototype three-main-bearing MGB engines were made under toolroom conditions in a prototype shop that was crewed by the best machinists who were using the best equipment that British Motor Corporation had. In finalized form, these precision-made prototype engines produced 98 BHP, and when the original MGB advertisements were released, this is the power output that was claimed. However, in mass production such strict adherence to the engineering ideal was not possible. Due to “tolerance stacking”, some of the mass-produced high compression versions of the engine produced as little as 93 BHP, while the overall average was in the neighborhood of 95 BHP. The advertisements and publicly released performance specification figures were quickly changed accordingly. Considering the 1.8 Liter (110 Cu. In.) displacement of the engine, for their time, they were impressive indeed:

Engine Speed	**Torque (Ft-lbs)**	**BHP**
1,000 RPM	72	13
1,500 RPM	89	26
2,000 RPM	101	38
2,500 RPM	107	51
3,000 RPM	110	63
3,500 RPM	108	73
4,000 RPM	106	81

4,500 RPM	104	88
5,000 RPM	98	94
5,500 RPM	91	95
6,000 RPM	81	92

Since the difference between a B Series engine produced to the engineering ideal and an ordinary, mass-produced one can be as much as 5 BHP, as well as its contribution to an increased lifespan, it is easy to see why the extra investment in the higher cost of a shop that is capable of such precision machinework can thought to be worthwhile.

If you are having trouble finding a decent machine shop, ask the racers at the local MG club. They should be able to point out the right shop for you. If such a local resource is not available, just call around to different machine shops and ask them what kind of tolerances they can work to. If they do not give a direct answer and instead ask what kind of tolerances it is that you need, do not tell them, just repeat the question instead. If they do not state that they can easily work to +/- .0005" (+/- .0127mm), then keep looking. Perhaps one of the best and most revealing tricks is to tell them that the specified clearance gap between the cylinder bore and the piston skirt is .0021" (.05334mm) to .0033" (.0838mm) at the top and .0006" (.1524mm) to .0012" (.3048mm) at the bottom. If then they ask how many degrees of taper is needed, then they do not know how to calculate the trigonometry necessary to calculate the needed angle of taper. That automatically means that you are dealing with mere semi-skilled machine operators, not real Machinists or Tool & Diemakers, since such mathematical training is a standard part of the formal classroom training curriculum of every apprentice machinist. No shop owner in his right mind would allow such inadequately-skilled personnel to operate sensitive fine-tolerance machinery, so they will not have any such machinery in the shop, no matter what claims they make to the contrary. Also, a knowledgeable engine building shop would be aware that clearances are specified at the top and bottom of the piston skirt, which means that the pistons are tapered, not the cylinders. Such a shop is a general-purpose machine shop and obviously not one that specializes in building engines. Walk away and keep looking. Once you have found a likely prospect, ask if you can tour the shop at the end of their work day. The shop should reflect pride of profession. It should be well-lit and clean, including the machinery. That means no large amounts of metal chips scattered on the floor or lying in sodden heaps under the cutting stations. The tools should be clean and in proper storage with nothing, including dirty shop rags, left lying around. Check to see if they have a heat treating furnace. Fine-tolerance machinery has to be maintained in excellent condition in order to consistently produce such fine tolerances, plus fine-tolerance machinery is more expensive. The skill level required to operate such equipment is higher, and thus the labor is also more expensive. In the end, you get what you pay for. If they really can do fine-tolerance machining, then the grinding of crankshaft journals and the installation of valve seat inserts should be mere child's play to them.

As a former Machinist and later a Tool & Diemaker, I well understand the significance of the level of organization and cleanliness in a shop. Here in the USA we have we have a social stereotype of a shop that's run by a guy who doesn't care about quality in anything. The owner is referred to as "Greasy Joe." His shop is dark and filthy. His equipment is ancient, won't hold tolerance, or repeat size. Stuff is lying everywhere, and much of it is covered with dust and old oily rags. Only Greasy Joe can find anything in this pigpen, and it often takes a while for even him to locate anything. Greasy Joe walks around chomping on an unlit cigar butt and looks as though he hasn't bathed in about two weeks. His clothes are shabby and his attitude is grouchy. Greasy Joe reuses old, metal-fatigued parts like burned valves and valve springs with 100,000 miles on them. He gives a thirty-day labor-only warranty on every engine that he rebuilds, knowing that the engine will sit on a garage floor waiting to be installed for at least a part of that thirty day period. Only the ignorant, the fools, and the suckers ever go to him for a rebuild, and that's how he secretly sees his customers. If the engine fails, he'll say that you abused it and so the warranty is as void as his workmanship. I won't do business with Greasy Joe's Shop for any purpose under any circumstances.

During the course of an engine rebuild it is common to find that the engine block is warped along its longitudinal axis, so engine builders are always prepared to line-bore the main bearing and camshaft journal mounts. Warped mating surfaces are the major contributing factor in leakage and in the development of cracks in the cylinder head casting. This warpage is normally the result of the repeated expansion and contraction of the engine block (i.e., thermal cycling) having gradually relieved the stresses remaining from the original process of casting. However, we rarely stop to consider that this warpage should also extend to the mating surfaces elsewhere on the engine. The necessity of skimming them flat just as one would the deck of the engine block and the mating surface of the cylinder head should always be explored. To check for warpage in your garage, simply clean the mating surfaces and smear a very, very thin, even stain of machinist’s bluing or petroleum jelly on them. In a smooth, perpendicular motion, place a clean plate glass or a mirror on the surface and then gently pull it away. Hold it up to a light and look for any gaps in the bluing / petroleum jelly outline. If you find any, you have warpage. This technique will work with any mating surface. Get the mating surfaces flat and you will have gone a long way towards having an oil-tight engine.

Remember that machine resurfacing of a gasket area does not necessarily guarantee either flatness or the proper surface finish. That is why the flatness and surface finish should always be checked before installing a new gasket. As a general rule, the smoother the surface finish is, the better it is. When the surface finish is rougher than about 100 RA microinches, there are too many peaks and valleys on the surface of the metal to permit a proper seal to be achieved. The cylinder head gasket may not cold seal and could leak coolant, oil, and / or combustion gases. Using a thicker cylinder head gasket that has increased conformability and / or a thicker soft facing can compensate somewhat for a rougher surface, but such cylinder head gaskets do not retain torque well and are less durable. Too rough a surface finish has more "bite", digging into the cylinder head gasket more aggressively, increasing the scuffing and shearing that the cylinder head gasket undergoes as the engine expands and contracts. In bimetal engines that pair cast iron engine blocks and aluminum alloy cylinder heads, this can be especially hard on the cylinder head gasket because of the difference between aluminum alloy and iron in their coefficients of expansion and contraction. Too smooth a finish may not provide enough bite to seal the cylinder head gasket securely. There can also be movement between the gasket and metal, causing the cylinder head gasket to abrade and leak.

When using a resin-impregnated cylinder head gasket, the surface finish for both the mating surface of the cast iron cylinder head and that of the deck of the cast iron engine block should be 80 to 100 RA microinches. When using a steel-reinforced cylinder head gasket that combines either a fiber composite or expanded graphite layers, the surface finish should be 60 to 100 RA microinches. If a rubber-faced multilayered steel cylinder head gasket is used, then the surface finish should be 30 RA microinches maximum, but there is no minimum. The smoother it is, the better its seal will be. When the mating surfaces of an aluminum alloy cylinder head, intake manifold, or exhaust manifold are resurfaced for use with a resin gasket, the surface finish should be 50 to 60 RA microinches. When using an annealed copper cylinder head gasket the surface finish should be 60 RA microinches for a cast iron cylinder head and 40-50 RA microinches for an aluminum alloy cylinder head. In all cases, the surface finish should be fairly uniform across the entire face of the cylinder head and deck of the engine block, not varying by more than 20% from one area to another. In addition, there should be no more than +/- .001" (.0254mm) of out-of-flat across for 3" (76.2mm) in any direction. Pay particular attention to the areas between the cylinders on the engine block, between the combustion chambers on the cylinder head, and where the cylinder head gasket seats around the cylinders on both of these surfaces, as these are the most highly stressed sealing areas. Any surface flaws that are found should be eliminated by resurfacing.

Welding

Do not be tempted into trying to repair a cracked cylinder head by taking it to a welder. Welding cast iron is a very tricky thing, requiring the right equipment. Contrary to what some welders might tell you, as a former Tool & Diemaker I can explain why it cannot be done on a bench in the garage. The problem lies in the fact that a casting is essentially just a bunch of bubbles held together by metal. There is always the risk, even though the alloy of the engine block and the alloy of the welding rod may be the same, that the density of the weld will be different from that of the density of the casting. This results in different rates of expansion and contraction when the casting heats and cools. If the density of the weld is not the same as that of the cylinder head, the casting will crack where it adjoins the weld and you will find yourself right back where you started.

However, because creating a weld is nothing more than a matter of heating the metal alloy of the rod to the point that it flows into and heats the metal of the casting to the point that it liquefies and blends with the molten alloy of the welding rod, it is possible to achieve the same density if certain conditions are met: First, the temperature of the molten metal of the welding rod should be no higher than that necessary for attaining a molten state. Second, the casting should be heated in a heat-treating furnace until it almost melts (about 1,600°). The white-hot iron casting then is removed and the weld applied with an iron welding rod only, and then the casting is quickly placed back in the furnace and very slowly brought down to room temperature in controlled stages. Although this controlled cooling process will help to allow stresses to even themselves out, the casting may become warped and require machining.

Why is it so necessary to heat the casting to a predetermined temperature in a furnace instead of just heating it with a torch on a welding bench? Simply so that the temperature of the weld will be as close as possible as that of the casting. Why is that so important? First, because of the density issue already described above. That requires a degree of precision control that a welder cannot attain with a blowtorch on a workbench, even though he may sincerely believe that he can. Face it, the man is a welder, not a trained Metallurgist or a trained Tool & Diemaker. He simply does not know any better. Secondly, due to the fact that the heat differences are not as localized, the localized thermal stresses created by the extreme heat of welding will be minimized and not be isolated to the area immediately around the weld. Cast iron conducts heat very slowly, so the closer the temperature of the iron of the casting to that of the weld when the welding process begins, the less thermal stress will be generated in the areas adjacent to the weld. This elaborate procedure is necessary in order to eliminate the possibility of cracking due to induced thermal stress, which is a separate issue from that of weld density. The whole idea behind the process is often called "stress relieving", a process that I am sure that you have heard of. Now you understand just what it is.

Needless to say, this process is expensive, but justified when working with a rare and irreplaceable cylinder head, such as would be the case with one from a 1910 Rolls-Royce. If the problem is with a crack in a cylinder head from an MGB, I would just scrap it. There are many used heads available in good condition for far less money than what the above-described process costs. You would have to pay for the additional machining costs on the cylinder head either way that you choose to go, so why bother with the expense and risks of doing a proper welding job when there are good used heads available?

The better shops will do most or all of the aforementioned machining and engineering procedures as a matter of course. If the shop you are considering cannot provide these services, they are merely tradesmen rather than professionals: go elsewhere.

Painting the Engine

After the machinework on the engine block has been completed and the interior and the oil passages have been cleaned out thoroughly, be sure to paint the inside of the crankcase area of the engine block with Glyptal, a coating that is highly resistant to oil. This is recommended to seal the pores of the iron, thus preventing any deposits or metallic machining dust remaining inside of the pores of the iron after cleaning from leaching into the oil. This is what the factory did. It will also promote drainage of the oil down the inside of the engine block and out of the oil sump as a deterrent to the buildup of carbon and sludge. Likewise, the interior of the rocker arm cover and the covers of the tappet chest, as well as the interior of the oil sump should also be coated with Glyptal for the same reason. However, their flanges should be left unpainted so that they will form a more effective seal with their gaskets.

In order to etch the surface for superior paint adhesion, always use a good quality metal conditioner. I use POR-15 Metal Ready, as it will remove light surface rust, and will etch the surface of the iron in order to create an anchor pattern so that the engine paint can adhere to it, as well as leaving a zinc phosphate coating on the iron metal surface that will inhibit the formation of rust. Being rinsed off with water or wiped with a wet rag, it leaves no residue that needs to be removed prior to paint application, and can be used on aluminum as well. Best of all, being non-toxic, non-corrosive, non-caustic, non-flammable, and bio-degradable, it is very safe to use, even indoors.

Prior to painting the exterior of the engine components, be sure to mask off the outer walls of the cylinders, all bearing mounting surfaces, the crankshaft main bearing cap seating surfaces, and the gasket areas, and then apply a coat of thermoconductive enamel engine paint onto the exterior surfaces of the engine block before it has a chance to rust again. There is something that you can do in advance that will turn your engine rebuild into an easier experience that will also produce a more satisfying finished product. Purchase all of your seals and gaskets well in advance before you start the reassembly rather than waiting until just before you start (as most people do). Using the gaskets and seals as templates, draw around them on wax paper in order to make cheap cut-out silhouette copies of all of them. Why go to the effort of making cheap silhouette copies? Because the copies can be used to precisely mask off all of the sealing surfaces prior to painting the engine block. Tape or glue stiff tabs onto the edges of each of them so that they can be easily peeled off of the surface immediately after the painting of the engine block. While it is true that many people simply spray paint the entire surface of the engine block, including the gasket sealing areas, in reality this is a bad practice. Such engines tend to ooze oil around their gaskets because the gaskets cannot achieve an effective seal against a smooth, glossy surface. Gaskets do a much better job of sealing when applied to bare metal surfaces that are machine-finished to the proper surface texture that is achieved during manufacture at the factory. Instead, all gasket areas of the engine should be masked off prior to painting so that the gaskets will have a metallic surface to seal upon. Failure to take this extra step will likely result in oil oozing out from underneath the gaskets. Simply smear a coat of petroleum jelly to one side of the copy and place it over its sealing area on the engine block, and then apply the paint. After applying the paint, remove the gasket and allow the paint to dry thoroughly, and then remove the remaining film of petroleum jelly with alcohol. Getting this detail of the rebuild prepared in advance will make for a more oil-tight engine. Unfortunately, few commercial shops will take the time to make this extra effort, despite the fact that they know full well that factories do not paint an engine until after it is assembled. Most owners who are faced with doing the reassembly themselves usually omit this step of advance preparation in ignorance of proper procedure, and simply paint the entire engine block and cylinder head prior to any assembly work being done, only to end up with the disappointment of living with an engine that oozes and leaks. Be sure to put rubber stoppers (available at good hardware stores) into all of the holes in order to prevent paint from getting in.

The outer walls of the cylinders should then be prevented from rusting by smearing their surfaces with WD40, and then covering them with WD40-soaked paper until the time arrives for final reassembly of the engine. Because the electric starter motor needs a solid electrical ground (earth) in order to work properly, do not paint either the gasket area of the rear of the engine backplate where the electric starter motor mounts or to the area of the front face of the engine backplate where it mates up with the gasket on the backside of the engine block.

Hirsch has an excellent thermoconductive engine enamel that, being unique in that it was originally formulated for use on jet engines, will withstand temperatures up to 600° Fahrenheit (315.5° Celsius) and is an exact duplicate of the shade of red ("MG Maroon") used on the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18 engines. It remains glossy almost indefinitely and can be applied directly to cast iron without primer. It is famous for being used on about 90% of all of the winners of the Pebbles Beach Concours, so you can rest assured that its appearance will be first-rate. I will not paint an engine with anything else. Hirsch has a website that can be found at http://www.hirschauto.com/ .

The Engine Block

The most desirable engine blocks for a high output engine are the early 18V engine blocks of the 1972 through 1974 Chrome Bumper models. These later engine blocks have thicker, stronger crankshaft main bearing caps, most notably the rear crankshaft main bearing cap (MG Part # 12H 1951) in which the oil drainage slot was eliminated, and Grade 8

½”-20 UNF machine bolts instead of studs for securing their crankshaft main bearing caps. These can be readily identified by their engine numbers: 18V-581-F-H, 18V-581-Y-H/L, 18V-582-F-H, 18V-582-Y-H/L, 18V-583-F-H, 18V-583-Y-H, 18V-584-Z-L and 18V-585-Z-L from the 1972 model year, and 18V672-Z-L and 18V-673-Z-L from the 1973 through 1974 model years.

It should be noted that while the torque reading on a machine bolt partly reflects the twist of its shank, the torque method is sometimes inaccurate because of the uncertainty in the coefficient of friction at the interface between the bolt and the rod. This inaccuracy can be minimized by using the lubricant supplied by ARP. On the other hand, the torque reading on a stud reflects only the stretching force exerted on the stud. Because a bolt stretches in proportion to the tension in it, it is this “stretch” that clamps things, so the more accurate reading obtained with a stud is the better way to go. British Leyland switched from the original stud design (BMC Part # 51K 1482, Moss Motors Part # 328-195) to machine bolts (BMC Part # 12B 2356, Moss Motors Part # 322-145) for securing their crankshaft main bearing caps during a time of cost cutting. These later crankshaft main bearing caps have shallow recesses in their mounting bosses in order to accommodate the heads of their mounting bolts, while the earlier crankshaft main bearing caps have deeper recesses their mounting bosses in order to accommodate the machine washers and nuts of the mounting studs. These later crankshaft main bearing caps can be used in the earlier engines if their appropriate Original Equipment mounting bolts are also used, and only after they have been line-bored in place. However, if these mounting bolts are unavailable, ARP makes a set of high strength studs that may be used as an upgrade (Moss Motors Part # 322-878, APT Part # MS5B54). The Original Equipment mounting bolts cannot be used to mount the earlier crankshaft main bearing caps as the threaded portion of the engine block is too short to allow the head of the machine bolt to properly bear against the mounting boss of the main bearing cap.

The crankshaft main bearing caps should always be carefully inspected beforehand for any signs of cracking and their edges smoothed in order to preclude cracks from forming under the stress of prolonged high engine speeds. Advise your machinist that both the crankshaft main bearing caps, as well as the connecting rods and their end caps, are individually matched paired sets and hence are not interchangeable. The crankshaft main bearing mounts are bored with the bearing caps in assembly. Most of the time the alignment of the split line will not be exactly on the center of the bore, so crankshaft main bearing caps always have to be mated according to the original pairing. If the crankshaft main bearing caps are switched to a different location, or from a different engine block, the bore size will almost always be changed. If the crankshaft main bearing caps were lost, and replacement caps were procured, then they will have to be line bored in order to restore their concentricity of their bores. In order to determine how much it must be reduced, you should first assemble the crankshaft main bearing caps and measure their bores in order to disclose the size of the largest bore. The typical procedure is to shave 0.010" off of all of the crankshaft main bearing caps in order to reduce the size of the bore. When line boring the block, you need to pay particular attention to the bore size and clearance for the mechanical rear seal. Next, reassemble the crankshaft main bearing caps to the block and then rebore the crankshaft main bearing mounts in line 0.005" higher than original to restore original bore size. When line boring both the crankshaft main bearing mounts and the bores for the camshaft bearings, tolerances should be held to +/-.0005" (+/- .0127mm). However, it is well proven that precision machining is beneficial to lengthening the lifespan of a high performance engine, so after line-boring, these bores should then be line-honed to a machining tolerance of +.0001" / -.0000" (+.00254mm / -0mm). This will enable the fitting of the main bearings with just the right amount of "crush" when being torqued to their specified 70 Ft-lbs and minimize distortion in order to ensure concentric running. It should be noted that both front and rear crankshaft main bearing caps must always be installed flush against the engine block. The bores of the mounts for the camshaft bearings should also be carefully measured in order to ensure press-fitting of the camshaft bearings without the need for any subsequent modification to their Internal Diameters. If they are found to be out of tolerance, then sleeves may have to be custom fabricated and press-fitted, although it should be noted that this is a very exceptional occurrence.

In order to ensure correct connecting rod alignment, the axis of the bore of each of the cylinders should be located perpendicular to the plane of the rotational axis of the crankshaft as well as directly over the middle of the axis of the crankpin journal of the crankshaft when the crankpin journal is at Top Dead Center. This axis can be established with the necessary precision only after the thrust washers have been installed and the endplay (endfloat) of the crankshaft has been eliminated. Crankshaft endplay (endfloat) also can effect oil control. If the endplay (endfloat) of the crankshaft is excessive, then the resulting back and forth movement will misalign the connecting rods. This effects their ability to assist the pistons and rings in controlling both combustion gases and oil, as well as causing excessive side wear of the pistons and rings. When a connecting rod is misaligned, two things happen that will effect oil economy. First, the piston rings will not be perpendicular to the axis of the cylinder bore and will consequently both rotate and wear rapidly. Second, the connecting rod big end bearings will be tilted on their crankshaft journals with the result that excessive oil will be slung onto the cylinder wall. In addition, consequent abnormal bearing and journal wear will also occur.

Some engine blocks had their cylinders improperly located at the factory, while others will suffer from this condition as a result of warpage of the engine block. The geometric relationship between the axis of the crankshaft and those of the cylinders is critical to both performance and the engine's life expectancy. After line-boring of the main bearing saddles, the engine block is set upside down on the machine bed and shimmed so that the heights of the main bearing saddles are equal. Next, a skim cut is taken off of the bottom surface of the engine block in order to make it parallel with the centerline of the crankshaft. This having been accomplished, the engine block is then inverted with a centerless-ground bar of the same diameter as the main bearing saddles installed in them in order to act as a reference point for establishing the center point of the cylinders. This critical center point having been achieved, the cylinders can then be bored to size. By using this method, the cylinder bores will be at the required right angle to the crankshaft.

Next, a skim cut is taken off of the deck of the engine block in order to make it parallel with the axis of the crankshaft. All machine work on the crankshaft must be completed and the width of its throws recorded. At this point the crankshaft, connecting rods, and pistons with a worn top compression ring on each are trial built using selective assembly in order to attain equal piston heights. The oiled assembly is then installed to tolerance with its appropriate thrust bearings. A degree wheel and a dial gauge can be used in order to aid in establishing Top Dead Center for each throw. Crane Cams makes an excellent degree wheel expressly for this purpose (APT Part # 99162-1) that you can see at http://aptfast.com/Images_Parts/Cams_Valve_Train/A_Parts/99162-1.jpg An edge finder is then used in order to determine the axis of the crankpin as well as both of the edges of each of the throws. Their centers are then computed in order to establish the correct axis of the cylinder bores above the crankshaft. After boring and honing, a final skim of the deck of the engine block is performed in order to attain the desired piston to the deck clearance.

Because of the close proximity of the cylinder head studs to the cylinder bores, the bores of the cylinders tend to distort slightly when the cylinder head is torqued down. Some machinists will try to compensate for this by sizing the bores of the cylinders to their maximum factory-specified clearance, but this approach will result in a shorter piston and bore life by decreasing their load bearing surface area. The proper approach to boring under such circumstances is to mount a torque plate to the deck of the engine block along with a new gasket, and torque it to the same specifications as would be used when mounting the cylinder head in order to simulate the distortional stress imposed by a torqued cylinder head, and only then proceed to bore the cylinders. Prior to boring the cylinders, the crankshaft main bearing caps should also be installed and torqued down to specification. In addition, each piston should be measured with a micrometer in order to establish its own individual optimum bore size. Bore finish roughness prior to honing should be approximately 15 AA. The diameter of the bore should be left with approximately .002" to .003" less than that of the finished diameter of the bore in order to leave enough material to be removed during the honing process.

Following this, the crankpin journals of the crankshaft should all be carefully indexed and the lengths of its throws matched. This latter operation is essential not only in order to equalize the swept volume of each cylinder, but also to permit the dynamic balancing to match the opposing dynamic effects from one cylinder to another, thus making for a smoother engine. Indexing of the throws of the crankshaft will also assure that the valve timing will be precisely in proper phase with the movements of the pistons, thus making for optimum breathing.

When cleaning out the passages inside of the crankshaft, be sure to remove all of the oil flow restrictors that have been installed inside of the journals of the crankshaft throws. These were installed into the crankshaft in order to prevent centrifugal force from causing excessive oil flow outward from the crankshaft at high engine speeds, combining with capillary action to result in too much oil being drawn away from the bearings on the main journals of the crankshaft. This danger is greatest in the case of the journals of the crankshaft throws for the connecting rods of the #2 and the #3 cylinders as they both must share the oil flow fed from a single 5/16" (7.9375mm) oil passage from the high pressure gallery via the center main bearing of the crankshaft, while each of the main journals of the crankshaft throws for #1 and #4 cylinders are each supplied by their own 5/16" (7.9375mm) individual oil passage from the high pressure gallery via the main bearings. This being the case, these oil flow restrictors were occasionally omitted from the journals of the crankshaft throws for #1 and #4 cylinders, and may not be found at all inside of the journals of the crankshaft throws of some later crankshafts as their installation was discontinued by the factory in order to reduce costs. However, if the engine is fitted with a camshaft that produces its peak power output at higher engine speeds, then their installation would be an obviously prudent choice in order to prevent too much oil from being drawn away from the low pressure gallery, even if the single oil passage from the high pressure gallery to the center crankshaft main bearing is enlarged from 5/16” (7.9375 mm) to 7/16” (11.1125 mm) in order to ensure an adequate oil supply to both it and the bearings of the crankshaft throws for the big ends of the connecting rods for the #2 and the #3 cylinders. I suspect that the factory used the same size oil passage drills just to simplify production and minimize tooling costs. Two different diameter drill bits would require two different feed and speed settings, which cannot be done on a normal multi-spindle drill machine.

The Oil Pump

Just as gasoline is the food of an engine and its cylinders are its lungs, so oil is the lifeblood of an engine and the oil pump is its heart. I cannot overemphasize the importance of this fact. If your engine is to live a healthy life, its oil pump must be immaculately rebuilt. Unless you are building a highly stressed high output engine with an extensively modified oiling system, your Original Equipment-specification oil pump will be adequate to the task. This is due to the fact that its design is of the Holbourne-Eaton positive displacement eccentric rotor type, the rate of flow of which increases in direct proportion to the engine speed. Any increase in pressure beyond that of the oil pressure relief valve spring rating results in the opening of the oil pressure-regulating valve and the excess oil discharging into the oil sump. Properly rebuilt, it should deliver 60 to 70 PSI at idle speed when the oil temperature is 200° Fahrenheit (93.3° Celsius).

Depending upon the viscosity of the oil being used, an unmodified oil pump will cavitate at an engine speed of about 6,500 to 7,000 RPM. The dangers inherent in this cannot be understated. It means that the oil pump will draw air bubbles inside of it, and the oil itself will suddenly stop flowing through the rotors of the pump. No oil output implies no oil pressure and thus no oil flow, the bearings and journals of both the crankshaft and the connecting rods suffering sudden catastrophic failure within a matter of seconds. However, this danger can be avoided. Remove any and all burrs that you can find in the pump body, and then make sure that the passageways in the body and the delivery arm have no sudden steps or angles to inhibit oil flow. These can often be removed with a Dremel tool and a polishing bit. Doing so should eliminate the risk of a loss of oil pressure resulting from cavitation at operating speeds up to 6,800 to 7,200 RPM.

Understand that a large increase in oil pressure will not, in and of itself, result in a large increase of oil flow through the system. This is due to the fact that oil pressure increases only exponentially with oil flow, thus the increase in oil flow will actually be very small. A high-volume / high-pressure oil pump will require more power in order to function and increase stress and consequent wear on both the gear teeth of its spindle gear and the teeth of its drive gear on the camshaft, as well as increasing torsional stress upon the drive shaft of the oil pump. Because such an oil pump is of additional benefit only at low engine speeds when employed in the system of an Original Equipment-specification engine block, it will do little for any engine other than one whose oiling system has been comprehensively modified to suit an ultra-high performance specification. Racing engines have a greater amount of clearance between their crankshaft main bearings and the journals of their crankshafts in order to reduce the possibility of overheating their lubricating oil, and that application is what a high flow volume / high pressure oil pump is intended for. However, there is a downside to this approach: because of the greater loss of oil volume and oil pressure that results from the greater clearances, the bushings of the rocker arms often suffer from decreased lubrication, especially at elevated engine speeds where the crankshaft can centrifuge oil outward into the crankcase. This forces racers into a conversion to special rocker arms that are supported by needle roller bearings. Now you know why they use them.

Should you decide that you have sufficient reason to justify the pursuit of this objective, it would be prudent to install a bronze spindle gear in order to prevent the rapid wear of the gear teeth of the camshaft that so often attends such applications, as well as to preclude their breakage at high engine speeds that results from the greater loadings induced by hairy camshaft lobe profiles. This bronze spindle gear can be obtained from Cambridge Motorsports. It should be noted that while a bronze spindle gear absorbs the increased gear tooth strain caused by the increased loading that is an unwecome by-product of a high-volume / high-pressure oil pump or a hairy camshaft lobe profile, as with any bronze gear, it must be checked regularly for wear. The faces of new gearsets tend to "mate" as they wear in, so be warned that replacing one gear in a mated set with a new one can be asking for serious trouble that attends accelerated wear of these components. Go ahead and replace the spindle gear with one of the alloy of your choice if you are installing a new camshaft. If anything is worth doing, it is worth doing right.

Amongst other modifications, the Special Tuning Manual mentions machining an extra feed port into the bottom end cover of the oil pump in order to improve oil flow. Today's replacement pumps already incorporate some of these modifications that the factory chose to incorporate into the later versions of the oil pump, but do not include the extra feed port. Some specialist suppliers offer pumps fully modified with this extra feed port according to the Special Tuning Manual specifications for use in engines that attain very high engine speeds. The disadvantage of this modification is that when the engine is shut off the extra feed port then also becomes a drainage passage. Oil that is inside of the body of the pump flows back into the oil sump . At each cold startup, it will require an extra second or two for oil pressure to build up. In addition, after every oil change it will take longer to build up oil pressure (about 20-30 seconds or more) because draining the oil sump exposes the oil pickup, and this helps drain the oil out of the pump through the extra port. While this is not a problem on a racing engine that will be disassembled and inspected several times during a season, on a street driven car it can contribute to severely shortening the life of the engine bearings as well as that of the journals of the crankshaft. Unlike racers at a track, few owners of street-driven cars will be willing to go through the procedure of repriming the oil pump in order to protect their bearings every time that they want to start their engines after an oil change.

Be aware that circular scoring of the face surface of the lower body of the pump is common, being caused by hard particles becoming wedged between the rotors and the face surface of the lower body of the pump, and then being dragged around by the rotor. Fortunately, the face surface of the lower body of the pump can often be reconditioned by lapping it on a sheet of 400 grade or finer wet and dry sandpaper that is secured to a flat, solid surface. Lubricate the sandpaper with kerosene and, while applying even downward pressure, use a mixture of circular and straight movements. After a few minutes, inspect the surface and then wash the sandpaper out with kerosene in order to remove any loose materials. Repeat this process until the face surface of the lower body of the pump is free of all scoring, then clean it carefully and thoroughly. Having accomplished this task, insert the rotors into the lower body of the pump. Use a feeler gauge to check the diametrical clearance of the outer rotor in its recess in the body of the pump. This clearance figure must not exceed .010" (.254mm). Check to be sure that the clearances between the projections of the inner rotor and the lobes of the outer rotor in all positions do not exceed .006" (.152mm). Lay a straightedge across the face of the body of the pump and check to be sure that the depth clearance of the rotor does not exceed .005". If the clearance depth exceeds .005" (.127mm), then the clearance depth can be reduced by removing its locating dowels and lapping it in the same manner as used to remove scoring on the face surface of the lower body of the pump.

If the pump passes these inspections, coat the exterior of the outer rotor with an assembly lubricant, then insert it into the body of the pump with its chamfered end at the driving end of its recess in the body of the pump, and then coat the inner rotor with Vaseline and place it into the outer rotor. This will aid in priming the pump on the first start up of the engine. Smear any excess into the gaps - no need to pack it solid. Should new rotors be found to be needed, new ones can be obtained from Octarine Services (Octarine Part # 51K881KIT). Octarine Services has a website that can be found at http://www.octarine services.fsnet.co.uk/octarine.htm . Bolt the two halves of the body together, and then insert the drive shaft (BMC Part # 11G 12) into the top of the pump with a little oil in order to assist in holding it in place, ready for fitting. Be sure to rotate the drive shaft in order to ensure that its dog is engaged in its mating slot of the rotor shaft and that the pump turns freely.

Be aware that the design of the oil pump was modified during the course of its production life, resulting in two slightly but significantly different sealing gaskets included with new oil pumps or with rebuild gasket sets. This is due to the fact that the oil passages are in slightly different locations on these two engine designs. The early version of this pump (BMC Part # 88G 296) used on the three-main-bearing versions of the engine (18G and 18GA) had a problem of its pressure falling off above 5,500 RPM, an issue that was addressed on the oil pump of the five-main-bearing engines (BMC Part # 12H 1429) by machining a recess into its cover. The gasket for the oil pump of the five-main-bearing engine (BMC Part # 12H 1018) has a large semi-rectangular or D-shaped cutout while the smaller gasket for the oil pump of the three-main-bearing engine (BMC Part # 88G 420) does not. The gasket for the early version of the pump will block the intake passage of the later version of the pump, depriving the engine of oil and causing the oil pressure gauge to show no pressure reading, nor will there be any oil flow at the line going from the rear of the engine block to the oil filter / oil cooler. This is the reason why you must match the gasket to the oil pump, not to the three studs in the engine block. Do not use any sealant on an oil pump gasket as it is both unnecessary and also presents a possible hazard to the precision parts inside of the engine should any of the sealant break loose. The 5/16”-18 UNF machine bolts that secure the oil pump to the crankcase should be torqued to 14 Ft-lbs.

Unless there is something drastically wrong with the oil pump or its fitment, such as a cracked pump body, non-flat mounting surface, mounting bolts too long, split gasket, etc., it will not drain down simply because the intake and outlet ports are above the centerline, near the top of the pump as it is mounted to the block. A delay in the rise of oil pressure during start up is usually caused by an air leak in the oil pick-up system from the end of the oil delivery arm to the intake side of the pump. This could be the consequence of a badly sealing gasket, a heavily silted up oil pick up screen, or a pin hole in the oil delivery arm.

Be aware that two different diameter oil pick up screens (105mm and 135mm) were used to protect the oil pump of the B Series engine, the smaller of the two being of a side-mount type that used only on the three-main-bearing 18G and 18GA engines. The larger of the two (BMC Part # 12H 1644) was introduced on the five-main-bearing 18GB engine, being the more desirable due to its larger strainer area. The oil pump screen is the only part of the engine that the oil pump relies upon in order to function properly. The oil pump is also the only component of the engine that makes contact with the oil before it reaches the oil filter. Any solid particles that fall into the oil sump go directly into the oil pump pick up screen. Normally these particles are captured by a layer of varnish on the screen where they will not cause a problem until someone tries to clean the pickup screen with solvent. When the varnish is softened or partially dissolved, it frees the foreign material. Often in the process of cleaning, the wire mesh is distorted which in turn causes the ferrule to become unseated from the cover plate. The majority of wear in the oil pump is caused by a solid particle jammed between the rotor and pump body or between the rotor and the cover plate, pushing the rotors into the cover plate and destroying the .002" to .004" (.0508mm to .116mm) normal clearance between the cover and the rotors. In a few rare cases, this can even cause a rotor to jam momentarily, resulting in the breakage of the shaft and an instant loss of oil pressure! Because the mounting plate of the pickup screen has a tendency to fracture with old age, plus the fact that proper disassembly and reassembly of the pickup screen for cleaning is difficult and always presents the hazard of damage to the pickup screen, never reuse one. Instead, replace it with a new pick up screen (Moss Motors Part # 460-760).

The mounting plate of the oil strainer should be carefully inspected for cracking. When mounting the oil strainer to the oil pickup extension of the oil pump, take care to ensure that its top surface is flat against its gasket and is well sealed so that no air leakage can incur. Under normal operating conditions, this area is below the level of the oil, but it can become exposed to the air under hard cornering, resulting in air bubbles being pumped into the bearings with the consequence of hammering of the bearing surfaces. If you have a tendency to push the car hard through curves and turns, have a baffle plate welded into the oil sump pan in order to prevent oil surge and thus ensure a ready supply of oil for the pump. With vehicle motion, oil is constantly moving in the oil pan. Even with vertical baffles in the oil pan, oil waves can be splashed onto the crankshaft to be "wind whipped" by it and, as a consequence, to be fed into the fuel induction system. It is then burned in the combustion chambers, resulting in a buildup of carbon inside of the combustion chambers, which in turn leads to preignition and even to that great destroyer of engines, detonation. A baffle plate installed into the oil sump, between the oil in the pan and the crankshaft, is generally effective in preventing the above conditions. A blueprint for an oil sump baffle plate can be found on page 457 of the Bentley manual. If you do not have access to the means to create your own baffle plate, one may be purchased from Cambridge Motorsports. They have two versions available, one compatible with the 105mm oil strainer of the 18G and 18GA engines and the other with the 135mm oil strainer of the 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, 18GK, and all 18V engines.

Make sure that the oil pressure relief valve operates freely. Its oil pressure relief valve plunger (BMC Part # 12H 865) should have a satin finish chrome plating on it in order to prevent galling. Take care to lap it into its seat in order to ensure idle pressure and to flush out the lapping compound completely prior to reassembly. The oil pressure relief valve spring (BMC Part # 1H 756) should have a free length (uncompressed length) of 3.0" (76.2mm) and a resistance of 15.5 to 16.5 Ft-lbs (7.0 to 7.4 kg) when compressed to a fitted length of 2 5/32" (54. 8mm). Be aware that the interior of the plunger terminates in a truncated cone. A packing piece (BMC Part # AEH 798) fits inside of the plunger and seats inside of this truncated cone, giving the spring a solid seat upon which to exert its thrust. If the packing piece is not sandwiched between it and the spring, then the spring will not engage the plunger squarely, making the spring bow and consequently causing the plunger to tilt inside of the bore, thus causing it to bind.

Bearing down against the resistance of the oil pressure relief valve spring while trying to get the ½" BSP threads of the domed spring cap engaged is well known to be one of the most frustrating tasks on the B Series engine. However, there is a way to make this somewhat onerous task easier. Simply refit the spring cap without the spring, then slowly unscrew it and mark both the spring cap and the engine block at the point where the threads disengage. When reinstalling the assembly into the engine block, align the marks, compress the spring, and then turn the spring cap in order to engage the threads. Easy is good, hard is bad! Once you have the domed spring cap tightened down, torque it to 43 Ft-lbs.

Many racers choose to install a separate switch for the fuel pump in order to allow the engine to crank and thus build up oil pressure without drawing fuel into the engine in order to consequently protect the bearings prior to cold starting the engine. When shutting down the engine, the fuel pump is switched off prior to switching off the ignition system, and the vacuum created by the fuel induction system is permitted to draw the fuel out from the carburetor float bowls until the engine stalls. This permits the engine to be cranked in order to build oil pressure without flooding the engine. While this makes for a less convenient starting procedure and is not necessarily required for most high performance street engines, it can be a worthwhile modification in the cases of very high power output engines that place high loadings on their bearings.

Bearing clearances determine how much oil leaks out to be spun off by the rotating crankshaft, and some of that oil is consequently deposited onto the walls of the cylinders in order to lubricate the pistons and their rings. If the oil film thickness becomes too thick, then the piston rings will hydroplane on the oil film. This oil will then migrate into the combustion chamber and be burned, resulting in carbon buildup on the piston crowns and top compression ring lands, as well as the combustion chambers, increasing the risk of preignition and / or detonation, and even sticking and breakage of the top compression rings.

If you wish to slow the rate of oil temperature rise under high stress operating conditions, it is possible to install the larger capacity oil sump (BMC Part # 12H 3541) of the five-main-bearing 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines onto a 18V engine. This will give you the advantage of its 50% larger oil capacity (9 pints, 9.6 pints with oil cooler system vs. 6 pints, 6.6 pints with oil cooler system). Although the earlier oil sump has a bulge at its rear in order to allow for drainage from a slot in the earlier rear crankshaft main bearing cap, its sealing lip will still match the flange of the later engine. Both oil sumps have the same bolt hole pattern and use the same gasket (BMC Part # GEG 504), as well as the same 18 bolts and washers. However, the later 18V oil sump is not usable with the earlier 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines due to its lack of the drainage bulge at the rear. In all cases, the oil sump machine bolts should be torqued to 6 Ft-lbs. Be aware that the oil sump of the 18G and the 18GA three-main-bearing engines (BMC Part # 12H 395) and its gasket (BMC Part # GEG 503) will not interchange with the later oil sumps due to their having a different bolt hole pattern, and uses 19 bolts in order to secure it to the oil sump mounting flange of the engine block.

If you change to a different capacity oil sump, you will need the appropriate dipstick and dipstick tube. There were three different dipstick tubes, all of which were flared at the top in order to accept the oil seal on the dipstick, and three different corresponding dipsticks. The dipstick peculiar to the three-main-bearing 18G and 18GA engine blocks, as well as to the five-main-bearing 18GB, 18GD, and 18GF engine blocks, is cranked (BMC Part # 12H 74), uses a large, bulky oil seal (BMC Part # 1B 1735) that fits both into and over the very shallow flair of the dipstick tube (BMC Part # 1B 1063), and is the longest of the three dipsticks. Often the large rubber seal will be found to have contracted and shrunk with age, allowing the tip of the dipstick to sink to the bottom of the oil sump. If this problem is left unattended to, the tip of the dipstick can actually wear a hole in the bottom of the oil sump! In any case, the dipstick should not be so loose that it moves about, and if it does, engine vibration will cause it to make a noise that will drive you insane in chasing down. However, the factory became aware of the problem, and a new dipstick was designed that eliminated this problem. The second dipstick, which is used on the five-main bearing 18GG, 18GH, 18GJ, and 18GK engine blocks, is also cranked (BMC Part # 12H 2964), but uses a notably smaller oil seal that fits into the deeper flair of its slightly shorter dipstick tube. The third dipstick, which is used on the 18V engines (BMC Part # 12H 3963), is straight, uses the same oil seal (BMC Part# 1B 1735) as that used on the dipsticks of the 18GG, 18GH, 18GJ, and 18GK engines, and as such fits into the longer dipstick tube with a deep flair that is identical to that of the second dipstick tube. Its corresponding dipstick tube (BMC Part # 12H 3351) is the longest of the three. In all cases, these seals should be kept in good condition in order to maintain the proper sealed vacuum level inside of the engine. In order to get an accurate reading on the dipstick when using the earlier larger-capacity oil sump on an 18V engine block you will need to use the dipstick (BMC Part # 12H 2964, Moss Motors Part # 451-355) and the dipstick tube (BMC Part # 12H 2966) of the 18GF, 18GG, 18GH and 18GK engine blocks. On 18V engine blocks with their original smaller-capacity oil sump, you will need to use the later dipstick (BMC Part # 12H 3963) and dipstick tube (BMC Part # 12H 3351, Moss Motors Part # 460-035).

Of course, oil sump gasket leaks develop over time as the gasket deteriorates from both heat and its constant contact with oil, in time becoming a real nuisance. Fortunately for them, the engineers at Fel-Pro have come up with a solution called the PermaDryPlus® Oil Pan Gasket (Fel-Pro Part # OS20011). Constructed of high temperature resistant, edge-molded silicone rubber on a rigid carrier, it provides a superior fit, as well as both high heat and vacuum resistance, while the included Oil Pan SnapUps speed installation.

The oil supply generated by an Original-Equipment oil pump and the oiling passages in the engine block are quite adequate for use within the normal operating speeds of a Original Equipment specification-output engine. However, if an increased-output engine is called upon to operate at higher than normal engine speeds or under heavier loadings, such as when a Piper BP285 camshaft is installed or the engine is modified to Big Bore specifications, it becomes prudent to modify the oiling system. This is due to the fact that the oil flow from the front main bearing supplies the number one cylinder's connecting rod big end bearing, oil flow from the rear main bearing supplies the number four cylinder's connecting rod big end bearing, and the oil flow from the center main bearing of the crankshaft supplies both of the connecting rod big end bearings for cylinders numbers two and three. The oil passages from the high-pressure oil gallery to the main bearings are all the same diameter, thus for the same oil pressure they all have the same oil flow capacity. However, the passage feeding the center main bearing of the crankshaft has almost twice the oil flow requirement due to the fact that it is oiling three bearings (the center main bearing plus two connecting rod big end bearings) as opposed to those feeding only the two main and the connecting rod big end bearings for each of the front and rear cylinders (one main bearing and one connecting rod big end bearing).

In order to compensate for this demand, the oil passage from the pump to the oil outlet at the rear of the engine block should be enlarged to ½" (.500"), which is the same size as the outlet on the oil pump. A special ½" Internal Diameter oil feed line using -10 Aeroquip adapter fittings will need to be custom-fabricated in order to enable the increased oil supply to flow efficiently to the oil filter stand. The oil feed passage to the center main bearing will then need to be enlarged from its original 5/16" (.3125") diameter to 11/32" (.34375") diameter and the crankshaft main bearing journals #2 and #4 crossdrilled and center grooved. Unless the crankshaft is to be stress relieved, as in the case of by nitride hardening, this grooving should be accomplished by grinding rather that by turning on a lathe in order to prevent the creation of stress risers that could result in breakage of the journals. In order to prevent lubrication failure resulting from centrifugal forces at high engine speeds, the journals for the connecting rods should then be crossdrilled 110° back from Top Dead Center with the drilled passage intersecting the original oil passage. These modifications were standard practice amongst the engines used by the factory racing team and were to be found on all early MGB crankshafts in order to comply with the homologation rules of racing associations. Remember that whenever any journal has new passages drilled into it, the mouths of the passages will always need to be chamfered in order to eliminate stress risers and that the journals containing them will need to be reground afterwards. Ideally, the crankshaft should then be Nitride hardened in order to extend its service life.

Nitriding is a chemical hardening process in which the part is heated in a furnace, the oxygen is vacuumed out, and a chemical gas that penetrates the entire surface is introduced.

The depth of hardness is dependent upon the time the part is exposed to the gas. Typically, a nitride-hardened crank will have a depth of hardness of about 0.010". Nitriding is a low-heat process compared to Tuftriding, but it shares the advantage of avoiding the introduction of the localized stress zones that occurs during induction hardening. The Nitride heat treatment process offers several advantages over carburised treatments, such as excellent levels of hardness and retainment of dimensional accuracy. The Nitride-hardened component is resilient to softening and will retain its surface hardness up to temperatures of 1,100° Fahrenheit (500° Celsius), whereupon cooling the whole component will revert to its original degree of hardness. A Nitride-hardened component is less easily heat damaged by temporary lubrication failure than a carburised counterpart. Nitride hardening improves fatigue strength and achieves a mean hardness of 750-950 HV. As a former Tool & Diemaker, I assure you that stresses on the surfaces of the material are an unavoidable result of machining. A cutting process tears and compresses metal, resulting in microscopic hills and valleys from which cracks can develop. In heat treating, the metal is brought up to a temperature whereat the crystals of which it is composed are able to reorient themselves in a more uniform manner, then cooled under controlled circumstances so that the form of the resultant material is stabilized.

With these modifications, a high flow volume oil pump becomes useful as the extra oil flow through the bearings provides additional cooling under conditions of high loads and sustained high engine speeds, enabling the engine to be reliably run to 7,000 RPM. However, if you desire higher operating speeds than 6,500 RPM, you will have to fit rocker arms that run on needle-roller bearings, as the Original Equipment rocker arm bushings will ultimately fail at such elevated engine speeds. Cambridge Motorsport offers these items as roller rocker arms in either the Original Equipment lift ratio of 1.426:1 or in a 1.625:1 high lift ratio with the option of either central or offset oil feed in the rear rocker shaft pedestal, thus making the system available for use with all of the variants of the heads used on the BMC B Series engine. Of course, these will need to be inspected for wear on a more frequent basis than would otherwise be the case were you employing bushings. The rocker arms of both types are located by tubular steel rocker arm spacers in order to prevent the rocker arms from longitudinal "walking" at high engine speeds. Such tubular steel rocker arm spacers were used by the MG factory race team for the same purpose and were available to the public as special-order competition equipment (Long, Special Tuning Part # AEH 764; Short, Special Tuning Part # AEH 765). However, the Original Equipment rocker arm spacer springs (BMC Part # 6K 871) are quite adequate for this task at engine speeds below

6,500 RPM and have the dual advantage of less friction as well as damping valvetrain vibration and its resultant noise.

If your valve train seems to be excessively noisy, either use a degree wheel or mark off from the Top Dead Center mark on the harmonic balancer pulley wheel a series of additional marks at 10° intervals. These will need enough to go about 40° past your camshaft lobe profile's opening point. Use a dial indicator to measure the lift at every 10° mark. It is better to measure the lift at the top of the pushrod, although to do this, you will need to move the rocker arm aside, or remove it completely, depending on which method of rocker arm location it is that you employ, springs or tubular spacers. The lift that you measure at each 10° interval is not the actual lift velocity, but is an expression of the lift velocity. At some stage as you move towards Top Dead Center there should be a phase where the velocity (as shown by lift per 10°) is constant for about 20°. This is the opening ramp of the lobe contour, also known as quietening ramp, and it is the part of the lobe which is designed to eliminate the clearance. This ramp typically lasts between .009" total lift and .015" total lift, but may vary somewhat from that figure depending upon the profile. Do these figures sound familiar? Yes, that is approximately the tappet clearance range of most camshafts. Most opening ramps are lifting at a rate somewhere in the range of .003" to .005" per 10°. BMC used a lift rate of .003" per 10°, and newer performance cam profiles can be up to .005" per 10°. If the rate of lift exceeds this amount, then the valve clearance will be taken up with a hammering action, and consequently the valve gear will be noisy and accelerated wear will occur. The higher the rate of lift, the worse these effects will be. Even at .005" per 10°, there is a noticeable increase in valve gear noise. If the opening ramp velocity is above the range, then that is likely to be where the source of the tappet noise is. A related problem can occur if the maximum velocity of the lobe profile is too great for the size of the tappet. In this case, the lobe contacts on the corner of the tappet rather than its actual contact face. The corner of the tappet digs in an causes rapid wear of both the tappet and the lobe, then noisy operation becomes ever worse as nothing around the lobe or the tappet is the right shape any more.

Oil Coolers

Enhanced-performance engines produce more power by creating more heat in order to expand the atmosphere inside of their cylinders in order to force the pistons, connecting rods, and crankshaft to do more work. This in turn places increased pressure upon their bearing surfaces inside of the engine. This increased heat and pressure places additional strain upon the lubricating oil, shortening its useful lifespan. While the radiator performs the function of cooling both the cylinder head and the cylinders, it should be noted that it is the oil that cools the internal parts of the engine. While mineral-based oils are fairly efficient at absorbing and transferring heat, the more heat-resistant synthetic oils are relatively inefficient at this task.

In order to assist in this function, as well as to help protect the lubricating qualities of the oil from breakdown, an oil cooler was fitted to all MGBs except during the 1975 through 1980 production years when power output was chopped in an effort to meet emissions regulations. The correct running temperature of the oil is perhaps even more important than coolant temperatures. North American market cars had a 13-row cooler, and this should be considered to be the absolute minimum for an enhanced-performance engine. If your car has one, be sure that it is hot tanked along with the other components and thoroughly cleaned out before reinstalling it. The importance of giving it a thorough cleaning cannot be overestimated as it will contain sediment that will ruin your new engine. If you are replacing it or installing one for the first time, use one that has at least 16 rows and install a 200° Fahrenheit (93.3° Celsius) thermostatic bypass valve as overcooled oil can rob power and lead to accelerated wear. Because overcooled oil is thicker than it would be at normal operating temperatures, the piston rings will tend to “hydroplane” over the oil and, on the upward stroke of the piston, scrape it into the combustion chamber where it will be burned, leading to carbon deposits and an increased risk of preignition. Thermostatic bypass valves have a dual function in which the flow of oil is either channeled directly to the oil filter when the oil temperature is below 200° Fahrenheit (93.3° Celsius), or is channeled directly to the oil cooler when the oil temperature is above 200° Fahrenheit (93.3° Celsius). An excellent thermostatic bypass valve with ½” NPT threads is available from Perma-Cool (Perma-Cool Part # 1070). Perma-Cool has a website that can be found at http://www.perma-cool.com/ . Personally, I use Mobil 1 in all of my cars and I agree that it resists molecular shear better than the petroleum-based oils. However, it is always best to think in the long-term. Just because modern oils can stand the heat without breaking down as rapidly as they did in the past does not mean that the consequences of the heat can be thus ignored. As motor oil degrades with use, it becomes less thermally efficient and thus cannot remove heat as well as it did before. An oil cooler helps to get rid of the excess heat that can destroy the additives that were added by the refiner in order to help protect the engine. Engines last longer when their operating tolerances stay within engineering specifications, even when the oil is of the best lubricating quality. Thus we see that the crucial deciding factor behind installing an oil cooler is the fact that it is the oil that carries away the heat from the moving parts. This being the case, An oil cooler can be considered to be a wise move for a car that is going to be kept and run for many years.

For those who wish to monitor their oil temperature, an oil temperature gauge can be substituted for the oil pressure gauge in the dashboard of the MK II models and a sensor incorporated into either the oil line in the case of the original equipment system, or into the oil filter head of a remote oil filter. If you wish to retain the ability to read the oil pressure, the combination gauge that incorporates both the water temperature and the oil pressure readout used on the MGB MK I can be substituted for the water temperature gauge. This will require fitting an adaptor to the cylinder head so that the capillary sensor of the earlier gauge can be installed. The capillary pipe should exit the fitting on the cylinder head with a P turn about 2 inches in diameter in order to route it back towards the cylinder head where it follows the cylinder head gasket line to the number plate, whereupon it dips down under the plate boss and runs level back underneath the heater valve where it is clipped to a small plate that is secured under the lower bolt of the heater valve. It then runs straight back from there into a movement-absorbing coil approximately 4" (100 mm) in diameter before running across the firewall (bulkhead) shelf and being fixed by a P clip before going through the grommet on the shelf. There is a further P clip on the inside of the firewall (bulkhead).

Note that the 1968-1971 models used an electric oil pressure gauge (BMC Part # BHA 4687) powered by an electric transmitter unit (BMC Part # ?, Moss Motors Part # 131-580) that was mounted behind the alternator. This electric transmitter unit is both expensive and is prone to failure in use. This electric transmitter unit failure is usually characterized by the oil pressure dropping to a low reading, sometimes zero, while driving. If you put a mechanical direct pressure reading gauge on the engine, you can verify if the problem is the electric transmitter unit. MG came to recognize this problem, and chose to return to a mechanical direct pressure gauge system on the North American Market models in August of 1971. Fortunately, this later system, complete with a same-size gauge (BMC Part # BHA 5091 or BHA 5092, Lbs and Kg reading, respectively), can be readily installed on the 1968-1971 models.

Another item that is used in order to help reduce oil temperature is a larger capacity die cast finned aluminum alloy oil sump. These are particularly popular with the racing crowd, as they eliminate the need for the complexity of an oil cooler system. They have integral vertical internal baffle plates that preclude oil surge. The optional removable aluminum alloy baffle plate covers are available for both 105mm and 135mm oil strainer sizes. Primarily intended for racing use, these are rarely seen on street engines as they are both expensive and, being an aluminum alloy casting, are more vulnerable to damage by debris thrown from the front tires. While this may seem to be a matter of grave concern for owners who operate their cars on public roads, it should be noted that prior to the absorption of MG into the British Motor Corporation automotive empire, MG engines commonly had aluminum alloy sumps, as in the case of the XPAG and the XPEG engines of the MG TF. This was partly due to the fact that they convey an additional advantage of imparting greater rigidity to the engine block. This being the case, it is a good idea for those who choose to bore their cylinders beyond the factory's +.040" maximum oversize. They are available for the B Series engine in both LM24 aluminum alloy and in magnesium alloy in both baffled and unbaffled form from Cambridge Motorsport. Both are quite rugged, having been cast with a minimum thickness of .125" (3.175mm) and possessing both internal and external reinforcing ribs, and weighing in at about 4 Kg complete with the optional baffle plate. They also have the dual advantage of a slightly larger capacity of about 1 pint and offer improved heat dissipation. As a result of this improved heat dissipation, the size of the oil cooler can be reduced, thus presenting reduced obstruction of airflow through the matrix of the radiator for the engine coolant. Should you decide to use one, be mindful of the fact that they use a large rubber O-ring for sealing instead of a gasket and as such will require that you have the mounting flange on your engine skimmed flat and the mounting holes chamfered and retapped with a ¼"-20 UNF rethreading tap, otherwise it will certainly leak and possibly crack.

Oil Filtration

When it comes to protecting your engine, there is no substitute for an effective oil filtration system. The felt elements used in the early oil Tecalamite and Purolator oil filters are technically obsolete. They simply cannot filter oil as effectively as the filtering mediums that are used in the modern, spin-on cartridge-type oil filters. Fortunately, the oil filter stand first introduced on the late 18GG and 18GH engines that uses a cartridge-type oil filter can be fitted onto the older engines. If you have an earlier engine, obtain the oil filter stand (BMC Part # 12H 3273) and its rubber O-ring (BMC Part # 8G 619) that fits to the engine block, the machine bolt and copper seal that attaches the adapter to the engine block, and the copper washer and adapter for the oil hose (flexible pipe) that goes in front. This last item may not be needed, as there are two types of oil hose (flexible pipe) adapter fittings: one that uses a large banjo bolt (BMC Part # 1K 2142) along with an inner copper washer (BMC Part # 1H 898) as well as an outer copper washer (BMC Part # 6K 501), and one that uses a screw-on adapter fitting (BMC Part # AHH 6701) with a gasket (BMC Part # 6K 431). You will need the oil hose (flexible pipe) adapter fitting only if yours does not use the banjo fitting.

Make sure that the oil filter stand still has the anti-drain tube fitted in the stanchion. Avoid using oil filters that are taller than necessary, otherwise when the engine is shut off any oil above the anti-drain tube will drain out of the oil filter into the engine, leaving an air pocket that must be filled before oil pressure can be achieved, an event that can take place only after the then-displaced air enters the system of oiling passages (not a good thing!). Also, beware of short oil filters that are about the same height as the length of the anti-drain tube. These can cause the anti-drain tube to come into contact with an internal support inside of the oil filter, causing it to bend. This is sure to damage the combination anti-drain-back and by-pass valve. A difference of 1/8" (3.175mm) in the height of the internal construction from what is correct could give normal performance at low engine speeds, yet produce total disaster at full throttle. Consequently, any oil filter should be examined to be certain that all of the internal components of the oil filter should be at a minimum a full diameter of the anti-drain tube away from the anti-drain tube when fully tightened. Do not attempt to avoid this by cutting down the anti-drain tube by ½" (12.7mm). This will allow more oil to drain out of the oil filter, thus requiring it to be refilled before proper oil pressure can be achieved, just as if you had installed an oil filter that is taller than is necessary.

Unfortunately, the filter mounting threads on the stanchion of the oil filter stand occasionally become cross-threaded as a result of the clumsy installation of a new oil filter. So, here's an old Tool & Diemaker's tip: you can do a reasonable job of repairing the bunged-up threads with a purpose-made thread file. Be very conservative with that thread file. Once you remove the metal, you can't put it back on. After you finish with the file, run a ¾-16 sizing die over the threads to form them up properly.

The choices of quality oil filters available are almost endless, the best of these including those from Mann (Part # W917), Purolator Pure One (Part # PL20081), AC Delco (Part # PF13C), Motorcraft (Part # FL300), Volvo 3517857, Wix 51362, and NAPA 1068, but the most effective is also the easiest to install: the K&N Performance Gold Oil Filter (K&N Part # HP2004). They all use the same mounting thread of ¾"-16 threads per inch (SAE). Note that even with a good quality oil filter, you can have a bit of a hassle getting the oil filter to screw on. If you attempt to simply spin it on, you risk cross-threading with the result of damage to the threads on the stanchion of the mount. Instead, lubricate the sealing ring with clean oil and then try turning it in the opposite direction, with a little downwards pressure, and you will feel where the threads meet. At that point, turn it down onto the oil filter stand. It is all a bit of a learning experience, one that many of us have gone through. if the only oil filter that you can get for the B Series engine is the K&N oil filter, then you have definitely lucked out. Its resin-impregnated filtration element filters down to both 2 and 3 microns and its construction is as rugged as it can be. That makes it unique as all of the other filtration elements only filter in one size only. It even has a spring-loaded bypass valve in case of a clogged filtration element instead of a simple flap-type plastic bypass valve. Even then, it can handle up 550PSI before the reinforced filtration element collapses under the pressure. A high-pressure oil pump in a B Series engine can only attain a bit more than 100 PSI, so you are quite safe there. The 1" nut on top makes it a snap to get off when the swollen seal of other filters would cause it to refuse to budge. Needless to say, it is the oil filter that I always use, even if it does cost more (you get what you pay for).

If yours is a pre-1968 Lucas C41/2 generator (dynamo)-equipped engine, you might prefer to convert to the spin-on oil filter adapter offered by Moss Motors (Moss Motors Part # 235-940) as it mounts the oil filter cartridge in a downward position and can accept long oil filters with a substantially greater filtration media surface area. This adapter will accept oil filters made by AC Delco (Part # PF60), Purolator (Part # L20064 and Part # L24457), and K&N (Part # HP 2009). When installing a new oil filter, always check to be sure that the old sealing ring from the old filter has not separated from the body of the old oil filter canister and consequently has become stuck in place inside of the oil filter stand mount. If it is, you will experience a very serious oil leak immediately after firing up the engine. Make sure that the new sealing ring of the new oil filter canister is properly oiled before installation and is seated along its entire circumference in its mounting groove base in the oil filter body. Note that this oil filter adaptor will not fit on the MKII models, as they use a pre-engaged starter that is too long to permit it to fit.

Although the engine block is invariably cleaned out and painted after all of the machining operations are done, sometimes this cleaning is not done as diligently by commercial shops as one would hope. In any case, there is always a little of machining dust bit left lurking in the recesses that are the most difficult to clean, just waiting to do harm at some future date. I have seen these particles appear in the oil of engines that had over 50,000 miles on them. Dating back to the days when oil filters consisted of little more than steel wool in a can, an Old-Timey-Mechanic's trick for protecting the engine during its break-in period is to use a large elastic band to secure a powerful magnet to the oil filter in order to capture ferrous metal particles that circulate in the oiling system, thus better protecting the finely machined surfaces of the engine. Since a casting may be considered to be a large number of holes held together by metal, the use of magnets is particularly beneficial during the break-in period as there is always some fine metal dust remaining from the machining processes that is wedged into the porous face of the cast iron engine block. When the engine block heats up, these pores expand and the fine metal dust is released. A fine-straining oil filter may stop them, but such an oil filter gets clogged up earlier and then its bypass valve opens, allowing everything to circulate throughout the engine along with the oil, be it dirt, grit, metal particles, bits of old dinosaur bones, you name it. If the oil filter does not have a bypass valve, then the oil pressure crushes the filtration element, pulling its ends away from their sealing seats, and then the oil simply flows around the filtration element into the engine, often taking contaminants washed from the oil filter element along with it. A device called the "FilterMag" is available in diameters from 60mm to 140mm in both standard and heavy-duty versions. FilterMag has a website at http://www.filtermag.com . Another good precaution is to install a ¼" BSP magnetic oil sump plug (Moss Motors Part # 328-282). I have been using both methods for over thirty years and am always surprised at what is caught by the magnets. Of course, it goes without saying that magnets are no substitute for a good, fine-straining oil filter!

Oil

Always check the API ratings of any oil that you are considering for purchase. If you look at the specifications on the containers, amongst the other markings you will see an "API", followed by a series of letter, S for gasoline (petrol) and C for diesel. Most gasoline (petrol) engine oils these days are either API SL or SM. Our B Series engines were originally designed when SB was that rating that was in force, and were produced through the SC, SD and SE ratings, with the SF, SG, SH. and SJ ratings for more recent engines (there are no SI or SK ratings). Up to and including the SJ rating they were all backwards-compatible, i.e. earlier engines got the benefits of the later ratings from improved formulation as long as you used the original viscosity, the modern very low viscosity oils not being suitable for our engines. After the SJ rating oils, the environmentalists managed to get the proportions of ZDDP (zinc dithiophosphate), phosphorus and other additives reduced, partly for environmental reasons, and partly because they tend to poison catalytic converters over time, resulting in a reduction of its lifespan. SM has very low ZDDP and should be avoided, as it is known to cause problems. Therefore you should stick to API SJ or earlier, but the later the rating, the better. Castrol XL 20W-50 is API SE thus is suitable, as is Halfords Classic 20W-50. Be aware that the lower viscosity oils do not contain enough ZDDP to provide proper pressure protection for the tappets of our engines, Castrol GTX High Mileage 15W-40 is API SL for example. A ZDDP additive should always be used whenever such oils are used during the break-in period. Once the break-in phase is over, the use of this additive should be discontinued.

The best petroleum-based oil for the B Series engine of an MGB is Castrol XL 20W/50, while the best synthetic oil is Mobil 1 15W/50. I use the former in my transmission and the latter in my engine. Why do I not use the less expensive Castrol in my engine? Simple: most of the wear that takes place in an engine occurs during the warm-up period because the oil is too thick to flow easily. Once the engine gets up to operating temperature, it flows freely and does its job outstandingly well. The Mobil 1 synthetic flows just as well when it is cold as it does when it is hot. It also does not thin out at high temperatures, which is a serious plus in an engine that has been modified for higher power output. Why do I not use Mobil 1 in my transmission? Because the baulk rings (synchro rings) of transmission require a certain amount of friction in order to perform their function properly, so it seems to shift a bit better with the Castrol petroleum-based oil, and the sliding annular clutch (conical clutch) of the Overdrive unit does not try to slip as it does when filled with the super-slick Mobil 1. How often should you change your oil? Read on-

The concerns of many about acid buildup and moisture condensation in the oil are right on the money. However, there is another factor that needs to be mentioned: the effects of blow-by. Just because a static wet compression test may give readings that seem up to specification does not automatically imply that the compression rings are doing an adequate job of containing the enormous pressures of combustion. When partially burned fuel blasts past the rings and onward into the crankcase, the oil becomes contaminated with carbon, one of the hardest substances known to man and the enemy of all precision-machined surfaces. It is the stuff of which a lovingly-blueprinted engine's nightmares are made of. How to tell the condition of the compression rings without putting the car on a dynometer and running it against a heavy load? Simple: If your oil turns an opaque black within 3,000 miles, then you have a contamination problem. How to protect that big investment that you have made in your newly rebuilt prize?

First, be picky when it comes to your choice of oil. Use only the best. True, you can use lesser quality oil and never experience an oil-related catastrophic failure. Today's oils are far better than what was available thirty years ago, and outright failures that are oil-related are all but unknown today. Nevertheless, a better oil can mean a longer engine life.

Second, always change the oil filter whenever you change the oil, and use the best, finest-straining oil filter that you can get. The fewer solid particles there are circulating inside of your engine, the better off your engine is.

Third, be ruthless when it comes to oil changes. If the oil is opaque, it is seriously contaminated, so change it. If it has 3,000 miles on it, change it. If it has been in the engine for six months, change it. If you are putting the car in storage for the winter, change it. When you change the oil, do not be hasty and replace the drain plug when the drain flow slows to a drip. If you put a measuring cup under the drip and wait a few of hours, you will get about twelve additional ounces of the nastiest, grittiest stuff you will ever have the displeasure to see coming out of an engine. This cr@p will wear out your engine. Let it drain and get all of that old oil out.

True, you do not have to be as fanatical as I am. You can use a less-expensive, ordinary oil, inexpensive, ordinary oil filters, change your oil as quickly as you can, and still expect to get a good 80,000 miles out of your engine. Today's oils really are that good. But personally, I figure that if I do not get at least 110,000 miles out of an engine, then it is a lemon. To me, 140,000 miles is a lot more reasonable, and achievable.

Bearings

When selecting bearings, most commercial engine builders concentrate only on getting the proper clearances and maintaining adequate oil pressure. Durability is unquestioningly expected from any bearing chosen, and the advantages of different bearing material options are often left unconsidered. If engine operating conditions are taken into consideration and bearing materials chosen accordingly, then the likelihood of long-term durability is greater. Enhanced-performance engines make greater demands upon their bearings than Original Equipment specification engines do, requiring bearings with greater eccentricity. The term "Eccentricity" refers to the variation in the Inside Diameter (I.D.) of a bearing assembly when it is measured at different points around its bore. A properly designed engine bearing is not truly "round" when it is installed in the connecting rod or engine block. Under operating loads, a connecting rod or main bearing housing bore will distort, pulling inward at the parting line between the upper and lower halves. In order to keep the bearing from contacting the journal in these areas, it becomes necessary for the bearing design to include additional clearance at each parting end of the bearing. As engine loads increase, so does the amount of distortion, thus bearings that are highly stressed by increased engine speeds and heavier loadings require greater eccentricity than do bearings intended for more sedate use.

Be aware that there are essentially three types of bearings available to support the crankshaft. The first and best of them is a trimetal type with an Indium overlay, as made by Vandervall, the Original Equipment manufacturer that provided the bearings for the engines used in the MGB. These bearings lack the common white / gray color because the cosmetic tin plating has been eliminated. Tin can migrate across a bearing's steel backing under hard driving conditions, forming high spots on its Inside Diameter (I.D.) that will intrude into the oil clearance. This will result in concentrated load areas that are susceptible to premature fatigue. Lacking tin, these Indium bearings feature greater dimensional accuracy, reduced over plate thickness, and improved resistance to fatigue damage. The extra strong, but very thin overplate layer does have a cost - the bearing surface is more susceptible to damage from debris, making diligent oil and oil filter changes mandatory, but this is a small price to pay for the increase in durability. However, Indium is highly corrosion resistant and does a much better job of spreading oil over its surface than Lead / Bronze or Lead / Copper, a very real advantage at both high pressure loadings and low oil flow conditions, as well as at high temperatures where oil can lose its cohesion. It is these characteristics that make this the most desirable type of bearing for use at higher engine speeds, and this is the type of bearing that was selected by the factory engineers to be Original Equipment in those B Series engined destined for use in the MGB. The second type is Lead / Bronze or Lead / Copper, and third the A (Aluminum alloy) or SA (Silicone / Aluminum alloy) material normally used in Original Equipment Manufacturer engines for family car, primarily due to their lower cost, as well as the fact that their embedability allows them to withstand dirty oil better. However, it should be understood that when they become worn, the grit embedded within them will become exposed and result in more rapid wear of the bearing surfaces of the crankshaft journals. The first two types are acceptable for long-term use in a high performance engine, although the Indium bearing is considered to be the more desirable of the two.

Some very high performance applications may require different bearing clearances than factory specification engines. Many engine builders target a clearance range between .0022" (.5588mm) and .0027" (.6858mm) in order to keep running temperatures of the bearings within reasonable limits so as to prevent molecular sheer and consequent failure of the lubricating oil. Clearances greater than .0030" (.0762mm) are not normally recommended. Some engine builders require higher oil pressures and larger flow volumes than an Original Equipment oil pump can provide, particularly at lower engine speeds due to the fact that large bearing clearances will lower oil pressure, thus requiring a high flow volume / high pressure oil pump and modified oil feed passages inside of the engine block.

Crankshafts

The three-main-bearing crankshaft of the 1800cc engines was a redesign of the crankshaft that was employed in the earlier 1622cc engine. It featured larger-width bearings in order to enable them to absorb greater stress, but the crankshaft had an unfortunate tendency to "whip". In the end, warranty claims for broken crankshafts caused by driver abuse forced the redesign to the five-main-bearing version. The five-main-bearing crankshaft is heavier than the earlier three-main-bearing version (32 lbs vs. 25 lbs), so the three-main-bearing crankshaft accelerates to high engine speeds more quickly. However, one should be careful not to exceed 5,000 to 5,500 RPM on the three-main-bearing engine. The primary reason for this prohibition is not the crankshaft, but rather the diagonally split big end of the connecting rods. These were always the Achilles' heel of the three-main-bearing engines. The rapid reverse of direction at the end of each stroke of the piston puts a great deal of stress on the diagonally split big end design. Even worse is downshifting to decelerate the car, which develops a great deal of vacuum inside of each cylinder and again subjects the diagonally split big end to tremendous amounts of stress. It should be noted that every factory racing MGB used a three-main-bearing engine, even after the five-main-bearing engine was introduced. The factory racing team preferred the three-main-bearing engine because its lower internal friction was worth about 4 BHP, and its lighter crankshaft and flywheel made it more willing to quickly attain the high engine speeds demanded by racing circuits. Of course, in the interests of reliability, these were replaced on a routine basis.

It is not commonly known that the five-main-bearing B Series engines used in the MGB actually made use of a succession of four crankshafts. The first was peculiar to the 18GB engine, being of machined from a forged steel blank, which used a 10.75” (273.050mm) flywheel (BMC Part # 12H 1474) and ring gear (BMC Part # 1G 2874, Moss Motors Part # 190-040), neither of which are interchangeable with the later flywheels and ring gears used in the later five-main-bearing engines. The second crankshaft was a forged EN 16 carbon steel design that had a wedge-shaped taper to its counterweights. It may be found in 18GD, 18GF, 18GG, 18GG, 18GH, 18GJ, and 18GK engines. Both of these first two crankshafts had shoulders on their throws near the crankpins that were used for mounting during the machining process. New production techniques permitted the elimination of the shoulders, and made possible the third crankshaft, which was a slab-sided cast iron design. It may be found in 18V-581-F-H, 18V-581-Y-H/L, 18V-582-F-H, 18V-582-Y-H/L, 18V-583-F-H, 18V-583-Y-H, 18V-584-Z-L, 18V-585-Z-L, 18V-672-Z-L, and 18V-673-Z-L engines. The fourth was a forged EN 16 carbon steel design, easily recognized by its flared counterweights which did a better job of smoothing out vibration. It may be found in 18V779-F-H, 18V-780-F-H, 18V797-AE-L, 18V-798-AE-L, 18V-801-AE-L, 18V-802-AE-L, 18V-836-Z-L, 18V-837-AE-L, 18V846-F-H, L, 18V847-F-H, 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L, 18V-891-AE-L, 18V-892-AE-L, and 18V-893-AE-L engines.

The crankshaft with the best balance and wear characteristics is the slab-sided five-main-bearing cast iron version found in the early versions of the 18V engines (18V-584-Z-L, 18V-585-Z-L, 18V-672-Z-L, and 18V-673-Z-L). Although technically slightly weaker than the alternate forged EN 16 carbon steel crankshafts used in other versions of the five-main-bearing engine and seven pounds heavier than the earlier three-main-bearing steel crankshafts (32 lbs vs. 25 lbs), it is strong enough for the streetable enhanced-performance engine that is the goal of this article. Do not succumb to the temptation to use the less expensive crankshaft from a Morris Marina. I do not say this because of its too-small crankshaft spigot pilot bushing. The spigot bore can be easily machined to accept the larger-diameter crankshaft spigot pilot bushing of the MGB. I give this warning because this particular crankshaft is made of flow-cast spheroidal graphite iron and was intended for use in sedate family cars. It is simply not strong enough for use in an enhanced-performance engine. In addition, it uses a completely different, much heavier (28 lbs) flywheel that is ¼" (.250" / 6.35mm) thinner and so much larger in diameter that it will not fit inside of the bellhousing of the four-synchro transmission that is used in the MGB.

Although the five-main-bearing crankshafts found in MGB engines are all but unbreakable in an engine rebuilt to Original Equipment specifications, with a seriously power-enhanced engine that is equipped with a hot camshaft, it is merely prudent to take precautions. As the diameter of the main bearing journal is reduced, the fillet radius needs to be increased in order to ensure that a stress point that can result in a broken crankshaft does not develop. Be sure to tell the machinist that you want the journals fillet-radiused to .030" (.762mm) at the web and that they should be glass beaded afterwards in order to both increase the fatigue life of the crankshaft as well as to reduce the chances of breakage under heavy loadings at high engine speeds. Glass beading the fillet radius will reduce subsurface stresses that result from the machining process. It is the amount that the main journals must be reduced that is most likely to have an effect on whether or not oversized crankshaft thrust washers will be needed. Note that one side of the crankshaft thrust washers is faced with steel and that its opposite side is a bearing surface made of white metal alloy with oiling grooves in its face. The white metal alloy side should face against the crankshaft, and the steel side should face against the engine block. That is, the crankshaft thrust washers (BMC Part # 12B 120 (Upper), 12B 120 3 (Lower), Federal-Mogul Part # 65077BF) must always be installed with their white metal alloy facings and their oil grooves facing away from their adjacent crankshaft journal main bearings. That is, the white metal alloy side should face against the crankshaft, and the steel side should face against the engine block, otherwise rapid wear and possible failure of the crankshaft will result.

Racing crankshaft designs often have a larger fillet radius in the web area where the connecting rod journal meets the counterweight. This rounded inside corner increases crankshaft strength, but can interfere with the big end bearing of the connecting rod. Many high performance connecting rod big end bearings are chamfered in order to provide the side clearance necessary for such large-filleted crankshaft journals. The chamfers are used on the sides of the bearing alongside the crankshaft counterweight in order to provide as much surface area as possible as well as to provide added protection against side thrust forces. Even though these bearings are designed for this purpose, it is still important to check for adequate clearance in their chamfer area.

Do not blindly have faith that your machinist has created the correct journal diameters. Unless you have access to precision measuring equipment in order to assure that the critical dimensions are in fact correct, these clearances should always be checked with a Plastigauge. Although choosing the dimension that the journals should be reground to may seem to be a straightforward matter, in reality the purpose to which the engine is intended to be put to must be taken into account when making the choice. While an engine built to what is essentially an Original Equipment specification can have its journals ground to the middle of the tolerances set by the bearing manufacturer, an enhanced-performance engine is quite another matter. As a rule, most outright bearing failures are caused by a localized buildup of heat within the space between the bearing and the journal. This most commonly occurs at high engine speeds when the engine is called upon to work against heavy loads, an operating condition that an enhanced-performance engine faces often. Heat that is generated within the oil by friction of the moving parts is removed only by the oil flowing through the bearings. When the flow of oil is insufficient to carry away the heat, the oil inside of the bearing will overheat, causing the oil film to break down and lose its lubricating properties, consequently allowing metal-to-metal contact that will result in bearing failure and crankshaft damage, and even catastrophic failure of the engine. This can be countered to a limited degree by increasing the bearing clearances, which in turn means that the load supporting capacity of the bearing is reduced, resulting in a shorter service life. As a compromise, the journal should be sized .0003" to .0005" (.00762mm to .0127mm) smaller than the bearing manufacturer's recommended maximum clearance. This will have an additional benefit of also reducing oil drag, the result of which will be a very small increase in power output. The minimum diameter for the crankshaft journals is limited to 2.086" (52.9844mm) (-.040" / -1.016mm) undersize. Be aware that in order for this solution to be effective, the system must be modified for increased oil flow, otherwise inadequate delivery to the low pressure gallery will result.

When reground, the journal surfaces of a crankshaft will have microscopic peaks that are "tipped" in the direction that the sparks spray during grinding. If allowed to remain, lubrication will be interrupted when the engine is running and the bearings will wear prematurely. After the crankshaft has been ground, it is important that all bearing journals receive a three-step final polishing so that these peaks are tipped into the opposite direction than that in which the crankshaft rotates. This is referred to as the "favorable" direction. In the first step 280 grit paper should be used, followed by 320 grit paper, and finished with a very fine (400 grit) paper for the third and final step.

Be sure to check both ends of the crankshaft for any grooves worn into it by the old seals. If you can catch your fingernail on an oil seal wear track or shaft groove, then a repair sleeve should be installed in order to prevent leakage, especially if your machinist cannot polish them out. Chicago Rawhide produces a Speedi-Sleeve for the rear end of the crankshaft (Moss Motors Part # 520-515). SKF produces a Speedi-Sleeve for the rear end of the crankshaft (SKF Part # 520-515). The Speedi-Sleeve is made of high quality SAE 304 stainless steel. The surface is wear-resistant and machined without directionality to a finish of Ra = 10 to 20 microinches or 0.25 µm to 0.5 µm. This is, in fact, a better counterface than can normally be achieved on a shaft. A more advanced version, the Speedi-Sleeve Gold is an enhanced version of the popular Speedi-Sleeve which offers improved resistance to abrasive wear. Designed to be used in applications where extended seal system life is needed, Speedi-Sleeve Gold bridges the performance gap between the standard sleeve and expensive custom shaft treatments. A thin metallic film applied to the base stainless steel imparts a gold color and significantly increases durability and surface hardness (to approx. 2300 Vickers). For the front end of the crankshaft, National makes a Redi Sleeve (National Part # 99156). Although installation is simple, it should be done carefully in order to achieve the best results. Before starting, the sealing zone on the shaft should be carefully cleaned and any burrs or rough spots should be filed down and polished. Deep wear grooves, scratches or very rough surfaces should be treated with a suitable powdered metal epoxy filler such as Loctite Quickmetal 66010. The sleeve must be positioned on the shaft before the filler has hardened. It should also be noted that although Speedi-Sleeves can be easily installed within minutes on most shafts, they should not be placed over splines or keyways, etc. on the shaft. As the thin-walled sleeve has an interference fit, any disturbances on the shaft surface may create a similar pattern on the surface of the sleeve, and then the oil seal will leak. SKF has a website that can be found at http://www2.vsm.skf.com/usa_english/node920.aspx.com/ and National can be found at the website of its parent corporation, Federal Mogul, at http://www.federal-mogul.com/.

Installation of the cork seals for the front and rear crankshaft main bearing caps is often problematic for some owners. This is often due to the fact that although only two are required, rebuild kits are frequently supplied with several different cork seals of differing cross-sections and lengths. This is due to the fact that prior to the introduction of the 18V engines, a square cross-section cork seal was installed onto the rear crankshaft main bearing plate, while a rectangular cross-section cork seal was installed onto the front crankshaft main bearing plate. When the 18V engines were introduced, they were standardized on two square cross-section cork seals. One of the reasons that cork was chosen for use as a seal was its compressibility. Because this quality can vary, the seals are supplied in overlengths so that the mechanic can fine-tune their lengths by trimming.

Getting the cork seals for the front and rear crankshaft main bearing caps to the right length requires some patience. The groove of the flange into which they fit must be both clean and unpainted, otherwise the gaskets will eventually seep oil. Insert the ends of the cork seal into the groove of the flange so that its middle bows upwards, then gently press its ends outwards towards the ends of the grooves. If the cork seal still bows upwards in the middle when light pressure is exerted upon it, do not attempt to force it down. Instead, remove it and use either an Exacto knife or a single edge razor blade to cut 1/32" (.8mm) of material from one end. This should be done with care being taken to assure that the cut is perpendicular to the longitudinal axis of the seal; otherwise, there will be a likelihood of leakage from the poorly cut end. Repeat this process until the cork seal just compresses into its slot. The finished cork seals must extend slightly above the flange so that they will be compressed when the oil sump is bolted down. When this sizing operation has been accomplished, remove the cork seal and then coat it with a thin skim coat of Permatex Ultra Black RTV Gasket Maker. This is possibly the best silicone sealant on the market, able to withstand temperatures of up to 500° Fahrenheit (260° Celsius). Be sure to let the gasket cure for several hours before attempting to trim away any excess material using either an Exacto knife or a single edge razor blade or before installing it. This will result in a rubberized gasket that will not become saturated with oil and consequently ooze drippings onto your garage floor. Finally, use some Permatex Aviation Form-A-Gasket or Hi-Tack in order to secure it into position. Do not use silicone-based Permatex Blue RTV or Permatex Red Ultra RTV sealant on any of the engine gaskets as they are both prone to failure and contamination of the engine oil under hot operating conditions. In addition, they can also substantially contribute to swelling of the seals. Excessive swelling of a shaft seal's elastomeric lip is a good indicator that the lip material and the lubricant in use are not compatible. Swelling of the seal material can be particularly problematic if some materials, such as silicone, come into contact with oil at high temperatures. Under such conditions, softening, swelling, and reversion of the seal material can occur. If material swell becomes an issue, check to be sure that the elastomer in use is compatible with the lubricant and any other fluids coming into contact with the seal, either during cleaning, installation, or operation. This includes any solvents that may be used during teardown. You should also check to be sure that system fluids are not being contaminated in some way; contamination could cause an otherwise acceptable seal material to swell or degrade.

The front and rear crankshaft main bearing plates are a precision fit in order to enhance oil retention. Their exterior faces should be painted prior to installation, but no paint should be on their mating surfaces or inside of their seal grooves. Under no circumstances should any of their mating surfaces be cleaned with an abrasive, otherwise leakage will result.

When installing the main bearing plates into the engine block, do not attempt to expand the engine block by heating it with a blowtorch, as this uneven heating will place stress on the engine block that can lead to cracking. Do not yield to the temptation to simply place a block of wood against a main bearing plate and use a heavy hammer in order to force it into position, as this can result in damage to its precision-machined mating surfaces. I prefer to shrink-fit both the front and rear crankshaft main bearing plates in order to ease the assembly process. This will make for ease of proper alignment without any of the aforementioned risks. Spray the main bearing plates with WD-40, seal them inside of a Ziploc bag in order to prevent ice from forming on them, and then chill them thoroughly in a freezer overnight with the thermostat set as low as it will go. When they are well-chilled, fit them into place. Simply place the crankshaft main bearing plate into its respective slot at the end of the engine block, and then, using a pair of long ½”-20 UNF machine bolts in order to align the crankshaft main bearing plates, turn each retaining bolt about one-half turn at a time until the crankshaft main bearing plates are almost seated. This will produce a rate of descent of .025” (.64mm) per half-turn of the machine bolt, which will be within the distortion tolerance of the cast iron engine block. As they descend into place, change to shorter machine bolts. When they are almost fully seated, use the factory machine bolts, again turning them a half-turn at a time, in order to fully seat them. An alternative method consists of using two main cap studs from a studded B Series engine block in order to guide the crankshaft main bearing plate into place, use a soft mallet in order to gently tap it downward into place. As it reaches the bottom of its slot, use a soft hammer or mallet in order to gently tap it into its correct position with its outer face flush with the engine block.

The secret to measuring crankshaft endplay (endfloat) lies in remembering that steel is actually more elastic than rubber. If you drop a rubber ball and a steel ball bearing of identical diameters from an identical height, then the steel ball bearing will bounce higher every time! Consequently, it is always best to push the crankshaft as far as it will go rather than to tap it and risk it bouncing backward or forward. Remember, we are dealing in thousandths of an inch here! If you push (or lever) it as far as it will go, your measurements will be more accurate. These measurements should also be done dry, i.e., without oil, and with the center crankshaft main bearing cap loose on its machine bolts / studs, so that the crankshaft lines up the two sets of thrust washers. Only then should you tighten down the center crankshaft main bearing cap and check endplay (endfloat) with a dial gauge on the nose of the crankshaft. While too little play is worse than too much, it should be noted that whenever endthrust is applied to the crankshaft (such as when applying the clutch), the lateral acceleration of the crankshaft increases with its lateral movement. Thus, the greater the endplay (endfloat), the greater the impact loading against both the thrust washers and bearings and, as a consequence, the greater and more rapid the wear. The endplay (endfloat) should always be the same (.004" to .005" / .1016mm to .127mm) whenever the crankshaft is moved as far as it will go (this is what machinists call "repeatability").

As an alternative for sealing the rear end of the crankshaft, an uprated single-lip crankshaft rear oil seal that was used in the version of the B Series engine found in the Sherpa van will do an excellent job of keeping oil inside of the engine over the long term (Rover Part # LUF 10002). Contrary to rumor, this crankshaft rear oil seal is not a double-lipped design. A double-lipped seal is essentially a single lipped inner oil seal shrouded from dirt and dust by an outer second lip. This usually extends the life of the oil seal, and any extra sealing action is purely a bonus. The original crankshaft rear oil seal worked just fine before the advent of the sealed crankcase, but the vacuum present in the sealed crankcase of the later engines tends to draw stuff in, making the dust exclusion feature of a double lipped oil seal a plus. Most double lipped oil seals have the outer lip reversed in order to prevent ingress of dirt or air that can do damage to the seal and thus interfere with the maintenance of the vacuum inside of the crankcase. On the other hand, a true double-lipped oil seal would have both lips in the same direction, for good sealing against fluid escape. The seal dimensions are: Outside Diameter (O.D.) 4.125" (104.775mm), Inside Diameter (I.D.) 3.500" (88.9mm), Width .375" (9.525mm). Being made of Viton, they are not prone to failure until thermal conditions rise above 450° Fahrenheit (232.2° Celsius). This crankshaft rear oil seal can be obtained from Brit Tek (Brit Tek Part # AHU2242). When you examine the new crankshaft rear oil seal, notice that one side of the seal has a sharp edged lip while the other side does not. The side with the sharp lip also has a spring around the circumference of the rubber. The spring holds the rubber in contact with the shaft and the sharp edged lip runs on the shaft. The sharp lip is what actually seals the shaft. On the spring side you can see that oil under pressure would tend to assist the spring in keeping the rubber in contact with the shaft. Likewise, if installed backwards, oil pressure on the side opposite the spring would tend to lift the rubber and would oppose the spring, defeating its purpose.

Simply use a screwdriver to pry out the old crankshaft rear oil seal. Installation of the new crankshaft rear oil seal is quite straightforward. Make sure that the engine backplate is held in place with the machine bolts that are only finger-tight so that it can be moved by gently tapping it with a soft hammer. This may be necessary in order to properly center the new crankshaft rear oil seal. Clean up the rear of the crankshaft, as well as the hole in the engine backplate. Slide the plastic adapter that came with the crankshaft rear oil seal over the end of the crankshaft, with its big end first. Be aware that when installing an oil seal, many builders simply put a thin coat of oil onto the shaft and then slide the seal into place. This is an adequate method if a mechanism is to be put into service shortly, but if it is to await for several days, or even weeks, capillary action and gravity can interact to cause the oil can drain away, leaving the sealing lip of the oil seal to run dry against a shaft that is unlubricated until circulating oil reaches it. A better method is to put a thin smear of red rubber grease onto both the sealing lip of the seal and its seating area on the shaft. Next, apply a thin coat of oil onto the rest of the shaft. Oil the outside of both the adapter, and then slide the oil seal over the adapter until it meets up with the engine backplate. Grease the outer rim of the new crankshaft rear oil seal with red rubber grease. Do not tap the new crankshaft rear oil seal directly with a hammer when installing it onto the engine backplate as this can cause it to distort. Gently tap the adapter all around its circumference with a small hammer in order to drive the oil seal into place. Gently drive the new crankshaft rear oil seal in until it is flush with the engine backplate, and then pull off the adapter. The red rubber grease will stay in place indefinitely on both the shaft and the lips of the seal. A smear of oil on the outside surface of the oil seal will, depending on the material of which the oil seal is made, help to swell the seal shut against the shaft. Prior to pressing the engine backplate onto the locating dowels and before torquing the 3/8”-24 UNF machine bolts of the engine backplate, give the engine backplate a gentle rotational wriggle in order to guarantee that the lip(s) of the crankshaft rear oil seal are square against the journal of the crankshaft, thus ensuring a leak-free fit. This will also help to settle the new crankshaft rear oil seal and thus make certain that it is absolutely centered onto the crankshaft and is not pre-loaded by means of the engine backplate being slightly offset. Install the crescent-shaped locking plate (BMC Part # 1H 1021) with its four 5/16”-24 UNF machine bolts, and then bend the tabs upward in order to retain the bolt heads in position.

Flywheels and Ring Gears

The flywheels used on the B Series engine are pretty stout items, being highly resistant to warpage. However, the more material that you remove, the less heat-absorbing mass will be available, and thus the greater the tendency of the flywheel to warpage. The crankshaft’s period of harmonic vibration will also move higher up the powerband. Because the flywheel will synchronize its rotational speed with that of the clutch more rapidly due to its reduced inertia, stress on the flywheel mounting bolts will increase. As a result of this, stronger flywheel bolts from ARP are a wise precaution against breakage (Advanced Performance Technology Part # FBB716-6). It should be noted that the bolts holding the MGB flywheel to the crankshaft have significantly thinner heads than standard ¾” bolts. Standard sockets have a tapered lip on the inner edge to help guide the socket over the bolt head. Unfortunately, this presents a significant problem in that the taper on the mouth of the socket does not allow a full engagement on the head of the flywheel bolts, which in turn often results in damage to the flywheel bolt heads when attempting to remove or install them. The solution is to remove this tapered area on the mouth of the socket and allow the socket to completely engage the bolt head. It should be noted that the flywheel bolts must be installed into the flange at the rear of the crankshaft prior to placing the crankshaft in place in the engine block. Once the rear main bearing plate is in place, installation of the bolts is all but impossible.

Aluminum alloy flywheels are race-only items. A lighter flywheel requires less energy to increase its rotational speed, which is why they are popular with racers. While lightening of the flywheel in an attempt to reduce the engine’s resistance to acceleration can be of minor benefit, it should be borne in mind that such an approach is not without its own drawbacks. The purpose of a flywheel is to store energy through its own inertia. While it resists acceleration, it also resists deceleration, its kinetic energy thus smoothing the rotation of the crankshaft as each cylinder goes through its four phases. When seeking to meet North American air pollution standards, the engineers at the factory chose to initiate combustion earlier during the compression stroke by using more ignition advance. The consequent pressure increase as the piston approached the top of its compression stroke resulted in a more rapid deceleration of the crankshaft, increasing both vibration and stall tendency. Their solution to these undesirable effects was to increase the kinetic effects of the rotational weight of the flywheel by increasing its diameter from 10¾” to 11½“. This, in turn, necessitated the outwards relocation of the electric starter motor.

Although lightening the flywheel to a minimum weight of 16 lbs will cause the engine to pick up and lose RPM faster with the clutch disengaged and thus enable faster shifting, this will be achieved at the price of increased Secondary Vibration and a tendency for the engine to stall due to decreased flywheel inertia. Deliberate attention to throttle control and engine speed will also be required in order to accomplish smooth shifts unless the shifts are performed faster, which in turn will result in faster wear of the synchronizer hubs of the transmission.

The flywheel will definitely require dynamic rebalancing of a higher order of precision than normal because the lighter it becomes, the greater the effect of a small out-of-balance factor will be. Should you choose to have this done, advise the machinist that the material to be removed should be taken only from the front and back faces and not from the clutch friction surface. Although the hydraulic clutch mechanism is self-adjusting up to the limits of the original design specification, if too much material is machined away from the clutch friction surface of the flywheel, then the diaphragm springs will not be able to provide sufficient pressure. An original thickness lip should remain at the circumference of the flywheel in order to provide a stable mounting surface for the ring gear. No section of the flywheel should have a thickness of less than 7/16” (.4375” / 11.1125mm), and a 3/8” (.375” / 9.525mm) radius should be used on all corners. After machining, the entire flywheel should be coated with WD-40 in order to prevent rust. When you are ready to install the clutch assembly onto the flywheel, clean the WD-40 off of the friction surface with alcohol. All of the flywheels were stamped with a “1/4” mark in order to ensure that the flywheel would be installed with the mark in a vertical position when cylinders #1 and #4 were at Top Dead Center so that the engine would be dynamically balanced correctly. It is often obscured by light rust. Clean the flywheel using some steel wool or a ScotchBrite pad and you should find the markings. On the 10 ¾” flywheel of the 18G, 18GA, and 18GB engines the mark is on the outer edge of the flywheel on the engine side next to the ring gear, while on the 11½” flywheels of the 18GD, 18GF, 18GG, 18GH, 18GJ, 18GK, and 18V engines it is on the upper side edge of the rim that faces the clutch.

While you have the flywheel off, inspect the teeth of its ring gear carefully. Do not be concerned if you notice that two sections of the gear teeth, 180º apart, exhibit more wear than the rest of the teeth. This is due to the fact that the engine tends to stop as either #1 or #4 cylinder attains compression, or as #2 or #3 cylinder attains compression. Ring gears for 18GD and later engines (BMC Part # 12H 2900, Moss Motors Part # 190-050) should exhibit a notch machined into their teeth for the starter gear. These notches are preferable for longer starter gear life and less chance of the starter jamming in place and burning up. However, if the teeth are worn to the point that this groove is absent, or show a deep semicircular groove on the gear teeth of the ring gear (BMC Part # 1G 287, Moss Motors Part # 190-040) for the earlier 10.75" flywheel for the 18G, 18GA or 18GB engines, then replacement of the ring gear is a fairly easy task. Cut the ring gear almost all the way through with a cutting wheel on a Dremel tool, or with a hacksaw, then crack the last .030" or so with a chisel. By using this technique, you will not nick the flywheel. Coat the flywheel with WD-40 in order to prevent ice from forming on it, seal it in plastic, the place it inside of a refrigerator or a deep freeze with its thermostat set as low as it will go in order to contract its diameter so that the ring gear will be easier to put in place. Heat the ring gear in an oven in order to expand its diameter so that it will be easier to put in place. Note that the ring gear should not be heated above 482º Fahrenheit (250º Celsius) or its temper will be effected. Use heavy oven mitts, welding gloves or pliers to carry the hot ring gear to the flywheel. Be aware that older cars had no step on the ring gear but some of the new replacements do. The step faces the rear for inertia starter motors. The heated ring gear should drop freely onto the wheel and should be spun immediately in order to ensure that it seats upon the flange. At that point, you need only to allow the assembly to cool to room temperature, and then you are done!

Be advised that contrary to popular belief, the flywheels and ring gears used in the MGB are not completely interchangeable. The 10.75" (273.050mm) flywheel (BMC Part # 12H 713) and its ring gear (BMC Part # 1G 2874) of the three-main-bearing 18G and 18GA engines are not interchangeable with those of the later five-main-bearing engines, nor is its lockplate (BMC Part # FNX 506, Moss Motors Part # 460-710), nor are the mounting bolts (BMC Part # 51K 1022, Moss Motors Part #322-850). This flywheel had three dowels. Both the 10.75" (273mm) flywheel (BMC Part # 12H 1474) and ring gear (BMC Part # 1G 2874) of the 18GB engine were unique to it alone, although its lockplate (BMC Part # 12H 1303, Moss Motors Part # 460-715) and its six mounting bolts (BMC Part # 51K 1809, Moss Motors Part # 322-160) are common to the later, larger-diameter 11.5" flywheels. This flywheel also had three dowels. The engine backplate is also different due to the positioning of the Lucas M418G inertia-type electric starter assembly (BMC Part # 13H 4561, Moss Motors Part # 140-165) is closer to the crankshaft in order to accommodate the smaller-diameter flywheel. It can be readily identified by its 3.95" diameter hole for the rear seal of the crankshaft. All 11.5" (292mm) flywheels (BMC Part # 12H 2184) of the 18GD, 18GF, 18GG, 18GG, 18GH, 18GJ, 18GK and 18V engines and their ring gears (BMC Part # 12H 2900, Moss Motors Part # 190-050), along with their Lucas 2M100 pre-engaged-type starter assembly (BMC Part # 13H 6130, Moss Motors Part # 131-220), are all interchangeable, although not with those of the earlier engines. These engines had two dowels for locating the engine backplate. Interestingly, all of the engines used the same dowel (BMC Part # 1G 2984, Moss Motors Part # 325-045).

Be aware that failures due to impact shear occur in bolts loaded in single shear, such as flywheel and ring gear bolts. This usually occurs when the bolts are called upon to locate the device as well as to clamp it and, almost always, the bolts were insufficiently preloaded on installation. Fasteners are designed to clamp parts together, not to locate them. Location is the function of dowels.

The engine backplate for the four-synchro transmission with its lower mounting for the Lucas 2M100 pre-engaged type electric starter can be readily identified by its 4.12" diameter hole for the rear seal of the crankshaft. Although the 10.75" (273.050mm) flywheels of the 18G, 18GA, and 18GB engines may seem more desirable to some due to their lower rotational weights, their matching ring gears are not interchangeable with that of the later 11.5" (292.1mm) flywheel of the later engines, thus forcing the use of both the matching engine backplate, Lucas M418G inertia-type electric starter, and three-synchro transmission along with its appropriate-length driveshaft (propeller shaft).

Balancing

There are two factors involved in balancing an engine, respectively referred to as Primary and Secondary Factors. The term Primary Factor refers to vibration that is induced by reciprocating components (Primary Vibration), such as piston assemblies and connecting rods. The term Secondary Factor refers to vibration that is induced by rotating components, such as the crankshaft and the flywheel (Secondary Vibration). An in-line four cylinder engine will produce Primary Vibration, as will a horizontally-opposed four cylinder engine. This is why some designers of modern in-line four cylinder engines include a counter-rotating balance shaft in order to produce an equal but opposite vibration in order to cancel out the Primary Vibration produced by the engine. Obviously, careful attention to the balancing of the engine is warranted.

The effective length of the connecting rods (eye center-to-eye center distance) should be matched in order to establish a uniform connecting rod to stroke ratio. If possible, have the connecting rods balanced end-for-end. This means that the Small Ends of the connecting rods should all be matched for weight and their Big Ends should also all be matched for weight. The total weight of each connecting rod should then be matched to each of the others. In doing so, the equalized oscillating dynamic forces produced by their reciprocal movement inside of cylinders #1 and #4 will be equal to those produced inside of cylinders #2 and #3, effectively canceling each other out and thus reducing Primary Vibration. The Primary Vibration cannot be completely eliminated due to the fact that the throws of the crankshaft are rotating on separate, parallel planes. In addition, because of the equalized weights, the opposing centrifugal forces created by the connecting rods upon the opposing throws of the crankshaft will be equalized, thus reducing Secondary Vibration. This will aid in maintaining concentricity of the axis of the crankshaft with that of the bearing bores, not only reducing both bearing and journal wear, but also reducing pressurized stress on their lubricant. Primary Vibration includes all of the forces that occur at the same frequency as the rotation of the crankshaft. This includes cycles of acceleration and deceleration of the rotational speed of the crankshaft. Lighter pistons and connecting rods require less energy from the rotating assembly in order to change direction. This being the case, they decrease the amplitude of this primary frequency, and thus reduce Primary Vibration.

Although the prospect of balancing your own connecting rods may seem to be an intimidating task, it is actually a rather simple and straightforward operation to perform. If you work patiently, you can match the respective weights to within .1 gram, which is as good as or better than any professional shop can accomplish. You will need a scientific scale with excellent repeatability. The term "repeatability" means that every time that the same object is placed upon it; the scale should always give the same weight readout. Actual calibrated accuracy is unnecessary. Since we are dealing only with relative differences in weight, it is only necessary that the scale have excellent repeatability.

The small end of each assembled connecting rod is placed upon an oiled metal ball that acts as a fulcrum point and its big end placed on the middle of a scientific scale. Once the lightest big end of the connecting rods is identified, the adjacent balance pads on the heavier ones are lightly ground on a fine stone until the weight of their big ends are equal to that of the lightest one of the set. Should the connecting rods be of the type that has no balance pads, the metal must be carefully removed from the entire length of the curvatures on the opposing ends. All grinding must be perpendicular to the pivot axis of the connecting rod, never parallel to it. During the grinding process, care must be taken to not overheat the metal in order to avoid annealing the metal. Should the metal turn blue, it has become annealed, and thus has been unfortunately rendered useless. Scrap it.

Next, the big end of each assembled connecting rod is placed on the ball and its small end placed on the middle of the scientific scale. Once the lightest small end of the connecting rods is identified, the adjacent balance pads of the heavier ones are also lightly ground on a fine stone until the weight of their small ends are equal to that of the lightest one of the set.

Once the grinding process is complete, all grinding marks should be removed by polishing in order to prevent the formation of stress points in the surface of the material that can develop into cracks. This operation can be easily performed with a Dremel tool fitted with a polishing bit. After polishing, the ends of the connecting rods should be reweighed and any resulting small discrepancies in weight resolved by polishing. Once the balancing process is finished, all of the components should be disassembled and thoroughly cleaned.

If you desire lighter connecting rods in order to further reduce the amplitude of Primary Vibration and its attendant power loss, the third version of the horizontally-split Original Equipment connecting rods can be readily modified in order to accept the necessary small end bushings and will fit this requirement at minimal cost. These are to be found in the late 18V engines and have no balance pads, thus minimizing their total weight. They were installed into all of the production engines in matched sets that were reasonably well-matched for weight at the factory, so only minimal modification is necessary in order to bring them into precise end-to-end balance.

Pistons that use only three rings are lighter than the older-design four-ring and obsolete five-ring designs. However, the four-ring and five-ring pistons have longer skirts that provided a greater load-bearing area, thus reducing the pressure of the piston against the cylinder wall, which, in turn, reduces wear. As in all engineering considerations, the choice between the two is a matter of priorities. Fortunately, the engine lubricating oils of today do such a good job of preventing wear that the more modern three-ring piston designs wear at an acceptable rate. Both the piston / ring / wrist (gudgeon) pin assemblies as well as the connecting rod assemblies should be matched respectively to within .10 of a gram.

Matching the weight of the piston assemblies is a more involved and time-consuming process than that of balancing the connecting rods. Each bare piston, wrist (gudgeon) pin, and piston ring must be weighed and their individual weights recorded. The components are then assigned to a piston according to their weight, i.e., the lightest components are assigned to the heaviest piston, and the heaviest components are assigned to the lightest piston. The total weight of each assembly is then recorded. This approach allows the variation of the total weights of the assemblies to be minimized, simplifying the weight removal process by minimizing the amount of material to be removed.

Material is then lightly removed from around the wrist (gudgeon) pin bosses inside of the pistons as well as from the inside of the skirts of the bare pistons until the total weight of the heavier piston assemblies are equal to that of the lightest one. In order to avoid weakening the pistons, material removal should be distributed evenly across as wide an area as possible. This operation is best performed with a Dremel tool fitted with a polishing bit so that the finish will be as smooth as possible, thus precluding cracks from forming. After polishing, the pistons should be reweighed and any discrepancies in weight resolved by further polishing. Once the matching process is finished, all of the pistons should be thoroughly cleaned and immediately assembled in order to prevent mismatching of the components.

The reciprocating masses having thus been matched, the crankshaft, pulley wheel on the harmonic balancer (harmonic damper), and the flywheel should then be dynamically balanced separately and subsequently checked for balance as a complete assembly. The flywheel should be balanced only after its clutch friction surface has been skimmed smooth and its balance factor determined with both the ring gear and the clutch cover attached. A balance factor near 1.0 indicates an engine whose Primary Vibration is in line with piston motion (the engine vibrates up-down), while a value near 0.5 results in an engine that vibrates more in a circle (Secondary Vibration).

If you are using the flared counterweight cast iron crankshaft of the later versions of the 18V engine, balancing will need to accomplished either by drilling or by both grinding and polishing in order to remove material. However, if you are using the flat-sided steel crankshaft and you can afford it, advise the machinist that you would prefer that the dynamic balance of the crankshaft be achieved by wedging rather than by drilling. While the involved machinework is much more expensive than drilling, wedging will reduce both stress risers and oil drag, as well as greatly assist in prolonging main journal bearing life. In addition, the two to three pound reduction in rotational mass produced by wedging will produce some of the same positive result of rapid changes of engine speed as that of which reducing the weight of the flywheel can achieve, although without incurring the attendant liability of increased secondary vibration. The cause of the attendant vibration that results from lightening of the flywheel is often misunderstood, being the result of the reduction of the kinetic energy of the flywheel with its consequent oscillating acceleration / deceleration that results from reversing the travel of the reciprocating masses inside of the engine, which results in a consequentially increased amplitude of rapidly oscillating torque effect. By achieving a reduction of rotational mass in this manner, the flywheel can retain the same mass, thus retaining its ability to absorb heat without warping, as well as providing sufficient inertia to smooth the power impulses of the engine. It is not only unnecessary to knife-edge the corners of the crankshaft counterweights of a street engine in an attempt to reduce windage loss and atmospheric turbulence within the crankcase, it is actually undesirable. Be aware that knife-edging of the counterweight of the crankshaft can result in stress points that can lead to fracturing. Instead, a generous radius on each corner of the crankshaft will reduce both oil drag and aerodynamic drag to approximately 90% of that attained by knife-edging. These procedures are fundamental to producing the smoothest running engine possible and will provide a bit more power that would otherwise be lost to the production of vibration, in some engines amounting to perhaps as much as two horsepower. It will also result in reduced main bearing wear due to more concentric running. Due to their relatively superior machining characteristics, the alloy of the flat-sided five-main-bearing nodular cast iron version of the crankshaft found in the early 18V engines (18V-584-Z-L, 18V-585-Z-L, 18V-672-Z-L, and 18V-673-Z-L) is the easiest for a machinist to work with.

Connecting Rods

The obliquely-split connecting rods (BMC Part # 12H 998) for cylinders #1 and #3, (BMC Part # 12H 997) for cylinders #2 and #4 first used in the three main bearing 18G and 18GA engines used a smaller-diameter.750” (19.05mm) wrist (gudgeon) pin secured by a pinch-bolt small end, thus making them noninterchangable into later engines without using their wrist (gudgeon) pins as well as their accompanying obsolete four-ring pistons. Both they and the obliquely-split connecting rod (BMC Part # 12H 1019) of the five-main bearing engine (18GB, 18GD, 18GF, 18GG, and through early 18GH engines) weighed in at a ponderous 980 grams. Not only are they heavy, they are notoriously weak for use in highly stressed engines. However, this disadvantage can be partially overcome by using ARP connecting rod bolts (11/32" for 18G & 18GA engines, ARP Part # 206-6002; 3/8" for 18GB through 18H engines, ARP Part # 206-6003). Either should be tightened to a stretch of .0067".

The horizontally-split connecting rods with balance pads used in the late 18GH, 18GJ, 18GK, and through early 18V engines were a notably lighter 845 grams. There were two variants of this connecting rod. The first variant (BMC Part # 12H 2444) was introduced in June of 1970 and is found in the 18GH, 18J, and 18K engines. It has a bushed small end for use with a floating wrist (gudgeon) pin that is secured within the piston by circlips. It can be quickly identified by its large balance weights on both ends. The later variant that was introduced in March of 1971 is found in both the late 18GK and the early 18V engines. It has an unbushed small end designed for use with a press-fitted wrist (gudgeon) pin (BMC Part # 12H 3596). It can be quickly identified by its small balance weights on both ends. The final connecting rod design (BMC Part # CAM 1588) used in the late 18V engines (18V 801, 18V 802, 18V 846, 18V 847, 18V 883, 18V 884, 18V 890, 18V 891, 18V 892, 18V 893) had no balance pads and were the lightest, weighing 760 grams, which is slightly in excess of 1/3 more than the Arrow Precision connecting rod. The most desirable connecting rods for any normally-aspirated street engine are also the lightest ones as, due to their reduced inertia, their reciprocation will not only produce less Primary Vibration and power loss, but less stress is placed upon their big end bearings as well. These connecting rods can commonly be found on engines whose identification numbers start with 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L, 18V-891-AE-L, 18V-892-AE-L, or 18V-893-AE-L.

Be aware that the connecting rods used on the 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and through early 18GK engines use the larger 13/16" (.8125" / 20.6375mm) wrist (gudgeon) pins that floated inside of a press-fitted bushing in the small end of the connecting rod. This bushing was later eliminated in the connecting rods of the late 18GK through 18V engines. These later connecting rods also used the larger 13/16" (.08125" / 22.6375mm) diameter wrist (gudgeon) pins, but in their case, they were press-fitted into the connecting rods, so your pistons must be chosen accordingly. However, the small end of these later connecting rods can be machined in order to permit them to accept the earlier bushing if floating wrist (gudgeon) pins are desired so that a floating wrist (gudgeon) pin can be used. Due to the fewer parts involved with a press-fitted wrist (gudgeon) pin / piston assembly, more precise control is attainable than with a floating assembly. However, wear of the wrist (gudgeon) pin bores of the piston is greatly accelerated. Actual stresses, depending on cylinder gas pressures and engine speeds, are influenced by wrist (gudgeon) pin ovalization and bending, as well as by the hydrodynamic oil film pressure distribution inside of the pin bore. A floating wrist (gudgeon) pin configuration allows the wrist (gudgeon) pin to rotate during engine operation. As compared to a fixed wrist (gudgeon) pin, this prevents the floating wrist (gudgeon) pin from repeatedly flexing into a fatigue-inducing bow shape. Reduced wrist (gudgeon) pin clearances always have a positive effect. Wrist (gudgeon) pin to pin bore installation clearances can be reduced approximately 50 percent and configurations can thus be designed for higher specific pin bore surface pressures at peak cylinder gas pressure. Once the lower limits are established (example: scuff issues), upper installation clearances can be minimized. In the cases of both types, an assembly lubricant such as CRC Sta-Lub Extreme Pressure Lubricant must be applied in order to prevent galling during initial fire-up of the engine.

Shot peening and electropolishing of the connecting rods are necessary only if you are going racing. However, smoothing of all of the edges on a connecting rod can greatly aid in decreasing the possibility of pressure risers forming fissures that will develop into fractures. Note that exotic lightweight connecting rods such as those marketed by Carrillo (588 grams) and Arrow Precision (570 grams) (Arrow Precision Part # 146S) are primarily intended for racing use and are unnecessary for use in all but the most radical of street engines, although their lower reciprocating mass will reduce stress loading on other parts of the engine including the crankshaft and bearing assemblies, as well as reduce both horsepower loss and the amplitude of vibration caused by primary imbalance, although not its frequency. They also both make use of cap bolts in order to secure their crankshaft main bearing caps instead of through-bolts or studs, thus endowing them with the rigidity that is necessary for reliability at unusually high sustained engine speeds. It should be noted that both of these designs use floating wrist (gudgeon) pins.

Perhaps the connecting rods made by Arrow Precision deserve special mention at this point. These are made from vacuum de-gassed nickel chrome alloy steel (Double-Remelt) that has a very low sulphur content of .025%, which in turn means fewer inclusions and faults. Simply put, Double Remelt is a method of producing steels that are very low in non-metallic inclusions. Solid inclusions such as silicates and gaseous inclusions such as nitrogen, oxygen, and hydrogen form microscopic stress-raisers within the steel. Trace oxygen and nitrogen also form oxides and nitrides. For highly stressed components such as lightweight connecting rods, these inclusions need to be reduced to a minimum level, lower than that which can be achieved in the normal steelmaking process. These high-quality steels are therefore produced inside of an arc furnace in which they are degassed by way of bubbling argon through the molten alloy. They are subsequently cast into electrodes. These electrodes are then remelted under a vacuum, further reducing impurities. The production of Double Remelt steel is thus simply repeating this process in order to create an even higher level of quality. As these processes are obviously very expensive ways of producing steel, they are only used for critical applications where maximum strength is needed. When forged, this means that imperfections are reduced to an absolute minimum, ensuring excellent consistency. In the case of connecting rods, this permits a high level of strength to be achieved while using a minimum of material, thus reducing weight. This steel is referred to as EN24 (817M40 British Standard). EN24, as used by Arrow Precision, is comparable to 4340 AISI / SAE steel, both of which are available as both air melt and as vacuum remelted steel. This being the case, checks should be made when comparing with another manufacturer's product as to just which type of 4340 is being used. 4340 is higher in hardenability than any other standard AISI grade. Other names for 4340 include Werkstoff 1.6565 and Kurzname 40 CrNiMo6.

While many engine designers sought to reduce wear of the bore, rings, and piston by distributing side thrust forces over as large an area as possible, as in the case of engines having a large bore coupled with a short stroke, the attendant loss of thermal efficiency resulting from the consequentially large area of the roof of the combustion chamber was deemed to be unacceptably inefficient by the designers of the B Series engine. Instead, lubrication was seen to be the proper solution. In order to accomplish this, connecting rods with the uncommon feature of an oil squirt passage were devised. Contrary to what some amateur engine builders may believe, positioning the connecting rods so that the oil squirt passages face toward the camshaft is not necessary as the camshaft receives excellent lubrication from both the pressure galleries in which its journals spin plus residual oil flowing down the pushrod bores from the rocker arm assembles, as well as oil sprayed from the crankshaft's main bearings and the connecting rod big end bearings on the crankshaft. When installed, the oil squirt passage of all of the connecting rods must face toward the side of the engine opposite to the camshaft in order to both cool the piston and better lubricate the load bearing surface of the cylinder wall during the power stroke, as well as to throw oil over the wrist (gudgeon) pin bosses of the piston where there are holes drilled in order to allow the oil to reach the wrist (gudgeon) pin. Failure to install the connecting rod in its proper orientation will eventually result in extreme wrist (gudgeon) pin wear within the piston itself, plus create the very real likelihood of piston failure, not to mention consequent increased bore wear as well. This is due to the fact the oil squirt passage inside of the connecting rod aligns with two opposing drilled passages in the journal of the crankshaft. When installed with the oil squirt passage facing away from the camshaft, this alignment will occur halfway through the upward stroke and again halfway through the downward stroke. However, should the connecting rod be installed with the oil squirt passage facing toward camshaft, the stream of oil will not reach the thrust side of the cylinder on the upward stroke and will not even get inside of the bore on the downward stroke.

If a connecting rod bolt is installed without its proper preload (prestretch), then every revolution of the crankshaft will cause a separation between the body of the connecting rod and its cap. Under load, this insufficient preload will cause additional stretch in the connecting rod bolt. This stretch is removed when the load is removed on the upward stroke of each revolution. This cycle of stretching and relaxing can cause the connecting rod bolt to fatigue and fracture. In order for this cyclic stretching to be prevented from occurring, the preload of the connecting rod bolt must be greater than the cyclical load caused by the reciprocation of the connecting rod. A properly installed connecting rod bolt remains stretched by its preload and is not exercised by the by the cyclical loads that are imposed upon the connecting rod. A quality connecting rod bolt will stay stretched for years in this manner without fracturing. Many connecting rod bolt failures are caused by insufficient preload. When a fastener is insufficiently preloaded during installation, the dynamic load may exceed the clamping load resulting in cyclic tensile stress and eventual failure. Perhaps the best connecting rod bolts are available from ARP. Cycle testing shows their High Performance connecting rod bolts to be nearly five times more reliable than the Original Equipment connecting rod bolts.

When tightening, it is important to prevent the connecting rod bolt from failing as a result of the stress induced by tightening it to a load that is greater than that of the demand placed upon it by the engine. In other types of bolted joints, this careful attention to detail is unimportant. For example, flywheel bolts need only be torqued to their specified 40 Ft-lbs in order to prevent them from working loose. Flywheel loads are carried either by shear pins or by side loads upon the bolts; as such, they do not cause cyclical tension loads in the bolts. On the other hand, connecting rod bolts support the primary tension loads caused by engine operation and must be protected from cyclical stretching. That is why proper tightening of the connecting rod bolts is so important. These same conditions also apply to the fasteners of the crankshaft main bearing caps.

There are three methods that can be employed to determine how much tension is being exerted on a fastener: using a torque wrench, measuring the amount of stretch, and turning the fastener a predetermined amount (torque angle). Using a torque wrench involves a considerable number of variables. These include the calibrated accuracy of the torque wrench, the characteristics of the lubricant being used, the quality of the mating surfaces between the head of the connecting rod bolt and the connecting rod, and even the person using the torque wrench. Of these methods, the use of a stretch gauge is the most accurate. It is important to note that in order for a fastener to function properly it must be "stretched" a specific amount. The ability of the material to "rebound" like a spring is what actually provides the clamping force. Different materials react differently to these conditions, and engineers design bolts to operate within specific ranges. If a fastener is over-torqued and becomes stretched too much, then its yield strength has been exceeded and it is ruined. If the fastener is found to be longer than when it was originally manufactured, even if it is only .001" (.0254mm), it is in a partially failed condition and should be discarded. Therefore, fasteners are engineered with the ductility to stretch a given amount and then rebound to produce the proper clamping force. I highly recommend the use of a stretch gauge when installing connecting rod bolts wherever it is possible to measure the fastener as it is the most accurate way to determine correct stretch. If a stretch gauge is not available, a vernier caliper may be an acceptable substitute. The use of the torque wrench data alone should normally be used only for guide purposes.

The most critical time to measure stretch is during the process of final assembly. Unfortunately, that is also when the connecting rods are inside of the cylinders and being assembled onto the crankshaft. While it is a simple matter to measure stretch when the connecting rods are on the workbench, it is much more difficult process to perform when they are being assembled into the crankcase. There is, however, a solution to this difficulty. During the engine building process, you will doubtless assemble and disassemble the connecting rods several times. As you are doing this, you will be inadvertently polishing the threads of both the connecting rod bolts and their securing nuts, so always keep these in matched pairs. Always torque them all to the same number and record the stretch and torque numbers for each paired set every time that you assemble them. When you are ready for final assembly, you will then know that all of your fasteners will stretch at the same toque reading. You can them safely torque the rest of the fasteners to that number and be confident that there correct amount of stretch will be present.

Many current engine designs use bolts that are called "Torque To Yield". As the name implies, these bolts are designed to stretch a certain amount when tightened correctly. During the Heating / Cooling Cycle, these bolts often take a permanent set and are not capable of regaining the correct torque if re-used after disassembly, such as is the case of the bolts that join the two halves of the front brake calipers. Should you decide to use this type of bolt, it is recommended that they always be replaced with new ones every time that they are removed.

While the torque figure for the obliquely-split connecting rods is 35 to 40 Ft-lbs and is 33 Ft-lbs for the horizontally-split connecting rods, a torque wrench essentially measures the amount of torque necessary to overcome friction, not actual clamping force. Friction is an extremely challenging problem because it is so variable and difficult to control. The best way to avoid the pitfalls of measuring by friction is to use the stretch method. This way preload is controlled and independent of friction. Each time that the bolt is torqued and loosened, the friction factor gets smaller. Eventually the friction levels out and becomes constant for all following repetitions. Therefore, when installing a new bolt where the stretch method cannot be used, the bolt should be loosened and tightened several times before final torque. The number of times depends on the lubricant. For ARP recommended lubricants, five loosening and tightening cycles are sufficient. Regardless of the method used, be aware that optimum clamping loads can be achieved only by careful attention to reducing friction between the threads as well as on the clamping face of the connecting rod and the connecting rod bolt. I recommend using ARP bolt lubricant on both the threads and on the face of the nut.

Be aware that on the horizontally-split connecting rod bolts, only one side of the bolt head is beveled in order to provide sufficient clearance for the shank of the camshaft, so take notice of this fact when you reassemble them. The beveled edge of the bolt head must be aligned parallel to the crankshaft in order to assure adequate clearance of the shank of the camshaft. After you have assembled the connecting rods onto the crankshaft, do not innocently presume that your bolt heads will have sufficient clearance with the camshaft. At least one aftermarket manufacturer (County) is making their camshafts with .035" (.889mm) oversize diameter shanks, so be sure that you check the clearance between the heads of the connecting rod bolt and the shanks of the camshaft.

If your engine is of a very conservative specification in terms of engine speed, a set of the Original Equipment specification connecting rod bolts should prove to be of adequate strength. Prior to installation, use a powerful magnifying glass (5X or more) in order to carefully examine their threads. If you observe any signs of distress or malformation of the threads, do not use the bolt. In any case, do not reuse the original nuts. Instead, replace them with a set of twelve point ARP nuts.

Ensure that the mating faces are wiped clean and use solvent to remove any old oil from threads of the bolts and the mating surfaces of the connecting rod assembly. Apply ARP moly assembly lubricant to both the seating face of bolt and the threads of both the bolt and its clamping nut. Next, assemble the cap to the connecting rod and torque the bolt to 15 to

20 Ft-lbs, then tighten each bolt to recommended stretch value, i.e., loosen the first bolt, zero the stretch gauge, and tighten until required stretch is achieved. Finally, loosen the second bolt and repeat the process until both of the connecting rod bolts have the required amount of stretch. In order to optimize the accuracy of the big end bore size and roundness, as well as to achieve the correct bolt preload, each bolt should be stretch gauged. Remember: a torque wrench setting should be considered to be a guide only.

Pistons

Most owners have a simple idea of what a piston is. They think of it as that thing with the rings on it that combustion gases force down the cylinder in order to transmit power through the connecting rod. However, when building an enhanced performance engine, a more detailed knowledge of the subject can help in making an informed decision concerning what is appropriate to one's needs. Although pistons may seem to be simple to the naked eye, they are actually of quite sophisticated design. Piston skirts are designed so that they give optimum profiles at full operating temperature. Contrary to what many believe, they are not round. Instead, they are elliptical and their vertical profile is not straight and parallel. In areas where the piston runs hotter, expansion is greater and thus allowances for this have to be made in order to accommodate this in the skirt profile. This profile, both radial and vertical, is extremely critical and a diametrical adjustment of only five microns (0.0002" / .00508mm) can make all the difference in terms of achieving optimum results. The profile-turning lathes that produce this ovality and vertical profile on the piston skirt easily produce repeatability within +/- .0002" (+/-.00508mm).

The diecasting process of producing pistons requires the melting of a high silicon content alloy in an electric furnace with precisely controlled temperatures. The molten alloy is then poured into a multi-piece die, thus producing a very accurately shaped piston casting. The casting die is designed so that when the metal has solidified, the various pieces of the die can be removed separately. Because of this production method, undercuts and reliefs can be produced to minimize the weight of the piston. Unfortunately, while cast pistons are strong enough for most enhanced-performance engine applications, they have lower ductility than forged pistons and by the very nature of their cast construction are brittle and more prone to breakage under the stresses induced by heavy skirt loadings, as well as by the high cylinder pressures created by detonation.

The production process of forged pistons is more complex than that of the diecasting method. The forging process requires material to be bought in at closely controlled diameters, cut to billet size, and all cut faces are then machined to a smooth finish. The billet is preheated in an air-circulating furnace to a temperature quite close to the operating temperature attained by the piston crown when the engine is operating at full power. This temperature is critical and, together with the tightly controlled speed of the forging process, produces the dense and very fine horizontal grain structure which gives forgings their higher strength and greater fatigue life, enabling these pistons to better withstand the high cylinder pressures and skirt loadings that are imposed by high performance use. After forging, any excess material is removed and the forging is then heat-treated, followed by wet blast cleaning. This results in the forging having a smooth eggshell finish, while the casting has ribs and lines, some to assist the casting process, and others formed due to the casting tool being comprised of around nine different pieces. Forging eliminates porosity in the metal, thus improving ductility and allowing them to conduct heat away from the piston crown quickly so that piston crowns can run 75° Fahrenheit to 100° Fahrenheit (23.9° Celsius to 37.8° Celsius) cooler than comparable cast piston crowns, typically 450° Fahrenheit (232.2° Celsius) instead of 550° Fahrenheit (287.8° Celsius).

One of the most important advantages of forged pistons is what happens at the point of piston failure. Under extreme conditions, such as detonation, forgings tend to soften and fail gradually. You usually have time to replace them before the rest of the engine is ruined. Cast hypereutectic pistons, on the other hand, although relatively strong in terms of ultimate tensile strength, have less ductility and are prone to fracture when their limits are exceeded.

However, be aware that not all forged pistons are created equal. The aluminum alloy of a high quality cast hypereutectic piston is normally alloyed with silicone in order to endow it with the same expansion / contraction coefficient as that of the cast iron engine block in which it is intended to operate. The majority of forged pistons are usually made of 2618 alloy, which has a greater expansion / contraction coefficient than cast pistons due to their having a low (<2%) silicone content (silicone does not like being forged). A 2618 alloy piston expands 15% more while enduring severe operating temperatures than does 4032 alloy piston. As a consequence, they have to be fitted with greater cold running clearances that can accelerate wear somewhat during the engine's warm-up period. These greater cold running clearances lead to a condition known as "piston slap" under heavy loadings in which the piston rocks within the cylinder and causes an audible tapping noise that continues until the engine has warmed to operational temperature. Be warned that engines equipped with these pistons should not be revved when cold, or excessive scuffing can occur. Because of their greater ductility than that of 4032 alloy, they can get rid of the high levels of sustained heat incurred in racing. Pistons using 2618 alloy are preferable for employment in engines using superchargers, nitrous-oxide injection, or for pure race applications where frequent inspection and replacement are the norm. However, their lower silicone content alloy also makes for faster wear and makes them somewhat more vulnerable to scuffing.

The best forged pistons to use for a Big Bore B Series engine are the flat-topped JE pistons. Being made from high silicon content (13.5%) 4032 aerospace alloy, these are the only custom pistons sophisticated enough to have the same expansion / contraction coefficient as the Original Equipment Hepolite cast pistons, thus they can be fitted with the same clearances. This alloy expands from heat less than pistons with less silicon (such as 2618 alloy), but since its eutectic level of silicon is fully alloyed on a molecular level, it has the advantage of being less brittle and more flexible. As a result, these pistons can survive mild detonation with less damage than cast pistons. They come with faces that have been peened and polished in order to remove dangerous hotspots and thus prevent preignition and detonation. Each set is delivered matched for weight, complete with the needed state-of-the-art thin steel piston ring assemblies that better compensate for flexure of the Big Bore B Series engine's cylinder walls. Such is JE's attention to detail that their oil control rings employ an anti rotational locking detent that fits into an opening at the intersection of the ring groove and wrist pin hole. Due to the height of their piston crowns being .040" (1.016mm) greater than that of Original Equipment Hepolite cast pistons, when they are at Top Dead Center they are flush with the deck of the engine block and, as such, they will not require redecking the engine block to the point that there is a risk of weakening the engine block or of reducing the number of threads available in the mounting bosses for the cylinder head studs. In addition, their crowns are thick enough (.415" / 10.541mm) to allow the machining of the crown to a cold clearance height (.012"/ .3048mm) that is appropriate for producing ideal squish (quench) characteristics, as well as allowing the machining of an appropriate dish diameter in order to custom-tailor both the size of the squish (quench) area of the crown and the radial depth of the dish in order to produce the desired compression ratio, as well as to produce the desired contour of the dish to form the bottom of the combustion chamber to individual specifications in order to promote optimum flame propagation. It should be noted that the width of the squish (quench) area of the piston crown should be the same as the maximum distance that the mating surface of the cylinder head extends over the bore. JE pistons can be fabricated with skirt extensions that are beyond the normal skirt level of a Hepolite piston. The longer the skirt becomes, the less the piston will "tip" in the bore, and thus the better the stability and the sealing ability of the rings. The skirt of the JE piston actually projects below the bottom of the cylinders at

Bottom Dead Center (BDC). Obviously the skirt cannot be so long as to foul the crankshaft balance weights at Bottom Dead Center (BDC), so the bottoms of the skirt are machined in order to afford a sufficient running clearance of .060” (1.524mm). Their only drawback to the design is the extra weight created by the thick piston crown. While not everyone has access to such machining skills, there is a simple solution: JE offers the service of custom-machining their pistons to order, thus the piston can be made with an appropriate-width squish band as well as a dished piston crown which, when coupled with a professionally-reworked combustion chamber in the cylinder head, will accomplish the combustion chamber shape needed to promote efficient combustion while decreasing the tendency toward preignition when using a 10.5:1 Geometric Compression Ratio (GCR) with 93-Octane pump gasoline.

Although forged pistons can be as much as 25% lighter than a cast piston, there are also cases in which they are heavier, so in either case the balance factors of the crankshaft have to be modified. If the reciprocating mass is greater than that of the original equipment specification, then the engine will vibrate a bit more due to the oscillation of the speed of the flywheel as it overcomes the greater weight pumping up and down inside of the engine. Of course, the extra weight of the heavier pistons could be compensated for by using lighter H-beam Carrillo or Arrow Precision drop-forged EN24 chromium-molybdenum alloy A-beam connecting rods, but this is a very expensive ($$$$!) solution.

For most applications, hypereutectic pistons will perform quite well. These are cast pistons made from an aluminum alloy that has over a 12.5% silicone content to endow it with both strength, durability, and a coefficient of thermal expansion that closely matches that of the grey iron of the cylinders. Special melting processes are necessary in order to supersaturate the aluminum with the additional silicon content. Special molds, as well as special casting and special cooling techniques are required in order to produce finely and uniformly dispersed silicon particles throughout the material. This produces pistons that are very hard, but rather brittle. This being the case, they are longwearing, but their brittleness has proven them to be unforgiving of detonation. Their reduced coefficient of thermal expansion being similar to that of the engine block allows the piston to be run with reduced clearances, which in turn reduces scuffing as well as gas pressure losses due to combustion gases escaping past the sealing rings.

Although all piston manufacturers try to assure that their pistons are thoroughly cleaned of all grinding residue prior to packaging, upon occasion this part of the quality control process fails. If material is present in the sides and the back of a groove, as soon as the engine is started, this grit will begin to migrate out of the groove and come between the face of the piston ring and the cylinder wall, causing abrasive wear. This condition also restricts the ability of the piston ring to move, causing its outward expansion to become sluggish within the groove. This sluggish movement will result in an oil film on the cylinder wall that is too thick, resulting in poor oil control. In addition, this abrasive material left inside of the grooves will cause greatly accelerated wear and result in premature engine failure. Consequently, pistons should always be carefully examined and cleaned prior to installation of their rings.

Simply installing oversize rings onto old pistons of an oil-burning engine is a sleazy bodge that is used by unscrupulous owners with the aim of getting an engine running in order to sell the car to an unsuspecting buyer. It definitely is not something that you would do if you intend to keep the car. Piston slap that is caused by worn cylinder walls, which is at its worst at idle speed and silences as the engine speeds increases, will indeed cause your excessive oil consumption to soon reappear due to rounding of the sealing edges of the oversize rings, so it really is a waste of time and money. You will have to pull the engine again, disassemble it again, and have the cylinders bored and honed in order to accept new oversize pistons and rings, reassemble it again, install in back into the car again, etc., etc. As I said, it really is a waste of time and money.

The visual appearance of a compression ring is a matte black color, which is characteristic of the black manganese phosphate coating that is used to protect the metal. This coating is applied to the compression ring for two important reasons: (1) rust prevention and (2) oil retention for protection against scuffing during early engine operation. Rust prevention is necessary for the period of time that a piston ring is inside of the set box sitting on a shelf, sometimes in very humid areas. Scuff protection is most critical during early engine operation. Even when an engine has been pressure pre-lubricated the ring belt area of the piston receives little if any oil during the pre-lube process and receives oil only after the engine is running and oil is being spun from the crankshaft. The piston rings then must depend on the oil which was applied to them before they were installed into the cylinders for a brief period before the engine is initially started. The manganese phosphate coating has excellent properties to accomplish its task of scuff protection because it is porous and quite soft. It can be likened to a sponge in that it soaks up and retains oil for the period before the piston ring receives oil from the running engine.

The compression ring at the top of the floating four-ring floating piston receives less lubrication than in the later three-ring designs, the consequence being that both the top compression rings and the cylinder bores wear faster. However, with modern lubricants this difference is minimized, though still present. Consequently, the advantage of an oil-retaining molybdenum coating is obvious in such a case. The biggest objection to the earlier four-ring pistons is that, being 3” (76.2mm) long, their weight is greater than that of the later 2.4” (60.96mm) long three-ring design. It takes more power to make that extra weight reciprocate up and down inside of the engine, plus the amplitude of Primary Vibration produced is also increased. Due to the improved load-bearing capacities of modern engine lubricating oils, the reduced load-bearing surface area of the shorter three-ring pistons does not present a serious wear problem for either the piston or the cylinder wall. These are the reasons why the factory chose to redesign the assembly in this manner.

However, there is an even more important reason why the old four-ring piston design is inappropriate for a high performance version of the B Series engine. Because of both the combustion characteristics of the available fuels during the era in which they were designed, as well as the vulnerability of the grey iron of which the compression rings were then being designed, there was a risk of the compression ring being damaged by the heat of combustion. In time, advances in metallurgy resulted in a new generation of compression rings that were less susceptible to combustion damage, the most significant of which was the development of more suitable alloys of Ductile Iron, sometimes called “Nodular Iron”. The high strength of Ductile Iron, its wear resistance, as well as its elasticity combine to allow heavier loads with less deflection. Coupled with the advent of better fuels, the designers were able to relocate the compression ring further down from the crown of the piston in the design of the more modern three-ring pistons in order to further prevent damage from combustion. Unfortunately, under rich running conditions unburned fuel can condense above the compression ring and inside of its groove, lurking there to detonate should preignition occur. This leads to fracturing of the compression ring and, in extreme cases, breakage of the neighboring land. In addition, because of the structural weakness that stems from the drainage slot from the oil ring groove extending to the wrist (gudgeon) pin boss in some piston designs, this in turn can cause the upper section of the piston to fracture and separate from its skirt.

Be aware that all pistons used in the B Series engine are “handed”, and must be fitted facing the correct way round as the wrist (gudgeon) pin is offset in order to provide a constant side thrust against the thrust side of the cylinder. The amount of offset can vary between .030” to .060”, and varies by piston design since the loading of the wall/skirt is the prime determinant. This constant side thrust against the cylinder wall prevents the piston from slapping against the cylinder wall.

It should be noted that both floating wrist (gudgeon) pin and press fit wrist(gudgeon) pin pistons for the B series both have offset wrist (gudgeon) pins. The offset is quite noticeable if you look at the bottom of the piston. Press fit pistons have a recess around one end of the pin bore that faces forward in the engine. All pistons should always be fitted so that the offset is located towards the thrust side of the cylinder wall, i.e., with both the oil hole in the connecting rod and the offset of the wrist (gudgeon) pin facing way from the camshaft.

The Original Equipment Hepolite pistons have an arrow on their crowns in order to assist in installing them (with the arrow pointing toward the front of the engine). Arrows can and have been omitted or wrongly stamped on aftermarket pistons, so always check the offset against the marking.

I would suggest using in combination the more modern three-ring pistons that make use of press-fit wrist (gudgeon) pins and having the wrist (gudgeon) pin bores of the pistons machine grooved in order to accept wrist (gudgeon) pin retainers along with bushed small end connecting rods so that you can have a floating wrist (gudgeon) pin system. In that way you can have the best of both worlds!

The following tables may be useful as a guide in selecting your piston size:

Small Bore Engines	
Bore Oversize Bore Dimension	**Displacement**
+.000” 3.160” / (80.264mm)	1798 cc
+.020” 3.180” / (80.772mm)	1822 cc

Big Bore Engines	
Bore Oversize Bore Dimension	**Displacement**
+.080” 3.240” / (82.296mm)	1892 cc
+.090” 3.250” / (82.55mm)	1903 cc

+.030” 3.190” / (80.026mm)	1833 cc
+.040” 3.200” / (81.28mm)	1844 cc
+.060” 3.220” / (81.788mm)	1868 cc

+.100” 3.260” / (82.804mm)	1915 cc
+ .108” 3.268” / (83.0072mm)	1924 cc
+.130” 3.290” / (83.566mm)	1948 cc

If your engine has not been run for a long time, do not be surprised if you find that the pistons have seized inside of their cylinder bores. This is due to the piston rings having rusted onto the walls of the cylinders. Be aware that if the old piston rings have rusted to the cylinder wall, then the pistons will most likely be already useless. Removing them with a press will surely break the piston rings, which will then ruin their grooves in the piston, as well as badly score the cylinder wall. If the engine block has been disassembled, you can simply dissolve the rust as well as the stuck pistons with phosphoric acid. That way the damage to the iron cylinder bore will not extend far beyond the extent of surface rust. Just be sure to set the engine block on wooden rails out of doors before you pour the phosphoric acid into the bore(s) as the fumes that will result are extremely toxic! It emits a suffocating odor that can quickly burn the lining of the nose, the throat, and even the lungs.

Before an attempt to machine the cylinder bores is made, it must be ascertained if any sleeves (liners) have been installed. This was occasionally done at the factory in order to salvage an engine block casting that would have been otherwise unusable due to problems with core shifting during the casting process that occasionally resulted in engine blocks with either off-center cylinders or cylinders of insufficient cylinder wall thickness. If a sleeve (liner) has been fitted, be aware that it cannot be rebored to accept an oversize piston. Instead, the engine block will have to be machined so that new sleeves (liners) can be press-fitted into all of the cylinders and the previous size pistons installed. It should be noted that the factory supplied oversize pistons only up to +.040” (+1.016mm) oversize due to variations in the thickness of the cylinder walls and problems with porosity that were due to the limitations of the casting technology of the era. Cylinder bores beyond this diameter may require the fitting of oversize sleeves (liners). There is an advantage to this approach. Sleeves (liners) can be made of special wear-resistant material that is likely to have a longer service life than that of the original gray iron bore. Most sleeves (liners) have the advantage of being made of spun cast (centrifugally cast) iron alloy, which is denser and of better quality than the "block-type" cast grey iron, leading to a prolonged lifespan for the bore. Most manufacturers can supply them in a prefinished state so that no subsequent machining is required after installation. Optional hard-chrome, nickel, or silicon carbide bore coatings are also available. Naturally, when fitting them the cylinder bores are opened out to a much greater extent than is used for a normal rebore. However, the superior rigidity of the liners is sufficient to resist any tendency toward bore flexure. In view of the necessity of ensuring the correct interference-fit of the sleeve (liner) inside of the bore, plus the necessity of subsequent accuracy in the finishing-off of the bore of the liner, this method is more expensive than an ordinary "oversize" rebore as it requires precision grinding of the bore to an exacting diameter. On balance, I would say that if long-term durability is a priority and your checkbook can stand it, sleeves (liners) are the best answer. You will also need to check the cylinder bores after boring them out, as the mild steel straps that held the sand casting cores together are to be found embedded between cylinders #1 and #2 and between cylinders #3 and #4. Overboring to +.130" (3.302mm) occasionally exposes these straps and they often have voids around them, invariably making sleeving necessary. Be aware that in the case of chrome-plated cylinder sleeves (liners), chrome plated rings should not under any circumstances be installed into the pistons of an engine that has had chrome-plated sleeves (liners) installed. Catastrophic destruction of both the rings and the sleeves will result. However, plain cast iron, manganese phosphate coated, or molybdenum coated rings are compatible with chrome-plated sleeves (liners).

Note that the various Hepolite pistons found in the different versions of the B Series engine installed in the MGB make use of different piston / cylinder wall clearances. For example:

18G / 18GA Engines	
Top of skirt clearance	.00405" +/- .0045" (.10287mm +/- .1143mm)
Bottom of skirt clearance	.0021" +/- .003" (.05334mm +/- .0762mm)

18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, 18GK, and 18V Engines	
Top of skirt clearance	.0027" +/- .0006" (.06858mm +/- .01524mm)
Bottom of skirt clearance	.0009" +/-.0003' (.02286mm +/- .00762mm)

Depending upon differences in the design and the alloys used in their construction, the necessary clearances for aftermarket pistons may be different, so always adhere to the manufacturer's recommended clearances. Be aware that the cylinder bores are a very important factor if an engine is to have good oil control. Cylinder size, straightness, and finish are very critical to the pistons and the rings performing their functions properly. Cylinders must not only be bored round and straight; this integrity must also be maintained throughout the honing process. This will require the use of a torque plate during the honing process. Cylinders should be round to within +/- .0005" (+/-.0127mm) or less.

The top of the cylinder bore should be chamfered at an angle of either 60° or 75° to an Outside Diameter (O.D.) that is .020" (.508mm) greater than that of the finished bore in order to allow easier installation of the piston / ring assemblies, and also to prevent the development of "hot spots" on the top edge of the cylinders that are precursors of preignition. The bottom of the cylinder bore should always be chamfered at an angle of 60° to an Outside Diameter (O.D.) that is .020" (.508mm) greater than that of the bore in order to reduce wear of the skirt of the piston. Be advised that all chamfering must always be done prior to the honing of the cylinder.

When using a hone to crosshatch the cylinder bore, bear in mind that it is the fine, micro-channeled grooves created by honing that retain oil in order to lubricate both the pistons and their rings. Prior to honing, all oiling passages should be masked in order to preclude the incursion of the highly abrasive honing residue. Thermal distortion will occur when the engine is run; however, the total distortion of the cylinders will be lessened if the cylinders have been honed with a torque plate torqued in place. The blanking plate and the crankshaft main bearing caps should be left torqued in place during this process, and then, after the honing process is completed, removed and the surfaces of both the cylinders and the engine block thoroughly cleaned. With the blanking plate and the crankshaft main bearing caps torqued in place, the cylinder can then be honed round in the stressed condition with a #220 to #280 grit conventional stone (or a #325 to #550 grit diamond stone) to a 20- to 30-microinch finish until there is .001" left from final bore. The bore must be round to within .0002" when checked 360° from the bottom to the top of the bore. Continue with a 220 grit stone at 50% load until .0002" is left from final bore, then use the 220 grit stone at 20% load to the final bore size.

Do not attempt to hone the cylinder walls with anything other than honing oil. Honing oil will help to cleanly cut the surface as well as help to cool the brush hone. Under no circumstances should you attempt to hone the cylinders inside of a solvent tank using parts solvent. If you do this, then the hone will tear, fold, and rip the bore finish. A honing groove angle of 45° is optimal for both cast iron and chromium-plated piston rings. A honing groove angle of 20° to 22° is optimal for Molybdenum-coated piston rings. Too steep an angle promotes oil migration down the cylinder, resulting in a thin oil film that can allow piston ring and cylinder scuffing, while too flat a crosshatch angle can lead to piston ring rotation, which in turn will interfere with the seating of the rings. It can also cause the honed grooves to hold an excessive amount of oil, a condition that will additionally create thicker oil films that will cause the piston rings to hydroplane rather than to seal. A one-directional honing pattern or an uneven crosshatch pattern can also cause excessive piston ring rotation. Continual high-speed rotation of the rings will wear the sidewalls of the rings and their seating grooves in the piston as well, ruining them both. This will result in excessive clearance, piston groove pound-out, and eventual piston ring breakage. During the initial break-in stages of the engine, the piston rings can rotate inside of the cylinder bore as much as 5 RPM, even when everything is geometrically correct. However, as seating takes place, this rotation will gradually reduce to a snail's pace. Piston rings that are subjected to excessive rotation can be identified by a horizontal or diagonal wear pattern on the face of the piston ring rather than the vertical pattern that is produced under normal operating conditions.

Be aware that the honing must be done slowly in order to minimize heat build-up. No hand honing should ever be attempted. The final bore needs to be less than +/- .0002" out of round, checked 360° around the bore from the bottom to the top of the cylinder. This should be checked with a dial bore gauge.

After honing, use a #400 to #600 grit plateau hone in order to remove the jagged peaks of the ridges and folded or torn metal on the edges of the crosshatch grooves and thus facilitate easier and more precise seating of the rings. It will also minimize the "bore rifling" effect that can cause unwanted piston ring rotation during the break-in process. Although some shops will tell you that this is not a necessary procedure, they most likely do not have a plateau hone and do not care to bear the expense of purchasing one. If these imperfections that are created by the crosshatch honing are left in place, then the piston rings will have to wear them away, resulting in worn piston rings and inferior sealing of the combustion gases. Be sure to confirm with the honing equipment manufacturer that the recommended stone grit will produce the following Rz and Ra roughness recommendations: Rz = 59 - 138 µ in (=1.5 – 3.5 µm) or Ra = 15 - 35 µ in (=0.4 – 0.9 µm). After plateau honing, both the pistons and cylinder bores should be again precisely measured. The bore must be round to within .0002" when checked 360 degrees from the bottom to the top of the bore. The pistons should then be paired to their optimum cylinder bores. Once the rings have properly seated, this will help to ensure as oil tight a seal as possible in order to do away with (as far as possible) two major problems: oil contamination of the incoming fuel / air charge, and increased crankcase pressure resulting from blow-by. Tight control over both promotes maximum power output and greatly reduces emissions problems.

After all of the honing procedures have been completed, wash the cylinders and engine block with hot, soapy water. Cleaning of the cylinders after the honing process is important and cannot be overemphasized. The process of honing the cylinders leaves two types of "dirt" on the cylinder wall, honing stone residue, and cast iron dust. If not removed before the engine is reassembled, the world's finest lapping compound will be in place, just waiting to destroy all of the rings, bearings, and all other moving parts in the engine, plus the hard work of assembly, the instant the engine is started. To clean the cylinders correctly, use a stiff round brush. Afterwards, oil their bores in order to prevent rust, using only clean oil and a lint-free rag.

While all of the connecting rods that the factory used in the B Series engine make use of the same eye center-to-eye center length (6.500" / 165.1mm), thus producing a connecting rod to stroke ratio of 1.86:1, they differ greatly in design details. In light of the relatively slow combustion rate of the fuels available at the time of the engine's design, this "fast" connecting rod to stroke ratio chosen produced an usually fast piston acceleration rate that works well with the more volatile, faster-burning unleaded fuels that are produced today. However, it also increased the sidethrust loading of the piston, forcing the use of a three-inch long piston in order to provide a sufficient load bearing surface area necessitated by the lower pressure-rated oils then available.

The connecting rods of the 18G and 18GA engines use a wrist (gudgeon) pin that was secured by means of a clamping small end feature, while those from the 18GB, 18GD, 18GF, 18GH, 18GJ, and through the early 18GK engines all use floating wrist (gudgeon) pins that ride inside of small end bushings and are retained by circlips, and those of the late 18GK and all 18V engines use press-fitted wrist (gudgeon) pins, although the wrist (gudgeon) pins of the five-main-bearing engines are of the same diameter (.8125" / 20.637mm). The press-fitted pins typically require a 3 to 5 ton press to install. Prior to installation, the ends of the pins should be checked to be sure that they have been lightly chamfered in order to prevent them from damaging their bores in the piston. The safest installation technique is to spray the pins with WD-40 in order to displace any moisture on them, place them into a well-sealed Ziploc bag in order to prevent ice from forming on them, and with the thermostat on the deep freeze set as low as it will go, they should be left in there to chill overnight. That will shrink them to a smaller diameter. Boil the pistons in water in order to cause their wrist (gudgeon) pin bores to expand just prior to pressing in the chilled pins.

Although the pistons themselves are interchangeable between the five-main-bearing engines due to their identical small end and big end bearing sizes of the connecting rods, they must be installed complete with their appropriate connecting rods and wrist (gudgeon) pins. The only exception to this rule is when pistons with floating wrist (gudgeon) pins are installed into the later connecting rods of the late 18GK and all 18V engines that have been modified in order to accept the small end bushing that these pistons require. Be aware that the four-ring pistons of the three-main-bearing 18G and 18GA engines used wrist (gudgeon) pins of .7500” (19.05mm) diameter and as such cannot be used with the connecting rods of the later five-main-bearing engines.

Because of the B Series engine’s relatively short connecting rod to stroke ratio of 1.86:1, the engineers at MG originally insisted on forgoing the use of the usual split skirt Lo-ex piston normally installed in other versions of the B Series engine that were intended for use in the more sedate family sedans. They chose instead to specify 3” (76.2mm) long solid skirted pistons to both take advantage of their inherently enhanced oil control and to minimize the effects of the greater side thrust loadings resulting from the higher engine speeds attainable with dual carburetors, thus guaranteeing reliability. For use on small bore engines (1868cc or less), the Original Equipment Hepolite pistons, although a bit heavy at 476 grams, are of excellent quality and in most high performance street versions of the B Series engine they need not be superseded by specialty racing pistons. They also have the distinct advantage of having their oversize number impressed upon the forward part of their crowns in order to ease reassembly. These are available from Advanced Performance Technology. They have a website that can be found at http://www.aptfast.com/ .

If you choose to install the Original Equipment 8.8:1 high compression pistons with their 6.5 cc dished crowns of the earlier 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and early 18GK engines without their bushed small end connecting rods into the late 18GK and all 18V engines that do not have bushed small ends, then you will need to machine the small end of the Original Equipment connecting rods of the 18V engines in order to accept the small end bushing of the earlier connecting rods as this piston uses a floating wrist (gudgeon) pin system. The smaller 39cc combustion chamber volume of the North American Market version of the 18V heads will cause the Geometric Compression Ratio (GCR) to be boosted to about 9.4:1 with the Original Equipment piston clearance depth of .040”, presuming, of course, that the machinist has not removed too much material from the deck of the engine block or from the cylinder head, in which case it will be higher still. On the other hand, if the later low compression ratio pistons of the North American Market version of the 18V engine (8:1) with their 16.2 cc dishes are installed into an engine equipped with the cylinder head from an 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engine with their 43cc combustion chambers, then Geometric Compression Ratio (GCR) will be a very low 7.7:1 with the Original Equipment piston clearance depth of .040”. Fortunately, the UK/European market pistons for the 18V engines were produced in a 9:1 Geometric Compression Ratio (GCR) when used with their 43cc combustion chambers. When used with the 39cc combustion chambers of the North American Market version of the 18V engines, the Geometric Compression Ratio (GCR) will be 9.6:1, which is about as high as one would prudently choose to go with a cast iron cylinder head that has the benefit of professionally modified combustion chambers. County makes a flat-topped three-ring Aerolite piston in the standard oversizes that is a direct replacement for the Hepolite 20616 piston. It produces a 9.5:1 Geometric Compression Ratio (GCR) with the Original Equipment piston clearance depth of .040" when used with the 43cc combustion chambers of the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines. When used with the 39cc combustion chambers of the North American Market version of the 18V engines, the Geometric Compression Ratio (GCR) will be a lofty 10.1:1 with the Original Equipment piston clearance depth of .040", and 10.5:1 with a running piston clearance depth of .012", making them a good choice for use with an aluminum alloy cylinder head. In either case, high octane rated fuel will be a necessity. Although their compression height (wrist pin-to-crown dimension) is the same as that of the Original Equipment Hepolite dished crown pistons, these use the floating wrist (gudgeon) pin of the 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and early 18GK engines, thus requiring that the small end of the connecting rods of the 18V engines be opened up and bushed with the bushings of the earlier connecting rods. It should be noted that, given equal compression ratios, a dished crown piston is a more efficient power producer than a flat-topped piston. This is a result of its concave crown presenting a greater surface area for the expanding combustion gases to act against. This being the case, dished crown pistons are preferable and any increase toward a desired higher compression ratio of the engine could be attained by machining either the deck of the engine block or the mating surface of the cylinder head, or by combining both methods.

Bear in mind that it is very important to keep your combinations of compression ratio and camshaft properly balanced. For example, you would not use an 8:1 compression ratio with a camshaft lobe profile that produces a duration of 270°, as a 9.5:1 compression ratio would be more appropriate. Conversely, you should not have a 10:1 compression ratio and use a camshaft lobe profile with 256° duration! Be aware that as soon as the duration increases above 264°, the Original Equipment exhaust system will increasingly restrict the breathing ability of the engine. It is often difficult to tune the carburetor to make the engine idle properly due to the extra back pressure from the exhaust system.

The following table should be of assistance in selecting AE/Hepolite or County pistons-

BMC Part #	Manufacturer's Part #	Oversizes Available	Number of Rings	Comments
12H 961	**AE/ Hepolite Part #** 16201V* **County (Aerolite) Part** # CP546K*	+.020" +.040" +.060"	5 (O.E. Hepolite) 4 (County Aerolite)	Used from 1962 thru 1964, 18G & 18GA engines, 6.5 cc dish, wrist (gudgeon) pin secured by a pinch bolt in the small end of the connecting rod
8G 2578	**AE/ Hepolite Part #** 17358V* **County (Aerolite) Part** # Not Available	+.020" +.040" +.060"	4	Used from 1964 thru 1966, Long skirt, 6.5 cc dish, floating wrist (gudgeon) pin, bushed small end of the connecting rod
8G 2643 and 12H 5161	**AE/ Hepolite Part #** 17525V* **County (Aerolite) Part** # CP308K*	+.020" +.030" +.040" +.060"	4	Used from 1967 thru 1971, Short skirt, 6.5 cc dish, floating wrist (gudgeon) pin, bushed small end of the connecting rod
12H 5163H/C	**AE/ Hepolite Part #** 18802V* **County (Aerolite) Part** # CP307K*	+.020" +.030" +.040" +.060"	3	Used from 1972 thru 1980, 18V UK/European Market specification High Compression Engines, Short skirt, 6.5 cc dish, press-fitted wrist (gudgeon) pin
12H 5163L/C	**AE/ Hepolite Part #** 19354V* **County (Aerolite) Part**	+.020" +.040"	3	Used from 1972 thru 1980, 18V US Federal specification Low

	# CP307AK*			Compression Engines, Short skirt, 16.2 cc dish, press-fitted wrist (gudgeon) pin
Not Original Equipment	**AE/ Hepolite Part #** 20616V* **County (Aerolite) Part** # CP548K* **Accralite Part #** 1195xc8179	+.060”	3	1868cc Flat-top piston
Not Original Equipment	**AE/ Hepolite Part #** 20771KRV **County (Aerolite) Part** # CP551K*	+.080”	3	1892 cc Press-fitted wrist (gudgeon) pin, sleeving of cylinders recommended
Not Original Equipment	**AE/ Hepolite Part #** ? **County (Aerolite) Part** # ?	82.5 mm 3.250”	3	1901 cc +.000” Lotus Twin Cam piston, floating wrist (gudgeon) pin, bushed small end on the connecting rod, requires lateral offset boring of cylinders due to non-offset wristpin, sleeving of cylinders recommended
Not Original Equipment	**AE/ Hepolite Part #** ? **County (Aerolite) Part** # ?	83 mm 3.268”	3	1925 cc +.020” Lotus Twin Cam piston, floating wrist (gudgeon) pin, bushed small end on the connecting rod, requires lateral offset

				boring of cylinders due to non-offset wristpin, sleeving of cylinders strongly recommended
Not Original Equipment	**Accralite Part # 1196xc835**	83.57mm 3.288”	3	1945 cc Flat-Top piston, offset floating wrist (gudgeon) pin, bushed small end on the connecting rod, sleeving of cylinders very strongly recommended
Not Original Equipment	**AE/ Hepolite Part #** 21093V* **County (Aerolite) Part** # CP717K*	83.57mm 3.288”	3	1945 cc +.040” Lotus Twin Cam piston, floating wrist (gudgeon) pin, bushed small end on the connecting rod, requires lateral offset boring of cylinders due to non-offset wristpin, sleeving of cylinders very strongly recommended
Not Original Equipment	**AE/ Hepolite Part #** ? **County (Aerolite) Part** # ?	84mm 3.307”	3	1971 cc +.060” Lotus Twin Cam piston, floating wrist (gudgeon) pin, bushed small end on the connecting rod, requires lateral offset boring of cylinders due to non-offset

				wristpin, requires mandatory sleeving of cylinders

* Note: the suffix "K" or "V" is used to refer to an order for a full set of four pistons, complete with rings, wristpins (gudgeon pins), and circlips if required. Individual pistons, complete with rings, wristpins (gudgeon pins), and circlips if required are ordered by deleting the suffix. Oversizes can be ordered by simply adding the required size (or oversize) to the end of the part number preceding it with a space. I.e., CP717 STD, CP358 020, CP 307 040, CP308 060, 17525 030, etc.

The pistons should be individually and carefully fitted to their respective wrist (gudgeon) pins. Chamfering of any sharp edges on the piston crowns reduces possibility of the development of localized hot spots that cause preignition and / or detonation. Each piston should be carefully matched for clearance with its selected bore. Too little clearance will result in scuffing and too much clearance will reduce the sealing effectiveness of the rings. The basic piston diameter measurement is taken at 90° to the wrist pin (gudgeon pin), immediately the bottom of the lowermost piston ring and above the wrist pin (gudgeon pin). Pistons with an oil ring on the skirt below the wrist pin (gudgeon pin) are another matter. This is the "top of skirt measurement" referred to in the books. Measurements in or above the ring groove area will be considerable smaller. Likewise, any measurements not at 90° to the wrist pin (gudgeon pin). Measurements that are taken further down the skirt will be found to yield larger figures, giving the reduced clearance figures that found in the books. These published reduced clearance figures are the ones mistakenly taken to indicate tapered cylinder bores by some folks. When assembling the engine, for a quick measurement of the piston to cylinder bore clearance, insert the piston upside down into its bore, and then while holding it against the cylinder wall at the bottom of the skirt 90° from the pin, then insert a feeler gauge between the opposite side of the skirt and the wall. Whatever size gauge gives you a light drag when inserted will be very close to your actual clearance. While a micrometer and a bore gauge will give more accurate results, you do need to be reasonably well practiced with them in order to use them accurately.

Piston Rings

All piston ring sets contain information pertaining to their proper installation. The compression ring is the closest piston ring to combustion gases and thus is exposed to the greatest amount of chemical corrosion as well as the highest operating temperatures. The compression ring transfers 70% of the combustion chamber heat from the piston to the cylinder wall. While most compression rings appear similar, there are many subtle design features that dictate their correct installation. Not only must the compression ring be installed with the proper side facing toward the piston crown, it is also imperative that the compression ring be installed into its appropriate groove.

There are essentially two basic types of compression rings commonly encountered on the top groove of the pistons of a BMC B Series engine: the simple barrel faced type and the torsional-twist type. A barrel faced compression ring has a .001" radius curved running surface that compensates for any slight misalignment of the ring groove in order to provide consistent lubrication of both the compression ring and the cylinder wall. It is the fastest-reacting type of compression ring to combustion pressure, and therefore is the quickest to seal. The barrel face also provides a wedge effect that optimizes oil distribution throughout the full stroke of the piston. In addition, the curved running surface reduces the possibility of an oil film breakdown caused by excess pressure at the edge of the compression ring or by excessive piston tilt during operation. This type of compression ring remains in contact with the cylinder wall when it is not under pressure during the intake stroke, thus making it the preferred choice for use as the top compression ring. Because of both its location at the top of the piston, and its reduced sealing area, molybdenum coating for enhanced oil retention is desirable. The torsional-twist type features Increased side sealing for superior performance in high output engines. It typically produces better oil control than a simple barrel faced compression ring, but it may have either a barrel or a straight face.

The reverse torsional twist type compression ring is the most common type used as a second compression ring. Be warned that if reverse torsional type compression rings are installed upside down, they will not be able to take advantage of Torsional Twist, a built-in imbalance between the way that the upper and lower sides compress that causes a slight twist in the compression ring when being compressed. This twisting feature enables this type of compression ring to seal itself both inside of the groove as well as against the cylinder wall. Reverse torsional twist along with a tapered ring face creates bottom-edge scraping. It also promotes side sealing that prevents oil from being forced under and behind the piston ring, resulting in better oil economy. Having a wiper profile consisting of an additional groove machined into the bottom edge of the face of the oil ring provides both twist and an oil accumulator for improved oil scraping and oil economy volume. Be aware that installing a reverse torsional type compression ring upside-down can lead to excessive oil pumping, excessive blow-by, and in some cases completely dry up the bore, causing both piston ring and cylinder scuffing as well as accelerated wear. Being a low-tension type of piston ring, it expands outward against the cylinder wall only when it is subjected to pressure from above. This being the case, during the intake stroke it contracts into its groove in the piston, minimizing both friction and consequent wear.

Be aware that cylinder leakage tests are conducted with the piston in a steady state. They do not account for time, piston movement, or true operating pressures. While gimmicks make for good advertising, multi-piece gapless compression ring designs add unnecessary weight and complexity, neither of which will make your engine more powerful.

There is some confusion amongst owners as to what type of piston ring set should be used. The single most important factor to be considered in selecting the proper compression ring face coating material is the service conditions that the engine will be operated under. The three popular types of top compression ring face coatings, cast iron, chromium, and molybdenum, each has its own advantages in respect to operating conditions. Cast iron is a durable wear surface in normal operating conditions and is less costly than the molybdenum- or chromium-faced piston ring. Molybdenum's porosity holds oil on the Outside Diameter (O.D.) face of the piston ring, giving it a very high resistance to scuffing and scoring. However, the pores of the material also can serve as a trap for foreign materials, so a fine-straining oil filter is advisable when using molybdenum coated piston rings. To identify a molybdenum coated piston ring, look for a silver-grey plated finish with black phosphated top and bottom surfaces. Chromium has good resistance to scuffing but lacks molybdenum's oil retention capabilities.

For typical light duty service where the vehicle is not subjected to long periods of high speed or high load operation and is run primarily on paved streets, plain cast iron is a good choice because the type of cast iron used for the manufacture of piston rings is very durable when not subjected to unusual dirt or heat conditions.

Chromium has more resistance to scuffing and scoring than cast iron, but has somewhat less resistance than that of molybdenum. To identify a chromium plated piston ring, its face will have chrome plating and its top and bottom will be reddish-brown in color. Chromium on any of the piston rings is not advisable with Nikasil bores. Because in a dusty environment the incoming fuel / air mixture probably will contain some abrasive contaminant, the smoother, more abrasion-resistant surface of chromium makes it the logical choice. Chromium's extreme density and hardness resists the Impingement of abrasives into the face of the piston ring, which accelerates cylinder wear of the cylinder bore, and actually contributes to the exhaust gases carrying some of the airborne contaminant out through the exhaust system. These are the reasons why track racers who routinely run their engines without air filters prefer to use chromium plated compression rings.

However, a street engine always uses air filters. When subjected to continuous high speed or severe load conditions, the engine will be subjected to long periods of high temperatures. Molybdenum is inherently porous in its applied state, resulting in excellent retention of oil on the face of the piston ring, making it the better choice. It also has the additional advantage of having the highest melting point of the three popular face coatings, giving it the capability to survive better under more severe operating conditions, more specifically to resist scuffing and scoring. Being a softer material than chromium, it inherently causes less wear of the bore of the cylinder. It is the obvious choice for a high performance street engine. It is also what the engineers at MG chose for the engine of the MGB.

The primary function of the second compression ring is oil control. Maintaining compression pressure is a purely secondary function. A tapered face design allows this compression ring to work as a "scraper", reducing the potential for oil migration into the combustion chamber.

The design of an oil control ring includes two thin rails that act as running surfaces. Holes or slots cut into the radial center of the oil control ring allow drainback paths for excess oil back to lubricate the piston ring groove. Oil control rings are commonly of one-piece construction, incorporating all of these features. Some one-piece oil control rings utilize a spring expander in order to apply additional radial pressure to the oil control ring. This increases the unit pressure (measured amount of force and running surface size) that is applied at the cylinder wall. This being the case, these oil control rings have the highest inherent expansion pressure of all of the piston rings. Some engines, such as the BMC B Series engine, use a three-piece oil control ring consisting of two rails and a circumferential expander. The oil control rings themselves are located on the top and bottom sides of the circumferential expander. The circumferential expander usually contains multiple slots or windows in order to permit a drainback path for oil in order to lubricate the piston ring groove. This type of oil control ring design makes use of inherent piston ring pressure, that is, expander pressure, and the high unit pressure provided by the small running surface of the thin rails in order to effect sealing.

When it comes to an oil control ring that is to be located directly beneath a compression ring, the best type for street use is the vented design that employs twin rails and a circumferential expander in order to provide a drainback path for the removal of excess oil. Due to their low tension, these reduce internal engine friction while their sturdy, box-like construction eliminates oil control ring flutter and deformation, thus maintaining positive oil control at high engine speeds. Oil drainage from the piston land is critical to oil control. If the oil scraped by the oil control ring cannot be drained rapidly from behind or under the oil control ring, it will hydroplane, so check to be sure that the oil drainage holes inside of the grooves of the piston are of uniform size and shape. If they are not, they should all be carefully reamed to a smooth finish. In the case of an additional oil control ring located on the skirt of the piston, a one-piece slotted cast iron design is best.

As a general rule, the following combinations of piston rings will give satisfactory performance:

Ring Size & Type				
Top Groove	**2nd Groove**	**3rd Groove**	**4th Groove**	**5th Groove**
1/16" Barrel-faced compression ring	1/16" Reverse Torsional compression ring	5/32" 3-piece Vented oil control ring		
1/16" Torsional compression ring	1/16" Reverse Torsional compression ring	5/32" 3-piece Vented oil control ring		
1/16" Barrel-faced compression ring	1/16" Reverse Torsional compression ring	1/16" Reverse Torsional compression ring	5/32" 3-piece Vented oil control ring	

1/16" Torsional compression ring	1/16" Reverse Torsional compression ring	1/16" Reverse Torsional compression ring	5/32" 3-piece Vented oil control ring	
1/16" Barrel-faced compression ring	1/16" Reverse Torsional compression ring	1/16" Reverse Torsional compression ring	5/32" 3-piece Vented oil control ring	5/32" One-piece slotted cast iron oil control ring
1/16" Torsional compression ring	1/16" Reverse Torsional compression ring	1/16" Reverse Torsional compression ring	5/32" 3-piece Vented oil control ring	5/32" One-piece slotted cast iron oil control ring

Not surprisingly, the type of material used for the construction of the piston ring is a significant factor. Cast grey iron is a perfectly adequate piston ring material as long as the piston rings are of sufficient size to handle the loads. However, the change to thinner low-tension piston rings combined with efforts to squeeze more power out of smaller displacement engines has increased the operating loads on the piston rings, especially on the top compression ring that receives the brunt of the punishment. Uncoated cast grey iron is compatible with cast iron cylinder walls and will not gall or scuff, but it is also brittle. If you bend a piston ring that is made out of cast grey iron too far, it will snap in two. The material has limited "give" as a consequence of the nature of its microstructure. When the grain structure of cast grey iron is examined under a microscope, it exhibits sharp rectangular grains that easily fracture if the metal is shock loaded (as in detonation), or is bent too far (a good reason for always using a piston ring expander when installing piston rings on a piston).

In the case of narrow low-tension piston rings (5/64" or 1.5mm), cast grey iron piston rings can break when the engine is subjected to heavy or continuous detonation. The hammer-like blows produced by the colliding flame fronts shock-loads the piston rings and can break them, resulting in a loss of compression, cylinder wall damage, and oil consumption problems. Various alloys of cast grey iron are available, including "intermediate" alloys that are both somewhat harder (28 to 38 HRC) and stronger. In many applications, piston rings made of these special alloys are used uncoated for the second compression ring. Chromium-coated or molybdenum-coated intermediate cast grey iron piston rings are also used for top compression rings. Low-tension piston rings minimize the tension at their interface of the piston ring with the cylinder wall, thus reducing both wear and energy loss due to friction. However, in order for these piston rings to function efficiently, the cylinder wall must be carefully machined. The maximum permissible cylinder taper must not exceed .006" and ovality must not exceed .005".

That brings us to "ductile" iron piston rings, which are roughly twice as strong as piston rings made of cast grey iron. Ductile iron is also called "nodular" iron because its microstructure contains rounded or nodular shaped grains. These increase strength and allow the metal to bend without breaking. Consequently, ductile iron compression rings can take much more severe pounding than cast grey iron compression rings can without breaking. It is also has greater hardness, but this does not necessarily mean that it is more wear resistant. Most of the abrasives that cause premature piston ring wear will wear a ductile iron piston ring just as fast as an ordinary cast grey iron piston ring. It is the molybdenum or chromium coating on the ductile piston ring that helps to retard the wear rate. Piston rings constructed of this material have also been a popular choice for racers because of its ability to hold up in a high engine speed, high stress racing environment. Unfortunately, ductile iron has two drawbacks. The first is that it is more expensive than cast grey iron. The basic material costs more and, due to its almost 75% greater hardness, it is more costly to machine. The second drawback is that it is not as compatible with cast iron cylinder walls as cast grey iron. It has a tendency to scuff and gall unless it is has been coated with either chromium or molybdenum. Be aware that there is absolutely no advantage to using high strength ductile iron piston rings in an engine that does not require it. It is like putting 100 octane fuel into a car with a 7.5 to 1 compression ratio.

How do you tell a ductile iron piston ring from one that is made of cast grey iron? You cannot tell by appearance alone because both materials have the same appearance. In addition, both kinds of piston rings are usually coated with a protective black phosphate coating at the factory in order to protect them from corrosion during storage and shipping. A cast grey iron piston ring will makes a dull thud if tapped or dropped on the floor. Ductile iron (as well as steel), however, makes a ringing sound, much like a bell.

The next step upwards from ductile iron is steel. While ductile iron is roughly twice as strong as cast grey iron, steel is roughly twice as strong as ductile iron. Consequently, steel piston rings can really take a pounding without failing. They are the material if choice for those manufacturers of forged pistons that are designed for use in high performance applications.

Here is how the three alloys compare:

Material	Material Hardness	Tensile Strength	Failure Strength
Cast grey Iron	22 – 23 HRC	45,000 PSI	30,500 PSI
Ductile Iron	38 – 40 HRC	180,000 PSI	87,300 PSI
Steel (SAE 9254)	44 – 53 HRC	240,000 PSI	138,600 PSI

As you can see, steel is harder, has a higher tensile strength and higher fatigue strength that either ductile or cast grey iron. How this actually translates into piston ring strength and wear resistance depends on the size and shape of the piston rings. However, generally speaking, steel piston rings provide better mechanical stress, breakage, and heat resistance as well as reduced piston ring side wear and reduced groove side wear. They seal better under adverse conditions and resist breakage and wear under heavy loadings. They are ideal for any application that involves higher combustion temperatures, higher compression loads and tougher emission standards. The longer-wearing characteristics of the SAE 9254 high alloy steel that is typically used in such piston rings also assists in lowering engine oil consumption through a reduction in piston ring side wear and ring groove pound out. Being lighter, they provide a more effective seal against the bottom of the ring groove. Their smaller cross section, permitted by their greater strength, also improves the ability of the piston ring to conform to less-than-perfect cylinder bores.

Like ductile iron, steel is not entirely compatible with cast iron cylinder walls, so it must be either chromium-plated, or gas nitride-hardened, or face-coated with plasma molybdenum that is inserted into a recess in the face of the piston ring. Steel piston rings are made from preformed steel wire, much in the same fashion as the steel rails of oil control rings are. Most of the steel piston rings currently in production have a width of 1.2mm (0.0472”). Some are as small as 1.0mm (0.0393). 1.2mm steel piston rings are usually barrel faced, having contoured Outside Diameters which gives the piston ring a center contact with the cylinder wall. The extremely narrow 1.0mm piston rings usually have a tapered face. The 1.2mm piston rings are approximately as thick as two oil ring rails stacked together, so there is not a lot of space in which a groove can be machined for a molybdenum facing. That is why manufacturers choose to usually either chromium plat or gas nitride-harden such piston rings. The amount of machining that is required in order to finish a steel piston ring is far less than that which is required to finish cast grey iron or ductile iron piston rings, so steel piston rings are actually less expensive to manufacture, at least in large batches.

According to most piston ring manufacturers, steel and ductile iron piston rings can be considered virtually interchangeable as far as rebuilding most passenger car gasoline engines is concerned. So if a steel replacement piston ring is not available for a certain application that uses a steel piston ring as original equipment, you can usually substitute a ductile iron piston ring. However, you should never ordinary cast grey iron piston rings as a substitute for ductile iron or steel top compression piston rings because the cheaper piston rings will not hold up nearly as well in a high performance application.

Along with the change to steel piston rings for high power output engine applications may come another new technology: gas nitriding. Gas nitriding, which should not be confused with the black phosphate coating that is currently used on most piston rings in order to prevent rust during shipping and storage, is a heat treatment process that impregnates the surface of the metal with nitrogen in order to case harden the metal. When used on piston rings, it case hardens the entire surface of the piston ring to a depth of about .005" which greatly improves its resistance to side wear as well as face wear. Gas nitride-hardened piston rings have a hardness of about 68 HRC, which translates into about 1100 on the Vickers scale. This is almost 50% more than steel piston rings and four times that of cast grey iron piston rings! These piston rings are so hard that ring wear is virtually nonexistent. In fact, the cylinder walls will wear out long before the piston rings will.

The dimensional setting of piston ring gaps is often a confusing and misunderstood art. There are minimum and maximum piston ring gap specifications that must be observed for the best performance of a new set of piston rings. Minimum gap tolerances must be observed in order to prevent the piston ring ends from butting together while the piston ring expands as the engine approaches operating temperature. The Society of Automotive Engineers recommends a minimum of .0035" gap per inch of cylinder diameter. For example, the proper minimum piston ring gap for a set of +.020" oversize pistons would be: [(3.180") + (.020") bore clearance] x .0035"] = .011" minimum gap. Maximum piston ring gap is an important part of piston ring performance in that too much gap results in lost compression, power loss and ultimately poor oil control. Manufacturers rigidly adhere to these tolerances and the piston ring gaps are inspected in gauges accurate to .0001" (.00762mm) at the cylinder diameter the piston ring is manufactured for. An increase in the cylinder diameter beyond the designated size that the piston ring is designed for results in an increase of approximately .003" (.0762mm) in piston ring gap for each .001" (.0254mm) increase in cylinder diameter. For this reason, each piston should be individually matched to its most appropriate diameter cylinder bore with a torque plate installed onto the deck of the block, and the gap of each piston ring should then be measured only in the particular cylinder bore in which it is going to be used. In order to check piston ring gap, the piston rings should be placed at the lowest possible part of the cylinder below the piston ring travel area, as this is the portion of the cylinder for which the piston ring is sized. The gap of the compression piston rings should be precisely set by carefully working with a fine toothed ring filing tool. The gap should be filed in an inward direction and square to the sides. Oil control piston rings should never be filed. If their gap should prove to be less than .015"(.381mm), then they should be exchanged. It should be noted that if the pressure between the second compression ring equals or exceeds the pressure above the top compression ring, it could cause the top compression ring to lift off of the bottom of the piston ring groove and lose contact with the sealing surfaces, resulting in a loss of compression and "blow-by". This also inhibits the ability of the rings to transfer heat away from the piston. Most amateur engine builders (and some tradesmen) are unaware that the gap of the second compression ring should be .004" (.1016mm) greater than that of the top compression ring in order to vent away gases and thus increase the top compression rings' ability to seal off the combustion gases, especially at high engine speeds. This larger escape path reduces inter-ring pressure and thus assists in keeping the top compression ring seated against its groove, preventing combustion pressure from escaping. Without this adequate escape path, the trapped pressure beneath the compression ring will nearly equalize with that above it, thus allowing the compression ring to become unseated from its seal against the bottom of its groove as the piston travels down the bore. These conditions will cause reduced cylinder sealing and ring flutter at high engine speeds. In addition, a fluttering ring cannot transfer heat away from the piston to the cylinder wall, a condition that will be aggravated by the blow-by of combustion gases blasting the heat-conducting film of lubricating oil on the cylinder bore downwards, away from where it is needed at the worst possible moment in the piston's cycle of travel. These conditions can result in piston overheating, top piston ring groove "pound-out", piston ring side-wear, and scuffing. If the gap is not perfectly both symmetrical and vertical, blow-by and blowback gases may generate thrust, causing the piston ring to rotate inside of its bore.

Check the end gap of the compression rings by both oiling and inserting each of them into the cylinder into which they will each be installed. This must be done after squaring the compression rings to the cylinder wall by turning a piston upside down and using the top of the piston to push each of the compression rings with down into the cylinder approximately one inch. Measure the end gap with a feeler gauge. Check this against the end gap specification chart at the back of the Deves Piston Ring catalog or on the instruction sheet that is included in each set of piston rings. If these gaps are too small, the ends of the piston rings need to be filed. On the other hand, if the gaps are too large, you may have the wrong size piston rings, or your cylinder may have been accidentally bored oversize.

Secure the butt end of a small sharp file into a vise. File only one end of the piston ring in order to allow you to verify that you are keeping the gap straight and parallel. File from the outside face toward Inside Diameter (I.D.) in order to avoid chipping the coating on its face or leaving burrs on the edges of the Outside Diameter (O.D.). Remove any burrs created by the gapping process with a fine stone. This will produce a smooth finish that will preclude the formation of stresses in the material that can lead to fracturing.

Piston Oversize	**Top Compression Ring Gap**	**Second Compression Ring Gap**
+.020”	.0112”	.0152”
+.030”	.0112”	.0152”
+.040”	.0113”	.0153”
+.060”	.0113”	.0153”
+.080”	.0114”	.0154”
+.100”	.0115”	.0154”
83mm (3.268”)	.0114”	.0154”
83.57mm (3.290”)	.0115”	.0155”

Take care to keep the rings and cylinders consistent with each other as there might be some variance in the diameter of the individual cylinders and the compression rings must be individually gapped as per the cylinders. There is no need to check the end gaps on the Deves four-piece oil ring. The rails are made for specific bore sizes and are pre-gapped at the factory. Just make sure you have the correct set for your particular application.

It is important to use proper piston ring installation tools and to exercise care when opening the piston ring prior to putting it onto the piston. If the gaps of the piston ring are not kept strictly in line with each other, then the piston ring will be bent or otherwise distorted. In order to perform properly, a compression ring must seal not only on the face of its Outside Diameter (O.D.), but also on its side faces as well. If the compression ring is bent it offers a leakage path for both the blow-by of combustion gases as well as oil. This is due to the fact that both the piston rings and the piston are engineered to complement each other from a standpoint of clearance between the back of the piston ring and the bottom of the piston groove. This clearance is referred to as back clearance. If the clearance is insufficient, severe engine damage can result when the piston and rings are installed into the engine, so they should always be checked before assembly. To check compression ring back clearance, place the outer edge of the compression ring fully into its land. No portion of its Inside Diameter (I.D.) should protrude beyond the piston land. The back clearance of the oil rings are more difficult to check. However, by placing both the oil rails and the expander together as they would be assembled and inserting them into their groove, it can be determined if back clearance exists. The minimum back clearance should be .015" (.381mm). Be aware that pistons are designed to have considerably more clearance in the piston ring land area than in the skirt area.

Be aware that due to advances in metallurgy, some manufacturers have reduced the cross sectional size of their piston rings over the years. These thin piston rings are better able to cope with bore flexure and the out-of-round bore profiles that result from wear, thus enabling better sealing under adverse conditions, permitting less blow-by of combustion gases. In reducing the cross-section, it becomes necessary to reduce the depth of the groove in order to facilitate installation of the oil ring assembly. However, some piston manufacturers have chosen to stay with a deep groove piston, while in other cases they have matched the Original Equipment piston and used a shallow groove. The depth of a deep groove will generally be .190" (4.826mm) or more while that of a shallow groove will be .190" (4.826mm) or less. To cover the possibility of differing two groove depths in one engine application, piston ring manufacturers have had to issue two differently dimensioned piston ring sets for the same engine. If a deep groove piston ring set is used on a shallow groove piston, then there is a good possibility the piston ring will bottom in the groove and result in severe engine damage. To determine if this condition exists, install the oil ring in the groove in the normal manner. Push the assembly in as far as it will go into the land. If the rails protrude, the oil control ring is incorrect for the groove. A straight edge held against the rails and squared along the piston will aid in revealing if this condition exists. If it does, do not install it. Consult the manufacturer to obtain the correct set of piston rings. If a shallow groove oil control ring is used in a deep groove piston, installation of the oil control ring onto the piston will be difficult and the oil control ring assembly will "pop off" of the piston, resulting in severe engine damage.

When assembling the piston rings onto the piston, always work from the bottom to the top. This means that the oil control ring must be assembled into its bottom groove first. Be aware that if the expander gap and rail gaps of the oil control ring are installed in staggered positions, rather than in aligned positions, then the installation will be much easier. Position the oil control ring expander in the center of the groove and align the gap of the oil control ring expander with the axis of the wrist pin. In a spiral motion, place the bottom Rail over the oil control ring expander at the bottom edge of groove. Align the gap of the bottom rail with the side of piston that is opposite its thrust side. In a spiral motion, place the spacer over the expander and above the bottom rail. Align the gap of the spacer with the axis of the wrist (gudgeon) pin, on opposite side of piston from the gap of the oil ring expander. In a spiral motion, place the top rail over the expander and between the spacer and the top of the ring groove. Align the gap of the top rail with the side of the piston that is 180^{o} opposite from the gap of the bottom rail.

Prior to installing the first and second compression rings onto a piston, always examine them carefully. Piston rings that have a "pip" mark or dot on the side of the compression ring must always be installed with the "pip" mark or dot towards the top of the piston. Compression rings that have a bevel on their Inside Diameter (I.D.), but no "pip" mark or dot, must be installed with the bevel towards the top of the piston. Compression rings that have a groove in the Outside Diameter (O.D.) and no "pip" mark or dot must be installed with the groove toward the bottom of the piston. Compression rings that have no dots, bevels, or grooves can be installed either way. Note that after installing all rings onto the pistons it is always a good idea to recheck each piston ring on each piston for correct installation.

A piston ring expander should always be used whenever installing the compression rings onto the piston. Expand the piston ring just far enough to slide down and fit into the grooves of the piston; compression rings are flexible to a point, but beyond that point, permanent distortion will occur that will prevent the rings from functioning properly. Installation of the piston / ring assembly into the engine block requires the use of a piston ring compressor (A ratchet type compressor is recommended for Deves piston rings) in order to lessen the risk of damaging the cylinder walls or the piston rings.

Before using the piston ring compressor, check the installing band for nicks, dents and other damage by running a thumb and forefinger completely around the band. When preparing to install a piston, position the gap on the top compression ring perpendicular to the cylinder wall on the distributor side of the engine, then stagger the rest every 120º. Once the piston is set into the cylinder with the piston ring compressor installed, it common practice to tap on the top of the compressor in order to "square it up" to the piston. The usual procedure is to then use the handle of the hammer to tap the piston home. However, this is a mistaken procedure. The inside area of the compressor band becomes rolled over, nicked, etc. and causes damage to the piston rings when they are installed. Instead, after the piston is set into the cylinder, the piston should be gently tapped with the handle of the hammer in order to square up the piston. The piston can then be gently tapped or pushed into the cylinder while holding the piston ring compressor firmly against the top of the cylinder. Do not rush, as damage most frequently occurs at this step. It is not uncommon for an oil ring rail to become caught on the edge of the cylinder when the piston is installed. Forcing it into the bore will damage the rail, requiring a replacement.

Although there are enough manufacturers of piston rings to confuse the novice engine builder as to which set of piston rings would be best to procure for use in his personal engine, there is one manufacturer whose reputation for product quality is widely known and highly respected: Deves. Deves has been manufacturing piston rings since 1928 and offers a large product line. They can rightly be called masters of advanced metallurgy, creating special composition alloys that are designed for high heat transfer, high flexibility, and low wear rates.

Compression

Should you decide to raise the compression ratio by milling the cylinder head, be sure to caution the machinist to be careful to not skim too closely to the ¼"-28 UNF bottom mounting hole for the heater valve. However, be warned that it is unwise to go much over a 9:1 Geometric Compression Ratio (GCR) with unmodified combustion chambers or you will most likely regret it when it preignites on America's newest federally-mandated development: Oxygenated Gasoline. Without professionally modified combustion chambers, any increase in Geometric Compression Ratio (GCR) beyond 9.5:1 with a cast iron cylinder head will give only a modest increase in power at the expense of streetability. Typically, Geometric Compression Ratio (GCR)s on the order of 9:1 to 9.5:1 are acceptable for use in engines fitted with cast iron cylinder heads and most mild street camshafts, while engines fitted with aluminum alloy cylinder heads can tolerate 10-10.5:1 Geometric Compression Ratios (GCRs) under the same operating conditions. In milling the cylinder head in order to increase compression ratio, be advised that the 43cc heart-shaped combustion chambers of the 18G, 18GA, 18BG, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines have a depth of .425" (10.8mm). This being the case, removing .0099" (.25146mm) of material will result in the reduction of the volume of the combustion chamber by 1cc. On the other hand, the kidney-shaped 39cc combustion chambers of the 18V engines have a depth of .375" (9.5mm), requiring the removal of .0096" (.24384mm) of material in order to result in the reduction of the volume of the combustion chamber by 1cc. Because it is common to find that the sealing surface of cylinder head has been previously skimmed, it is important to use a depth gauge in order to establish the actual volume of the combustion chamber. However, it is far better instead to seek an increase in compression ratio by milling the deck of the engine block as this will result in the piston being closer to the cylinder head, thus creating more optimal squish (quench) characteristics, as well as requiring a shallower depth for the valve clearance counterbores should a high-lift camshaft lobe profile or oversize valves be used.

The effect of compression ratio on preignition and detonation is greatly dependent on the duration and intake closing point of the camshaft. Engines with hotter camshaft lobe profiles can tolerate higher compression ratios due to the fact that at low engine speeds, as a result of their greater period of overlap characteristics, some of the fuel / air charge escapes out of the exhaust valve before it fully closes, thus reducing compression. This explains why engines fitted with such camshafts are typically equipped with higher compression pistons. The increased Geometric Compression Ratio (GCR) is necessary in order for the smaller fuel / air charge to be efficiently burned. The easiest way to determine if a particular camshaft is compatible with a particular compression ratio is to measure the compression while cranking the engine. On pump gasoline, anything over 180 PSI can be a problem when using an Original Equipment cast iron cylinder head with unmodified combustion chambers.

Rocker Arms

After having had the mating surface of the cylinder head skimmed flat, be sure to also have the top of the cylinder head skimmed flat and parallel to the plane of the bottom mating surface of the cylinder head. This is necessary for proper alignment of the rocker arm assembly. Upon reassembly, take care that you insert .0045" shims (BMC Part # 12H 3960) under each of the two center rocker shaft pedestals in order to impart a very slight arc to the rocker arm shaft. This slight arc is well within the elastic limit of the rocker shaft. This will aid in preventing excessive wear of the rocker shaft pedestal bores by keeping the rocker shaft from rotating and moving fore and aft atop the cylinder head. The problem is caused by having the rocker shaft locked at one end rather than at its middle. Replacement of the rocker arm spacer springs with tubular steel rocker arm spacers (Special Tuning Part #s AEH 764 and AEH 765) is highly advisable when using a high lift camshaft that extends the powerband beyond 6,500 RPM in order to preclude 'walking' of the rocker arms at such high engine speeds. Tubular steel rocker arm spacers are normally found only on race engines, but their omission in a street engine that is intended to be operated at higher than normal engine speeds can be a mistake. Because a race engine is torn down and carefully inspected quite often, there is no problem with crud accumulating inside of the tubular steel rocker arm spacers as well as between their shims and the rocker arms. The factory used spring spacers as Original Equipment partly because of their quiet operation and partly in order to allow crud to be washed free from the rocker shaft. Due to their potential for inducing rapid wear on the side faces of the rocker arms and the rocker shaft pedestals should their shims wear through before being replaced, they are rarely found on a street engine. Their .004" +/- .001" (.1016mm +/- .0254mm) gapping should be checked with a Plastigauge every time that the valves are adjusted in order to monitor for this eventuality.

The 18V models of the B Series engine progressively underwent several changes in order to reduce production costs. Amongst these was the deletion of the oiling passages in both the rocker arm and in the tappet adjusting screw which had provided ample lubrication to the cupped upper end of the pushrods. There are two small passages inside of each of the previous-style rocker arms (BMC Part # 12H 3377). One runs laterally through the rocker arm from the rocker arm bushing to the adjuster screw passage, and is plugged with a rivet at its outer end on the mounting boss of the adjuster. Oil flows through this passage from the bushing out around the waist of the adjuster screw, through a radial passage into the center of the screw, and out from a passage in the bottom end of the screw in order to lubricate the ball and socket interface at the top end of the push rod. The second oil passage in the rocker arm exits at an angle from the top shoulder of the rocker bushing area so that oil will be sprayed onto the thrust face of the rocker arm. This oil lubricates the thrust face of the rocker arm and the tip of the valve stem, as well as the top of the valve stem in order to lubricate its interface with the valve guide. This second passage was retained on the later-style rocker arms. These 18V engines had a camshaft (BMC Part # CAM 1156) that was made with the same lobe profile as before, but its position was advanced by 4°. In its initial versions with a cylinder head (BMC Part # 12H 2520) that had a larger (1.625" / 41.275mm diameter) intake valve and larger intake ports than previously used in order to compensate for the retiming of the camshaft, the 18V-584-Z-L, 18V-585-Z-L, 18V-672-Z-L, and 18V-673-Z-L versions made slightly more power at the same engine speed as the previous 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines. However, the subsequent versions reverted to the smaller original 1.565" diameter intake valve size, which subsequently relocated both the torque and the horsepower peaks to a lower engine speed range and thus the additional lubrication provided by the passages in the rocker arms and ball end adjusters were deemed unnecessary.

However, in an enhanced-performance engine with radical lift camshaft lobe profiles, the fitting of these later, stronger specification items (APT Part # RAS-2) can, in some cases such as when the later short tappet / long pushrod assembly is used in concert with the taller heads of the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines, be a wise move for prolonging the lifespan of these components due to their decreased angle of deflection reducing the probability of breakage at high engine speeds.

The B Series engine has long had a reputation for tappet clatter. While mechanical tappets are indeed noisier than hydraulic tappets, if your engine sounds as if a Spanish castanet dancer is inside of it, the principle reason for the noise level is not the mechanical tappets, but the result of wear of the rocker shaft and the rocker arm bushings. This is the most wear-prone system in the B Series engine. As the pushrod lifts the fulcrum end of the rocker arm, the bottom of the inner face of the bushing is also lifted upwards against the rocker shaft. The resulting pressure loadings can become quite heavy, especially when high-lift camshaft lobe profiles are compounded with high engine speeds, inducing accelerated wear. This is worsened if both the oil and the oil filter are not changed regularly because metallic particles will become embedded into the surface of the rocker arm bushings, thus hastening wear. As wear progresses, a notch develops on the rocker shaft where the rocker arm bushing slides against it and the bottom of the inner face of the rocker arm bushing gradually becomes pear-shaped as well. The resulting slop in the valvetrain thus induces tappet clatter. Being fed from the low pressure gallery through the rear rocker pedestal and then into the rear end of the rocker shaft, a worn and leaking rocker arm system delivers progressively decreasing amounts of oil to the rocker arm bushings as the oil flow travels toward the front end of the rocker shaft. In order to reduce wear, the hardened surface of a nitride-hardened rocker shaft is highly desirable. In addition, when purchasing new rocker arm bushings, try to avoid purchasing ones made of the softer, faster-wearing Silicon bronze alloy. Silicon bronze alloy also has a problem with embedability, its surface becoming impregnated with stray particles of material that a cheap oil filter has failed to remove. This in turn results in wear of the rocker shaft. Instead, use ones that are made of Manganese bronze alloy. When they arrive, be sure to inspect them to be sure that the oil groove is present and that they are of the correct Outside Diameter (O.D.) of .7490" +/- .0005" (19.0246mm +/- .0127). If your rocker arms do not have the additional oiling passageways for lubricating the valve stem and the ball end of the adjuster, the necessary holes in the bushings can be created with a #47 drill bit. Afterwards, ream them to an Internal Diameter of .62775" +/- .00225" (15.9449mm +/- .05715mm). The rocker arm bushings must be pressed into the bore in exactly the same position as the Original Equipment rocker arm bushing. The factory specifies that the bushing split should be at the top, which leaves the oil trough at the bottom, thus providing an oil film under operating conditions. On one side of the bushing a hole is drilled to accept a 1.6mm OD x 6mm solid pin. Installation of this pin ensures that the bushing will be prevented from rotating in its bore. For all new or rebuilt engines with domed tappets, proper Molybdenum Disulphide (MoS2) Extreme Pressure assembly lubricants and a ZDDP (zinc dithiophosphate) oil additive (Moss Motors Part# 220-815) should always be used during the break-in phase. This additive provides extra protection at the point of contact, helping the domed face of the tappet to properly mate with the lobe of the camshaft. Once the break-in phase is over, the use of this additive should be discontinued.

High-performance modified engines benefit from oils with superior film strength and anti-wear properties. The interface of the domed tappet and the lobe of the camshaft is the one area of an engine that has an extreme Hertzian contact load. That load increases significantly when high-pressure valve springs or higher-lift camshaft lobe profiles are employed. The use of properly formulated engine oils for this application will help reduce wear and extend the service life of both the domed tappets and the camshaft lobes. There are many more ways to achieve good anti-wear performance than simply adding zinc and phosphorus compounds alone. Zinc and phosphorus are widely used because they are the most cost effective solutions to achieve anti-wear properties.

Be careful to maintain the symmetry of the thrusts of the opposing ends of the rocker arms in order to avoid excessive side thrust loadings of the valve stems against their valve guides and of the rocker pads against the tip of the stem. Failure to do so will result in accelerated wear of the valve stem and the bore of the valve guide, as well as of the rocker arm bushing and the rocker shaft.

A precise technique for establishing this symmetry is to use a depth gauge in order to obtain the needed installed valve stem height measurement. It will be easiest to make these measurements by inserting and removing the valves in their valve guides one at a time without their valve springs in order to provide clearance for the depth gauge. This calculation can be accomplished by subtracting the amount of material that was removed during the of the cylinder head from the original height of the cylinder head casting, and then adding the measured height of the installed valves, and then factoring in the difference between the installed height of the original valves and that of the new valves. .0045” shims (BMC Part # 12H 3690, Moss Motors Part # 460-255) can then be used in order to create the optimum height location of the rocker shaft. Those who lack precision measuring instruments can also use a less precise alternate technique. The thrust faces of the rocker arms should be laterally central over the valve stem and centered longitudinally on the tip of the valve stem when at half lift. This can be easily established by placing machinist’s bluing on the tips of the valve stems and examining their print marks on the thrust faces of the rocker arms. If necessary, shims can be used between the rocker arm and the rocker shaft pedestal in order to create this symmetry. Should it prove to be necessary to shim the rear rocker shaft pedestal in order to achieve the desired height, note the position of the oiling hole on the bottom of the rocker shaft pedestal and modify the shim(s) accordingly so that the rocker arm bushings and rocker shaft will receive adequate lubrication.

Note that on the rear rocker pedestal there is a small screw (BMC Part # 2A 258, Moss Motors Part # 460-270) that is held in place with a locking plate (BMC Part # 2A 259, Moss Motors Part # 460-280), maintaining the proper alignment of the oil feed passage inside of the rocker pedestal to the one in the rocker shaft. Movement of the rocker shaft and consequent misalignment will result in oil not flowing through into the shaft and thence onward to the individual rocker arm bushings.

When installing the rocker arm studs, be aware that they have different threads on each end. The top end of the stud is 5/16”-24 UNF, while the lower end has 5/16”-18 UNF threads. Do not succumb to the temptation to install a set of aluminum rocker pedestals from the smaller displacement version of the B Series engine that was used in the MGA. Because of the greater coefficient of expansion / contraction of aluminum, setting of valve clearances will be a problem. Aluminum being a soft material, these pedestals have a tendency to either gradually spread under load or to collapse upon being over-torqued, causing inadequate clamping force to be exerted upon the cylinder head gasket. A blown cylinder head gasket will be the subsequent result. As if that is not bad enough, distortion of their bores also distorts the rocker shaft, leading to its eventual breakage. The top of the pedestal is relatively thin where it passes over the rocker shaft, making it prone to cracking. If a thin or non-hardened washer used, the top of the aluminum pedestal will be deformed with a slight depression, which in turn will cause small stress cracks to form in the aluminum pedestal, the eventual result being that the pedestal will break, subsequently allowing the rocker shaft to lift upwards as the pushrods lift the adjacent rocker arms. This upward bending of the end of the shaft will dramatically increase the stress on the nearest pedestal, which is then very likely to fracture as well. Small wonder that these rocker pedestals were discontinued with the advent of the 1800cc version of the B Series engine.

Because the rotating side faces on the sides of the rocker arms have mated to their adjacent rocker shaft pedestals over the years, be sure to keep them all in the same order as that in which they were previously installed or you may have problems aligning the centerline of the thrust faces of the rocker arms over the valve stems. If the centerlines of the rocker thrust faces are not centered over the valve stems, uneven wear of the rockershaft (BMC Part # 11G 62) and rocker arm bushings (BMC Part # 1G 2295) will result, shortening their lifespan. Never reuse the old rocker arm spacer springs (BMC Part # 6K 871) when rebuilding an engine. If they are weak, the rockers will “walk” on the rocker shaft at high engine speeds, and the rocker shaft as well as the rocker arm bushings will both wear more rapidly. In severe cases of wear, it is possible for a pushrod to become disengaged from its cup on the adjuster of the rocker arm, often with severe engine damage following immediately thereafter.

Rocker Arm Ratios

Fortunately, an increase in rocker arm lift ratio is a relatively simple approach which, as an alternative to changing the camshaft and ignition timing, is considered by some to be one of the better options for increasing power (aside from headwork). The use of a set of high lift ratio rocker arms will allow the valve to open further without changing the opening and closing points, as well as also keeping the valve closed during periods when it would be desirable, thereby increasing cylinder pressure and making for a broader, and hence more tractable, increase in power than would possible with the use of longer-duration valve timing. The advantage of this more expensive alternative to changing the camshaft is that because the valves will still open and close at the same time as before, you can retain your Original Equipment specification ignition curve while gaining as much as roughly 10% more power.

Because the change in lift ratio also increases both the spring and inertia loadings of the pushrods, the use of the more rigid tubular chrome-moly pushrods should be considered to be mandatory. In addition, a special camshaft with an altered Lobe Center Angle will be necessary. This is due to the fact that while the breathing of the engine is improved, the valve timing has remained unaltered, resulting in insufficient time available for exhausting the larger quantity of combustion gases. This can often be resolved by increasing the Lobe Center angle by 2° to 4°. Due to this being an expensive modification, this method of attaining more power output is normally resorted to only after a three-angle valve job and professional headwork. Should you choose to employ this method, you will find that it complements a three-angle valve / valve seat and headwork well.

There are three different basic types of high lift ratio rocker arm systems. The first type is the simplest, consisting of the use of rocker arm bushings with offset bores. These are sometimes referred to as "eccentric bushings." Using this approach, it is possible to achieve an increase in valve lift. They must be press-fitted into the bore of the rocker arm with their thick section oriented toward the valve. It is advisable to use a bearing-fit compound such as Loctite in order to fill out bore wear in the old rocker arms as well as to ensure that the bushing does not rotate inside of the rocker arm while under load. Once installed, they need to be reamed to an Internal Diameter (I.D.) of between .616" to .620" (15.6464mm to 15.748mm) for the optimum fit on the rocker shaft. However, in order to achieve the correct alignment of the thrust face of the rocker pad over the valve stem, it is necessary to use a set of rocker pedestals that also has offset bores. Because this approach to attaining increased valve lift usually produces only a 5%-10% increase in power, it should be considered to be the least cost-efficient.

The second and least expensive type of system consists of a set of rocker arms in which the lift ratio is increased by means of a shorter pushrod lift arm. While this simple approach permits the use of the Original Equipment rocker shaft pedestals, due to the necessarily increased inclination of the pushrods, it has the disadvantage of increasing the side thrust forces on the tappets as well as on the valve guides and valve stems, resulting in accelerated wear of the valve train. When used in conjunction with high lift camshafts, it is possible to damage the tips of the valve stems. In addition the unsupported ends of the rocker shaft can suffer from the effects of flexure. With this type of high lift ratio rocker arm, you will also need to relieve the pushrod passages in both the cylinder head and the deck of the engine block in order to avoid bending a pushrod. Extreme care will be needed in doing so in order to prevent breaking through the walls of the coolant passages. In order to achieve this, the pushrod passages should not be bored, but instead should be elongated toward the centerline of the engine to a length of .660" (.764 mm).

The third and most expensive type is a system which uses special rocker shaft pedestals of which the axis of the rocker shaft is relocated to a new offset position that is further away from the valves, as well as different rocker arms in which the lengths of the arms have been altered in order to achieve the desired increase in valve lift while reducing the side thrust forces on the tappets by keeping the pushrods closer to their original orientation than in comparison with the simpler, short lift arm system. Because the valve arm of the rocker has a longer radius to its arc of travel, side thrust loads on the valve stem are reduced in comparison with the simpler short lift arm system, thus keeping valve guide and valve stem wear within acceptable limits for street use. While obviously the most expensive, this is the preferred system for long-term use. If you decide to employ this type of rocker arm, make sure that they use bushings to ride on the rocker shaft. Needle-roller bearing rocker arms are a for-race-only item due to their short operational life. Before installation, look to see that the oiling grooves of the bushings are on the bottom and that their ports are aligned properly with the oil passageways. Once installed, the bushings will need to be reamed to an internal diameter of .616" to .620" (15.6464mm to 15.748mm). These rocker arms are manufactured by Piper and are available from Brit Tek (Brit Tek Part # PRROO1). Due to their having a higher lift ratio (1.625:1) than that of the Original Equipment rocker arms (1.426:1), these will achieve the goal of opening the valves further (about 14%) and more rapidly, but will require either lightening of the mass of the valvetrain or, as an alternative, the fitting of stiffer valve springs in order to handle the greater stresses that will result from the increased acceleration loads of the valvetrain mass, the latter approach to the problem also increasing pressure loadings at the camshaft lobe / tappet interface by 14% beyond that created by an Original Equipment ratio rocker arm. In either case, matching of the valve spring rates to the inertial needs of the valvetrain is critical. If the springs are too strong, then rapid wear of the interface of the tappets and lobes of the camshaft will result. If the valve springs are too weak, then the tappets will hammer against the lobe of the camshaft and both will be quickly ruined. These rocker arms also make use of tubular spacers rather than the Original Equipment spring spacers in order to preclude "walking" of the rocker arms at high engine speeds.

If you are installing high lift ratio roller rocker arms in conjunction with a set of 5/16" tubular pushrods, it is important to take care that the pushrod cup does not contact the underside of the fulcrum end of the rocker arm. This contact commonly occurs at full valve lift if the rocker clearance adjusting screw is screwed too far out, leaving very little of the adjusting screw underneath for permitting pushrod cup articulation. Therefore, in order to prevent this contact, care must be taken that there is an adequate portion of the adjusting screw protrudes from beneath the rocker arm.

It needs to be understood that if high lift ratio rocker arms are employed, then the valve adjustment clearance gap used will need to be revised by whatever the lift ratio increase is. That is, if an Original Equipment specification camshaft is employed in conjunction with a high lift ratio rocker arm that gives a 14% increase in lift, then the Original Equipment clearance gap specification of .015” (.381mm) would need to be increased by 14% to .017” (.4318mm). The clearance gap used with a current-production Piper BP270 camshaft is set at .013” (.3302mm) for the intake valves and .015” (.381mm) for the exhaust valves when used with Original Equipment specification rocker arms and thus their clearance gap would need to be revised to .015” (.381mm) for the intake valves and .017” (4318mm) for the exhaust valves in order to accommodate a high lift ratio rocker arm that gives a 14% increase in lift.

Many of these systems make use of a roller bearing on the valve end of the rocker arm to reduce friction. Although the bodies of such rocker arms are almost invariably made of high strength aluminum alloys, the heavy steel roller bearing located at the end of the rocker arm results in them actually having greater rotating mass. Consequently, the use of the both lighter and more rigid tubular chrome-moly pushrods, lightweight valve spring retainer caps (cups), and late model 18V bucket tappets are also advisable in order to contain valvetrain inertia whenever such rocker arms are employed. There is a notable disadvantage to the use of roller rocker arms. Because the roller contacts the tip if the valve stem across a very narrow area, the pressure loading exerted on the area of contact is so great that it prevents the valve from rotating during its initial break-in period. This is evidenced by the fact that the rollers make a mark on the tip of the valve stem over a period of time. If the valves were rotating, then the wear pattern would be radial rather than the lateral pattern observed on the tips of valve stems actuated by a roller rocker. Due to the lack of rotation, the life expectancy of both the valves and that of the valve seats is shortened. Be aware that when doing a valve clearance adjustment on roller tipped rocker arms, you have to insert the feeler gauge from the side, not the front as you would with regular rocker arms.

Do not try to combine high ratio rocker arms with a high lift camshaft lobe profile such as that employed in the Piper BP285 design. At full lift, the rocker arm geometry will create excessive side thrust loadings on the valve stem that will also result in galling of the thrust face of the rocker arm. If a roller rocker arm is used, eventual breakage of the pin that supports the roller is inevitable. In any event, even with 100% volumetric efficiency, there would be no advantage in terms of airflow potential to opening the valve such an extreme amount since the maximum airflow through any valve occurs when it has been lifted to a height that is 25% of its diameter. This is due to the fact that the curtain area of the valve then equals its diametric area. Due to the horizontal port configuration inherent to the Heron-type cylinder head, with the possible exception of a highly modified large-valve racing cylinder head, there is no worthwhile advantage in terms of airflow to a valve lift of more than .455" (11.557mm).

The Original Equipment rocker arms use a very wide thrust face with a very gentle thrust face radius that is quite accommodating when it comes to minor variations in the length of the pushrod. However, a roller thrust face concentrates the entire thrust loading onto a very tiny area of the tip of the valve stem. Given this fact, it becomes very important to position the roller properly over the tip of the valve stem. If the pushrod is not of the correct length, the roller will not be properly located at the center of the tip of the valve stem at half lift. Obviously, the possibility that the engine may require custom-length pushrods must be explored. Crane makes an adjustable-length pushrod for just this purpose.

Even with proper pushrod length, the roller starts from the outer side of the rocker. As valve lift begins, the roller moves toward the center of the tip of the valve stem, reaching the center of the tip of the valve stem at approximately half of maximum valve lift. As valve lift continues, the roller moves past the valve center toward the inner edge of the valve stem at maximum valve lift. Then as the valve begins to close, the roller retraces its movement back to where it began. With improper pushrod length, the roller starts in the wrong position and travels much farther, increasing sidethrust, which will accelerate wear of the valve guide, especially in the case of increased rocker arm ratios, which create an increased side load on the valve stem. The only way to minimize this is to minimize the amount of sidethrust caused by the rocker arm. The best way to accomplish this is to establish an optimum geometry by installing pushrods of the proper length.

Wait until you have assembled the engine for the last time to check for pushrod length so that a change to installed height will not require a pushrod change. These are just some of the variables that will effect pushrod length: engine block deck height, rocker shaft pedestal height, rocker arm ratio, a smaller or larger base circle camshaft, differing intake and exhaust camshaft base circles, valve installed height, and variations between manufacturers' rocker arm designs.

In order to check for proper pushrod length when using roller rocker arms, all that you really need is machinist's bluing, a degree wheel, and a method of turning the crankshaft. After adjusting the clearance between the roller and the tip of the valve stem, clean the top of the tip of the valve stem, and then paint the roller with a thin smear of machinist's bluing. Rotate the crankshaft until the rocker arm is at half lift. Remove the rocker arm and examine the position of the stain of the bluing on the tip of the valve stem. If the pushrod length is correct, the edge of the machinist's bluing should be located on the center of the tip of the valve stem. If the edge of the machinist's bluing is off-center towards the camshaft side of the engine, the pushrod is too short. If the edge of the machinist's bluing is off-center towards the distributor side of the engine, then the pushrod is too long.

The best way to determine the proper length for a custom pushrod is to use a Crane adjustable-length pushrod. With this tool, you merely adjust the length of the pushrod until the bluing stain at half valve lift is on the center of the tip of the valve stem, and then measure the length of the pushrod in order to establish the specification to which your new pushrods need to be fabricated.

Should you choose to install a camshaft that will produce greater valve lift, then the cylinder head should be flow-tested in order to determine at what amount of valve lift its airflow rate declines, as too much valve lift is actually counterproductive in that it will result in reduced velocity of the fuel / air charge. The reduced velocity inherently implies reduced inertia of the fuel / air charge, thus a smaller fuel / air charge will enter the cylinder. Should you choose to employ a more radical camshaft that extends the powerband to 6,500 RPM or higher, a nitride-hardened rocker shaft and stronger outer rocker shaft pedestals that support the outer ends of the rocker shaft will be mandatory in order to withstand the bending thrust loads on the rocker assembly at high engine speeds. These outer rocker shaft pedestals that support the outer ends of the rocker shaft were originally developed by the factory race team specifically for this purpose (Special Tuning Part # AEH 762, front, and Special Tuning Part # AEH 763, rear). The use of Manganese bronze alloy rocker arm bushings to deal with these increased stresses would be advisable as well. When used with the cylinder heads of 18V engines, the increased valve lift will require further counterboring of the deck of the engine block in order to prevent the heads of the exhaust valves from hitting it, as well as that of the cylinder head in order to recess the valve spring seating surfaces in order to accommodate the required longer valve springs so that they will not bind. In addition, you will need to shorten the upper section of the valve guides to provide the necessary clearances needed in order to accommodate the increased valve lift and avoid damage caused by valvetrain compression. Note that the Original Equipment valve guides have a length of 1 5/8" (1.625" / 41.275mm) for the intake valve guide (BMC Part # 12H 2222) and 2 13/64" (2.203125" / 55.9594mm) for the exhaust valve guide (BMC Part # 12B 1339).

Valve Guides

When installed, all valves and valve guides should be of equal respective heights. An Original Equipment intake valve guide has a length of 1.875" (47.625mm), while an Original Equipment exhaust valve guide has a length of 2.219" (56.3626mm). Both have an Internal Diameter of .3443445" +/- .00025" (8.74635mm +/- .00635mm) and an Outside Diameter (O.D.) of .56375" +/- .00025" (143193mm +/- .00635mm). Both have an installed height of .625" (15.875mm). Valve guides with shortened lower ends should not be used in an attempt to reduce interference with airflow through the port as this will increase the load bearing on the valve stem and result in accelerated wear, especially if greater valve lift is sought. Instead, a tapered (bulleted) valve guide should be used to accomplish the same end. A slight chamfer on both the top and bottom of the valve guides' Internal Diameter should be present in order to prevent sharp edges from wiping oil off of the stem. The bores in the cylinder head for the valve guides should all be carefully reamed smooth in order to maximize their contact areas against the shank of the valve guides and thus maximize heat transfer from the valves, as well as preventing thermal distortion of the bore of the valve guides from resulting in premature wear, or, even worse, a sticking valve. This heat transfer is important for keeping the head of the valve from becoming a "hot spot" that can trigger preignition. If the resultant bore is too large for a valve guide of Original Equipment diameter, under no circumstances should a valve guide sleeve be used to make up the difference as it will interfere with heat transfer. Use an oversize valve guide instead. Because valve guides will frequently distort when being pressed into their bores in the cylinder head, they should always be reamed to their manufacturer's recommended clearances after installation in order to assure a consistent Internal Diameter. If this is not done, distortion of the bore will increase as engine temperatures rise and possibly result in a seizure of the valve stem in its valve guide.

Many valve guides are sold as “pre-sized” and have bores of a rather large Internal Diameter in order to permit them to be slightly crushed and distorted upon installation. As a result, valve guides should be shrink-fitted into the cylinder head and afterwards each valve guide bore in the cylinder head must be precision-reamed in order to minimize the chances of galling during insertion and distortion of the Internal Diameter of the bore of the valve guide. If simply press-fitted into place at room temperature, then their bores may distort and after reaming will have too loose a clearance where distortion remains. If the shop doing the assembly cannot perform these operations, such “pre-sized” valve guides should be avoided.

Heat flow is through the Back-of-Head (piston side) by radiation during combustion and by conduction through the Back-of-Head, valve seat and valve stem during the exhaust stroke. Valves with a single 45° angle seat pass 75% of their accumulated heat through the valve seat and 25% through the valve stem to the valve guide, so be sure to use manganese silicon bronze valve guides on at least the exhaust valves in order to help get rid of the extra heat that always comes with extra power, thus preventing the valve heads from becoming “hot spots” and consequently triggering preignition. Manganese silicon bronze transfers heat some two and a half times more efficiently than the Original Equipment close-grain cast iron valve guides, resulting in greater reliability, especially when used with the higher combustion temperatures attendant with the use of lead-free fuel. They are especially valuable when used in concert with valves that have a three-angle valve seat. Because such valves have a narrower contact area with their valve seats, they are less able to transfer their accumulated heat to the valve seat, thus making improved conduction of heat out through their valve stems a significant priority, particularly when the engine is run hard. An additional advantage in their higher heat conductivity lies in the fact that the intake valves can also run cooler, thus reducing the amount of heat that they transfer to the incoming fuel / air charge. Because the charge will be cooler, it will be more dense, thus allowing for a greater amount of charge to fill the cylinder. This in turn increases potential power output.

Be aware that while bronze valve guides have the ability to run tighter clearances at operating temperatures, they do need to be reamed to a larger Internal Diameter because of their greater rate of expansion. Although this may seem implausible, consider that if the valve guide is standing alone in the air, heating will cause its Internal Diameter to increase. However, if the expansion of the valve guide is constricted by it being inside of a cylinder head that has a lower coefficient of expansion / contraction than that of the alloy of which the valve guide is composed, as in the case of a manganese silicone bronze valve guide in a ductile cast grey iron cylinder head, then the Internal Diameter of the valve guide will decrease because the metal has to have some unoccupied volume in which to be displaced by the heat-induced expansion. The Original Equipment intake valve stem diameter was .3422" to .3427" (8.69188mm to 8.70458mm), while the bore of the Original Equipment cast iron intake valve guide was .3442" to .3447" (8.69188mm to 8.75538mm), giving a minimum clearance of .0015" (.0381mm), a nominal clearance of .0020" and a maximum clearance of .0025" (.0635mm). As a general rule of thumb, use a micrometer to measure the diameter of your valve stems and then use a plug gauge to measure the internal diameter of the valve guide. After installing them into the cylinder head, always ream the bronze valve guides to the maximum clearance specified for that of the cast iron valve guide (.0025" / .0635mm). This will prevent the valves from sticking. Some shops, out of sheer ignorance or an unwillingness to purchase the correct-size reamer for a job that they rarely do, will ream the valve guides to the average diameter for a cast iron valve guide, and sticking valves will become a problem. Of course, they blame the material that valve guides are made of, never themselves.

Contrary to common belief, there are actually several types of bronze valve guides- Silicon bronze, Manganese bronze, Phosphor bronze, Tin bronze, Aluminum alloy bronze, Aluminum alloy silicone bronze, Nickel aluminum alloy bronze, and even Manganese silicone aluminum alloy bronze, each with its own inherent advantages for its intended special application. Most of these are either extruded or produced by the continuous casting method, while special low-production-volume valve guides intended for special applications are usually produced from solid stock on a lathe, then centerless ground and reamed. I prefer to use Peter Burgess' manganese silicone bronze tapered (bulleted) valve guides that he recommends always be fitted with an installed maximum internal bore size (.3447" / 8.75538mm) due to the greater expansion / contraction coefficient of the manganese silicone bronze alloy. However, it should be noted that the alloy that he uses is unique to his specification. It is compatible with either the chrome plated or the unplated valve stems of stainless steel valves, as well as tuftrided finishes. Once the engine is broken in, the surface of the valve stems will be impregnated with manganese silicone load bearing needles, which are very efficient at holding oil and reducing wear. The only cases of valves sticking with these stems have been in full-race engines that were run under the most arduous conditions. In all cases, once the engine had cooled to ambient temperatures and the cylinder head removed for inspection, the valves slid out of their valve guides as normal, and no damage to either the bore of the valve guide or to the valve stem was found. This makes these sophisticated alloy valve guides unique in their suitability for high performance applications.

Tapered (bulleted) valve guides deserve special mention at this point. They present a smaller profile to airflow and effectively increase the volume of the port without increasing its size, thus maintaining maximum fuel / air charge velocity and volume while creating less turbulence that can lead to fuel condensation and consequent loss of combustion efficiency. Also, the greater the valve lift that the camshaft lobe profile produces, the greater their contribution to increased power output. They should be considered to be a mandatory modification for even the most moderately enhanced output engine, including those with unmodified ports and Original Equipment specification camshafts.

Whenever major headwork is done in an effort to increase port flow capacity, the first major modification to be performed is the reduction or the removal of the valve guide boss inside of the port. While this projection does interfere with the flow of the fuel / air charge, it is present for a reason: to prevent the valve guide from moving in place by providing a sufficient surface area for its shank to grip. Because of the reciprocation of the valve within the valve guide, the valve guide would otherwise slowly be worked downward, further blocking the flow of the fuel / air charge within the port. While today's machining techniques are usually precise enough to size both the valve guide and its bore in the cast iron cylinder head casting so that this is unlikely to occur, this is still a considerable risk where aluminum alloy cylinder heads are concerned due to their higher coefficient of expansion and contraction when compared to those of suitable valve guide materials. However, the way to guarantee that this cannot happen is the installation of flanged valve guides. These are available from Peter Burgess.

Valve Springs

If coupled with new Original Equipment-specification dual valve springs and their valve spring retainer caps (cups), as well as their valve spring collars as used in the pre-18V engines and early 18V engines, this reduction in reciprocating mass should be sufficient to enable the valve springs to easily protect the engine from valvetrain float and valve / piston clash up to at least 6,700 RPM when used in concert with camshafts and rocker arms that have the Original Equipment amount of valve lift, plus reduce both camshaft lobe and tappet wear as a result of their inherently lower inertia loads. These valve springs should have a free length (uncompressed length) of 1 31/32" (inner springs, BMC Part # 12H 176) and 2 9/64" (outer springs, BMC Part # 12H 1679), and for proper preload they should have an installed length of 1 7/16" (inner springs) and 1 9/16" (outer springs). Not surprisingly, they also have different resistances: 72.5 lbs. for the outer springs and 30 lbs. for the inner springs. Taken collectively, all this should ensure more accurate valve timing resulting in a smoother, more powerful output at high engine speeds.

Just to make things a bit simpler, here is a chart of the parts that were used in the valve spring mechanisms of the MGB's engine:

Engine	**18G 18GA**	**18GB* 18GD* 18GF***	**18GD** 18GF** 18GG 18GH 18GJ 18GK**	**18V 581, 18V 582, 18V 583, 18V 584, 18V 585, 18V 672, 18V 673, 18V 779, 18V 780, 18V 836, 18V 837**	**18V 797, 18V 798, 18V 801, 18V 802, 18V 846, 18V 847, 18V 883, 18V 884, 18V 890, 18V 891, 18V 892, 18V 893**
Intake Valve	12H 435	12H 435	12H 2115	12H 2520	12H 4211
Intake Valve Guide	12H 2222	12H 2222	12H 2222	12H 2222	12H 2222
Exhaust Valve	12H 436	12H 436	12H 2116	12H 2116	CAM 1377
Exhaust Valve Guide	12B 1339	12B 1339	12B 1339	12B 1339	12B 1339
Circlip	1K 372	1K 372	NONE	NONE	NONE

Cotters	1K 800	1K 800	12H 2177	12H 2177	12H 2177
Spring Retainer Cap (Cup)	1H 1320 or 12H 992	12H 992	12H 3309	12H 3353	12H 3353
Outer Valve Spring	1H 111	12H 1679	12H 1679	12H 1679	12H 3352
Inner Valve Spring	1H 723	12H 176	12H 176	NONE	12H 176
Valve Spring Collar	1H 1321	12H 1321	1K 1321	12H 3354	12H 3354

* 18GB/101 to 9200

* 18GD/We/H 101 to 835

* 18GD/We/L 101 to 1,045

* 18GD/RWE/H 101 to 1,712

* 18GD/RWE/L 101 to 1,136

* 18GD/Rc/H 101 to 103

* 18GD/Rc/L 101 to 106

** 18GD/We/H 836 to 6700

** 18GD/We/L 1,046 to 7,000

** 18GD/RWE/H 1,713 to 7,000

** 18GD/RWE/L 1,137 to 7,000

** 18GD/Rc/H 104 to 240

** 18GD/Rc/L 107 to 230

** 18GF/We/H 2,159 to 13,650

* 18GF/We/H 101 to 2,158

* 18GF/Rwe/H 101 to 530

* 18GF/Rc/H 101 to 102

** 18GF/Rc/H 103 to 105

Be aware that due to the different depths of their combustion chambers and redesigned coolant passages of the 18V engines, the heads used on the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines, and those used on the 18V Series engines, are of different thicknesses. As a result, the cylinder heads used on the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines are taller (3.172" / 80.6mm) than those of the shorter (3.125" / 79.4mm) cylinder heads used on the 18V engines. As a consequence of this, their pushrod / tappet combinations have different included lengths (277mm for the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines, and 274mm for the 18V engines). As a result, if you should choose to install the later 18V bucket tappets and longer pushrods into an engine equipped with one of these earlier heads, it will be necessary to screw their rocker arm ball adjusters 3mm further towards the bottom of their travel. This will result in an increase in the effective length of the fulcrum arm of the rocker, with a consequential slight decrease of valve lift. This will also increase stress on the threaded shank of the rocker arm ball adjuster (BMC Part # 48G 207), in which case it would be expedient to install the stronger solid ones (BMC Part # 12H 3376, Advanced Performance Technology Part # RAS-2) that were originally developed by the factory's Special Tuning Department as competition parts expressly for this purpose.

Although the 18V-672-Z-L and later versions of the 18V engine sacrificed dual valve springs (BMC Part # 12H 1679, Outer, BMC Part # 12H 3352, Inner) for single valve springs (BMC Part # 12H 176) in an effort to reduce production costs, it should be remembered that due to changes in their valve timing, these later engines reached their maximum power output at the notably lower engine speed of 4,800 RPM instead of the 5,400 RPM of earlier engines, and thus spring surge was deemed to not be a problem. However, at the greater valve lifts and higher peak operating speeds that a power-enhanced engine attains, the performance of a single valve spring is inadequate to avoid either spring surge or valve bounce. Spring surge is caused by the inertia effect of the individual coils of the valve spring. At certain critical engine speeds, the vibrations caused by the cam movement excite the natural frequency characteristics of the valve spring and this surge effect substantially reduces the available static spring load. In other words, these inertia forces oppose the valve spring tension at critical speeds. This being the case, spring surge can result in a valve failing to close rapidly enough to avoid clashing with the piston on the upstroke, while valve bounce can lead to a broken valve, particularly in the collet area of the valve stem. The area above and / or below the groove of the valve stem is the first place to look for the chief symptom of valve float. If the keepers (collets) are leaving scuff marks on the valve stem above and below the keeper (collet) groove, or the edges of the collet groove are rounded, then the valve is bouncing on the valve seat and the valve keepers (collets) are separating and scuffing the stem. Spring surge can also cause bent pushrods and broken rocker arms. The inner and outer springs in a dual valve spring set are wound in opposite directions so that their harmonic vibrations can cancel each other out, thus decreasing metal fatigue and helping to resist spring surge.

Dual valve springs are thus a necessity for an enhanced-performance engine in order to control spring surge at the high engine speeds that it can achieve, especially if a higher lift camshaft that relocates the power output peak to a higher point in the powerband is utilized. In addition, as the valve opens, the pressure of the rocker pad against the stem prevents the valve stem and its attached spring retainer cap (cup) from rotating. To protect the valve spring retainer cap from being grooved by the friction of the flat ends of the valve springs and the consequent danger of fracturing, the factory provided valve spring collars with a lower coefficient of friction than those of the valve spring retainer caps (cups) for the lower ends of the inner springs to slide upon as they compress. These valve spring collars also perform the dual function of concentrically locating the inner valve spring, thus preventing it from wandering and rubbing against the inner side of the outer valve spring. As the springs compress, the bottom end of the inner springs slide ever so slightly on their valve spring collars. The outer spring is made from heavier gauge wire that is wound at a different rate than the lighter gauge wire of the inner spring, thus it has a greater surface area bearing against the vale spring retainer cap (cup) than the inner spring. Being located further from the axis of the circular motion, it also has the advantage of possessing greater concentric leverage. As a result, due to their differing resistances (72.5 lbs. outer, 30 lbs. inner), the valve spring retainer cap (cup) twists along with the valve as the thrust loading produced by the thrust pad of the rocker arm ceases and thus the springs have a natural fine ratcheting effect on the valve spring cup that is transferred to the attached valve. Hence, a counterclockwise (anticlockwise) valve rotation in the valve guide is assured during the critical break-in period, contributing to effective sealing and an extended lifespan of the valve seat. Due to the critical nature of the grade of finish and the alloy used in their composition, it is highly important to replace these valve spring collars with new ones of Original Equipment specification whenever the valve springs are replaced with new ones, otherwise valve rotation during the break in period will not occur. The new valve springs should always be carefully inspected beforehand, any burrs removed, and the leading edges of their flats carefully radiused so that no damage to the mating surfaces of the valve spring collars or valve spring cups will occur. It should be noted that after the break-in period is completed, the friction surfaces of the valve spring collars will have worn to the point that valve rotation ceases. This should not be cause for concern, as the purpose of the system is to ensure the optimum bedding-in of the sealing areas of both the valves and their valve seats. However, due to the variations in quality attendant to mass production, this system seems to work better in theory than it does in practice. In order to guarantee proper valve seating of the valves against their valve seats, it is wiser to lap them together than it is to depend upon this system. Just be sure that all of the lapping compound is removed prior to final assembly, otherwise it will ruin your engine!

Be aware that the early type valve spring retainer caps (cups) (BMC Part #'s 1H 1329 and 12H 992) with square-groove cotters (BMC Part # 1K 800) used on the 18G, 18GA, 18GB, 18GD, and through 18GF/2159 non-Overdrive and 18GF/530 Overdrive engines will not work with the later type valve spring retainer caps (cups) (BMC Part # 12H 3309) and their round-groove cotters (BMC Part # 12H 2117). The larger size (1.625" / 41.275mm diameter) intake valves (BMC Part # 12H 2520) are not available with the square groove that is machined for the earlier size cotters. This is just as well, as the later round-groove design does not require a spring clip in order to retain the cotters in place, is more resistant to fatigue, and is better at permitting the valve to slowly rotate slightly during the break-in period and thus extend the lifespan of the valve seat. You will therefore need to use the later type dual spring valve spring retainer caps (cups) (BMC Part #12H 3309) and cotters (BMC Part # 12H 2117) used on the 18GF/2160 non-Overdrive and 18GF/531 Overdrive on through the 18V-585-Z-L engines in order to use the round-groove valve stems. You will also need the valve spring collars (BMC Part # 1H 1321) of the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines to go under the inner valve spring in order to concentrically locate the dual valve springs properly.

Although simply fitting a stiffer set of valve springs as a less expensive alternative to reducing reciprocating mass in the valvetrain is a possible option, in reality it is a poor practice. The additional pressure on the camshaft lobe / tappet interface and the increased stress on the camshaft drive chain and sprockets will result in accelerated wear of these components. In addition, the additional pressure on the valve spring cups can prevent valve rotation during the break-in period unless an engine lubricant designed for high load pressures is employed. However, such a lubricant will most likely interfere with the seating of the piston rings. In addition, the pressure at the interface of the load-bearing surfaces of the rocker arm bushing and the rocker shaft will be increased, resulting in more rapid wear of these components. In extreme cases, the increased torsional stress created by the increased pressure on the opening ramp of the camshaft can also cause the camshaft to distort along its axis at high engine speed, playing havoc with valve timing and risking the eventual metal fatigue-induced breakage of the camshaft itself.

Should you elect to use a camshaft lobe profile that produces valve lift greater than .450" (11.43mm), you should consider substituting a set of lightweight titanium alloy valve spring retainer caps (cups) for those of the heavier steel Original Equipment items in order to accomplish further reduction of the reciprocating mass. Should you choose to employ them, light titanium alloy valve spring retainer caps (cups) should be checked for deformation at the time of every valve adjustment in order to prevent the valve from pulling through the valve spring retainer cap (cup), resulting in a dropped valve. Under no circumstances should aluminum alloy valve spring retainer caps (cups) be used in anything other than an engine intended for exclusive use on a racetrack.

If you are using Original Equipment tappets and pushrods, always use valve springs with rates and lengths that are recommended by the manufacturer of the camshaft. Be aware that their installed height is of critical importance. Should an installed height (preload setting length) prove to be too long, the result will be an increased likelihood of a valve failing to be withdrawn towards its valve seat quickly enough to avoid clashing with the piston crown, a usually catastrophic event. Should an installed height prove to be too short, the result will be accelerated wear and possible failure of the valve head, the valve seat, the valve stem tips, the thrust face of the rocker arm, the rocker arm bushing, the rocker shaft, the ball end adjuster of the rocker arm, the seating cup of the pushrod, the face of the tappet, and the lobe of the camshaft. In extreme cases, the pushrod may flex or even bend at high engine speeds. If the installed lengths of the new springs are to be greater than that of the Original Equipment items (Inner: 1 7/16", Outer: 1 9/16"), it will be necessary to counterbore the spring seat surfaces in the cylinder head to the proper depth in order to attain the manufacturer's recommended preload setting length for the springs. Even if Original Equipment valve springs are used, their counterbores in the cylinder head should be refaced with an end mill perpendicular to the axis of the valve stem and its valve guide in order to ensure that they provide a proper base for the valve springs. In a few rare cases where very large amounts of valve lift are produced, extra-long valve springs must be employed. In such cases where there is not enough material in the cylinder head casting to permit counterboring the valve spring seat areas without breaking into a coolant passage, it will be necessary to fabricate custom-made valves with longer-than-standard valve stems. This in turn will require shimming the rocker shaft pedestals in order to maintain proper valvetrain geometry. Unless the mating surface of the cylinder head casting or deck of the engine block have been appropriately end milled in order to reduce the height of the rocker arm assembly, the fabrication of custom-length pushrods will also be required.

Many amateur engine builders will attempt to prevent the springs from binding by being sure that when they are installed they have a certain minimum of .XXX" clearance between the coils. Unfortunately, there is no such "magic clearance figure" that will universally insure against binding. Always follow the spring manufacturer's recommendation on this issue, just as you would on the issue of installed height. Peter Burgess recommends a minimum .050" (1.27mm) difference between the compressed height of the valve spring when the valve is at full lift and its fully assembled uncompressed height in order to avoid valvetrain compression damage.

Adjusting the valve clearances on the MGB engine is not unduly difficult as long as the correct procedure is followed.

The engine must be cold, so it is best to allow it to set overnight.

Remove all of the spark plugs so that the crankshaft can be rotated with minimal effort. You might also want to loosen one or all of the V-belts as well in order to reduce resistance. If you do not have a large wrench to turn the crankshaft, leave a V-belt properly tensioned so that you can use it to rotate the crankshaft.

Now, remove the rocker arm cover and retorque the cylinder head using the sequential pattern shown on page 77 of the Bentley manual. This is to make sure that the clearances will be as accurate as possible. Do this by backing off all of the nuts only a few degrees (do not loosen them entirely), and then tightening them down in proper sequence 5 Ft-lbs at a time so that the pressure at each torquing point will be equal. Stop torquing when you reach 45 to 50 Ft-lbs.

Look down at the pulley wheel of the harmonic balancer (harmonic damper) (the pulley wheel that is attached to the front end of crankshaft). If you inspect the pulley wheel on the harmonic balancer (harmonic damper) closely, you will notice a notch on the outside edge. This is used for setting the both the valve timing and the ignition timing. You might want to clean the notch with some of your wife's nail polish remover and paint the edges of it with some of her lightest colored nail lacquer in order to make it more visible (She won't mind you borrowing these items for such a noble purpose!).

Near the pulley wheel on the harmonic balancer (harmonic damper) you will also see a sheetmetal stamping that has what resembles saw teeth. These are ignition timing marks. The point of each tooth represents 10° in the rotational position of the crankshaft, as does the rotational distance between each of its notches. When the notch on the flywheel is aligned with the point on the left, the crankshaft is at 0°, also called Top Dead Center. That is when the pistons for #1 and #4 cylinders are at their maximum height of travel. When the crankshaft is rotated 180°, pistons #2 and #3 are at their maximum height of travel.

At this point, you have to establish just when a valve is fully closed so that you can properly set the gap. I do this by using a Starrett dial indicator mounted on a magnetic stand placed on the cast iron cylinder head. This is the best way to establish precisely when a valve is fully open. However, I am sure that unless you are a former Machinist or Tool & Die maker, you probably do not have these expensive tools. The quickest method is to affix a degree wheel to the harmonic balancer (harmonic damper) pulley wheel on the crankshaft, but this requires that you know in degrees exactly when a given valve is fully closed. I would suggest rotating the crankshaft and then using the Top Dead Center notch on the harmonic balancer (harmonic damper) pulley wheel in order to find Top Dead Center. Next, install the degree wheel that you used when you timed the camshaft, and then use it in order to locate the correct positions for adjusting the valve lash clearances. That way there will be no doubt as to the appropriateness of the camshaft position when you perform the valve adjustments. If you want to make things easier in the future, each time that you get the crankshaft into the proper position for setting the clearance on a particular valve, you can remove the degree wheel and use a punch in order to mark the circumference of the pulley wheel on the harmonic balancer (harmonic damper). However you may no longer have a degree wheel, so here is an Old-Timey-Mechanic's trick for a less expensive way to figure it out: Get out your set of blade-type feeler gauges and use your wife's nail polish to paint a mark on the middle of the blades that you will use. These are the ones that you will use to set the clearances on both the intake and exhaust valves when using an Original Equipment specification camshaft.

For the purpose of reference, consider the valves to be sequentially numbered from left to right when facing the engine from the spark plug side. (#1 on the far right, #8 on the far left.)

Remove the spark plugs and then rotate the crankshaft either by using a large wrench on the nut in front of the pulley wheel on the harmonic balancer (harmonic damper) of the crankshaft (Never, ever, use a pipe wrench!), or by pulling on a V-belt, until Valve #8 appears to be fully open. Although a bit of a hassle to attach to the end of the crankshaft, the use of a degree wheel is the most accurate way to tell when the valves are fully open. However, if you do not have access to a degree wheel, there is a cruder method. Slowly and gently rotate the crankshaft back and forth a very few degrees, using the gauges to determine when the gap is at its greatest. When you have the crankshaft in the position in which the gap is the greatest, you can proceed to the next step.

Using a ½" wrench, loosen the jam nut on the pushrod end of the rocker arm and then use a flat-tipped screwdriver in order to loosen the valve adjusting screw as far as it will go. Pull the pushrod away from the cup on the end of the adjuster screw. Lift the valve end of the rocker arm as high as it will go and, using your fingernail, feel the thrust face of the rocker arm. If you find a groove on the thrust face of the rocker arm, then using flat blade gauge obviously will not provide an accurate measurement of the gap between the thrust face of the rocker arm and the tip of the valve stem. You will need to acquire a ClickAdjust tool in combination with a ½" socket in order to accurately set the valves. You can find information on this tool at http://www.mgcars.org.uk/MG_Elec-Tech/Clikadjust_0.html . However, you need to be aware that the hardening on the external layer of the original working surfaces of the rocker arms is only a few thousandths of an inch thick. The material beneath this hardened layer is soft and will wear rapidly, making valve adjustments more frequent.

Now, make sure that the valves are adjusted in the following order:

1) Adjust Valve #1 when valve #8 is fully open.

2) Adjust Valve #3 when valve #6 is fully open.

3) Adjust Valve #5 when valve #4 is fully open.

4) Adjust Valve #2 when valve #7 is fully open.

5) Adjust Valve #8 when valve #1 is fully open.

6) Adjust Valve #6 when valve #3 is fully open.

7) Adjust Valve #4 when valve #5 is fully open.

8) Adjust Valve #7 when valve #2 is fully open.

Of course, it is best to not guess at what the proper clearance settings are when using a certain camshaft. The following table should be helpful-

Camshaft Type	Intake Valve Clearance	Exhaust Valve Clearance
Original Equipment	.015" (.38mm)	.015" (.38mm)
Piper BP255	.012" (.30mm)	.014" (.35mm)
Piper BP270	014" (.35mm)	.016" (.40mm)
Piper BP285	.014" (.35mm)	.014" (.35mm)

Using a new gasket, replace the rocker arm cover, and then torque the securing nuts to a mere 4 Ft-lbs. Any more torque than this will risk crushing the cork gasket. If you crush the gasket, it will leak. Be aware that cork gaskets have their own particular characteristics. Cork will not fill gaps, nor will it compensate for misalignment. Both of the sealing surfaces must be clean, flat, and parallel. Most owners are aware that a stamped steel rocker arm cover is susceptible to warpage if it is overtightened and thus tend to suspect this component whenever they have a problem with leakage. However, many forget that as a safeguard there are two rubber compression bushes under the securing nuts of the rocker arm cover, and that it is these that apply the pressure to the cork gasket by means of the rocker arm cover in order to enable it to form an effective seal. These rubber compression bushes(BMC Part #

2A 150), should always be replaced if they are found to be compressed, deformed, or hardened.

Note that the 18GD and later engines have a different securing nut design for the rocker arm cover. This was originally intended to prevent over-tightening, by tightening it down onto the top of the rocker gear nut. This later rocker arm cover securing nut (BMC Part # 12H 2557), as well as having the stud on top of the nut for the heater return pipe, has a deeper cylindrical section below its hexagonal section. This passes through a 1/8" spacer, a cup washer (BMC Part # 1A 2156), a rubber compression bush (BMC Part # 2A 150), and the rocker arm cover onto a long cylinder head stud that is used in order to secure the rocker gear. The rubber compression bush performs two functions: it seals the fixing holes in the rocker arm cover in order to prevent leaks, and it also applies pressure to the rocker arm cover and from there to the gasket on the cylinder head. Even with new rubber compression bushes and new cork gaskets, once the securing nut has taken up all of the free play between it and the top of the spacer, it can only be tightened one more turn before the bottom of the cover nut contacts the top of the rocker gear nut, thus preventing overtightening. However, unless all of the needed parts are present and of the correct thickness, including the rubber compression bush that is between the cup washer and the top of the rocker arm cover, as well as the cork gasket, it will limit the amount of downward pressure that can be exerted upon the gasket of the rocker arm cover. Old rubber compression bushes compress and harden with age a lose their resiliency. compressing and deforming over time, and thus reduce the pressure applied to the cover gasket. Old cork gaskets similarly compress, and can rarely be reused successfully once disturbed. Insufficient pressure coupled with the porosity inherent to cork gaskets will permit leaks to develop.

Using antisieze compound on the threads, reinstall the spark plugs. You will find that Loctite Marine Grade antisieze compound does an excellent job, being formulated for use in harsh environments where the items it is applied to are exposed to fresh or salt water, thus making it essentially “goof proof”. Clean all of the contacts of both the High Tension (HT) leads (spark plug leads), ignition coil lead (king lead), as well as the distributor cap with CRC QD Electronic Cleaner, and then reinstall the High Tension (HT) leads (spark plug leads) and the ignition coil lead (king lead) into the distributor cap and onto the spark plugs, using dielectric grease to prevent corrosion of the contacts.

Valves

A word about valve materials- For many years the standard exhaust valve steel was EN52B. This steel was first introduced over 70 to 75 years ago and has a hardness of 25 to 31 HRc. Improved engine design has lead to increased compression ratios and higher operating temperatures, and improved fuels with an increased octane rating and the addition of tetra-ethyl-lead have lead to an increasing tendency to prematurely burn out the exhaust valve. This steel is classed as semi-corrosion-resistant as it is attacked by Chlorine and Sulfur compounds. As a result, this material is no longer considered suitable for exhaust valves, although it is still perfectly satisfactory for intake valves when used with leaded gasoline.

About 1960 a new steel, Austenitic 214N (Stainless), was developed. This steel is also referred to as 349S42. It has a hardness of 30 HRc, retains its hardness even up to temperatures of 1,472° Fahrenheit (800° Celsius), and possesses excellent rupture strength under high temperature conditions combined with good creep and impact values. Its high Chromium content gives it good scaling resistance, and it has greater corrosion resistance against Chlorine although it is still not immune to sulfurous attack. In terms of creep strength, austenitic stainless steels are superior to all other types of stainless steel. This is the preferred material for use with the higher combustion temperatures attendant with unleaded gasoline. It is also more resistant to carbon buildup on its head than the more common EN52B valve material.

A more recently developed material, Nimonic 80A, has a hardness of 32 HRc and has an increased operating temperature over Austenitic 214N as well as higher corrosion resistance, thus making it suitable for engines with high operating temperatures. Due to its high cost, it is commonly seen only in very high compression ratio engines built expressly for racing. This is the alloy that the MG factory race team used for the valves in their engines.

Hard Chrome Plating imparts an added durability to a valve stem by depositing a layer of chromium onto the guide area of the stem of the valve of approximately 32 to 72 microns in thickness. This gives good compatibility if the valve is entirely made of Austenitic 214N (Stainless) and is to be used in a cast iron valve guide. This type of treatment is only applied to the valve stem. A Stellite 6 deposit can be applied to the exhaust valve seat face in order to enhance the valve seat hardness (Rockwell "C" of 38 to 42 HRc) that will enable it to be used with unleaded fuel or in highly stressed engines. A Stellite 12 deposit can be applied to the tip of the valve stem that will further enhance the tip hardness (Rockwell "C" of 48 to 52 HRc), thus reducing wear and consequent need for frequent valve gap adjustment. These coatings can be performed by the Doro Stellite Company, which has a website at http://www.stellite.com/ .

Be aware that Austenitic 214N valves come in two varieties: one-piece and two-piece. A two-piece valve uses EN52B alloy for its stem, which is then friction welded to the valve head which is made of much tougher 214N steel. Why would a manufacturer use two types of steel in one valve? Simply because 214N is a more costly alloy and, as it is only needed for the head of the exhaust valve, there is a small cost saving in using EN52B alloy for the stem. The saving in the cost of materials surpasses the cost of friction welding. It might seem to be a minor cost savings, but when you are producing millions of valves, it all adds up. Unfortunately, the friction weld can fail, the result being that the head of the valve drops into the combustion chamber as the piston is pumping up and down, there to do all sorts of evil things to your engine. One-piece valves may be slightly more expensive, but their additional reliability makes them worthwhile.

Tuftriding (AB1 or TF1, the process used depends upon the specification of the valve) gives a hard layer over the complete valve of between 72 to 74 Rockwell "C" approximately 10 to 20 microns in depth, and gives excellent wear properties in both either a cast iron or a bronze alloy valve guide, along with the added benefit of stress relieving the valve. In Tuftriding, the item is immersed in hot cyanide compounds, creating a tough, resistant surface that improves fatigue resistance. This type of treatment produces a black mottled finish over the entire surface of the valve. Before installation of any tuftrided component, it should be thoroughly scrubbed until it is free of any salts that remain from the tuftriding process. These salts are highly abrasive and will circulate within the engine, causing severe damage. Be warned that simply rinsing or flushing with petrochemical cleaners will prove to be an inadequate method of removal.

Do not be tempted into paying more for valves that have a swirl polished finish on their valve heads. Swirl polishing of the valve head is merely a convenient way to polish a valve head on the production line. It actually does nothing for performance. Long ago, somebody in a marketing position noticed the "swirl" effect and chose to make a false claim about it, leading people to mistakenly believe that a swirl polished finish causes has some effect on the swirl motion of the airflow as it passes over and beyond the valve. This is a false assumption, as the texture of the finish is too fine to do anything more than give a neat appearance. Most high performance specialty valves are polished in some manner in order to remove the tool marks left from the lathe process in order to reduce possible stresses. The shape of the valve head and diameter of the stem having more to do with cross sectional area than with anything else, only the very edges of the valve have any real influence over the flow pattern of the fuel / air charge. The width of both the valve seat and the margin, and whether or not a back cut is used, makes a greater difference than does the surface finish of the valve head.

Simply installing an oversize intake valve in an attempt to increase the flow of the fuel / air charge is a common but serious mistake. This will place the edge of the head of the valve closer to the wall of the combustion chamber. Without unshrouding the valve by modification of the adjacent wall of the combustion chamber, such a modification will actually reduce the flow of the fuel / air charge as a result of the increased shrouding.

Also, never eliminate the entire margin of the head of the valve in an attempt to reduce reciprocating mass. The resulting sharp edge and loss of heat absorbing mass will cause the edge of the valve head to become a hot spot during hard running, thus triggering preignition and even detonation. In addition, the intake valves should always have a 45° bevel at the juncture of the face and the margin in order to help reduce backflow during their overlap period. However, the exhaust valves should be generously radiused and polished at the juncture of their faces and their margins in order to facilitate exhaust flow.

Do not waste your money on exotic tuliped valves. Due to the side draft configuration of the ports of the B Series Engine and the vertical orientation of the valves, the incoming fuel / air charge flows across the head of the valve instead of around it. This being the case, the broader profile at the base of the stem on a tuliped valve will actually flow less than an Original Equipment flat-topped valve, and the extra material will present unnecessary complications involved with increased reciprocating mass in the valvetrain. This is why the engineers at the factory chose to use as Original Equipment an intake valve design that has a sharper underhead radius in order to provide a less broad shoulder at the junction of the valve stem than that of a tuliped valve, and also is why the design of racing valves for the B Series engine tend to take on the famous “penny on a stick” configuration.

Be aware that not all valves are equal, even though they are made of the same alloy. The undersides of the heads of Rimflo valves are of extreme concave design with a deep and broad antireversion groove near its circumference in order to reduce backflow during the overlap period. However, the increased thickness of the valve head necessary to allow the existence of this groove does not keep their reciprocating mass to a minimum so that they would be appropriate for use in engines that attain high engine speeds. Of course, their increased surface area, especially in the groove in the face of the head, makes them a natural trap for carbon and heat, a condition that can trigger preignition. In addition, once the carbon collects, they become even heavier. While these valves have excellent antireversion properties so that they can reduce the camminess of the engine when used with camshaft lobe profiles that feature long-duration valve timing, my opinion is that they are appropriate for use in race engines only. Race engines are torn down and decarbonized on a regular basis, while street engines are not. For long-term use in a street engine, they are impractical and thus inappropriate.

Valves made of EN52B alloy tend to have a rather stout shoulder joining the head of the valve to its stem in order to facilitate heat conduction, while the shoulders of valves made of the more heat-tolerant Austenitic 214N Stainless Steel are noticeably thinner, making for superior airflow characteristics and less reciprocating mass. However, this increased tolerance to heat should not be taken as an indication that the heat that they absorb in a high performance version of the engine can be ignored. Manganese silicone bronze valve guides should be installed in order to assist in conducting heat out of the valves.

It should be noted that mass producers of valves usually create one design that functions adequately well in the widest variety of engines and merely machine it to the required head diameter and angle of the valve seat. This being the case, such designs are not optimal for any given application. The Heron-type cylinder head of the B Series engine used in the MGB has its own special airflow requirements. The design of Manley valves have the necessary airflow characteristics that make them superior performers in the Heron-type heads of the B Series engine and do not have the problem of rapid carbonization, although they are slightly heavier than most of their competitors' valves. However, the design of the valves made by Peter Burgess, being tailored specifically to meet the airflow needs of the B Series engine, have the most optimal airflow characteristics of all.

Like a waisted throttle shaft, waisted valves are nice, but of themselves, they will not really have much of an effect in a street engine. They are primarily for very high engine speed racing use with a camshaft like a Piper 300 and full race heads. The risk with waisted valve stems is that due to their reduced rigidity, they can vibrate like a tuning fork at maximum valve lift during high engine speeds, especially when a high-lift camshaft is employed. This vibration can cause metal fatigue to set in prematurely and the valve stem will then ultimately fracture, the valve head being sucked into the combustion chamber, there to do all sorts of evil things. That is why they are never reground and reinstalled by racers: Short Fatigue Life. If you choose to disregard this warning, do not ever try to recycle them. Once the valve seating faces are worn, play it safe and toss them into the scrap metal recycling bin.

I would make one suggestion that Peter Burgess does not mention in his book: for use in a street engine, once you have had the three-angle face made on the valve, and after lapping it in, it should be either stellite-plated or, preferably, tuftrided. Neither of these improvements is overly expensive and will help to ensure a long, long service life in street use.

Valve Seats

The commonly used single 45° angle used for the valve seating area is the result of the desire for simplification of the manufacturing process, this simple configuration being both convenient and inexpensive for mass production. It also offers the advantage of self-maintenance of the concentricity between the sealing surface of the valve seat and that of the valve due to the wedging action of the 45° valve seat angle. This is important due to the fact that heat from the area of the exhaust valve has a greater effect on the area of the intake valve seat nearest the exhaust valve, causing as much as .004" (.1016mm) distortion in the case of cast iron heads, and more in the case of aluminum alloy cylinder heads. However, in terms of airflow potential, it is comparatively mediocre.

Perhaps one of the most cost-efficient methods of achieving increased airflow capacity is the use of a three-angle cut on both the valve seat and the valve. This is because the velocity of the flow of the fuel / air charge at the valve seat is greater than that in the port during the first half of the valve's travel. At no point in the further travel of the valve does the velocity of the flow of the fuel / air charge in the port become greater than its velocity at the valve seat, so improvements in the streamlining of the valve seat area is a matter of the highest priority. However, air does not like to change directions in such a radical manner, and turbulence is the result, especially as velocities of the flow of the fuel / air charge increase. Turbulence being the enemy of efficient airflow, a three-angle cut on both the valve seating area of the valve head and its valve seat offers a more efficient approach to the problem. Because air can change direction by 15° with almost no loss of inertia and consequent turbulence, a valve head angle of 30°, followed by an angle of 45° for the valve seating area, followed by an angle of 60° above the margin, presents a far more efficient configuration. On the other hand, if a simple 45° valve seat is used, then the juncture of the valve seat and the port should have a radius of .030" to .040" (.762mm to 1.016mm). If a three-angle valve seat with a 30° sealing angle is used, then a 75° angle should blend the port to the inner 60° valve seat angle. When compounded with a valve seat angle of 60° at the throat, followed by an angle of 45°, followed by an angle of 30°, this simple approach can increase power output by as much as 10% on an engine built to otherwise Original Equipment specifications.

15° / 30° / 45° / 60° / 75° angle valve seat configurations have long been common for racing use, going back to the glory days of Huffacker in the 1970s. However, while these are of advantage in racing engines that attain very high engine speeds, it is doubtful that there is any practical advantage for such a multiple angle configuration in a street engine.

The most commonly used three-angle configuration takes advantage of the wedging action of the 45° valve seat sealing angle. The optimum angles for the intake valve seat in the cylinder head should thus be 60° at the throat, 45° for a seating width of .050" (1.27mm) for the sealing angle, then 30° for a width of .010" adjacent to the roof of the combustion chamber, while the optimum angles for the intake valve should be 20° nearest the stem, 30° for a width of .060" (1.524mm), then 45° for a width of .050" (1.27mm) for the sealing angle at the margin of the valve head, followed by an angle of 60° above the margin, thus giving excellent streamlining to the intake airflow. The optimum angles for the exhaust valve seat in the cylinder head should thus be 60° at the throat, 45° for a width of .070" (1.778mm) for the sealing angle, then 30° for a width of .020" (.508mm) adjacent to the roof of the combustion chamber, while the optimum angles for the exhaust valve should be 30° nearest the stem, then 45° for a width of .070" (1.778mm) for the sealing angle at the margin of the valve head, thus giving excellent streamlining to the exhaust airflow. In any case, do not be tempted to narrow the width of the 45° valve seating area of the exhaust valve. The wider valve seating area is necessary to provide sufficient conductivity for the removal of heat from the exhaust valve.

However, as a simple matter of geometry, when compared to a 45° angle on the sealing surface of the intake valve seat, a 30° angle on the sealing surface of the intake valve seat will automatically increase the size of an opening by 21% during the first .100" to .150" (2.54mm to 3.81mm) of valve lift. This increase in the flow of the fuel / air charge at low valve lift can partially compensate for the partial shrouding of the valve that results from the close proximity of the wall of the combustion chamber. It will also help to extend the powerband as well. Because airflow rate is uneffected by making a mere 15° turn, the subsequent angles can thus be 45° and 60°. The optimum angles for the valve seat in the cylinder head should thus be 60° for a width of .040" (1.016mm) at the throat, 45° for a width of .020" (.508mm), then 30° for a width of .050" (1.27mm) for the sealing angle adjacent to the roof of the combustion chamber, while the optimum angles for the valve should be 30° at the margin of the valve head for the sealing angle, then 15° nearest the stem. This still leaves the reduced wedging action of the 30° valve seat to be dealt with so that an effective seal can be accomplished under the conditions of thermal distortion of the valve seat. This can be accomplished by machining a .050" (1.27mm) radius confirmation groove into the face of the valve .010" to .015" (.254mm to .381mm) from its margin.

Extensive research by such luminaries as Peter Burgess and David Vizard has established that in terms of airflow, the optimum respective widths for both a 30° and a 45° sealing angle of the valve seat is .055" and .065" (1.397mm and 1.651mm), respectively, for mild camshafts such as the Piper BP270 and .050" and .060" (1.27mm and 1.524mm) for hotter camshafts such as the Piper BP285. Any further decrease in the width of the sealing area will produce a valve seat that cannot adequately conduct away the heat stored in the valve, the valve in consequence becoming a hot spot that can trigger preignition.

Properly executed, a three-angle modification with a 30° sealing angle can increase airflow by as much as 25% at some points of valve lift when compared to a 45° sealing angle, especially when employed in the case of the variants of camshafts with shorter duration valve timing commonly found in streetable B Series engines. This high rate of airflow at low valve lift also facilitates a more rapid drop in cylinder pressure, reducing power lost due to the piston pumping out the exhaust gases. The resultant improvement in efficiency can increase power output by as much as 10% beyond that of a single-angle 45° valve seat. In addition, the increased flow of the fuel / air charge at low valve lift extends the power output beyond its previous peak, adding perhaps yet another five horsepower beyond that of a three-angle 45° valve seat, while also making for a less precipitous drop-off of power. This means that a Piper BP270 camshaft will produce almost as much midrange and top end power output as that of a Piper BP285 camshaft that is paired with the more conventional single-angle 45° valve seat, while not sacrificing any of its reliability or tractability at low engine speeds.

It should be noted that such a valve seat configuration is impractical unless top quality components are used for both the valves and the valve guides in order to assure long-term maintenance of the concentricity between the sealing surface of the valve seat and that of the valve. Plain EN52B alloy valves and cast iron valve guides will give an unacceptably short service life because of wear, especially when used with the higher sidethrust loadings on the valve stem that are inherent in the use of a higher-lift camshaft. The combination of manganese silicone bronze valve guides coupled with tuftrided Austenitic 214N stainless steel valves should give a more than adequate service life in such an application. Above all, precision machining is a requisite whenever such a modification is performed.

You need to understand the reason for the necessity of installing lead-free fuel compatible hardened valve seat inserts into the cylinder head. When valves are reground and their valve seats in the cylinder head recut, the old deposits of Tetra-Ethyl-Lead remaining from the era of leaded fuel are removed. Without Tetra-Ethyl-Lead to both cushion and lubricate the valve seating surfaces, the head of the valve impacts upon the raw cast iron of the valve seat and forms a series of micro-welds which are torn loose the next time that the valve opens, resulting in the erosion of both the cast iron valve seat and the valve. This also occurs when a cast iron cylinder head that was induction hardened has had new valve seats cut into its surfaces, removing the layer of hardened metal. This is because in induction hardening, the surface is heated by a high-frequency alternating electromagnetic field wherein eddy currents are generated within the metal and the electroconductive resistance of the metal leads to Joule heating of the metal into the austenitic crystal phase before it is quickly quenched in order to impart ductility. The quenched metal undergoes a martensitic transformation, resulting in a hardness penetration of 0.060”-0.080” (1.524mm-2.032mm) below the surface. The low cost, ease, and speed of this process makes it the favored hardening technique for production of Original Equipment cast iron heads. While induction hardening has the advantage of selectively hardening areas of a part or an assembly without effecting the properties of the part as a whole, once the hardened metal is removed by machining, the installation of lead-free fuel compatible hardened valve seat inserts, though a more expensive production procedure, becomes mandatory, but eliminates this problem.

As a rule, a valve seat should be replaced if the specified installed valve height cannot be achieved without excessive grinding of the valve stem tip (less than .030” / .762mm), or if the specified installed valve spring height cannot be achieved using a .060” (1.524mm) valve spring shim. This applies to integral valve seats as well as to nonintegral lead-free fuel compatible hardened valve seat inserts. The only other alternative to replacing the valve seat is to install an aftermarket valve that has an oversized head. This type of valve rides higher on the valve seat in order to compensate for excessive valve seat wear or machining, and can eliminate the need to replace the valve seat if the lubricating properties of Tetra-Ethyl-Lead are available.

There is a lot more to replacing a valve seat than simply prying out the old insert and then simply driving in a new one. If the cylinder head is cast iron with integral valve seats that have been machined into the cylinder head, then the cylinder head has to be machined in order to replace the valve seat (sometimes called installing a “false” valve seat). If the cylinder head is of aluminum alloy, the valve seat counterbore may have to be machined in order to accept an oversize valve seat if the bore for the insert is loose, deformed or damaged. Either way, a machinist has to figure the amount of interference that is required for the new valve seat before cutting the cylinder head on a seat-and-guide machine. Replacing a valve seat, therefore, involves a number of decisions and steps, all of which effect the ultimate outcome of the process.

As you might have guessed, there are differing opinions about the right way and wrong way to replace valve seats, particularly with respect to the amount of interference fit that is required in order to retain valve seats in aluminum alloy cylinder heads. A common fear expressed by many engine rebuilders is concern over the possibility of a valve seat falling out, particularly in aluminum alloy cylinder heads where the difference in the coefficients of thermal expansion between that of the cylinder head and that of the valve seats can cause valve seats to loosen should the cylinder head overheat. Consequently, engine rebuilders express differing views as to whether or not locking compound and / or peening or staking should be used as “insurance” when installing valve seats in aluminum alloy cylinder heads. In reality, the correct solution is strict adherence to engineering tolerances during the machining process.

The valve seats in an aluminum alloy cylinder head may also loosen or fall out when the cylinder head is being cleaned in a bake oven or preheated in an oven for straightening. The same thing can happen to the valve guides. Whether or not this occurs depends on the amount of interference fit between them and cylinder head. The less the interference, the more likely the valve seats are to loosen and fall out when the cylinder head is baked. If you do not want the risk of the valve seats falling out, then turn the cylinder head so that its combustion chambers are facing upward prior to baking it.

The one point that everyone does seem to agree upon is that the valve seats play a critical role in the longevity of the valves. The valve seats draw heat away from the valves and conduct it into the cylinder head. This provides most of the cooling that the valves receive and is absolutely critical with exhaust valves. Anything that interferes with the valve seat’s ability to cool the valves (such as a loose fit or deposits between the valve seat and its counterbore) can lead to premature valve failure. This being the case, do not use a locking compound because it can create a thermal barrier between the valve seat and its counterbore in the cylinder head.

Nonintegral valve seats can fail for a number of reasons. The majority of the valve seats that end up being replaced are replaced because they are either cracked or too worn to be reground or remachined. Valve seats can crack from thermal stress (usually as a result of engine overheating), thermal shock (a sudden and rapid change in operating temperature), or mechanical stress such as detonation or excessive valve lash that results in severe pounding, etc.

In order to remove worn valve seat inserts, some rebuilders heliarc weld a bead around the inner circumference of the valve seat insert. As the bead cools, it shrinks and contracts the valve seat insert, causing it to loosen. Another, less sophisticated technique that is sometimes used to remove a valve seat insert is to insert an old valve that is somewhat smaller than the valve seat into the cylinder head and then weld the valve to the valve seat insert. The valve stem can then be used like a driver to push out the valve seat insert. However, damage to the counterbore where the old valve seat insert has resided is often the result, thus requiring new counterbores to be machined into the cylinder head.

When machining the counterbores for the valve seat inserts into an old cast iron or aluminum alloy cylinder head, it must be understood that the material being removed can vary, altering the required rate of cut and can actually pull the cutter aside, dig the cutter in, etc. etc. You must be aware that the machinist is not dealing with the removal of virgin material. Even so, new materials can also vary. The machinist is not dealing with billet forged material where the grain of the metal should be consistent. If you smell old castings you can smell the oils that have permeated throughout the casting, demonstrating that the castings are not as solid as they may seem to be to the naked eye, but a series of clumped molecules with god knows what in between! In truth, a casting may be described as a collection of holes that are held together with metal. There is also another problem: the case of a cylinder head that has already been fitted with the commonly installed 3mm to 4mm deep inserts. The solution is to fit deeper valve seat inserts at a depth of 6.5mm to 7mm. As a general rule-of-thumb, the roofs of the combustion chambers of Original Equipment cast iron heads can be safely counterbored to a depth of 9mm without incurring the risk of breaking into a coolant passage. If wider valve seat inserts cannot be fitted in order to replace worn valve seat inserts, then the machinist is presented with the nightmare of making an identical-diameter counterbore deeper in order to accommodate a thicker valve seat insert, yet with its central axis perfectly concentric with that of the valve guide. If the cylinder head has distorted (and it frequently is found to have done so), then the positional relationship of the valve guide and that of the valve seat may have relocated by as much as .004” (1.016mm). Under such circumstances, even with the best piloted end mill in the world, the central axis of the new counterbore cannot be concentric with that of the old, lessening and unevenly distributing the mounting pressure around the periphery of the valve seat insert in the counterbore! The counterbore for the new valve seat will be seen to be offset inside of the old counterbore, so special measures will have to take up the space. The usual method is welding, as Peter Burgess does.

Most aftermarket valve seats need about .005” (.127mm) interference fit when installed into iron heads and about .007” (.1778mm) interference fit when installed into aluminum alloy cylinder heads. Valve seat suppliers usually build the required press fit into the Outside Diameter (O.D.) of the valve seat. A 1.500” (38.1mm) Outside Diameter (O.D.) valve seat will measure 1.505” (38.227mm) for cast iron applications and 1.507” (38.2278mm) for aluminum alloy cylinder heads. This is why it is best to consult your valve seat suppliers catalog and use the correct part number listed for that specific application. Most valve seats are finished on the Outside Diameter (O.D.) to about a 15 Ra surface finish. The finish in the counterbore should be equally smooth and round to within .001” T.I.R. This will ensure good contact area and excellent heat transfer properties for the valve to operate against.

The counterbore in the cylinder head must be absolutely clean and have a smooth surface finish. The valve seat inserts should be placed with the radius or chamfer side down and lubricated (Automatic Transmission Fluid works fine) prior to being pressed or driven in with a piloted driver (strongly recommended in order to prevent cocking). If the replacement valve seat insert has a sharp edge, it should be chamfered or rounded so that it will not gaul and scrape any metal off the cylinder head as it is being driven into position. If metal gets under the valve seat insert, it will create a gap that will form a heat barrier. This, in turn, will interfere with the ability of the valve seat insert to cool the valve, thus making premature valve failure the most likely result. Preheating the cylinder head in an oven and / or chilling the valve seat inserts will make installation easier and lessen the danger of broaching or galling the counterbore as the valve seat insert is being installed.

There is an alternative method of installing valve seat inserts. Instead of press-fitting the valve seat insert, the base of the valve seat insert is coated with an electroconductive base and then placed into the recess that has been machined into the cylinder head. A wedge-like conical electrode is then forced against the valve seat insert and a high energy pulse of electricity is then shot through the valve seat, the cylinder head acting as the ground (earth) for the electrical circuit. The electrical power is so intense that the coating literally fuses the valve seat to the cylinder head. When the electrode is withdrawn, the valve seat can be seen to be glowing red hot! Laymen often refer to this process as “welding the valve seat in place”, but to apply the term “welding” to it is really a bit of a stretch, as no exchange of metals takes place. This is the method most commonly used in mass production, and a few of the better specialty shops use this technique as it guarantees optimal heat dissipation through the valve seat into the cylinder head. Sometimes, however, the valve seat does not fuse properly, usually as a result of surface contamination, and becomes loose.

There are two schools of thought on the subject of the required hardness of a valve seat. One simply equates hardness with quality. The other school of thought also recognizes the importance of hardness, but realizes that other factors are just as important. Those of the hardness school of thought contend that you need a valve seat with a hardness of Rockwell C 37 to 45 for unleaded fuel and 40 to 50 Rc for very high compression and / or turbocharged applications. However, toughness and durability are actually the better measures of quality. This alternate school of thought emphasizes the metallurgical aspects of selecting a valve seat material in the replacement valve seat. The next step is to install the valve seat insert.

While most engine builders continue to install the traditional nickel-chromium lead-free fuel compatible hardened valve seat inserts such as the J-Loy inserts that are premium (Hard) valve seats with a typical hardness range of 35-40 Rc, Dura-Bond / Snyder has achieved a significant step forward in the technology of these items. Its Dura-Bond 30000 (Gold) Series is a sintered valve seat insert intended expressly for use in engines that run on unleaded fuel, the microstructure of which contains a blend of finely dispersed tungsten carbide in a tempered martensitic matrix of tempered tool steel and special graphite alloy iron particles. This gives superior wear resistance to both pounding and abrasive wear at the elevated temperatures commonly encountered in the combustion chambers of engines that run on unleaded fuel. Their superior machinability is the result of using special processing techniques to infuse a special high-grade, graphite-rich iron alloy that imparts high temperature lubrication properties as well as cutting tool lubricating properties. Because of this special processing, it is possible to cause very fine, spheroidalized (round shaped) tungsten carbide particles to evenly disperse within the tool steel. These minute sphereoidalized tungsten carbides are easier to machine because the tool bit can wedge in-between, with less cutting force and less friction. The smaller these "balls" of spheroidalized tungsten carbide particles, the easier it is on the cutting tools, because they will not be hitting any big irregular shaped "iceberg chunks" of tungsten carbide. Because of the special high temperature sintering and post heat treat processing, this valve seat material has metal alloy oxides called "cer-met" style because they are similar to ceramic (they do not soften at elevated temperature), but retain the machinability of metal. They prevent the "micro-welding" of the valve seat material to the valve face, therefore eliminating the primary cause of valve seat erosion. It is this high-tech, new-generation processing that achieves such high, hot hardness without having to put in massive amounts of expensive alloys that would otherwise be required in order to achieve equal performance. Normal foundry techniques do not allow this type of structure. Dura-Bond / Snyder has thus taken full advantage of the new powder metal technology in order to produce a "hard" valve seat that will machine almost like cast iron.

The following chart will help both of you when you consult with your machinist about the optimal choice of valve seats for your desired engine specification:

Aluminum Alloy Cylinder Head

Valve Head Diameter	Dura-Bond Part Number	Outside Diameter	Internal Diameter	Interference Fit
1.344" (34.1376mm)	77393	1.3484" (34.2494mm)	1.126" (28.6004mm)	0.0059" (.14986mm)
1.565" (39.751mm)	31318	1.570" (39.878mm)	1.243" (31.5722mm)	0.0059" (.14986mm)
1.625" (41.275mm)	31687	1.632" (41.4528mm)	1.375" (34.925mm)	.0071" (.18034mm)
1.690" (42.926mm)	30664+5	1.697" (43.1038mm)	1.438" (36.5252mm)	0.0071" (.18034mm)

Cast Iron Cylinder Head

Valve Head Diameter	Dura-Bond Part Number	Outside Diameter	Internal Diameter	Interference Fit	Depth
1.344” (34.1376mm)	31011	1.3484” (34.2494mm)	1.126” (28.6004mm)	0.0043” (.10922mm)	0.250” (.635mm)
1.565” (39.751mm)	31286	1.5695” (39.8653mm)	1.265” (32.121mm)	0.0043” (.10922mm)	0.280” (7.112mm)
1.625” (41.275mm)	31859	1.6299” (41.3995mm)	1.398” (35.5092mm)	0.0051” (.12954mm)	0.197” (.50038mm)
1.625” (41.275mm)	31482	1.630” (41.4020mm)	1.406” (35.7124mm)	0.0051” (.12954mm)	0.250” (.635mm)
1.69” (42.926mm)	31701+5	1.695” (43.053mm)	1.250” (31.75mm)	0.0051” (.12954mm)	0.375” (9.525mm)

To replace their older grinding systems, more and more shops are changing over to seat cutting equipment. In order to ensure good tool life with these systems, it is necessary to keep close control over the feed and the speed rates wherever possible. The spindle speed should be adjusted from intake to exhaust valves, especially where large diameter differences are involved. The cutting speed decreases with the increase in the diameter that takes place in switching from the exhaust side to the intake side.

Generally speaking, uncoated carbide inserts work best for the cutting of valve seat inserts. A sharp cutting edge (no hone) on the uncoated carbide will provide lower cutting forces overall. Although C2 grade carbide can provide satisfactory results, research suggests that C4 carbide will provide the best overall tool life and process flexibility. Check with your tool supplier for availability of both of these grades. Carbides that are intended to be used for cutting steel (Carbide grades C5 to C8) do not work well with valve seat insert materials. If ceramics can be obtained they will offer increased productivity, but they are more fragile and need more careful handling. Cermet cutters also will provide excellent results on iron-based heads.

The basic formula for the recommended spindle speed and feed rates for the different valve seat materials and different carbide grades is: [Surface Speed x 12 / 3.142] / Valve Seat Diameter (in inches) = Engine Speed. Cutting speeds are always calculated in feet per minute, while Feed rates are always calculated in inches per revolution. Feed Rate (inches per revolution of the cutter) x Spindle RPM = Inches per Minute. The following table should prove useful as a general guide:

	C2 Carbide		**C4 Carbide**	
Valve Seat Internal Diameter	**Spindle Speed**	**Feed Rate**	**Spindle Speed**	**Feed Rate**
1.375” (34.925mm)	550 RPM	165 Inch / Minute	700 RPM	210 Inch / Minute
1.500” (38.100mm)	500 RPM	150 Inch / Minute	650 RPM	190 Inch / Minute
1.625” (41.275mm)	470 RPM	140 Inch / Minute	600 RPM	180 Inch / Minute

Valve Stem Seals

Do not use an oil seal on the exhaust valve guide in a mistaken attempt to reduce oil consumption. High gas pressures within the exhaust port momentarily restrict oil from going down the valve stem to both that induced by mechanical transference resulting from valve stem movement and that by capillary action. However, the inertia of the hot exhaust gases traveling nearly perpendicular to the nose of the valve guide prevents these gases from traveling up the valve stem and consequently reducing lubrication. As the pulse of hot exhaust gases passes out of the port it leaves a partial vacuum in its wake, ambient pressure within the rocker arm cover then forces oil down the valve stem in order to both lubricate the bore of the valve guide and provide a heat-conducting medium. Thus, the absence of a valve stem oil seal on the exhaust valve guide will have no practical effect upon oil consumption. The film of oil on the valve stem is an essential part of the cooling of the exhaust valve as it fills the gap between the stem and the bore of the valve guide, acting as a medium for conducting heat out of the valve. Because bronze valve guides have closer operating clearances, valve stem seals are not only unnecessary on the exhaust valve guides, but are actually undesirable as they reduce lubrication of the valve stem, accelerating wear of not only the valve stem and its valve guide, but also of the valve seating surfaces as a consequence of attendant misalignment.

However, the opposite is the case where the intake valve is concerned. The low atmospheric pressures in the intake ports draw oil down the stem of the intake valve quite readily, leading to high oil consumption and carbon buildup on the head of the valve and inside of the combustion chambers. Additionally, the oil being mixed with the incoming fuel / air mixture consequently interferes with combustion efficiency and actually lowers its octane rating, making preignition a very real risk. Always install the highest quality valve stem seals on intake valve guides.

Deflector type valve stem seals grasp the valve stem, moving up and down with the valve, shielding the valve guide like an umbrella. Positive valve stem seals remain in a fixed position on the valve guide, acting like a squeegee to control lubrication of the valve stem as it slides in the valve guide. An insufficient supply of oil causes premature wear of both the valve stem and valve guide, while too much oil entering the valve guide results in excessive oil consumption, faster buildup of carbon deposits on the piston crowns, valves, and inside of the combustion chambers, and faster spark plug fouling, all of which are sometimes blamed on worn rings or worn valve stems.

Unlike the other seals of the engine where the goal is zero leakage, the valve stem oil seal must produce a controlled flow (regulated) leak. It is much more difficult to achieve controlled flow leakage because the margin for error is so small since it is so important for a thin film of oil to remain between the valve stem and valve guide. However, the amount of oil used to form this film must be strictly controlled. For this reason, do not use the Original Equipment O-ring type valve stem seals (BMC Part # AEK 113, Moss Motors Part # 297-108) on the intake valves. The design of this type of oil seal only permits it to prevent oil trapped inside of the valve spring retainer cap (cup) from draining down the valve stem and puddling atop the valve, a condition that induces smoking upon startup and leads to heavy, flow-restricting carbon deposits atop the heads of the valves, as well as carbon accumulation on the roofs of the combustion chambers and the crowns of the pistons. They actually work by making the oil coming down from the rocker arm flow outward to the edge of the vale spring retainer cap (cup) and then down the outside the diameter of the spring. It does not seal the valve stem so much as it directs the oil away from it. This archaic design characteristic is based on the theory that whatever oil goes down the valve stem into the valve guide and from there into the combustion chamber occurs by gravity flow only. the theory disregards the tremendous vacuum forces acting upon the lower end of the intake valve guide and the valve stem, as well as the mist and spray effect that the rapidly reciprocating springs, rocker arms, and pushrods have upon the oil inside of the rocker arm cover when the engine is running, so go ahead and install them. Being made of Nitrile, the Original Equipment O-ring type valve stem seals are prone to failure when operating under thermal conditions above 200° Fahrenheit (93.3° Celsius), a temperature commonly attained in even Original Equipment specification engines when working under a heavy load or in high ambient temperatures. This being the case, they should be periodically checked for signs of deterioration. Every 20,000 miles is probably often enough. Despite the misleading exploded-view diagrams that have been published for many years implying that the O-rings are to be placed below the valve spring retainer cap (cup), the O-rings actually are meant to be fitted inside of the valve spring retainer cap (cup), and are squeezed tight by the retaining cotters. All that you need is a valve spring compressor. I check them by inserting and swiping them with a thin, blunt sewing needle. If they're bad, they will tear or chip. The O-rings do in fact prevent oil in the valve spring retainer caps (cups) from puddling atop closed valves and in the combustion chamber when the engine is at rest, so go ahead and install them,

Instead, install onto the intake valve guides a set of Fel-Pro valve stem seals that require no modification of the valvetrain components (Fel-Pro Valve Stem Seal Part # SS 70373 for Chevrolet Vega 4 cylinder 140, 1986-1992 Ford 351 Windsor; also Advanced Performance Technology Part # 70373). These seals have to be slipped over the valve stem using a protective cover before being pressed onto the valve guide with a special tool. Being made of Viton, they are not prone to failure until thermal conditions rise above 450° Fahrenheit (232.2° Celsius), which is much higher than that which any properly maintained street engine experiences. In addition, these positive guide design valve stem seals do a far superior job by eliminating vacuum loss. As a side benefit of the elimination of this vacuum interference with flow through the ports, the fuel-air mixture is more stable and can be more accurately metered to a finer degree, thus increasing both power output and fuel economy.

If you fail to use the O-ring type stem seals on the exhaust valves, you may expect to see a puff of smoke upon starting the engine up from cold. If you have no seals of any kind on the exhaust valves, a small amount of oil may be expected to drain out of the valve collars and run down the valve stems while the engine is off. Most of it will puddle atop the closed valves and be burned inside of the exhaust system. However, if an exhaust valve is open, the oil will drip into the combustion chamber and form some carbon on the piston crown when combustion starts. If the engine is permitted to run for a while, the carbon will hopefully be burned away. However, if the engine is used for short trips and not allowed to reach full operating temperature, then you will be faced with carbon buildup and all of its attendant problems. The proper sequence for fitting the O-ring type stem seals is to first fit the valve springs and their vale spring retainer caps (cups), compress the valve springs by means of pressure on their valve spring retainer caps (cups), then fit the O-ring onto the shoulder of the valve stem. There is a recess provided in the valve spring retainer cap (cup) for retention of the oil seal. Next, fit the collets, then release the valve spring.

Cylinder Head Studs

Replacement of your ancient-and-probably-stretched-by-now cylinder head studs (BMC Part # 51K 245 (Short), 51K 246 (Long)) with new Original Equipment ones from Brit Tek (Brit Tek Part # HSK001) in the USA or from Octarine Services (Octarine Part # 51K281KIT) in the UK, or stronger ones made of 8740 steel from ARP (Brit Tek Part # HSK002, ARP Part # 206-4202) is also recommended. This material is heat-treated to provide a tensile strength in the 200,000 psi range, which is substantially stronger than the Original Equipment cylinder head studs. The ARP studs and nuts are rated at 190,000 PSI, which is considerably greater than a Grade 8 machine bolt, which is rated at 150,000 PSI. The nuts are of the twelve-point type so that a socket will hold on to them at much greater torque than is possible with a hex head nut, which will either round over or cam out of the socket. Because ARP rolls the threads onto their studs rather than cutting them, this “forging” not only strengthens the steel in the vulnerable areas of the threads, it does not leave sharp edges that create stress risers that can lead to thread fractures. The ARP cylinder head stud kits include hardened steel machine washers and twelve-point nuts. Although the nuts may be smaller than the Original Equipment nuts, they are of much greater tensile strength.

When working on an engine that is at least 30 years old, it is not uncommon to encounter a stud that refuses to come loose. Most owners simply douse the offending stud with penetrating fluid and let it soak in overnight. The following morning, two nuts are then placed on the stud and the bottom one used as a jam nut so that the stud can be turned. All too often, the tired old stud breaks off, usually at its juncture with the engine block. This unfortunate cause of much foul language can be avoided. An `Ol-Timey-Mechanic's little-known trick that works quite well is to heat the stud / engine block area as hot as you can get it with a torch, and then touch the junction of the stud and the engine block with candle wax (I use leftover little birthday cake candles for this) until you have a puddle of wax around the stud, then let the area cool a bit. The heat-thinned wax seems to seep down along the threads as the area cools and lubricates the stud so that you can back the studs out.

Steel studs have a different coefficient of expansion than that of a cast iron engine block, and preloading them will aggravate the effect of this factor by increasing stress on the engine block. Due to this different coefficient of expansion, if the steel studs are bottomed out against their shanks inside of the engine block, then the consequent preloading can cause the deck area around them to distort upwards as they expand more than the engine block, and that could lead to a blown cylinder head gasket, or even a cracked deck of the engine block. When the engine block cools, being a casting, it will tend to return to its original flat shape if it has not cracked. This, in turn, will drive you to despair when you try to figure out why your cylinder head gasket has been leaking. This becomes increasingly likely if material is removed from the deck of the engine block in order to significantly raise the compression ratio, thus decreasing its rigidity.

Chamfer and retap the threads in the engine block prior to installing the cylinder head studs. If they are not chamfered, then the threads that are flush with the deck will lift when the cylinder head stud nuts are torqued and distort the cylinder head gasket enough to cause it to blow. This crucial machining procedure is often overlooked. Even if it is not, a dull chamfering bit can leave a ridge at the top of the hole and that too can cause a cylinder head gasket to blow. Should the cylinder head stud spin or wobble in its threads when installed dry, check to be sure that the studs are not undersize. Do not make the all-too-common mistake of attempting to torque the cylinder head studs down onto the engine block as this may lead to cracking of the mounting lugs inside of the engine block. Because the cylinder head studs extend into lugs that serve the secondary purpose of reinforcing the deck of the engine block from the interior of the coolant jacket, any cracking of the lugs will allow coolant to leak past the cylinder head studs and undermine the sealing of the gasket. It is always possible that a previous owner may have already made this mistake, so coat the bottom threads of each of the cylinder head studs with a flexible sealer such as Fel-Pro Gray Bolt Prep immediately prior to the torquing of the cylinder head. Torquing of the nuts of the cylinder head studs to their specified 45-50 Ft-lbs will accomplish the task of securing the studs in the engine block just fine. Never use a thread locking compound as it will result in damage to the threads whenever the studs are removed, thus rendering them useless. Coat both the upper threads and the smooth shanks of the cylinder head studs with copper antisieze compound or any other heat-resistant antisieze compound in order to prevent corrosion. This will make removal of the cylinder head in the future into an easier task.

Be aware that it is not unknown for suppliers to accidentally ship the wrong cylinder head studs. The eleven cylinder head studs of B Series engines are 3/8" (.375" / 9.525mm) in diameter with the upper sections of having 24 threads per inch and the lower sections having 16 threads per inch. Seven of these are 4 ½" (4.5" / 114.3mm) in length, while the remaining four are 6 ¼" (6.25" / 158.75mm) in length.

When the engine is started and begins to heat up the cylinder head stud must stretch as the cylinder head grows in thickness. When the engine is stopped and allowed to cool, the cylinder head stud must shrink to its original length in order to keep the cylinder head gasket sealing for the next cold engine start. When the cylinder head stud has been submitted to years of repeated stretching and shrinking, it will eventually fail. If the cylinder head stud is removed for overhaul and reused, then the re-torque procedure can further increase the chances of failure of the cylinder head studs. When failure occurs and the blown cylinder head gasket subsequently allows antifreeze and lubricating oil to enter the combustion chamber, serious engine damage will result. Stretched cylinder head studs will not hold their torque settings and will lead to a leaking or blown cylinder head gasket and possibly a warped and / or cracked cylinder head.

Repeated retorquing of stretched cylinder head studs will likely result in a cracked cylinder head. Do not attempt to replace them with bolts. When a bolt to is torqued, it is reacting to two different forces simultaneously: stretching, and twisting. This being the case, a torque reading does not accurately reflect the amount of stretch of the fastener. On the other hand, when torque is applied to the nut, a properly installed stud will stretch only along its vertical axis. I never reuse the Original Equipment cylinder head studs because after 30 to 48 years of service, I cannot see the point in putting my trust in them. The financial cost of the implied mechanical complications that can be consequent to a blown cylinder head gasket just is not worth the savings on the price of a set of new cylinder head studs. Vendors that sell what they claim is the equivalent of Original Equipment cylinder head studs never seem to be able to give any specifications on what they are selling, probably because they simply do not know and cannot be bothered to find out if what they are selling is really equivalent to the Original Equipment item. That is why I use (and recommend) the ARP items, not simply because of their superior tensile strength.

Prior to lowering the cylinder head onto the engine block, follow the aircraft mechanic's practice of wrapping the threads of the cylinder head studs with tape in order to protect them from damage. However, do not tape the center cylinder head stud on the distributor side of the engine. It is a very tight fit in the cylinder head as it serves to locate the cylinder head fore and aft. Do not make the common Beginner's Mistake of presuming that because you have installed extra-strong cylinder head studs you can apply huge amounts of torque to their compression nuts in order to attain a more effective seal on the cylinder head gasket. This will most likely result in distortion of their mounting threads in the deck of the engine block. As a result, the clamping force will be reduced and the cylinder head studs will consequently loosen, leading in turn to a blown cylinder head gasket. In addition, over-torquing can crush the cylinder head gasket, clamping the fire-rings down so hard that they form a sharp edge that either splits or gets burnt away by combustion chamber temperatures. In the case of the steel-fillet insert type of cylinder head gasket, it twists the fillets out of place. This crushing also leads in turn to leakage of combustion gases and coolant, as well as a blown cylinder head gasket. Alternatively, particularly on Original Equipment specification cylinder head studs and nuts, increasing the torque value will deform the threads on the nut-end of the cylinder head stud, inverting the thread rather than tightening the cylinder head down. This weakens the thread engagement capability, causing the nut to slacken off further when the engine gets hot. Always use the torque sequence and torque values recommended by the manufacturer of the gasket.

Use either the Original Equipment hardened thick cylinder head stud machine washers (BMC Part # PWN 106, Moss Motors Part # 324-175) or replacement items of the best machine quality (thick and with machined faces) (APT Part # W3834) on the cylinder head studs, never thin mild steel ones from a hardware store. Make sure that the washer seating surfaces on the mounting bosses of the cylinder head are machined flat with an end mill after the cylinder head has been skimmed so that they will be on a parallel plane to the mating surface. That way the torque readings will accurately reflect evenly distributed pressure. Be sure to put an anti-seize compound on their threads prior to installing the cylinder head compression nuts and torquing them to the cylinder head. While oiling of the threads is commonly done in order to protect them from rust, the antisieze compound will do an adequate job of protecting the threads from corrosion. If you are really paranoid about the exposed sections of the threads corroding, then use 3/8"-24 UNF acorn nuts (dome nuts).

When a cylinder head gasket is installed between the cylinder head and engine block, tightening the cylinder head stud compression nuts compresses the gasket slightly, forcing the soft facing material on the gasket to conform to the small irregularities on the mating surfaces of both the cylinder head and the deck of the engine block. This allows the gasket to "cold seal" so that it will not leak coolant before the engine is started. The cylinder head gasket's ability to achieve a positive cold seal, as well as to maintain a long-lasting leak-free seal, depends on two things: its own ability to retain torque over time (which depends on the design of the gasket and the materials used in its construction), and the clamping force applied by the cylinder head stud compression nuts. Even the best cylinder head gasket will not maintain a tight seal if the cylinder head stud compression nuts have not been properly torqued down in the appropriate sequence. The amount of torque that is applied to the cylinder head stud compression nuts, as well as the order in which the machine bolts are tightened, combine to determine how the clamping force is distributed across the surface of the gasket. If one area of the gasket is under high clamping force while another area is not, it may allow the gasket to leak at the weakly clamped point, so the cylinder head stud compression nuts must be tightened to a specified value in a specified sequence in order to assure the best possible seal.

Another consequence of failing to torque the cylinder head stud compression nuts properly can be cylinder head warpage. The uneven loading created by unevenly tightened cylinder head stud compression nuts can distort the cylinder head. Over a period of time, this may cause the cylinder head to take a permanent set. Use an accurate torque wrench to tighten Original Equipment type cylinder head stud compression nuts in three to five incremental steps, while following the recommended sequence and torque specifications. Gradually tightening down the cylinder head stud compression nuts creates an evenly distributed clamping force on the gasket and minimizes distortion of both the cylinder head and the cylinder head gasket. It is a good idea to double-check the final torque readings on each cylinder head stud compression nut in order to assure that none of them have been missed and that the nuts of the cylinder head studs are retaining torque normally. If a cylinder head stud compression nut is not coming up to normal torque or is not holding a reading, it means trouble. Either the stud is stretching or, even worse, the threads are pulling out of the engine block. If a gasket requires retorquing, run the engine until it reaches normal operating temperature, then shut it off and retighten each cylinder head stud compression nut in the same proper sequence as before while the engine is still warm. Should the engine happen to have an aluminum alloy cylinder head, however, do not retorque the cylinder head stud compression nuts until the engine has cooled back down to room temperature. In the case of either type of cylinder head, when being removed their cylinder head stud compression nuts should be systematically loosened using the same pattern and manner as that used when they are torqued down.

Cylinder Head Gaskets

Clough & Wood in the UK made the Original Equipment cylinder head gasket on the early engines. These were asbestos-faced with copper on one side, and on its other side it was faced with either steel or copper that was colored in order to look like steel. While technically obsolete, they work very well with cast iron cylinder heads at standard compression ratios if correctly installed between a redecked cylinder block and a skimmed cylinder head, and accurately retorqued after initial running-in. For those who insist that their car be as "Original" as possible, The Roadster Factory sells them here in the USA.

While today's sealants are excellent and today's modern cylinder head gaskets possess greater compressibility than those of the past, they can compensate for warped mating surfaces only to a very limited degree. Instead of using an Original Equipment Clough & Wood cylinder head gasket, use either a Payen or a Fel-Pro cylinder head gasket or one that is marked "FRONT/TOP" so that installation will be a straightforward affair. These resin-impregnated cylinder head gaskets have copper sealing rings in order to better resist excessive crush pressures, and require no additional sealing coatings as they are surface-coated for microsealing. Best of all, they provide a clean release upon removal so that you will not have to endure the onerous chore of scraping away bits of old fiber gasket material. They are also particularly appropriate for use on engines that have been converted to aluminum alloy cylinder heads as they handle the differing coefficients of expansion of a cast iron engine block and an aluminum alloy cylinder head quite well. These resin cylinder head gaskets produce a better seal by softening when it gets hot, heat-bonding to the mating surfaces of the cylinder head and the engine block. However, if the mating surfaces are not scrupulously clean, or if you allow the coolant to pressurize before the bonding has fully taken place, then the resin-impregnated cylinder head gasket will not bond properly to the mating surfaces and subsequent leaks will be more likely. Consequently, you should run the engine without any coolant inside of it until the engine gets hot so that a good bond can be achieved. While doing so, the V-belt that drives the coolant pump should be removed so that the seal of the coolant pump will not be damaged by its shaft running against it without the benefit of the lubricants contained in the coolant.

Because of the differing coefficients of expansion, a copper cylinder head gasket should never be used in combination with a cast iron engine block and an aluminum alloy cylinder head. Racers like copper cylinder head gaskets for two reasons: First, because they have high crush resistance, thus permitting the application of higher levels of torque onto the nuts of the cylinder head studs in order to deal with higher pressure levels of both compression and combustion common to racing engines. Second, because racers frequently tear down their engines for inspection. This being the case, they do not want to have to spend time scraping bits of torn fiber cylinder head gasket from the mating surfaces. Beyond those two advantages, copper cylinder head gaskets are obsolete. Certainly the break-in procedure for them can be challenging. Many of these copper cylinder head gaskets have the letters "DV" stamped onto them which stands for "Double Varnish". These varnished cylinder head gaskets require that the engine be run without coolant until the cylinder head becomes hot enough for the varnish to soften and form a seal, then left to cool down for several hours, or, ideally, overnight. Of course, the trick is to reach the appropriate temperature without exceeding it and risking damage to the valve seats or the seizure of a piston. Other copper cylinder head gaskets require that they be annealed prior to installation in order to soften them to the point that they will conform to surface irregularities. This may defeat your purpose if you are trying to optimize on cylinder head gasket strength. Since there is oxygen within the copper it can only be annealed (using flame heat) a few times before it work hardens and becomes somewhat brittle. Ideally, a temperature of about 900º Fahrenheit (482º Celsius) is optimal. Afterwards, the copper cylinder head gasket should be allowed to air cool. A flaky post-annealing residue is normal after annealing in air. Once cool (in about five to ten minutes), it should be brushed on a hard, flat surface with a Scotch-Bright pad in order to clean and flatten it. Copper gaskets must always be re-torqued.

Never allow a cylinder head gasket to overhang into the bore of the cylinder, as this will lead to a blown cylinder head gasket and / or internal damage to the engine. The Fel-Pro and Payen cylinder head gaskets have a slightly larger bore diameter than that of the Original Equipment cylinder head gaskets, thus permitting displacement to be increased to as much as 1925cc without the risk of overhanging the bore of the cylinder. They also do not cost an arm and a leg as the Competition cylinder head gaskets do. If you have chosen to build an engine that has a bore diameter that is larger than 3.268" (83.0072mm), then you will need to install a specialized Big Bore cylinder head gasket. Cambridge Motorsport offers a Competition cylinder head gasket for bore sizes of up to 84.4mm (83.5mm bore size = 1950cc displacement). Its triple layer steel shim construction offers reliability even with the highest compression levels. This cylinder head gasket is, to my knowledge, the only reliable solution for Big Bore competition engines, and can be reused up to three times. Cambridge Motorsports has a website at http://www.cambridgemotorsport.com/ .

You will need to retorque the cylinder head immediately after the initial running of the engine. Do not completely loosen the nuts during retorquing. Instead, just back them off enough to get them moving, otherwise you will be measuring the torque to break the stiction between the nut and the stud rather than the torque that results in the necessary stretching of the stud. Note that a cast iron cylinder head should be retorqued while the engine is hot, while an aluminum alloy cylinder head should be retorqued when the engine is cold. Be aware that the manufacturer of both the Fel-Pro and the Payen resin cylinder head gaskets recommends that they not be retorqued after initial installation.

Pushrods

Old pushrods can be trouble. Any sign of pitting or nippling on their ends instantly qualifies them for the scrap bin. In addition, because of the fact that the central axis of each of the tappets is offset from that of the camshaft and the tappets have a .002” (.0508mm) dome on their faces that bear against the surfaces of the camshaft lobes. These surfaces are obliquely slanted in relation to the rotational axis of the camshaft, the tappets rotate in their bores when being lifted by the lobe of the camshaft, thus reducing both friction at the tappet / lobe interface and consequent wear. Should a pushrod become bent, it will prevent the tappet from rotating in its bore, resulting in uneven wear of the domed bottom end of the tappet, which in turn will make accurate setting of the valve clearances impossible and result in eventual ruination of both the domed face of the tappet and the lobe of the camshaft. When installing them, make sure that you put some fresh engine oil onto the cupped upper ends of the pushrods, as well as down the pushrod passages to lubricate both the cupped top of the tappets and the domed lower ends of the pushrods. (You weren’t really going to reinstall those ancient pushrods into a freshly blueprinted engine, now were you? Know what metal fatigue is?)

If you should choose to reuse your old Original Equipment pushrods, they should be inspected for signs of bending and excessive end wear. Remember that the ball ends of the pushrods have mated to their individual tappets and that the rocker arm ball adjusters (11/32") have mated to the cupped ends of the pushrods over the years, so when you take them out, keep them all in ordered sets and make sure that they are oriented the same as they came out of the engine (cup end up). Whenever you remove a pushrod, take care to always wriggle it in order to release it from its tappet, otherwise the suction on its ball end inside of the top of the tappet may pull the tappet completely out of the top of its bore, often resulting in the tappet becoming cocked above its bore. If this should occur, in all probability you will then need to remove the cover of the tappet chest in order to replace it straight into its bore. Clean the pushrods thoroughly, and then use a strong light while examining both ends of each pushrod for signs of scoring inside of their cupped ends and / or flat spots on the ball ends. If you find such a defect, scrap it. Next, apply a very thin coat of machinist's bluing or petroleum jelly onto their shafts. Roll each pushrod on a clean piece of quality plate glass, and then examine the stain on the glass. The presence of any gaps will reveal whether or not the pushrod is bent and should be replaced.

Unlike Original Equipment pushrods, tubular chrome-moly alloy pushrods do not flex at the higher engine speeds that an enhanced-performance street engine can often achieve, plus they have less reciprocating mass and thus will give more accurate valve timing at high engine speeds. This is a problem for both the early short pushrods (72 grams, BMC Part # 11G 241) used in the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines and the later long pushrods (88 grams, BMC Part # 12H 1306) used in the 18V engines as they tend to deflect as much as 5/64" (1.98438mm) at high engine speeds, even when new. This deflection is partially the result of the camshaft being offset away from the cylinder head in order to provide room for the siamesed intake ports while allowing the use of short arm rocker arms in order to reduce their rotational mass. The greater angularity resulting from the earlier long barrel tappet / short pushrod combination created such high side-thrust loadings on the walls of the outer bores in which the tappets rotated that it was necessary to incorporate an oiling cavity into the design of the tappet to ensure adequate lubrication at high engine speeds and so ensure its rotation. In addition, the greater the angularity of the pushrod, the greater its arc motion becomes. The high load area on the pushrod moves closer to the tappet as the tappet travels up the ramp of the camshaft. This makes it even more important to use a tubular pushrod design when using large roller bearing diameters, increased valve lash, very high engine speeds, high rocker ratios, rapid valve train acceleration and high spring pressure. The reduced angularity of the longer pushrods (10.787" / 273.99mm) that are used with shorter, lighter bucket tappets results in decreased side-thrust loadings on the tappets and thus enhances their lifespan, as well as also permitting them to rotate more freely at high engine speeds.

The shorter length (1.500" / 38.1mm), lighter (47.2 grams) bucket tappets (BMC Part # 2A 13) introduced on the 18V-584-Z-L engines will also assist in the goal of reducing reciprocating mass. Due to their having identical diameters of 13/16" (.8125"), the early long barrel tappets (79.7 grams, BMC Part # 1H 822) and the later short bucket tappets are interchangeable when paired with their length-appropriate pushrods. The later Original Equipment short bucket tappet / long pushrod (10.656" / 270.662mm) assembly is 13% lighter than the earlier Original Equipment long barrel tappet (2.298" / 58.3692mm) / short pushrod (8.750" / 222.25mm) combination used in the earlier 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines. Crane's lighter (64 grams) chrome-moly tubular pushrods (Crane Part # 905-0004) will also reduce inertia in the reciprocating mass of the valve train by about 20% when compared to that of the later Original Equipment 18V short bucket tappet / long pushrod combination and by 30% when compared with the earlier Original Equipment 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines' long tappet / short pushrod combination. This reduction of reciprocating mass permits most high-lift camshaft lobe profiles to be employed without the expedient of using stronger valve springs in order to prevent valvetrain float at the upper end of the powerband, thus reducing wear of both the tappets and the lobes of the camshaft.

The hollow, tubular construction of the Crane pushrods also endows them with the structural advantage of greater rigidity over that of the Original Equipment solid pushrods. Because the tappets of the B Series engine are offset outward from the rocker arms, the pushrods are forced to reciprocate at an angle that increases as the tappets move upward. As this angularity increases along with tappet lift, so does the arc motion of the upper end of the pushrod, and thus bending stress on the pushrods also increase along with compression load, especially in the case of solid pushrods employed with high-lift camshaft lobe profiles. Pushrod deflection is the consequence. In turn, deflection results in the shortening of the effective length of the pushrod, and thus the accuracy of the valve lift is sacrificed. The superior rigidity of tubular pushrods all but eliminates this unwelcome deflection, as well as moving their period of harmonic vibration to an engine speed that the Original Equipment B Series engine cannot attain, thus ensuring not only accurate valve timing, but reducing fatigue stresses on the valve springs. Both end fittings are heat-treated, making for superior wear characteristics. Crane can also supply them in custom lengths if necessary in order to compensate for skimming of the deck of the engine block or of the cylinder head. Their website can be found at http://www.cranecams.com/ .

Due to their larger diameter (.3125" / 79.375mm vs. .280" / 7.112mm), it will be necessary to relieve the passageways in the cylinder head for the pushrods in order to eliminate interference. Be aware that simply boring out these passageways to .660" (17.664mm) diameter in order to accomplish this may leave insufficient material to permit portwork to be done, as well as running the risk of breaking through into a coolant passageway. Instead, the pushrod passages should merely be elongated toward the centerline of the engine.

Sprockets

The reuse of old sprockets is false economy. Always replace both of the sprockets and the camshaft drive chain at the same time. When new, the links of the camshaft drive chain perfectly engage all of the teeth of the sprocket, spreading the load of drive energy across all of the teeth of the sprocket. A worn camshaft drive chain only pulls against the end tooth of the sprockets, the concentrated drive load inducing accelerated wear of both the teeth of the sprocket and the camshaft drive chain. Conversely, a worn sprocket no longer matches a new camshaft drive chain, with the same result. A set of worn sprockets will result in uneven and accelerated wear of a new camshaft drive chain, thus causing its length to oscillate. This in turn will accelerate wear of the camshaft drive chain tensioner and the gear teeth of both the ignition drive shaft and its mating gear on the camshaft. The oscillation of the camshaft drive chain will cause both the valve and the ignition timing to "wobble" inconsistently, playing havoc with performance. Timing scatter induced by a worn camshaft drive chain can reach up to 15°. In some high performance engines, 1 BHP is lost for every degree that the timing of the camshaft is out of phase with the crankshaft, with the ratio of power loss increasing if it exceeds 6°.

It should be understood that the higher the desired state of tune of an engine, the more highly necessary it becomes that the camshaft be operating in proper time with the crankshaft. The timing of an Original Equipment camshaft was a simple procedure when using Original Equipment sprockets. These components were made with such precision that all that was necessary was to bring the piston of the front cylinder to Top Dead Center and align the dots on both the drive and the driven sprockets. However, such is not always the case with the aftermarket components available today. The lobe profiles of these camshafts are frequently machined onto either blanks whose cast profiles are frequently of milder design, or onto camshafts with lobes that are built up by welding (both types are sometimes referred to as "regrinds") and as such it is common for the manufacturer to be forced by the geometries involved into relocating the lobe centerline to a position that has a different relationship to the keyway. Consequently, using the factory technique of simply aligning the dots on the sprockets in order to time the camshaft is foolhardy. Instead, the use of a degree wheel and a magnetic stand with a dial indicator is essential to achieving a proper in-phase setting. Do not be taken by surprise if the keyway of the camshaft and that of the driven sprocket of the camshaft are so misaligned that offset Woodruff keys cannot be made to fit, forcing you to purchase an adjustable camshaft driven sprocket.

Using the manufacturer's recommended timing when first installing a camshaft is usually a very good starting point. However it should be borne in mind that the manufacturer of the camshaft was unaware of the exact specifications of your particular engine, and so the recommended setting is actually for a theoretical "average" high performance engine in which it is expected to be employed. It is not unknown for the best results to be attained at a setting that is as much as 5° from the manufacturer's recommended setting. A series of runs on a dynometer will permit you to zero in on the optimum setting for your particular engine. Because dynometer time has to be paid for, the extra cost of an adjustable camshaft driven sprocket may be more than compensated for as a result of its quick adjustable capability resulting in less expense for time spent using the dynometer.

When installing both the drive and the driven sprockets, take care to be sure that they are properly aligned on the same plane by inserting spacer shims onto the nose of the crankshaft; otherwise, the sideloading of the camshaft drive chain will cause it to wear rapidly, both varying and retarding both the camshaft and the ignition timing. Push the crankshaft to its rearmost position against its thrust washers, and then rotate the crankshaft so that its keyway is in the Top Dead Center position. Next, rotate the camshaft without the pushrods and tappets installed so that its keyway is at approximately the One o'clock position. Place both the drive and the driven sprockets onto a clean, flat surface with their keyways oriented to the same positions as those of their corresponding mounting shafts, then carefully put the camshaft drive chain onto them so that the keyways maintain the same orientation as before when the sprockets pull the camshaft drive chain taught. Slide both the drive and the driven sprockets onto their respective shafts and rotate them slightly so that their keyways align with those of their respective mounting shafts. Now, push both the drive and the driven sprockets as far back on their shafts as they will go and install their Woodruff keys. Place a metal straight edge onto the face of the camshaft driven sprocket and use a feeler gauge in order to determine the gap between it and the face of the camshaft drive sprocket on the crankshaft. The skewed gears on the camshaft and the oil pump drive impart a rearward thrust to the camshaft, so I do this with both the camshaft and the crankshaft at their rearward positions. Because the face of an Original Equipment camshaft driven sprocket projects outward from its teeth by .005" (.127mm), subtract .005" (.127mm) from the gap figure in order to determine the required total thickness of the needed distancing shims. Once the shims are in place, install the oil thrower, along with the crankshaft drive sprocket and its lock washer. Be aware that it is easy to confuse the lock washer for the camshaft driven sprocket with the lock washer for the lock washer for the camshaft drive sprocket. The lock washer for the camshaft driven sprocket has a tongue that fits into the keyway and the central hole is fractionally smaller than that of the crankshaft washer. On the other hand, there should be no tongue or slot in the central hole of the lock washer for the crankshaft. If you look at the camshaft drive sprocket, you will see a moon shaped depression on the side that is opposite from its key slot. The crankshaft lock washer should be folded into this depression and then folded forward over one flat of the machine bolt on the opposite side in order to secure the machine bolt. At this point, you can install the camshaft drive chain tensioner.

The Camshaft Drive Chain Tensioner

A duplex-type camshaft drive chain tensioner (Advanced Performance Technology Part # BCT-1) , the 3/8" pitch duplex camshaft drive chain (BMC Part # 2H 4905, Advanced Performance Technology Part # TC-BA) and both the drive and the driven sprockets of the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GJ, 18GK, and through 18-V-584-Z-L and 18-V-585-Z-L engines (crankshaft drive sprocket BMC Part # 12A 1553, camshaft driven sprocket BMC Part # 11G 203), plus a nitride-hardened rocker shaft (Advanced Performance Technology Part # RSB-T) will aid in achieving long-term durability. In addition, an adjustable camshaft driven sprocket (Brit Tek Part # PGS001), although expensive, will enable you to easily keep the camshaft operating in phase with the crankshaft as all camshaft drive chains wear and thus "stretch." However, the same objective can be attained in a less expensive manner by using offset Woodruff keys (Advanced Performance Technology Part #'s OSK62, 2°; OSK66, 4°; OSK68, 6°; OSK70, 8°) in order to adjust the timing of the Original Equipment camshaft driven sprocket, although adjustments made in this manner are far more troublesome and tedious, not to mention far more time-consuming. Remember, the higher the state of tune of an engine, the more sensitive it will become to variations and inconsistencies in valve timing, thus the more frequent the need to readjust the timing of the camshaft will be in order to maintain the engine's performance at a level of peak efficiency.

The camshaft drive chain tensioner is an intriguing and often-misunderstood item. Due to the fact that it receives pressurized oil from the front bearing of the camshaft, it is usually presumed to be a device that uses hydraulic pressure in order to maintain tension on the return circuit of the camshaft drive chain. In fact, the pressurized oil flowing from the low pressure circuit into it through the spigot in its rear face merely ducts the oil flow outward onto its slipper pad in order to both lubricate the camshaft drive chain and reduce its friction against the slipper pad, thus extending the service life of both components. In reality, it functions in a rather simple and purely mechanical manner.

As the camshaft drive chain wears, it becomes longer and increasingly slackens on its return circuit. As the slipper plunger extends under the pressure of its compressed coil spring, the limiting peg thrusts against the smooth top of the helical slot in the cylinder and causes it to rotate. When the next indentation in lower edge of the helical slot aligns with the limiting peg, the plunger is prevented from moving back into the body of the tensioner mechanism, thus maintaining the correct tension of the camshaft drive chain.

When assembling the camshaft drive chain tensioner, it is very important that all of its components be clean. Install a new slipper pad onto the camshaft drive chain tensioner and check that the mechanism is functioning properly. Be sure that the high end of the slider ramp of the tensioner pad is presented to the feed side of the camshaft drive chain. Also, be sure to inspect the bore of its adjuster body for ovality (+. 003" / +.0762mm max.). Should it prove to be worn out, a new one can be obtained from Advanced Performance Technology (APT Part # BCT-1).

Insert the coil spring into the plunger and place the helically cut cylinder onto the opposite end of the coil spring. Compress the coil spring until the helically cut cylinder enters into the plunger, then use an Allen head wrench to hold the coil spring static while you turn the helically-cut cylinder clockwise until its end is below the limiting peg of the plunger and the coil spring is held under compression. Now, remove the Allen wrench and slide the assembly into the body of the tensioner. With its backplate against the front plate of the engine, install the camshaft drive chain tensioner along with its tab washer (BMC Part # AEC 340), torque the machine bolts to 10 Ft-lbs, and then bend over the lock tabs in order to secure them.

When the tensioner is newly installed, its spring loaded ratchet mechanism must be released after fitting in order to tension the camshaft drive chain. If this is not done, the ratchet mechanism will not maintain the camshaft drive chain under proper tension after the engine stops and the oil pressure ceases, whereupon the camshaft drive chain will rattle at idle speed when the engine is restarted. Release the plunger by inserting and rotating the Allen wrench clockwise. Under no circumstances should you attempt to rotate the Allen wrench counterclockwise (anticlockwise) or attempt to force the plunger outwards. If the tensioner pad is tensioned by means of forcing the ratchet through via turning the ratchet mechanism itself, the Allen key drive will become detached. The spring will then become displaced and the tensioning mechanism will be free to float up and down, defeating its purpose. Once the camshaft drive chain has been properly tensioned, you may fine-tune the timing of the camshaft.

The Harmonic Damper

Each time a cylinder fires, enough force is applied to the associated crankpin to cause the crankshaft to not only turn, but to twist that crankpin out of alignment with the others as well. Because steel is elastic in nature, this twisting causes an accompanying rebound of the crankpin back into the opposite direction and into its original alignment with the other throws of the crankshaft. This creates what is known as torsional vibration. When the torsional vibrations of the multiple throws of the crankshaft combine, the result is referred to as harmonic vibration. At certain frequencies, harmonic vibration is damaging to the crankshaft, inducing metal fatigue and subsequent fracturing. Thus, the actual purpose of a harmonic balancer (harmonic damper) is to dampen the harmonic vibration (the first torsional natural frequency) of the crankshaft, not to dynamically balance the engine. The harmonic balancer (harmonic damper) also serves to dampen vibration that travels along the camshaft drive chain, effecting both valve and ignition timing and thus decreasing the power output of the engine.

The harmonic balancer (harmonic damper) (often referred to as a Torsional Vibration (TV) Damper by our British brethren) of the B Series engine is made up of three parts: an inner steel mount that attaches to the forward end of the crankshaft, a dampening medium made of rubber, and an outer pulley wheel for both mounting and driving a V-belt as well as having the timing marks for setting the ignition timing. The harmonic balancer (harmonic damper) has a heavy ring as part of the pulley, attached to its hub by a bonded rubber ring. Because the heavy ring gets out of phase with the resonance of the engine's vibration, it is the oscillation of the rubber ring on the hub that absorbs the vibrations. As it ages, this rubber medium hardens and eventually develops cracks, losing its ability to perform its function. In extreme cases, the rubber can actually separate from its adjacent steel components, destroying its function and allowing the outer wheel with its timing marks to slip. A separated harmonic balancer (harmonic damper) is definitely a major disaster. The harmonic balancer (harmonic damper) itself is rather bulletproof, but the ultimate evil is to put it in solvent for cleaning and then leave it to soak, as the solvent destroys the rubber. Replacement or refurbishment of your tired old harmonic balancer (harmonic damper) is highly advisable as it reduces torsional stress on both the crankshaft and the camshaft, as well as reducing wear of the camshaft drive chain, coolant pump, and alternator due to the reduced amplitude of oscillating stress loadings. It should be torqued with a 1-5/16" socket. Your Original Equipment harmonic balancer (harmonic damper) can be economically rebuilt to as-new specifications by a specialist such as Damper Dudes. They are located at 6180 Parallel Drive, Anderson, CA 96007, (800) 413-2673. They have a website at http://damperdudes.net/ . Another well-qualified source for this service is Damper Doctor. They have a website at http://www.damperdoctor.com/ . However, Advanced Performance Technology' stainless steel version (APT Part # 18CSP-2) is even better than the Original Equipment item as it has the additional advantage of having provision for easy removal.

There is a problem with the harmonic balancer (harmonic damper) that uses rubber as a medium for damping: the rubber is effective only at a specific range of engine speeds. This being the case, in order to serve its purpose, the harmonic balancer (harmonic damper) must be designed so that its rubber damping medium is attuned to the engine speeds during which harmonic vibration occurs. Fortunately, there is a solution to this problem: the viscous damper. A vviscous damper consists of an inertia ring in a viscous fluid. The torsional vibration of the crankshaft forces the fluid through narrow passages that dissipates the vibration in the form of heat. A viscous damper cancels harmonic vibration equally well regardless of the engine speeds at which it is produced. It also has the advantage of being amplitude sensitive, responding to the intensity of vibration regardless of its frequency.

Reducing weight that is attached to the crankshaft, as when drilling and chamfering holes in the camshaft driven sprockets, harmonic balancer (harmonic damper), and removing weight from the flywheel, can elevate the period at which harmonic vibration occurs to a part of the powerband not normally used, thus prolonging the life of the crankshaft. Be aware that the Original Equipment Hepolite camshaft driven sprockets were produced using a sintering process, and as such should never be drilled in an attempt to lighten them and thus reduce their rotating mass. Such machining operations should be attempted only with solid steel sprockets. This is just as well, since the teeth of solid steel sprockets enjoy the virtue of a longer service life than those of a sintered sprocket.

Exhaust Manifolds

Simply put, an engine creates power by inhaling a fuel / air mixture, combusting it, and then exhaling it. There is no point in trying to get more fuel / air mixture into an engine if the hot combustion gases cannot get out efficiently, so let us tackle the subject of exhaust systems first. Because its performance is critical to power output, it is necessary to regard the complete exhaust system as an engine component.

The Original Equipment pre-1975 factory exhaust manifolds, of which there were two models, are surprisingly good performers. The Original Equipment SU exhaust manifold used with the 1½" SU HS4 Series carburetors' SU intake manifolds (BMC Part # 12H 911, 12H 1397, 12H 2571, 12H 2575, and 8G 767) have a mounting flange thickness of 9/16" (14.2875mm) and can be readily identified by its external casting number of 12H 709 (BMC Part # 12H 709). On the other hand, the Original Equipment SU exhaust manifold used with the 1½" SU HIF4 Series carburetors' SU intake manifold (BMC Part # 8G 774) has a mounting flange thickness of 7/16" (11.1125mm) and can be readily identified by its external casting number of 12H 3911 (BMC Part # 12H 3911). While its designers allowed for the turbulence created by the roughness of the interior surface of the casting, I highly recommend electropolishing to improve the airflow capacity of the cast iron exhaust manifold.

Electropolishing is an electrochemical process used to smooth metal, usually prior to plating. It is commonly performed on a precision casting (such as a window winder handle) or on prepolished sheet metal after it has been formed to shape (such as a bumper) prior to plating it. The item to be electropolished is thoroughly cleaned and subsequently immersed in a chemical bath. A current is then run through and the highest points on the surface of the metal are removed. In a sense, it is the reverse of plating in that metal is removed instead of deposited.

The benefit of electropolishing a cast iron exhaust manifold lies in the fact that because the item is completely immersed into the electropolishing bath, the process can get inside of the runners of the exhaust manifold, reaching into remote areas and otherwise inaccessible curves where human hands and mechanical tools cannot reach so that it will polish the interior of the exhaust manifold quite nicely. This produces a smoother surface that makes for reduced turbulence in the exhaust gas flow, much like the smooth walls of an exhaust manifold that has been constructed of tubular steel. Because an overly smooth mating surface may give sealing problems when used with some gaskets, be sure to instruct the firm that is doing the electropolishing to protect the gasket surfaces with plater's tape.

Another technique for attaining a smooth interior surface in the exhaust manifold is called Forced Extrusion Honing. Extrusion Honing is great because it can remove metal from areas where it is otherwise impossible to do so, such as inside of the long runners of the exhaust manifold. In this technique, a dense mixture of abrasive clay is forced through the interior of the manifold, polishing the surfaces to an even greater degree than can be achieved on a casting through electropolishing. Smaller, more restrictive areas in the cylinder head act like a venturi so that the mixture of abrasive clay flows faster there. This faster flow causes more cutting action and thus the Extrude Honing process by nature removes material where its removal is needed the most. This cutting mechanism is very good at producing runners that flow equally. I have seen a cylinder head in which both the intake and exhaust ports have been subjected to this process and it is very impressive. This service is available from Extrude Hone. Their website can be found at http://www.extrudehone.com/ .

The Cost vs. Benefit factor of these processes is all a matter of personal values. It is true that a decent tubular steel exhaust manifold might be obtained for less money than that of extrude honing and / or electropolishing an Original Equipment cast iron exhaust manifold, but tubular steel exhaust manifolds have much thinner walls that can resonate under the fluctuating pressures inside of the exhaust system and thus are much, much noisier. At certain parts of the powerband, they can actually resonate into an annoying ringing sound. In addition, tubular exhaust manifolds are welded assemblies, and their welds have been known to crack under the repeated stresses of heating and cooling. Further potential problems of misalignment of the exhaust runners with the exhaust ports, and mounting flanges not being on the same plane, all contribute to making the purchase of a tubular exhaust manifold into something of a gamble. The Original Equipment cast iron exhaust manifold has none of these problems. Because of the lesser heat conductivity of cast iron as well as the decreased surface area, an electropolished exhaust manifold will radiate less heat into the engine compartment. Its greater mass will also have the side benefit of reducing noise to a level notably less than that possible with any tubular steel exhaust manifold.

I sincerely believe that a LCB (Long Center Branch) 1¾" diameter tubular steel exhaust manifold will not flow any better than an electropolished Original Equipment cast iron exhaust manifold if it has the same basic runner design. It can also be beneficial to polish exhaust ports, thus reducing carbon buildup that results in the creation of airflow turbulence, as well as the advantage of less heat being conducted into the cylinder head as a result of the decreased surface area of the interior of their ports.

It is important to understand why the pre-1975 Original Equipment exhaust manifolds and LCB (Long Center Branch) 1¾" exhaust manifolds have nominally the same performance. The cross sectional sizes of their internal passages are nominally the same, thus the velocities of the exiting combustion gases passing through them are also nominally the same. This high gas velocity is critical to power output at low engine speeds because the greater the velocity of the exhaust gases, the greater their inertia. Due to the high degree of directional inertia, the exiting combustion gases continue to flow exclusively in the direction of and out past the exhaust valve even though the intake valve is opening. In order to properly understand this phenomenon, one needs to view the cylinder as an being an extension of the combustion chamber. The combusting fuel / air charge exerts pressure upon the piston crown, accelerating the piston down the cylinder. It must be understood that due to the geometry of the crankpin and connecting rod, the piston is decelerated as it passes 90° After Top Dead Center, this geometry-induced deceleration becoming increasingly severe as Bottom Dead Center is approached. However, combustion pressure causes the combusting gases continue to accelerate downward, their inertia causing them to pile up on top of and thus increase the pressure of the atmosphere immediately above and upon the piston crown. Due to this inertia effect, at Bottom Dead Center the atmospheric pressure at the roof of the combustion chamber is actually less than the atmospheric pressure immediately atop the piston crown. Because all forces in nature tend to equalize, at this point the pressurized atmosphere atop the piston crown expands upward, increasing its upward inertia as it accelerates toward the roof of the combustion chamber. If the exhaust valve is open to the point that it has sufficient airflow capacity, then the inertia of the exiting exhaust gases will remain sufficiently high enough to literally scavenge the atmosphere from inside of the cylinder, creating a partial vacuum of as much as 7 PSI (Pounds per Square Inch) less than the ambient atmospheric pressure outside of the engine. This in turn allows the incoming fuel / air mixture to be pushed in not only earlier, but also at a higher velocity (and thus a larger quantity of fuel / air mixture with better fuel atomization) by the greater ambient atmospheric pressure outside the engine, thereby increasing power output. All other factors being equal, a larger-diameter exhaust manifold would decrease this critical velocity, and with it, its benefits.

Reducing temperatures inside of the engine compartment is beneficial for power output, as well as presenting less potential for detonation. Although most carburetors do not have a "brain" or a sensor to measure temperature, all carburetors function similarly in that it is pressure drop across a venturi that is the mechanism that pulls fuel into the venturi. The pressure drop itself is a function of air density. Air temperature also plays into the equation. In operation, carburetors are quite effective at adapting within the limited range of operating conditions that we subject them to. Without cold air induction, a carburetor must accommodate a significantly wider range of air temperatures than if you supplied air from outside the engine compartment. For every 5.4° Fahrenheit (3° Celsius) decrease in temperature that the air ingested by the engine is lowered by, power output is raised by 1%. In view of this scientific fact, although wrapping the exhaust manifold in insulating tape (sometimes called "lagging") may seem to be a good idea in principle, it is a very bad idea in practice. Peter Burgess mentions this problem in his book "How to Power Tune MGB 4-Cylinder Engines." Why is it a bad idea? Heat from the exhaust gases continue being conducted into the cast iron, the insulation factor will be such that heat cannot escape from a wrapped cast iron exhaust manifold. Consequently, both the cylinder head and the exhaust manifold will run hotter. The heat will continue to build up and up, far beyond what the factory engineers designed the exhaust manifold to handle, with the result that the exhaust manifold will warp, often resulting in a leak at the manifold gasket. In addition, the additional heat collected in the iron of the exhaust manifold will also be transferred into the cylinder head, heating the walls of the intake ports and thus reducing the density of the incoming fuel / air charge. Even worse, the coolant passages in the cylinder head were not designed to remove such an excessive amount of heat, thus preignition of the fuel / air charge can become a problem and valve seat life can be shortened. In extreme cases, due to the fact that the exhaust valves for the middle two cylinders share the same central exhaust port, the cylinder head can actually warp between #2 and #3 cylinders, resulting in a coolant leak or a blown cylinder head gasket. In the case of tubular steel exhaust manifolds, the metal will become so hot that it will often spall and form flakes that will eventually disintegrate to form a hole in the area where the heat accumulation is greatest, usually at the junction of the runners. The lagging tape also becomes a moisture trap, accelerating the rusting process that can plague cast iron exhaust manifolds.

Instead of wrapping the exhaust manifold, get it Jet-Hot coated. Jet-Hot coating is a ceramic coating that can be applied to coat both the interior of the exhaust manifold as well as the exterior. The heat will have nowhere to go except out through the exhaust system, thus it will greatly reduce underhood temperatures. This is a significant factor as exhaust manifolds often reach temperatures above 400° Fahrenheit (204.4° Celsius). Thus, the cooler air being inhaled into the engine being denser, more fuel can be mixed with it to result in a more powerful fuel / air charge. Another benefit is that the setting of the thermosensitive 1½" SU HIF4 Series carburetors can remain more consistent. At the same time, it decreases deceleration of the exhaust gases, the sustained gas inertia thus enhancing the pulsed-vacuum effect, resulting in a more effective scavenging of the cylinders. The increased velocity of exhaust gases produced by its higher exit inertia not only clears each cylinder more quickly; the improved vacuum effect inside of the combustion chamber draws in the next fuel / air charge more efficiently. Jet-Hot coating does not contribute to hydrogen embrittlement, a condition associated with chrome plating and other coatings in which microscopic cracking can lead to premature failure. One word of warning to those considering Jet-Hot coating or any other type of ceramic coating: Be sure that the entire surface of the exhaust manifold, both the interior as well as the exterior, and that of the flanges is coated so that the heat of the exhaust gases will travel onward through the system instead of being absorbed and trapped in the metal of the exhaust manifold, otherwise the absorbed heat will create the same problems as in the case of lagging the exhaust manifold with insulating wrap. Warping of the exhaust manifold would become something to be expected, and warping of the cylinder head would also become a distinct possibility. Jet-Hot has a website that can be found at http://www.jet-hot.com/ .

Should you decide to use a tubular exhaust manifold that is not Jet-Hot coated, be sure to use a synthetic rubber gasket (Moss Motors Part # 296-375) and the later rear tappet chest cover, as the cork gaskets that are used in this location tend to fail under prolonged exposure to the extreme heat radiated by such exhaust manifolds. Use of the more warpage-resistant rear tappet chest cover from the 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L and 18V-891-AE-L engines will assist in this as well. Always use the thin cork gasket on the front cover - it lays flat and is actually the right size. Both the synthetic rubber gaskets and the thick cork gaskets are too small to fit well.

As a consequence of the smaller radius turn leading to the bottom of the exhaust port, exhaust gases emerge out of the exhaust port traveling at higher velocities near the top of the port while those at the bottom of the port lose more inertia and thus emerge at lower velocities. Because of this, as piston speed and exhaust gas velocity both begin to decrease near the top of the stroke, the exhaust gases near the bottom of the exhaust manifold runner begin to tumble as a result of their interaction with the higher speed gases above them, and then begin to reverse direction back toward the exhaust port. This phenomenon is referred to as "reversion". For decades, it was believed that concentrically aligning a square or rectangular exhaust port with the exhaust manifold runner would provide the best protection against this phenomenon, and the design of the B Series engine of the MGB took place during that era. However, today we are aware that reversion occurs along the bottom of the port. By locating the entrance of the bottom of the exhaust runners as low as possible, the external area beneath the bottom edge of the exhaust port can then act as a dam to this reverse gas flow along the bottom of the exhaust manifold runners, interrupting it and preventing it from entering the combustion chamber where it would contaminate and partially displace the incoming fuel / air charge, decreasing power and causing the rough running that longer-duration camshaft lobe profiles are notorious for producing, especially at lower engine speeds. It will also help to reduce "Pumping Losses" at low engine speeds and smooth acceleration from low engine speeds somewhat, although it will do nothing for throttle responsiveness.

The exhaust manifold should be mounted with the vertical centerline of its center runner aligned with that of the center exhaust port and the upper edge of the runner profiles tangential to those of the exhaust ports. This alignment can be established by simply using a straight edge to scribe a pair of vertical lines onto the face of the exhaust gasket area through the center of the center exhaust port and a corresponding line on the horizontal surface of the center mounting flange of the exhaust manifolds, then smearing a thin coat of machinist's bluing onto the mating surface of the exhaust manifold and pressing it against the cylinder head until the proper position is located. At that point, a corresponding pair of lines should be scribed onto the horizontal surface of the center mounting flange of the exhaust manifold. This low mounting can be accomplished by carefully filing the upper edges of the mounting slots of the exhaust manifold. The use of 1/8" (.125") dowel pins to guarantee proper alignment during assembly is recommended. These dowel pins should be installed into holes sunk to a depth of no more that ¼" (.250") into both the cylinder head and the exhaust manifold flanges and located on the centerline of the adjacent outer manifold studs. A composite Big Bore manifold gasket can then be similarly modified for sealing purposes.

When using a gasket with a metalized face to install the exhaust manifold, it is wise to install the metalized side of the exhaust manifold gasket facing toward the exhaust manifold so that the mating surface of the exhaust manifold can expand and contract along the metalized face of the gasket. However, it is best to use the composite gasket available from Advanced Performance Technology (APT Part # CMG-02) as it has excellent compressibility and oversize holes for modified ports. Its graphite-impregnated material allows for superior ease of expansion and contraction of both the cylinder head and the exhaust manifold, as well as also making for very easy removal. Be advised that due to this gasket's requirement for greater torque values, it is wise to install a kit of higher-strength 5/16”-24 UNF 316 stainless steel alloy ARP manifold studs, machine washers, and nuts. It should be noted that the studs that secure the intake and exhaust manifolds are particularly prone to rusting in place. This problem can also be solved by replacing them with ARP’s 316 stainless steel alloy studs. This particular high grade alloy has very good corrosion resistance as well as good creep resistance, although it is one of the more difficult metals to machine. Whichever gasket you choose to employ, be sure to “port match” the gasket to the ports. Put the gasket onto the studs and trim the holes so that they match the ports in the cylinder head.

Exhaust manifold design is dependent upon the laws of physics. The exhaust valve begins to open during the last part of the power stroke when the gas pressure within the cylinder is still high, causing a rapid escape of the exhaust gases. As the valve continues to open, a pressure wave is generated within the flow of exhaust gases. Expanding exhaust gases rush through the port and down the runners of the exhaust manifold. At the end of the runners of the exhaust manifold, the gases and pressure waves converge at the collector point. Inside of the collector point, the gases expand quickly as the pressure wave propagates into all of the available spaces, including the other runners of the exhaust manifold. The exhaust gases and some of the pressure wave energy flow into the collector outlet and out into the exhaust system.

Based upon this, two basic phenomena are at work in the exhaust system: exhaust gas movement and pressure wave activity. The pressure differential between the gases inside of the cylinder and the outside atmosphere will determine the velocity of the exhaust gases. While the exhaust gases can flow at an average velocity of over 350 ft/sec, the velocity of the pressure wave is dependent upon exhaust gas temperature and travels at the speed of sound. As the exhaust gases travel down the exhaust pipe and expand, their velocity decreases. The pressure waves, on the other hand, base their velocity on the speed of sound. While the velocity of the pressure wave also decreases due to gas cooling as the gases travel down the exhaust pipe, the velocity will increase again as the pressure wave rebounds back up the exhaust pipe towards the cylinder. At all times, the velocity of the pressure wave is much greater than that of the exhaust gases. Pressure waves behave much differently than the exhaust gases do whenever a junction is encountered inside of the exhaust system. When two or more exhaust runners come together, as is the case in an exhaust manifold, the pressure waves travel into all of the available runners, both backwards as well as forwards. Pressure waves rebound back up the original runner, but with a negative pressure. The strength of the rebound of the pressure wave is based on the area change compared to the area of the originating exhaust pipe.

It is this rebounding, negative pulse energy that is the basis of pressure wave action tuning. The essential design ideal is to time the arrival of the rebounding negative pressure wave pulse to coincide with the period of valve overlap. As the intake valve is opening, this high pressure differential causes a fresh intake charge to push into the cylinder and also helps to remove the residual exhaust gases before the exhaust valve closes. Typically, the length of the runners of the exhaust manifold controls this phenomenon. Due to the "critical timing" aspect of this tuning technique, there may be parts of the power curve where more harm than good is done if the design is poor.

Gas velocity can be a double-edged sword as well, as too much gas velocity indicates that that the system may be too restrictive, inhibiting power output at high engine speeds, while too little gas velocity tends to make the power curve excessively "peaky", decreasing torque output at low engine speeds. This is the case in which large-diameter tubes permit the gases to expand. This expansion cools the gases, decreasing the velocity of both the exhaust gases and that of the pressure waves.

Exhaust Systems

As good as the Original Equipment exhaust manifolds are, the rest of the exhaust system can be improved upon. An Original Equipment set of mufflers (silencers) produces too much backpressure for an enhanced-performance engine to fulfill its potential. This is because backpressure does not increase in direct proportion to gas flow out of the exhaust ports. Instead, backpressure increases in proportion to the square of this gas flow. As backpressure increases, scavenging of exhaust gases from the cylinders decreases, then stops altogether, thus increasing the demand for the engine to expend power to pump the exhaust gases out through the exhaust system. These "Pumping Losses", as they are termed, thus rise dramatically as airflow (and power) increases. In addition, because of the fact that as backpressure increases, pressure within the combustion chamber increases, some of these pressurized combustion gases will escape out of the open intake valve, displacing the incoming fuel / air charge. These combustion gases then have to reenter the combustion chamber prior to the fuel / air charge, partially filling the volume. In consequence, a smaller volume of fuel / air charge enters the cylinder with the result that power output suffers. These factors can rob the engine of as much as six to eight horsepower. For practical purposes, this is almost equivalent to the difference between the output potential of a shorter-duration camshaft lobe design such as that employed in the Piper BP270 camshaft and that of a scavenging-dependent longer-duration camshaft lobe design such as that employed in the Piper BP285 camshaft. It is therefore imperative that the airflow capability of the exhaust system be improved upon in order for the potential of any serious power-enhancing modifications to be fulfilled.

On a very mildly tuned engine, simply removing the front muffler (silencer) and replacing it with a length of tubing often suffices. It will then be noted that the exhaust note becomes deeper. This is because the function of the middle muffler (silencer) is to dampen bass note frequencies, while that of the rear muffler (silencer) is to dampen the higher frequencies that give the exhaust a rasping tenor. However, when greater power outputs are being sought, a more efficient exhaust system becomes necessary. For this, the Peco exhaust system is a high-quality choice. There are plenty of aftermarket exhaust systems on the market, but those made by Peco seem to be the ones that live up to its manufacturer's promises and hence have become increasingly popular. Peco produces the only tubular exhaust manifold whose design takes into account the critical fact that the cylinder head uses a siamesed port for the exhaust valves of the middle two cylinders and hence has an oversize center branch pipe in order to accommodate the double rate of gas flow within the same time period. Their quality control is very tight for that of an aftermarket manufacturer, so their system always seems to fit without a lot of bending, hammering, and cursing. It also performs well on both modified and Original Equipment specification engines, which usually cannot be said for many of the others: they either work well only on Original Equipment specification engines, or work well only on engines that have been modified according to a specific recipe that will consist of other components made by the same company (which, of course, the advertisers never get around to pointing out before you spend your money on the #@!! thing!).

The Peco gray system will fit the Original Equipment 1¾" diameter pre-1975 exhaust manifolds without any modification while the red 2" Big Bore system will require the use of the Peco 2" diameter Big Bore tubular exhaust manifold. Having approximately a 30% greater cross section than a 1¾" diameter system, the Big Bore system is actually intended for use on larger bore engines (1868cc or larger) or smaller-bore engines fitted with flowed cylinder heads and hot camshafts such as the Piper BP285. It seems to be particularly beneficial when used on engines that are tuned to produce 125 HP or more. When fitted to smaller bore engines that have been equipped with Original Equipment camshafts, it will result in a bit more power at high engine speeds with the expense of some tractability at low engine speeds. This is due to the approximately a 30% greater cross section of the exhaust system reducing exhaust gas velocity, which in turn reduces scavenging effect in the combustion chambers at low engine speeds and thus increases "Pumping Losses". In addition, radiant heat from the tubular steel of the exhaust manifold is much greater, exposing the air in the engine compartment, the intake manifold, and the carburetors, as well as the fuel system, to more heat, thus reducing the density of the fuel / air charge and hence reducing power output. Jet-Hot coating of such exhaust manifolds is therefore highly recommended.

The Peco system is street legal. Often a performance exhaust will sound good at idle speed and while accelerating, but then turns into a howling monster while cruising on the highway and literally drives you out of the car, ears ringing. This might be acceptable in a racecar, but not in a street machine. At highway speeds, the Peco system is actually quieter than an Original Equipment system, emitting a rich baritone sound rather than the ear-pounding basso profundo or the rasping tenor of some other systems, proving that a good performance exhaust system need not be noisy enough to break down the structure of your internal organs.

The lack of a middle muffler (silencer) simplifies the Peco Big Bore system, allowing more ground clearance (something every MGB can use) and allows the fitting of a slip-and-clamp American-made performance catalytic converter where the middle silencer used to be, thus satisfying the emission laws of many localities.

However, if local regulations require it, or if you simply choose to rebuild the engine to Original Equipment specification, then an excellent quality replacement twin muffler (silencer) system is made by Falcon and can be had from Brit Tek (Part # FES001, MGB 1962-1974; Part # FES002, MGB 1975; Part # FES003, MGB 1976-1980). Brit Tek has a website that can be found at http://www.brittek.com/

Interestingly, the most common cause of backfiring in an MGB is the simplest to diagnose and fix: a leak in the exhaust system. As the pressure wave of a pulse of exhaust gases passes through the exhaust system, it leaves a partial vacuum behind it, sucking in cooler fresh air through the leaky joint in the exhaust system. Unburnt fuel then condenses in the exhaust system due to the induction of the cooler air and mixes with it, creating a condition rife with the potential for combustion. When a pulse of hot exhaust gases hits it- Bang! This problem can be aggravated by a too-lean or too-rich fuel / air mixture that will result in the production of increased amounts in unburned fuel. To find out if this is the origin of your particular problem, mix up a thick solution of water and liquid dishwashing detergent. Not the kind you put in the dishwasher, the other kind that your wife uses when she washes stuff in the sink. You know, the thick liquid stuff that she uses to cut grease with. With the system cold, squirt it on the joints of the exhaust system (do not forget the joint at the bottom of the exhaust manifold), then fire up the engine and look for bubbles. If you see bubbles, then you have found the leak. If tightening up the clamps does not cure the problem, your friendly local auto parts store or muffler shop can supply you with some exhaust system putty to take up the gaps in the connections that result from poorly matched exhaust system tubing diameters. A set of SuperTrapp T-bolt style exhaust clamps will give perfect 360º sealing in order to eliminate the leakage completely. They also have the advantage of being manufactured from durable stainless steel so that they will not corrode and are complete with Nyloc nuts. These are available from Summit Racing in both 1¾" (Part # SUP-094-1750) and 2" (Part # SUP-094-2000) through their website at http://store.summitracing.com/ . It should also be noted that the juncture of the exhaust manifold and header pipe is the most common location of such a leak. Because the interior of the bottom of the exhaust manifold has a conical sealing surface, the exhaust manifold gasket should be installed with the conically-shaped end facing upwards into the conical recess of the exhaust manifold. The gasket is squeezed between the exhaust manifold and the exhaust pipe flange in order to achieve its seal. Note that the headpipe has a flange welded on it to allow the triangular flange to compress the gasket (doughnut) when the machine bolts are tightened. If you use an old headpipe when you do a rebuild, then you will be running the risk that the welded-on flanges have significant rust damage and that they will eventually fail to the point that they will pull down through the triangular bolt-up flanges. If you have an indication of an exhaust leak that you cannot find, some backfiring while decelerating down a hill, and maybe even a loud squeaking noise on rough roads, then you might want to check and see if your headpipe is starting to migrate downward toward the pavement.

Air Cleaners and Intake Manifolds

Just as a less restrictive exhaust system is necessary to permit a high performance engine to breathe adequately, restrictions in the intake tract will likewise need to be reduced. For a Chrome Bumper car, this is not a problem. A pair of 5 7/8" Diameter x 3 1/4" deep K&N air filters (K&N Part # E-3190) will permit increased flow of the fuel / air charge without sacrificing protection. With proper fuel jet adjustment and a pair of SU AAA fuel-metering needles, when installed on an Original Equipment specification engine these larger air filters are worth about and additional three BHP on their own. When attempting to build a deeper-breathing engine, they are a prerequisite. A K&N airfilter is made of 4 to 6 layers of cotton gauze sandwiched between two epoxy-coated aluminum wire screens. The cotton is treated with a specially formulated grade of oil causing tackiness throughout the cotton's microscopic strands. The nature of the cotton allows high volumes of airflow, and when combined with the tackiness of the oil, creates an efficient filtering media that ensures engine protection. In contrast, most other filtering media cannot maintain the same balance of airflow and filtration throughout the air filter's life without sacrificing either one or the other. K&N air filters have the additional advantage of being almost infinitely reusable so long as reasonable care is taken when cleaning them. K&N makes a cleaning kit that does an excellent job for this specific purpose. Once cleaned, they should be allowed to air dry. Because the filtering medium is made of cotton, you should not use a hair drier to speed up the drying process. Should you make this mistake, the filtering medium will shrink.

In the smaller Original Equipment size, while some air filtering elements that are made with paper filtering media have an airflow capacity as little as 3.2 Cubic Feet per Minute, these reusable cotton element air filters have an airflow capacity of 6.5 Cubic Feet per Minute. However, switching from the Original Equipment airfilter elements to the same size K&N airfilter elements (K&N Part # E-2400) will have no effect on performance if you retain the use of any of the variations of the Original Equipment airfilter housings. The purpose of their intake tubes on the airfilter housings is first to accelerate the airflow beyond supersonic levels so that intake noise is literally sucked into the engine. This occurs because the air goes in faster than the sound goes out (sound only travels in air). This is a big issue since the advent of noise regulations, and it is illegal under US Federal law to alter or modify the air intakes on modern vehicles for this reason. In the case of the airfilter housing with offset air intake tubes for the 1½" SU HIF4 Series carburetors, their secondary purpose is to cause large particles of contaminants to be centrifuged out of the airflow and fall to the bottom of the airfilter housings. That leaves the airfilter element clean to filter out finer particles, plus the airflow slows with a small pressure drop, thus allowing the airfilter element to flow well and clean the air adequately. While successive versions of this basic airfilter housing became progressively quieter, they also became progressively more restrictive. The initial version (Front, BMC Part # AHH 7354; Rear, BMC Part # BHH 154) were originally intended for use with 1½" SU HS4 Series carburetors and are characterized by their large, straight intake tubes that direct the incoming air directly at the airfilter, while those of the later version that was intended for the UK/European market with the 1½" SU HIF4 Series carburetors were characterized by offset intake tubes being curved downwards on their ends (Front, BMC Part # 548; Rear, BMC Part # 549), while the North American Market counterpart for these were similar (Front, BMC Part # 665; Rear, BMC Part # 666), but more restrictive. In all cases, including Non-Original Equipment items, the base plates of the airfilter housings have holes that align with those of the gaskets that fit between the airfilter housings and the carburetors. Make sure that these holes are properly aligned, otherwise the vacuum pistons inside of the vacuum chambers (dashpots) of the carburetors will not assume their appropriate positions relative to the jets in order to cause the fuel-metering needles to meter fuel appropriately.

Possibly the worst air filters are the Stellings & Hellings items. These are the chrome items that have perforated chrome sleeves on both sides of the foam element. Its available airflow area is easily computed: 84 holes @ .190" dia. = 84 x $.095^2$ sq in. x 3.1416 = 2.38 sq. in. per airfilter. That is the maximum at the inner sleeve, reduced by alignment of the similar outer sleeve and foam. The airflow area of the Original Equipment paper airfilter, gross: 3 ¾" diameter X 3" deep x .8 estimated screen coefficient = 33 sq in. per airfilter. The actual element is Unfolded size = .550 pleat width x 2 sides per pleat x 3" depth x 76 pleats = 250.8 sq in. It is notable that a single pleat has nearly 50% more air filtering area = 3.3 sq in. than that of the entire Stellings & Hellings airfilter. The Stellings & Hellings airfilter has about 7% of the gross area and 1% of the element area that is present in the Original Equipment unit. This means that the Stellings & Hellings is not only very restrictive, but also that air velocity through the air filtering media is so high as to make any useful filtration impossible. The Stellings & Hellings unit also has a bad backplate design that further restricts airflow, and is so thin and thus flimsy as to not retain the gasket, allowing it to shrink into the intake, making it even worse. They may be flashy, but they are a true waste of money.

The B Series engine with its Siamesed intake port design causes some very powerful sonic shockwaves within the fuel induction system. The volume and depth of the large K&N airfilter dissipates these very effectively by creating a plenum over the opening. Sonic shockwaves simply dissipate within the confines of the plenum. Air entering the airfilter from the outside is slowed, smoothed and straightened. The large airfilter then becomes an endless source of calm, clean air. Both the 5 7/8” Base Diameter x 3 ½” Top Diameter x 2 11/16” Deep cone and the 5 7/8” Diameter x 1¾” deep “pancake”-type air filters reflect these sonic shockwaves back into the fuel induction system, causing induction pulse problems that will increasingly disrupt airflow above 3,500 RPM and cause the pistons of the carburetors to move independently of airflow, interfering with their consistency in their metering of fuel. In order for the carburetors to help the engine develop its maximum power output, the fuel / air ratio may vary no more than 6% from the ideal. For this same reason, if you should elect to install velocity stacks inside of an airfilter, then never use one that has a mouth that is closer to the inside of the outer cover than the size of the bore of the carburetor (1½” for the SU HS4 Series and the SU HIF4 Series, 1¾” for the SU HS6 Series and the SU HIF6 Series).

Velocity stacks that have walls with an exponential curve can dampen these shockwaves to a limited degree, but should not be considered to be a substitute for an airfilter assembly of adequate clearance and volume.

When installing these deeper air filters, one thing that I might suggest would be the fitting between the carburetors and these custom air filters of a pair of 1½" deep velocity stacks with a 7° taper and a .250" radius (APT Part # RP-4). These are proven to boost the flow of the fuel / air charge by a worthwhile 5.2% by means of drastically reducing the contraction of the airflow at the mouth of the carburetor. As a side benefit, this reduction in contraction will help to accelerate the velocity of the fuel / air charge, maintaining fuel suspension in the airflow and enhancing volumetric efficiency at high engine speeds. However, the carburetors will become more sensitive to the state of their synchronization. In order to minimize disruption of the flow of the fuel / air charge, they should be mounted with the bolt heads on the inside of the airfilter, while the mounting studs for the airfilter covers should be located as far as possible from the mouth of the carburetor. K&N makes a dandy set of chromed cover plates for the SU carburetors already set up with this worthwhile feature, along with the two necessary vent holes in order to enable airflow to the vacuum chamber (dashpot). These are available for both the 1½" SU HS4 Series and the 1½" SU HIF4 Series (K&N Part # 56-1390), as well as the 1¾" SU HS6 (K&N Part # 56-1400). You will need to purchase longer mounting through-bolts for them, however.

If you feel that you do not desire to include this modification, then at the very least install a pair of Advanced Performance Technology's stub stacks (APT Part # SS51). This additional refinement will not create a perceptible increase in power (about a 2-3 BHP increase), but they will make both the throttle response and the engine running characteristics slightly smoother by means of reducing both turbulence at the mouth of the intake tract as well as contraction of the airflow, thus providing more efficient fuel atomization at the fuel jet bridge and allowing the greater airflow potential of the larger air filters to be fully exploited. They might even eventually pay for themselves by thus slightly improving fuel economy (maybe). The Original Equipment airfilter boxes incorporate stubstacks into the airfilter housing design, so it is obvious that the engineers at the factory saw the value in this design feature.

When coupled with a Maniflow fabricated steel intake manifold (British Automotive Part # SUB4-2; Maniflow Part # L137), the improvements become even more impressive. It has high airflow potential coupled with good port velocity, enabling it to take advantage of the inertial effects of the fuel / air charge in order to better fill the cylinders at high engine speeds with the added benefit of maintaining excellent fuel suspension within the incoming fuel / air charge. In addition, because of the slower heat transference of steel as compared to aluminum alloy, it conducts less heat from the cylinder head into the incoming fuel / air charge, thus making for greater fuel / air charge density and hence more power potential. In the past, polishing and chrome plating the exterior of an intake manifold was often done in order to enable it to reflect away radiant heat emitting from the exhaust manifold, keeping it cooler. A more modern approach is to Jet-Hot coat them, as this does an even better job of protecting it from the radiant heat of the exhaust manifold.

If you elect to use the basic Maniflow intake manifold with 1½” SU HIF4 Series carburetors, then you will need to use either the early version of the UK/European Market 1½” SU HIF4 Series carburetors with the vacuum takeoff fitting on the carburetor body for provision for a ported-advance mechanism. However, if you use the North American Market 1½” SU HIF4 Series that lacks provision for a vacuum takeoff, Advanced Performance Technology also offers the option of welding in a nipple on the crossover balance tube that will allow the use of an intake manifold-advance distributor. Incidentally, do not let the small diameter of the crossover balance tube between the runners worry you. Many cars with dual SU carburetors have equally small crossover balance tubes. The large diameter of the crossover balance tube of the SU design was a carryover that had been necessary in order to assist in dissipating returning sonic shockwaves back in the days when air filter housings were of the thin, “pancake”-type design. With the deeper airfilter housings of modern designs, the shockwaves can dissipate external to the carburetors, so the smaller-diameter crossover balance tube of the Maniflow intake manifold produces less flow disruption, and that results in superior performance. If you wish to run an anti-run-on valve and have an earlier car that has neither the adsorption canister (BMC Part # 13H 5994) of the 18GK and later engined cars nor the later intake manifold that has a vacuum takeoff fitting for the application, you will then need to use the thinner Advanced Performance Technology’s phenolic carburetor spacers (APT Part # MFA338) that come suitably modified to provide fittings for a vacuum line, as well as the later exhaust manifold (Casting # 3911) as both have a mounting flange thickness of 7/16” (11.1125mm). This phenolic spacer with a vacuum takeoff incorporated into its design is a Maniflow item intended to be used with the Maniflow intake manifold that, unlike the Original Equipment SU intake manifold, has no provision for vacuum takeoff on its crossover balance tube. A companion unported phenolic spacer of the same thickness is also available from Advanced Performance Technology, although a second spacer with a vacuum takeoff may be substituted to allow the use of a vacuum-assisted servo for a power brake system. Because of both the angle of this intake manifold is angled 20° higher than the Original Equipment SU intake manifold in order to enhance its airflow characteristics, and variances in production tolerances of the bodyshell of the car, in a few cases longer and larger-diameter air filters will not allow the installation of an underhood insulation pad, hence the thinner design of the Advanced Performance Technology's spacers.

Why not stay with the Original Equipment SU intake manifold? Due to the sudden change of cross section that occurs in the area of the crossover balance tube intersection, the flow of the fuel / air charge is markedly disrupted into a vortex effect. The resulting turbulence causes the fuel / air mixture to condense somewhat, and also impedes airflow by causing the mass of the fuel / air charge that has been vectored into the upper section of the intake manifold to swirl 180º towards its bottom. Not only does this vortex effect cause the incoming fuel / air charge to lose some of both its velocity and inertia, when the incoming fuel / air charge reaches the turn into the throat of the port, its remaining inertia then causes it to careen into the opposite wall of the throat of the port instead of flowing along its contours as it should, thus impeding its own flow past the intake valve into the cylinder.

In addition, inside of the early examples of the Original Equipment SU intake manifold, when viewed from the carburetor side, a distinct edge can be seen at the top of the exit side. This abrupt ridge is formed at the intersection of the crossover balance tube and presents an obstruction to the flow of the fuel / air mixture. However, when later examples of the SU intake manifold are viewed from the carburetor side, a more radiused profile can be seen in the same area of the intersection. This more radiused profile is better for airflow than that of the sharp edge of the earlier SU intake manifolds. If an Original Equipment SU intake manifold is to be employed without any modifications having been performed upon it, then this later type is the preferred type. The first procedure for improving these intake manifolds is to smooth the passage as though the goal was to transform it into a round tube. The second procedure for improving these intake manifolds is to flatten off the top of the intake manifold where the crossover balance tube adjoins the port. The purpose of this blending is to distribute the inherent design inefficiency over a larger area, allowing more flow through the intake manifold with less overall disruption. With these procedures having been accomplished, the intersection of the crossover balance tube should be carefully given a .250" (6.35 mm) radius along its leading and trailing edges. With the exit side having been roughed out, proceed to the intake side in order to perform a similar sequence of procedures. Because its runner section is shorter and needs to be a good transition to the open area, the intake side need not be worked as heavily as the exit side. Be careful and work slowly, bearing in mind that excessive removal of material in this area will cause the flow to break away even earlier, resulting in the creation of eddies that will produce increased fuel condensation as well as an increase in the vortex effect which will reduce airflow capacity. Afterwards, polish the interior surfaces with # 60-grit sandpaper. While smoothing the cast surface of the inside of the intake manifold and blending the change of cross section, as well as making a .250" (6.35mm) radius at both the leading and trailing edges of both ends of the crossover balance tube can reduce the vortex effect, such compromise efforts cannot take the place of the better design of the Maniflow intake manifold.

It might be noted that the smaller-diameter crossover balance tube of the Maniflow design is not markedly different from others that are also used on more modern SU-equipped engines. The flow-disrupting large diameter of the crossover balance tube of the SU intake manifold design dates way back to the days when engines used thin pancake-style air filters. Unfortunately, such shallow filters reflect pulsations of sonic shockwaves back into the fuel induction system. However, with the thin, pancake-style air filters of yesteryear, some way had to be found to deal with these reflected sonic shockwaves. The large diameter of the crossover balance tube was the way in which the designers at SU dealt with the problem. As long as engine speeds are kept below 4,200 RPM, the reflected shockwaves will tend to dissipate inside of the large diameter of the crossover balance tube. The large-diameter crossover balance tube of the SU design has a reducer cast into its middle, and this effectively reduces the Internal Diameter of the crossover balance tube down to around 3/8" (.375" / 9.525mm). The design of the Maniflow intake manifold incorporates a more flow-efficient junction at a smaller-diameter crossover balance tube, but the smaller diameter does not provide sufficient volume to dissipate these pulsating sonic shockwaves, and consequently it requires the use of deep air filters in order to accomplish this. The volume and depth of today's larger modern air filters allows these pulsating sonic shockwaves to dissipate very effectively by creating a plenum over the opening. Sonic shockwaves simply dissipate within the confines of the plenum. Air entering the airfilter from the outside is then slowed, smoothed and straightened. The larger airfilter then becomes an endless source of calm, clean air.

Whichever intake manifold you elect to use, its runners should be smoothed with a #80 grit and then a #100 grit flap sander in order to maintain border turbulence. Afterwards its runners should be matched to the carburetors so that the mouths of its runners are flush and concentric with the throats of the carburetors, and then it should be likewise matched to the intake ports. It cannot be overstressed that the mouths of the runners of the intake manifold should be both concentric to and equal in diameter to the carburetors. Under no circumstances should they be larger as this will result in the formation of eddies at this critical juncture which will disrupt the flow of the incoming fuel / air charge and cause fuel condensation. The use of 1/8" (.125" / 3.175mm) dowel pins in order to guarantee this concentricity with the intake ports during assembly is recommended. These dowel pins should be installed into holes sunk to a depth of no more than 1/4" (.250" / 6.35mm) and located on the centerline of the adjacent manifold studs. The mouths of the intake ports, on the other hand, should be concentric with and .020" (.508mm) larger in diameter than those of the runners of the intake manifold so that the step thus created will invoke just enough border turbulence to maintain the suspension quality of the atomized fuel / air mixture.

Some gasket manufacturers make an embossed intake manifold gasket that uses a perforated core. Unfortunately, the embossings on a perforated core are often unable to stand up to the pressure when they are torqued into place. As a result, their embossings tend to collapse, losing their clamping force, and leakage ensues. As a response to this problem, the engineers at Fel-Pro have designed an intake manifold gasket using an embossed, solid core design coated with a plastic resin material as the carrier for a molded silicone sealing bead in order to permit far superior sealing and clamping force. While the molded rubber bead does the actual sealing, the resin carrier keeps the gasket in place underneath the embossing and assures uniform pressure. When properly installed, this type of gasket assures an even load around the ports, which avoids vacuum leakage and, best of all, its design stands up over time. It is available from Fel-Pro as part of a complete six-piece kit that includes high-quality phenolic spacers (Fel-Pro Part # 23555) for insulating the carburetors from heat that has been conducted through the intake manifold.

The Original Equipment dual carburetors are insulated by means of a pair of thick phenolic spacers (BMC Part # 12H 712) from heat conducted through the intake manifold from the cylinder head. Unfortunately, these are only partially effective at their task of keeping heat out of the fuel induction system. Because the Original Equipment SU intake manifold is made of aluminum alloy and is somewhat insulated from the conductive heat of the cylinder head by only a thin gasket, heat is rapidly transferred through its runners into the incoming fuel / air charge, reducing its density and thus decreasing performance. This condition is generally aggravated by modern fuel formulas containing highly volatile solutions with high vapor pressure (i.e., evaporate easily with a little heat). However, this rapid transfer of the heat, along with the insulation added by the phenolic spacers, does effectively prevent it from reaching the carburetors, both of which are also partially protected from the heat of the exhaust manifold by their heat shield. To eliminate this hindrance to performance, complete Jet-Hot coating of the intake manifold is highly recommended. If this is not possible, then the fabrication of U-shaped heat shields for the purpose of protecting the runners of the intake manifold from heat radiated by the exhaust manifold is advised. At a minimum, the intake manifold should be polished on its external surfaces in order to both reflect heat away and to reduce its radiant heat absorbing surface area.

When seeking improvements in airflow capacity, things become considerably more complicated when trying to fit higher capacity air filters onto a Rubber Bumper car that has been modified to make use of dual carburetors. Unfortunately, the manner in which the servo-boosted brake master cylinder projects from the firewall (bulkhead) forces most conventional owners into the use of conical air filters when installing dual carburetors. The problem with the conical air filters is that their shallowness creates induction pulse problems above 3,500 RPM, their small internal volume that will not allow the fitting of a set of velocity stacks, and their small surface area that offers insufficient airflow for an enhanced-performance engine. The K&N air filters all use the same airfiltering medium, so the smaller the surface area of the airfilter, the less the maximum airflow potential will be. Conversely, the bigger the surface area, the greater the maximum airflow potential. This is why those who go for serious power increases with a B Series engine prefer the 6" X 3 1/4" deep K&N air filters. Induction pulse problems aside, the airflow capacity of the little conical or pancake air filters is more appropriate to a mildly power-enhanced 1275cc A Series engine, such as is fitted to the MG Midget or the Austin Healey Sprite. In addition to this problem, the remote float bowls of the 1½" SU HS4 Series carburetor will usually interfere with the servo-boosted brake master cylinder, thus such a conversion requires the use of a set of SU HIF-type carburetors.

Retrofitting the earlier non-boosted brake master cylinder is the common solution, but this is not a bolt-on affair as its mounting flange is turned 90º, so the mounting holes of the pedal box will not line up, and the appropriate earlier pedal box assembly is radically different, even having a different mounting hole pattern at its base that requires drilling a new pattern of holes into the body of the car. This is just one of the reasons that it is unusual to see a Rubber Bumper model with an uprated B Series engine: It is much more work. When somebody wants to go for really dramatic power increases, he swiftly comes to think that he will need to retrofit the earlier brake master cylinder and pedal box assembly so that he can mount air filters that have a decent airflow capacity onto the carburetors like the Chrome Bumper model owners do. "After all," he reasons, "it is not really that difficult, it just requires some persistence and time, plus another brake master cylinder and the earlier pedal box assembly. If my boosting servo and brake master cylinder are in good shape, then I can always sell them as a unit to help cover the cost of the earlier brake master cylinder and pedal box assembly because the servo unit is getting harder and harder to find." Moreover, to the conventional, orthodox thinker, this reasoning holds true. However, read on-

Fabrication of a plenum chamber to go on the carburetors and running a large-diameter breather duct hose (flexible pipe) to a remote airfilter housing would enable the retention of the existing boosted brake system. From the airfilter housing the intake hose (flexible pipe) can be run to the air passages neatly provided beneath the bumper in the vented front valance of the 1972 through 1974½ models of the Chrome Bumper cars. You will need to do some scavenging in the junkyards to find an appropriate airfilter housing (more work) and then figure out a mounting system for it (still more work), but the larger, more commodious engine compartment of the later Rubber Bumper models should make it a relatively easy task. To equal the airflow capacity of a pair of 6" diameter 3 1/4" deep round air filters you will need an airfilter housing with an airfilter that has an area of about 122 square inches (11" X 11").

The Fuel System

Now for the subject of the fuel system: The minimum requirement for an Original Equipment specification engine is 8.4 US gallons (7 Imperial gallons) of fuel per hour. With a pumping capacity of 12 US gallons (10 Imperial gallons) per hour, a good-condition Original Equipment SU fuel pump is adequate for feeding the requirements of any streetable B Series Engine. However, when the fuel pump is badly worn, it may deliver as little as 7.2 US gallons (6 Imperial gallons) of fuel per hour.

The SU fuel pump is an impulse type of fuel pump. When power is supplied to the fuel pump, current flows through both the solenoid coil and the electrical contact breaker points. The electromagnetic force of the energized solenoid coil acts on the iron disk that is attached to the diaphragm, pulling both it and the its attached diaphragm toward the solenoid coil. This movement of the diaphragm develops a vacuum in the pump body, which pulls fuel in from the tank through an anti-backflow check valve and into the body of the fuel pump. The movement of the diaphragm also causes a shaft that is attached between it and the lower contact breaker points bridge or carrier to push the carrier upwards, causing the carrier to open the contact breaker points. Once the contact breaker points have opened, the flow of current through the solenoid coil is interrupted, allowing the diaphragm to be pushed back to its original position by the volute spring, which in turn pushes the fuel in the pump body out through another anti-backflow check valve and onward thru the fuel line to the carburetors. Once the diaphragm returns to its original position, the carrier of the contact breaker points “throws over” to the “points closed” position and the whole action is then repeated – thus the characteristic “tic, tic, tic” sound of the fuel pump. The fuel pump pressure is established by the strength of the volute spring that resides between the iron disk that is located on top of the diaphragm and the bottom of the solenoid coil. The anti-backflow check valves are a plastic sheet in a valve assembly that closes against the assembly’s valve seat and is held against its seat by system pressure. The system pressure is developed on the carburetor (outlet) side of the fuel pump, thus the valves act as anti-backflow check valves in order to keep fuel from flowing back to the tank. It should be noted that the SU fuel pump should be mounted with its outlet oriented towards the top because is designed to be self-bleeding when it is in that orientation. An SU fuel pump that has been mounted upside down (even the L type pumps with the gravity controlled valve disks) will work in an upside down position, but the efficiency will not be as high as possible, and one can wind up with an air bubble in the fuel pump that is hard to clear.

When installing a new or a freshly rebuilt fuel pump, it is prudent to replace the rubber fuel hose (flexible pipe) that connects it to the fuel tank. Should the fuel hose be found to be deteriorated and cracked, the fuel pump may be partially pumping air into the system and not properly pressurizing. This would be symptomized by very rapid clicking of the fuel pump. It should not click more than once every 20 seconds with the ignition on but the engine stopped. However, kinking the intake fuel hose between the fuel tank and the fuel pump acts like another valve (albeit a no-way valve in such a case) in the fuel line, so the problem could be with the intake valve inside of the fuel pump sticking in the open position. This condition is easy to diagnose without disassembling the fuel pump. Simply disconnect the intake fuel hose (flexible pipe) from the fuel tank and place it in a small transparent container of fuel. If you see the level going down and up repeatedly while the fuel pump is clicking like mad, then the intake valve is stuck in the open position.

One of the more interesting and somewhat quaint features of the MGB’s fuel pump is the use banjo unions. Sadly, these usually cannot be refurbished and must be replaced whenever they start to leak. The intake/outlet port threads are 3/8 BSP parallel thread. A tapered pipe thread should not be used as over-tightening it could split the alloy casting.

Be aware that the new AZX 1300 Series SU fuel pumps now use an O-ring (Burlen Fuel Systems Part # AUB 654; Moss Motors Part # 370-655) between the ports of the fuel pump and the banjo fitting. If your fuel pump has a recessed area on the intake and outlet ports, you will need to use an O-ring instead of the fiber washer that was used on the earlier fuel pumps. The order of assembly for the banjo fitting on these new pumps is as follows: Fit the O-ring into the recessed area of the intake and outlet ports of the fuel pump. The banjo fitting goes on top of the O-ring with its flat surface against the O-ring, a standard fiber washer goes into the recessed area on the top of the banjo fitting, and finally the banjo bolt is inserted through the fitting into the port, and is then tightened until the fitting is good and snug against the port. Be warned that if you install the fiber washer on the inside of the banjo connection, you will have a suction leak. If the fuel pump chatters away at a relatively high rate, caused by the points carrier repeatedly throwing over, it is an indication that it is drawing air into the intake side of the fuel pump. Here is an `Ol-Timey-Mechanic's fix for the problem of a leaking banjo union at the fuel pump: Take a new set of fiber washers and wrap them radially with Teflon plumber's tape (the type that is used for sealing pipe threads). Wrap the tape around, overlapping the wraps, in much the same wrapping manner as the paper that was once used to wrap new tires with. You should then have no more leakage after you install these modified washers.

The fuel pump should always be mounted with its outlet on the top. The outlet is the port that is closest to the top-hat style cover on the body of the pump. Note that it is not necessary for the holes on the inside of the banjo to be lined up with the outflow of the banjo union as there is a groove around the interior of the banjo fitting so that it will conveniently pass fuel when it is mounted in any orientation.

It should be noted that the SU fuel pump is vented. The purpose of these vents is to relieve the pressure built up behind the diaphragm each time the coil is energized, pulling the diaphragm upward. Without these vents, the air trapped behind the diaphragm would be compressed, acting as a stiff spring that that would then resist the movement of the diaphragm. If both of these vents were to become clogged, the pump will then become inconsistent in its operation. The main vent, which allows air to move both in and out, is located at the bottom of the coil housing. The vent on the end cover (if one is present), incorporates a check valve that allows air only to exit, but not to enter. It is not there for the purpose of admitting cooling air to the electronics, it is merely an additional vent that expels any pressure caused by the upward travel of the diaphragm.

A piece of 1/8" tubing should be attached to the vent line fitting at the base of the coil housing, and be terminated with a 'T' fitting (it can just as well be an 'L' fitting) inside of the boot (trunk). This tubing can be made of any material as its sole purpose is to vent air into and out of the space behind the fuel pump diaphragm where there are no fumes present.

The termination inside of the boot (trunk) is for the purpose of providing a relatively dry termination of the vent line, and to keep road debris and water out of the vent line and thus keep moisture out of the pump. In no case should the vent line be left to dangle under the car, as water splashed under the car can been drawn into the coil housing, causing severe rusting inside of the pump to the point where it will cease to function. If you do not feel comfortable with the vent being inside of the boot (trunk), it can be run to a high spot under the car (outside, top of the battery box comes to mind) and attached there. In any case, there should be no fumes coming from this vent line into the boot (trunk), as it is only venting the air from the area of the pump that is above the diaphragm. Except in the highly unlikely case of the diaphragm having developed a leak, no fumes should be present in this area.

As with any device that uses a set of electrical contact breaker points, there is wear of the contact breaker points, both mechanical (slight) and electrical arcing (major) that eventually causes its operation to deteriorate and, eventually, to cease completely. The maximum amount of (coil primary) current that can be switched by contact breaker points is about 4 amps. Above this level, points burnout may occur. Over the years, various methods were employed to suppress the electrical arcing at the contact breaker points. Originally, on the L type fuel pumps, the only suppressor used was a swamping resistor, taking the form of a resistance wire wrapped around the solenoid coil and attached in parallel with it. As stronger solenoid coils that drew more current were employed, a 0.47 microfarad capacitor was added in order to assist the swamping resistor in suppressing the arcing (by the way, even though it looks like an electrolytic capacitor, it is not, and therefore it is not polarity-sensitive). With the introduction of the AUF 300 and AZX 1300 Series fuel pumps, the capacitor was replaced with a diode to work in conjunction the swamping resistor. Be aware that his particular arrangement makes these fuel pumps polarity sensitive. The AZX 1307 negative (-) ground (earth) fuel pumps have black sealing tape, while the AZX 1318 positive (+) ground (earth) fuel pumps have red sealing tape. Fuel pumps with the prefix letters AUA, AUB or AUF are not polarity sensitive. All pumps with the prefix letters AZX require some modification if the polarity of the vehicle is changed. In non-electronic pumps, the only change that is required is to the terminals of the diode. If you have one polarity fuel pump and need the other, then just use a soldering iron to switch the terminals. On positive (+) ground (earth) cars the black wire goes to the Power Input and the red wire goes to the Contact Blade fixing screw. On negative (-) ground (earth) cars the reverse is the case, with the red wire going to the Power Input and the black wire going to the Contact Blade fixing screw.

All of the systems of arc suppression worked fairly well with the series of fuel pumps that they were designed for, giving the fuel pumps a reasonable life expectancy. Ultimately, solid-state versions of these fuel pumps were developed. In these, the contact breaker points were replaced with a Hall Effect solid circuit that makes use of a semiconductor in order to control the current flow within the solenoid coil. These fuel pumps have the same external appearance and operate in the same manner as the contact breaker points-style fuel pump, complete with the familiar tic, tic, tic sound. Because there is no longer the disadvantage of a set of contact breaker points, the life expectancy of the fuel pumps in those cars that are used regularly is established by the life span of the diaphragm and the anti-backflow check valves.

The contact breaker points-style SU fuel pump should be good for approximately 75,000 to 100,000 miles, depending upon how the car is used. A car that is driven regularly, all year around, should get 100,000 trouble-free miles. If the car is not driven regularly, or is put in hibernation over the winter, it will start experiencing problems somewhere along that time period. The alloy of the contact breaker points have a tendency to film over, and, if not driven regularly, the film will build up enough to prevent the dual contact breaker points from conducting current, whereupon the fuel pump will fail (as opposed to the car that is driven regularly so that the arcing of the contact breaker points keeps the film burned off). If your car falls into this category, then you should remove the fuel pump each spring and clean the dual contact breaker points. If you have a powerful magnifying glass, it is interesting to look at the contacts before cleaning them. Usually you will see what appears to be a waxy buildup on the dual contact breaker points that often looks like it has been pushed into a pile (not unlike snow pushed into a driveway by a snowplow) on one side of the contact. You need to remove this waxy buildup. To remove a heavy buildup of this film, it is necessary to remove the contact blade from the pedestal and then scrub its dual electrical contact breaker points with some 400 grit sand paper, then wash them off with CRC QD Electronic Cleaner. The dual lower contact breaker points on the rocker mechanism present more of a challenge to clean and is the reason why people often fail in their attempt to get the fuel pump to work properly. In order to accomplish this, is necessary to first remove the two pedestal mounting screws, and then to carefully fold the pedestal back in order to completely expose the rocker mechanism that has the two electrical contacts on it. 400 grit sandpaper also comes in handy here in order to sand the contacts. Afterwards, wash them with CRC QD Electronic Cleaner. A combination of the current flowing through the contacts and the small amount of arcing that occurs as the contact breaker points open will keep this buildup burned off when the car is driven regularly. On the other hand, if you have a solid state SU fuel pump as made by SU manufacturer Burlen

Fuel Systems, then there are no contact breaker points employed to trigger the fuel pump's solenoid, and therefore no such problem is possible.

Should you happen to have replaced the contact breaker points of your SU fuel pump, reassembly and adjustment is a fairly straightforward procedure. Note that the steel pin which secures the rocker mechanism to the pedestal is specially hardened and must not be replaced by anything other than a genuine SU part. Fit the rocker assembly to the pedestal by sliding it into place, and then push the steel pin through the small holes in the rockers and pedestal struts. Next, position the centre toggle so that when the inner rocker spindle is in tension against the rear of the contact point, the centre toggle spring is above the outer rocker spindle. This positioning is important in order to attain the correct "throw-over" action. It is also essential that the rockers are perfectly free to swing on the pivot pin and that the arms are not binding on the legs of the pedestal. If necessary, the rockers can be squared up into position with a pair of thin-nosed pliers. Do not fit the contact blade at this stage.

Place the armature spring into the solenoid coil housing with its large diameter towards the solenoid coil. Before fitting the diaphragm, make sure that the impact washer, which is a small neoprene washer that fits into the solenoid coil recess, is fitted to the armature. Do not use jointing compound or dope on the diaphragm. Fit the diaphragm by inserting the spindle into the hole in the solenoid coil, and then screwing it into the threaded trunnion in the centre of the rocker assembly. Screw in the diaphragm until the rocker will not "throw over". Note that this must not be confused with jamming the armature on the integral steps of the solenoid coil housing. On later-type rocker mechanisms with adjustable fingers, fit the contact blade and adjust the finger settings, and then carefully remove the contact blade.

While holding the solenoid coil housing assembly in the left hand in an approximately horizontal position, firmly but steadily push the diaphragm spindle in with the thumb of your right hand. Unscrew the diaphragm, pressing and releasing the diaphragm spindle with the thumb of your right hand until the rocker just "throws over". Now rotate the diaphragm back (unscrew) to the nearest hole and again a further four holes (two-thirds of a complete turn). The diaphragm is now correctly set.

By turning back the diaphragm edge and inserting an end lobe into the recess between the armature and the solenoid coil housing, fit the armature guide plate with its flat face towards the diaphragm. Follow this procedure until all four lobes are approximately in position, then firmly press each lobe home, finishing with the two end ones. This latter step is important in order to avoid distortion of the connecting arms between the lobes.

Fit the joint washer to the body, aligning the screw holes. Fit the solenoid coil housing onto the body of the fuel pump, ensuring correct seating between them. Line up the six securing screw holes, making sure that the cast lugs on the solenoid coil housing are at the bottom. Insert and tighten the six 2 BA screws finger-tight. Tighten the securing screws in correct sequence as they appear diametrically opposite each other.

Fit the contact breaker point blade and solenoid coil lead (king lead) to the pedestal with the 5 BA washer and screw. Where a diode resistor is fitted, it is in parallel with the solenoid coil connections. Be aware that this component is polarity-sensitive and that therefore all connections must be correctly made. A condenser (capacitor), where fitted, is not polarity-sensitive. Adjust the contact breaker point blade so that the contact breaker points on it are a little above the contact breaker points on the rocker when the contact breaker points are closed, also that when the contact breaker points make or break contact with each other, one pair of contact breaker points wipes over the centre line of the other in a symmetrical manner. As the contact breaker point blade is provided with a slot for the attachment screw, some degree of adjustment is possible. Tighten the breaker point contact blade attachment screw when the correct setting is obtained.

Check that when outer rocker is pressed onto the solenoid coil housing, the contact breaker point blade rests on the narrow rib or ridge that projects slightly above the main face of the pedestal. If it does not, slacken the attachment screw of the contact breaker point blade, swing the contact breaker point blade clear of the pedestal, and bend it downwards a sufficient amount so that when repositioned it rests against the rib lightly. Over-tensioning of the contact breaker point blade will restrict the travel of the rocker mechanism.

Check the lift of the tip of the contact breaker point blade above the top of the pedestal with a feeler gauge, bending the stop finger beneath the pedestal, if necessary, in order to obtain a lift of 0.035” +/- 0.005 “ (0.9mm +/- 0.13mm). Check the gap between rocker finger and solenoid coil housing with a feeler gauge, bending the stop-finger if necessary, in order to obtain a gap of 0.090” +/- 0.005” (2.3mm +/- 0.13mm). *Note: I usually adjust this lower rocker finger for a gap of 0.070” to 0.075” clearance at B.

Tuck all of the spare cable into position so that it cannot foul the rocker mechanism. Make sure that the diode resister or condenser (capacitor) is fitted snugly into the end cover at the correct attitude. Ensure that the seal washer of the end cover is in position on the terminal stud. Fit the bakelite end-cover and lock washer, and then secure it with the brass nut. Next, fit the terminal tag or connector, and then the insulated sleeve. The fuel pump is now ready for testing. After testing it, replace the rubber sealing band over the end cover gap and seal with adhesive tape. This tape may be removed in order to improve ventilation when the fuel pump is mounted internally in a moisture-free region, but must be retained otherwise.

The AUF 300 Series (now AZX 1300 Series) fuel pumps are found on all of the later MGBs plus many other British cars of the mid-1960s and later. They both have a plain air bottle on the intake side and a flow-smoothing device on the delivery side. Both types develop up to 2.7 PSI (AZX 1307) or 3.8 PSI (AZX 1308) and a flow rate of 2.4 pints per minute (18 gallons per hour). Because the absolute maximum pressure that any SU fuel pump manufactured puts out is 3.8 psi, which the SU carburetors are well capable of withstanding, a pressure regulator is not required, nor should one be used with an SU fuel pump / SU carburetor combination. Furthermore, because the output pressure of an SU fuel pump is controlled by a volute spring, it is absolutely impossible for them to develop an output pressure in excess of that stated pressure unless an owner either stretched the spring or replaced it with a homemade spring of greater strength.

Definitely fit an inline fuse that takes the Original Equipment tubular MGB fuses into the fuel pump circuit. This should be fitted as close to the power supply as possible, not close to the fuel pump as one might think, as the purpose of the fuse is not to protect the fuel pump itself, but rather to protect the wiring against short circuiting. A very convenient for the fuse position is where the rear wiring loom (harness) joins the main wiring loom (harness) in the mass of connectors by the fusebox. Put a couple of solder bullets (not the blue or red crimp type as neither are the correct size), and with an additional single bullet connector you can insert the fuse between the two harnesses. When the contact breaker points close and full current runs through the coil, it is drawing just over 4 amps (3 Ohms x 4 Amps = 12 Volts). Since the initial current draw might be a surge, use a 10 Amp fuse. You can also use a standard 17 Amp-rated, 35 Amp blow fuse. I know that the fuel pump does not need a fuse as big as this, but this would be to protect the wiring and that size is perfectly adequate for the purpose. This will spare the necessity of carrying multiple fuses around, and you should have two spare standard ones in the fusebox anyway. Be sure that the white positive (+) wire goes onto the terminal on the end cover and that the black ground (earth) wire goes onto the terminal on the flange of the solenoid coil housing. If the fuel pump works, but the engine then stops, remove the fuel cap from the fuel tank. If you hear a hissing noise, and / or if the fuel pump works without the fuel cap, then the vent inside of the fuel cap is blocked, forcing the fuel pump to pull against a partial vacuum in the fuel tank.

It should be understood that fuel pumps and carburetors are precision instruments that do not take well to the presence of dirt. As such, they should be well protected. While it is true that the SU AUF 300 type fuel pump does in fact have an integral fuel filter, it consists only of a small wire screen (BMC PART# AUB 617) that can be accessed only by disassembling the body of the fuel pump. On the other hand, the earlier SU HP type fuel pump has tubular fuel filter (BMC PART# AUA 1464) located behind a threaded plug in its pump body opposite its outlet fitting that can be readily removed for cleaning, but neither of these fuel filters can be considered to be anything more than coarse strainers that provide inadequate protection for the carburetors. Install a transparent fuel filter in the feed line just prior to the junction that feeds both of the carburetors, and then, if you want to be obsessive about it, install a second transparent fuel filter in the feed line that runs from the fuel tank to the fuel pump. If the transparent filters that you elect to use should happen to have glass housing bodies, these can be easily protected by sliding a short section of flexible transparent thick wall tubing over them. A petcock-type valve will simplify replacement in the future, preventing fuel from the carburetors from draining when the fuel line is disconnected from the filter. Should you see debris in this second fuel filter, simply replace it with the one that is before the carburetors and then install a new fuel filter in the line before the carburetors. By using this approach, you can best protect the carburetors and the fuel pump as well.

However, a word of warning is in order. The fuel tank needs to be free of internal rust. If not, instead of preventing any problems, a fuel filter on the intake side of the SU fuel pump can cause problems, the biggest being that it is an unseen problem. While the SU fuel pump will pass all but large chunks of rust without jamming, if a modern fuel filter is installed between a rusty fuel tank and the fuel pump it will then trap any fine rust particles, clogging up rather quickly. When it does, it will cause the fuel pump to stall in a “current on” condition. If left with the power on very long when this condition exists, it will then burn out the internal swamping resistor inside of the solenoid coil of the fuel pump. Once the fuel filter is replaced, everything will appear to be normal and the owner will go his way thinking that the problem has been solved. Unbeknownst to him is the fact that the burned out swamping resistor defeats the fuel pump’s arc suppression circuit and that the contact breaker points will burn out a short time later. The owner then installs a new set of contact breaker points, only to have them again burn out in a short period of time. This is the reason that a set of replacement contact breaker points will seem to burn out prematurely, so be sure to check the fuel filter on a regular basis. As a precaution, you should seal the inside of the fuel tank against rust and corrosion. Eastwood sells an excellent sealer that you simply slosh around inside of the fuel tank and allow to cure. However, be sure to blow out the screen inside of the fuel tank with compressed air before the sealant cures, otherwise the fuel pump will not be able to deliver the fuel to the carburetors. Eastwood has a website at http://www.eastwood.com/ .

When rebuilding your fuel system, do not under any circumstances use copper tubing to replace fuel lines (pipes). While it is true that the Original Equipment fuel and brake lines (pipes) were made from a copper / nickel alloy and are fine for use with the gasoline of yesteryear, today many unscrupulous vendors sell plain copper tubing for this purpose. However, vibration will cause plain copper tubing to work harden and crack. If you place a pure copper strip in gasoline (or LPG, or Diesel, or Jet Fuel) and warm it to 50° for two to four hours, a very colorful range of copper sulphides and oxides will appear on the surface if you leave it in for too long or if it is a bad batch of fuel. Typically, you will see purples, mauves, blues, and blacks. Yes, copper certainly is subject to chemical attack. This test has been retained by fuel companies because at present we use a lot more catalytically cracked gasoline, misleadingly called “stabilized gasoline” or more commonly referred to as “Stab gaso”. Catalytically cracked gasoline produced from relatively valueless heavy black “leftovers” is one of the main reasons why we have not run out of fuel despite all those predictions in the 1970's. Sadly, this is the component of gasoline that really attacks and corrodes copper. Copper catalyzes the auto-oxidation of fuel, which is the mechanism of sludge formation, thus it should never be in contact with fuel. At one time, it was quite commonplace to see a lot of copper used in the fuel system. Now, thanks to the use of stabilized gasoline, the gauge sender and certain parts of carburetors are just about the only copper parts left in use. Use stainless steel tubing instead.

Carburetion

The use of the Weber DCOE 45 carburetor on street MGBs came about as a result of their use on the factory team's track race cars. This fact, of course, produced a "monkey see, monkey do" mentality amongst some of those seeking more power for their street MGBs. Why did the factory race team choose the Weber over the tried-and-true SUs? It has to do with the design differences between the two. The SU is a Variable Venturi type, which in original form makes for a smooth, although slightly slow, throttle response and excellent fuel economy. The Weber DCOE 45, on the other hand, is a Fixed Venturi type. It has the advantage of having an injector pump to shoot raw gasoline into the venturi when the throttle opens rapidly and thus makes for very fast throttle response. This was a definite advantage on the racetrack, so that is part of the reason why the factory race team chose it over the SU. Remember that on a racetrack, smoothness and economy must be subordinate to responsiveness, as it is responsiveness that makes aggressive driving possible. Victory is what counts on the racetrack, and nothing else will substitute. However, the constant size of the fixed venturi of the Weber design makes for low air velocity as compared to the variable size of the variable venturi SU design, and so cylinder filling at low engine speeds, and hence low-RPM torque, suffers. However, there is a partial solution to the inferior part-throttle power output of the Dellorto DHLA and Weber DCOE carburetors vs. that of the SU. It is a dual system that uses a concentrically-mounted air horn to channel more airflow directly into the auxiliary venturi, which increases the air speed over the venturi, which in turn amplifies the mixture signal, thus bringing the engine onto the cam earlier. The underside of the air horn channels the airflow into the stub stack in order to smooth the entry of the airflow into the primary venturi with minimal turbulence and swirl, thus creating more power throughout the range. This intake system is available from MED engineering over in the UK. A Series engine expert Keith Calver, who owns and runs Mini Mania, helped with the developmental work. Dynometer results with an A series race engine showed a 4 to 6 BHP increase over that of a standard air horn setup. That is pretty wild for such a small engine, so you can just imagine what it does for a B Series engine equipped with a cylinder head that has been reworked by Peter Burgess to his Fast Road specification. They are not cheap, but definitely worthwhile.

This fast throttle response produces the illusion of more power, and so purchasers of this unit tend to experience what Psychologists call the "Halo Effect": they have paid out the big money, sweated the installation and calibration, spent more money to convert their ignition system to a racing type centrifugal advance distributor (Weber carburetors do not have provision for a vacuum takeoff for functioning with vacuum advance ignition systems: read the fine print!) and so they are already predisposed to feel a power increase even before they drive. When the quick throttle response creates the illusion of more power, they become like religious converts! In reality, all other factors being equal, there is no worthwhile difference between them in terms of power output on the dynamometer readouts unless a radical camshaft that produces its maximum power output at elevated engine speeds is being used.

However, it is not commonly understood that the SU carburetor design also has provision for pumping additional fuel into the venturi when the throttle is opened suddenly. The manuals have a distinct lack of detail on how this vital mechanism actually functions. The poppet valve feature of the damper is what allows the vacuum piston to rise and fall at two different rates, one for ascension and another for descension. When the throttle opens, the upward movement of the piston is delayed by both the poppet valve of the damper assembly inside of it and by the piston spring. This is due to the fact that the damper includes a one-way valve in which the seating element pops open to obtain free flow in one direction and immediately reseats whenever the flow reverses. With the piston thus permitted to rise only gradually as the throttle opens rapidly, the air velocity through the venturi increases dramatically, increasing the vacuum above the fuel jet bridge. Because the pressure differential between the atmosphere inside of the float bowl and the atmosphere above the fuel jet is increased, the higher atmospheric pressure inside of the float bowl "pumps" additional fuel through the fuel jet into the carburetor venturi, momentarily enrichening the fuel / air mixture in order to prevent a hesitation of response to the throttle change. Because the poppet valve of the damper works in one direction only, the vacuum piston is allowed to drop quickly, preventing an overly rich fuel/air mixture whenever the throttle is closed rapidly. Should the poppet valve be rendered non-functional as a result of clogging, the resultant rich mixture when the throttle is closed rapidly will be revealed as a popping sound in the exhaust note should there be an air leak in the exhaust system. The poppet valve of the damper is accordingly an important feature in terms of throttle response, and as such should be kept clean so that it will function properly. Fortunately, this is easily accomplished by periodically spraying the damper with carburetor cleaner. 10w/30 oil is only recommended for temperatures of 5° Fahrenheit (-15° Celsius) to 23° Fahrenheit (-5° Celsius), and 20w/50 is recommended for all temperatures above 14° Fahrenheit (-10° Celsius). When you get different results while using a different thickness oil, then if your fuel/air mixture is set to the lean side you will get better results with thicker oil, and if set to the rich side you will get better results with thin oil. In order to get the most out of the function of this design feature when driving hard, it is critical that the damper tube of the piston be kept filled with a 20W/50 oil that is relatively uneffected by temperature change, such as Mobil 1 synthetic oil. Why an oil that is relatively uneffected by temperature change? Because an oil whose viscosity is strongly effected by temperature change would cause an inconsistent effect on the primary function of the damper mechanism. However, be aware that while the engine is warming up you will have to take care to open the throttle very gently. This will be due to the fact that when using a normal engine oil that thickens when it is cold, the viscosity of the oil in the damper mechanism is high and the degree of enrichment attained when opening the throttle is therefore greater than when the engine has reached its normal operating temperature. A more stable synthetic engine oil will not provide this cold-start benefit, and hence the engine will tend to stall more easily when cold.

The other purpose of the damper feature is to prevent the pressure fluctuations occurring in the airflow of the incoming fuel / air charge from causing the vacuum piston to rapidly oscillate inside of the vacuum chamber (dashpot), playing havoc with accurate fuel metering. If you are using a high performance camshaft, be aware that these tend to produce more violent pulses in the induction system than milder camshafts do. The heavier oil helps to dampen this problem and prevent piston flutter under hard acceleration. This is easily visible under rolling road conditions. The lighter oils tend to squirt out of the top, too.

What is the reason that the factory diagrams show the oil level above the hollow damper tube? If you invert the vacuum chamber (dashpot) and examine it carefully, you will then notice a bushing inside of its neck. The damper tube of the piston is a precision fit inside of this bushing. When the piston rises, the air trapped inside of the damper mechanism forces oil downward around the neck of the piston in order to supply lubrication to this bushing. Should the vacuum chamber (dashpot) bushing and / or the piston damper tube become badly worn from lack of lubrication, the vacuum chamber (dashpot) will quickly duct the oil above the damper tube into the intake manifold. In addition, air will leak past the bushing of the vacuum chamber (dashpot) into the vacuum chamber (dashpot), decreasing the pressure differential and thus causing the piston to rise less than it should. An otherwise unexplainable lean running condition will result. Overfilling just spills over into the vacuum chamber (dashpot) and makes a mess. One easy way to check the oil level is to remove the damper and then re-insert it. If you feel resistance before you reach the threads on the cap, then you have enough oil for the damping effect to occur. Why use 20W/50 oil? Simple: for optimum acceleration response, use 20W/50. If too thin an oil is used, then the piston will lift too rapidly, resulting in an over-rich fuel / air mixture. This larger-diameter venturi will not only result in loads of air being ingested, but will also result in a lower intake charge velocity that produces an inadequate vacuum above the jet bridge to lift the needed amount of extra fuel for accelerating. The consequence will be that the engine will run lean and hiccup below 3,000 RPM.

Many SU carburetors have had their damper rods replaced with the wrong type by DPOs. It should be noted that the area above the hollow vacuum piston rod must be vented, otherwise internal pressure will build up on the upward movement of the piston and vacuum will occur on its downward movement. This will restrict normal piston travel. Venting may be done in one of two ways. The damper cap may be drilled to allow venting to atmosphere, or the gusset on the neck of the vacuum chamber (dashpot) may be drilled in order to allow internal venting back into the vacuum chamber (dashpot) and thus prevent dust from being sucked into the carburetor through the hole in the damper cap. It should be noted that the vent hole, be it in the damper cap or be it in the neck of the vacuum chamber (dashpot), should be periodically cleaned out with carburetor cleaner. You may use one type of venting or the other, but not both at the same time. If you have a solid damper cap and no internal drilling, then there is no vent and pressure / vacuum conditions will occur as aforementioned. If the damper cap is drilled and the web is also drilled, then there is a direct air leak into the vacuum chamber (dashpot) that will interfere with the lift of the vacuum piston. If the neck of the vacuum chamber (dashpot), such as found on 11/4" SU carburetors, has no webbing, then it cannot be drilled internally and thus it must make use of a vented damper cap. If the neck of the vacuum chamber (dashpot), as found on 1½" and larger SU carburetors, has a web or gusset, then it may or may not have been drilled. The only way to know for sure is to remove the damper and look inside of the neck. If you have a plastic damper cap which is incorrect, then you should simply replace it with the correct one. If you have a solid brass damper cap which is unvented, then you can drill the solid damper cap with a l/16" diameter drill. If you have a vented brass damper cap, then you can plug it with a short piece of 14 gauge copper wire. Cut the wire just barely longer than the thickness of the damper cap and peen it from the underside with a hammer and punch in order to keep it in place. Polishing the top surface with a fine wire rotary brush on a drill will make this improvised plug barely visible.

Upon occasion, a damper rod can become bent. This will force the piston of the damper rod into an off-center position. This in turn will create side thrust on the vacuum piston and thus restrict its normal movement. Remove the vacuum chamber (dashpot) and look at the end of the damper rod with its damper cap screwed snugly into the neck of the vacuum chamber (dashpot). If the end of the rod appears to be in the center of the bore, then it is most likely acceptable. However, if the rod is noticeably offset in one direction, then use a felt tipped pen in order to mark the damper cap in the direction that the rod needs to be bent. Remove the damper from the vacuum chamber (dashpot), and then gently bend the damper rod in the indicated direction. Several attempts may be required in order to get it into the right position. Visual centering is adequate as there is some lateral float in the damper piston. Fortunately, the new SU plastic-capped damper rods have the improved feature of a ball socket designed into its damper cap, which allows the rod to be self-aligning.

When air is passed through a venturi of fixed size, as is the case in designs such as those employed in the Weber DCOE and Dellorto DHLA, its velocity and amount of vacuum over the fuel-metering jet will vary with the demands of the engine. This varying amount of vacuum makes it necessary to employ compensating devices in order to produce the correct flow of fuel/air mixture and also forces a compromise on the choice of venturi size in that too small a venturi will result in a restriction at high engine speeds, while a too-large venturi will cause poor fuel metering and indifferent carburetion at low engine speeds. On the other hand, the operating principle of a Variable Venturi carburetor, such as is found in an SU, employs a design feature whereby the effective size of the venturi will expand as the need increases and contract when the need decreases. Such a variation in venturi area will produce a constant air velocity and vacuum over the fuel-metering jet.

Although the injector pump of the Weber DCOE and Dellorto DHLA carburetors endows them with fast throttle response, even this desirable advantage can be compensated for to a considerable degree in the SU design by three simple modifications. First, merely lift the vacuum piston to the top of its travel and scribe a line on both the front and back of it, using the bore of the body as a profiling template. Simply file or machine a 30° angle bevel on the leading edge of the vacuum piston and a 15° angle bevel on its trailing edge. While this will have no impact upon the airflow capacity of the carburetor, it will improve throttle response noticeably. It is an old MG Factory Racing Team trick.

Second, replacing both its vacuum piston with its spring-loaded biased fuel-metering needle and its vacuum chamber (dashpot) with the earlier vacuum piston with its fixed concentric fixed fuel-metering needle and a mating vacuum chamber (dashpot) (SU and BMC Part # AUD 9988) from the pre-1969 1½" SU HS4 Series carburetors (a simple "drop in" parts swap requiring only an inexpensive fuel jet / fuel-metering needle centering tool) improves both its immediate and its long-term performance further. This is due to the fact that the spring-loaded biased fuel-metering needle bears against and wears the Internal Diameter of the fuel jet, requiring both the fuel jet and the fuel-metering needle to be replaced every 20,000 miles. It should be noted that the spring-loaded biased fuel-metering needle was a development born of the need to meet engine emissions standards. Through research, it was found that while the fuel passing the front and sides of the earlier concentric fuel-metering needle used in the earlier versions of the carburetor was efficiently atomized by the air stream, the fuel drawn from behind it was subject to turbulence, condensing into an inefficiently burning fuel / air mixture. Bearing against the downstream Inside Diameter (I.D.) of the fuel jet, the biased fuel-metering needle has very little fuel drawn from behind it, thus resolving the turbulence problem at the expense of higher maintenance.

If you choose to rebuild a 1½" SU HS4 Series carburetor of unknown vintage for use on your engine, then be aware that the earliest versions had a damper (BMC Part # AUC 8103)) that has a piston that is .378" (9.60mm) in length. These were found to be a cause of uneven running, as well as a restriction of maximum engine speed. This was remedied on later carburetors by the use of a redesigned damper (BMC Part # AUC 8114) that has a piston that is .308" (7.62mm) in length. This redesigned damper piston should have the top of its octagonal damper cap stamped with an "O" for identification purposes. Fortunately, the damper rods are completely interchangeable.

These modifications will enable the 1½" SU HIF4 Series carburetor to meter fuel within a hair's breadth of a well set up Weber DCOE 40 carburetor at a fraction of the cost. Just be sure to refit the phenolic spacers and heat shield when you install them, otherwise the fuel will percolate in the float bowls, causing the engine to run lean and all but refuse to restart after being parked for a while when hot. Should you decide to reuse your old heat shield, be sure that its insulating pads on the side facing the intake manifold are in good condition. If they are not, new insulating material can be obtained at any Speed Shop frequented by the local Hot Rod set. Take care if any of the original insulating material is still on the heat shield. Be very circumspect about removing it as it contains asbestos that is a tremendous health hazard. Do not do anything to it that will cause it to produce dust that can be inhaled, as this asbestos dust can cause mesothelioma, a medical condition that is something that you definitely do not want. If you must do anything with this material, put the heat shield into a bucket of water and do whatever is necessary in order to remove it while it is submerged so that no airborne dust is generated, and then put the residue in a plastic bag and seal it before disposing of it. Be aware that the heat shields used with the 1½" SU HS4 Series carburetors (BMC Part # 12H 719, Victoria British Part # 10-35) and 1½" SU HIF4 Series carburetors (BMC Part # 12H 3607, Victoria British Part # 3-5742) are not fully interchangeable as their attachment points for the throttle cable are in different locations, causing wear of the throttle cable as a result of misalignment if used with carburetors that they were not designed to be used with.

Third, prior to selecting a fuel-metering needle, it is sometimes proved to be advantageous to replace the Original Equipment piston springs. It is not commonly understood that the factory deliberately specified 8 oz (yellow) piston springs in the 1½" SU carburetors in order to regulate the height of the piston. However, when radical high-lift camshaft lobe profiles are used in conjunction with cylinder heads that have been modified for increased rates of flow, the greater induced vacuum piston may cause the piston to reach its maximum height before the engine attains the engine speed at which it produces its maximum power output. This will result in the fuel/air mixture becoming too lean as engine speed rises beyond that point. When tuning the engine when it is not under load, the ideal strength of piston spring is one that permits the piston to reach its maximum upward position at that point in the engine speed range at which maximum power output is produced. When tuning the engine under load, at full throttle and 75% of the maximum permissible engine speed (redline), the vacuum piston should have just reached its maximum level of lift, with fueling at higher engine speeds being dictated by the vacuum created by the increased airflow drawing more fuel out from the jet. There should be no need to change the profile of the lower section of the fuel-metering needle as its purpose is to efficiently duct fuel through the fuel-metering jet. If you are using carburetors that have a maximum flow rate that is a bit on the large size, then be careful to start your experiments with the lighter-rate springs and gradually work your way to heavier-rate ones until the engine starts to hesitate upon opening the throttle, then use the next heavier-rate spring as your ideal choice. Conversely, if you are using carburetors that have a maximum flow rate that is a bit on the small size, then replace the springs with heavier-rate ones, typically of 2.5 oz to 5 oz. The springs available for the 1½" SU HS4 Series carburetors are the A-type in 2.5 oz at a height of 2 5/8" (Light Blue), 4.5 oz at a height of 2 5/8" (Red), 8 oz at a height of 2¾"(Yellow), 11.25 oz at a height of 3 7/8" (Red and Green), 12 oz at a height of 3" (Green), and 18 oz (Light Blue and Red). The springs available for the 1½" SU HIF4 Series carburetors are the B-type in 4.5 oz at a height of 2 5/8" (Red), 8 oz at a height of 2 ¾" (Yellow), and 12 oz at a height of 3" (Green).

As I am sure you are aware, the fuel-metering needle controls the fuel / air mixture in stages according to engine speed and vacuum. An engine that has been modified to breathe more deeply will have greater fuel needs as engine speed increases, so you will need the right fuel-metering needles to avoid problems with running performance. To help you get your fuel jetting and fuel-metering needles spot-on right, with over 350 fuel-metering needles for .090" fuel jets and over 200 needles for .100" fuel jets to choose from, you will find an investment in an SU Needle Profile Chart to be invaluable in making the engine sing as it should. Go to the Burlen Fuel systems website at http://www.burlen.co.uk/ and click on "View our latest news", then scroll down the page until you come to the yellow words "Catalogues and Merchandise" and click on that. The SU Needle Profile Chart is item # ALT 9601. After the mysteries of how these simple carburetors function have been dispelled from your mind, a session on a dynometer with the aid of an exhaust gas analyzer will have the carburetors metering your fuel / air charge to near perfection. Once you have got the carburetion properly fine-tuned, you will be amazed at how sweetly the engine will run!

It is possible that you will choose to rebuild your present type of SU carburetors and hope to later change to the other type. However, having already gone to the trouble of selecting the fuel-metering needle profile that is best for your engine, you might later hesitate to do so because you do not want to go to the trouble of selecting new fuel-metering needles for your new carburetors. There is a simple solution to this problem. There are two types of fuel-metering needles, spring-biased and fixed. The two types are essentially the same on their tapered sections, but the spring-biased and fixed fuel-metering needles have different bases. The fixed fuel-metering needles have a plain 1/8" diameter shaft. The spring-biased fuel-metering needles have a flange at the base in order to control how far to the side they are biased. At first glance, it appears that the flange of the spring-biased fuel-metering needles is machined as an integral part of the needle itself. However, upon closer scrutiny, it is revealed that the flange is actually a separate part that is press-fitted into place. Spring-biased fuel-metering needles are made the same way as fixed fuel-metering needles and then have their flanges simply press-fitted onto them. In order to make the change, you need to merely drill a hole the same size as the root of the flange, and then use a punch to drive the fuel-metering needle out of its flange. The end of the 1/8" base of the spring-biased fuel-metering needle has a knurled section that will need to be smoothed out with an ignition point file, but it is possible to interchange biased and fixed fuel-metering needles. All you need to do is to either press-fit or remove the flanged base of the fuel-metering needle. These later carburetors can be converted to fixed needles by installing a needle bush kit from Burlen Fuel Systems, Part # SU WZX-2003. It will also be necessary to install the earlier jet bearing which will permit centering the jet.

It should be noted that SU fuel-metering needles are identified by letters and occasionally by numbers that have been stamped into the shank of the needle. This causes an upset in the metal at the edges of the letters and may make it difficult to push the needle shank into the hole in the bottom of the vacuum piston. Never force the fuel-metering needle into its mounting hole in the bottom of the vacuum piston; otherwise, you may never get it out again without ruining it. Take an ignition point file and judiciously smooth the ridges around the letters until the base of the fuel-metering needle will slide in easily. In order to remove an old fuel-metering needle that you may want to re-use in the future, remove the jet locking screw and pull gently on the fuel-metering needle with your fingers. If it does not budge, then forbear the urge to grab the fuel-metering needle with a pair of pliers. Instead, spray some carburetor cleaner into the locking screw hole and leave it for about 30 seconds, followed by some penetrating oil. Gently tap the fuel-metering needle inward. The typical amount of movement is about 1/8". This should break the fuel-metering needle loose as well as lubricating a dry area of the shank of the needle. Pull gently again with your fingers. If it still does not move, then clamp the fuel-metering needle in a vise that has soft-faced jaws, and then gently pull and twist the piston in order to loosen the assembly.

Due to their greater airflow capacity (204 Cubic Feet per Minute), the larger 1¾" SU HS6 and 1¾" SU HIF6 carburetors coupled with a stub stack (APT Part # SS-52) might make for a bit more power (about 5 BHP) at high engine speeds (above 6,000 RPM) in a small bore engine (1868cc and smaller), but unless you are mounting them in order to meet the demands of either a Big Bore 1900+cc engine with ported cylinder heads or a smaller bore engine with a Piper BP285 camshaft and ported cylinder heads, you will get this minor power boost at the price of less power at the lower engine speeds below 2,500 RPM (which is where a street engine spends much of its operating life), a lumpy, vibrating idle, and difficult cold weather starting, all due to their lower port velocity resulting from their larger venturis. On a standard displacement engine with unmodified cylinder heads, they will sacrifice as much power below 4,000 RPM as they will gain above that point. They do, however, have the advantage of an additional metering diameter on the fuel-metering needle, making for more precise fueling control. Should you decide to use them on the aforementioned engine types, mount them on either a Maniflow intake manifold or the reproduction of the Special Tuning intake manifold (Special Tuning Part # 12H 2838) that is available from Burlen Fuel Systems.

Rather than go to the trouble and expense of purchasing, mounting, and tuning a pair of 1¾" SU HS6 or 1¾" SU HIF6 carburetors (Burlen Fuel Systems Part #s Gen HS4 & GEN HIF6, respectively) in an attempt to meet the airflow requirements of a hotter camshaft and resigning yourself to the resulting inferior starting and idle characteristics caused by their inherently lower port velocities at idle speed, it would be preferable to increase the airflow capacity of the 1½" SU HS4 Series carburetors or 1½" SU HIF4 Series carburetors in order to retain the advantage of their higher port velocities at low engine speeds. This increase in airflow capacity can be accomplished by a series of modifications.

Under no circumstances should you attempt to streamline or modify the fuel jet bridge of an SU carburetor in any way. To do so will damage or destroy part throttle running characteristics. Its square profile is intended to cause the incoming fuel / air charge to lift slightly and thus keep the atomized fuel in suspension, which it does quite effectively, thus the design of the fuel jet bridge should be considered to be already optimized. Instead, more can be accomplished by modifying the throttle disc and the throttle shaft.

The airflow rate of both the 1½" SU HS4 Series carburetors can be improved by retrofitting of the throttle disks from the earlier pre-1968 1½" SU HS4 Series (Burlen Fuel Systems Part # WZX 1323, BMC Part # AUC 3116), while that of the 1½" SU HIF4 Series carburetors can be improved by fitting the throttle discs used on the 1½" SU HIF4 Series carburetors found on the UK/European market 18V846 and 18V847 engines (Burlen Fuel Systems Part # WZX 1329, BMC Part # AUD 9808). Another option would be the fitting of the earlier throttle discs of the SU 1½" HS4 Series carburetors into 1½" SU HIF4 Series carburetors and filing the necessary pilot notch into its bottom edge. The SU HIF design has a small passage that runs from next to the main jet to under the throttle plate. The idle fuel / air mixture comes from the main jet through this passage. It provides a better fuel / air mix, thus resulting in more efficient combustion at engine idle speeds and lower emissions. Of course, the notch has no effect when the throttle is open. These throttle discs lack the airflow-obstructing poppet valve of the later versions and provide greatly improved engine braking as well. Next, the throttle plate should be thinned to one-half of its original thickness and knife edged. This should be accomplished by an angled cut along the upper forward face and the lower rear face to an edge thickness of .010" to .015" (.254mm to .381mm). Reduction of the section of the throttle shaft that the screw heads fit into along its entire length between the throttle shaft bushings to a thickness of .075" (1.905mm) will further improve airflow capacity. The throttle plate screws should be replaced with countersunk ones that have shallow dome heads and their holes in the throttle plate should be suitably modified by beveling in order to accept them. These new screws should preferably have Allen head sockets instead of slots, and any protruding section of their threaded ends should be removed and carefully filed flush with the throttle shaft. They should then be secured either with Loctite or, better yet, by brazing. Finally, the edges of the throttle bore section behind the piston should be enlarged into a square shape with radiused corners in order to remove their airflow-interrupting edges. Taken together, these modifications will increase the airflow capacity of the carburetor by approximately 30% to 169 Cubic Feet per Minute, about midway between the airflow capacities of unmodified 1½" SU HS4 Series carburetors and unmodified 1¾" SU HS6 carburetors.

However, in the case of your engine truly having a legitimate need for exceptional airflow capacity in its fuel induction system, such as in the case of a Big Bore engine of 1900cc or larger displacement with fully flowed five-port cylinder heads and a Piper BP285 camshaft, Burlen Fuel systems has available a conversion kit (Burlen Fuel Systems Part #

FZX 3063), complete with a matched pair of 1¾” SU HS6 carburetors (Burlen Fuel Systems Part # GEN HS6), a copy of the original Special Tuning intake manifold (Burlen Fuel Systems Part # AUE 103), and an appropriate heat shield (Burlen Fuel Systems Part # ABF 553), linkage, fuel lines, gaskets, and fitting instructions.

Do not buy SUs from an aftermarket outlet. Cut out the money-grubbing intermediaries and have Peter Burgess get them direct from the Burlen Fuel Systems factory and set them up to fit his headwork, or buy them yourself through their website at http://www.burlen.co.uk and follow his instructions on which fuel-metering needle and fuel jet combination to use.

There is another reason to use the SU: aesthetics. They look right, especially when used with K&N or Original Equipment air filters. The sidedraft Weber DCOE 45 looks as though it has been adapted and, due to clearance problems, changing an airfilter element of adequate airflow capacity is no fun at all. The fuel suspended in the incoming fuel / air charge is denser and heavier than the air, its greater inertia thus causing it to go towards the outside of the runners of the curved intake manifold, biasing the fuel towards the intake valves of #1 and #4 cylinders and thus creating a richer fuel / air mixture for those cylinders.

The intake manifold shape for the downdraft Weber DGV carburetors is actually even worse. The downdraft Weber DGV 32/36 makes the engine look as though it was pirated from a Russian tractor. It’s usually-included adapter manifold has the airflow characteristics of a bathtub with a hole in each side. This is due to Pierce Manifolds, its distributor, bundling their own poorly designed intake manifold with the carburetor and selling the resulting package as a kit. As a result, virtually every example of this combination that I have encountered or ever heard of had a “flat spot” in the powerband from 1,500 RPM to 2,500 RPM where throttle response was poor due to a weak fuel / air mixture. This “flat spot” can be eliminated by instead using a Cannon intake manifold that incorporates a pair of fittings that cycle the hot coolant from the radiator heater hose (flexible pipe) to the front of the intake manifold and from the rear of the intake manifold onward to the heater box. The result is smoother running and elimination of the infamous “Flat Spot”. However, this will not eliminate the problems imposed by the restrictive airfilter that Pierce Manifolds supplies with it in its kit. The Weber DGV 32/36 is a progressive-type carburetor. This means that the 32mm primary bore is opening with the throttle first, but the 36mm secondary bore (hence the 32/36 designation) does not open until the accelerator pedal is about 2/3rds of the way to the floor. Since 1½" equals about 38mm (1.496"), there are two considerations. First, due to the smaller bores of the Weber design, the two bores of a Weber DGV 32/36 Series carburetor together cannot possibly flow as much fuel / air mixture as twin 1½" SU carburetors can at full throttle. Second, when at less than full throttle, the progressive opening of the 32mm and the 36mm bores of the Weber design do not flow as efficiently as the constant-velocity airflow of the 1½" SUs. It is possible to convert the progressive throttle plate linkage to simultaneous throttle plate linkage which would provide a carburetor that would drive very much like the twin 1½" SU carburetors. However, with slightly smaller venturis than the 1½" SU carburetors, it is simply impossible for a Weber DGV 32/36 to perform as well as either twin 1½" SU HS4 Series carburetors or twin 1½" SU HIF4 Series carburetors. If your goal is performance, remember that the potential of the Weber DGV 32/36 is inferior to that of the twin 1½" SU Series carburetors. Its cousin, the Weber downdraft DGES 38/38, has twin 38mm synchronous butterflies. It is larger than the standard 32/36mm DGV, and thus has greater airflow capacity. It mounts onto the same intake manifold and gives more torque at low engine speeds, but can make the engine difficult to start in cool weather and has developed a reputation for troublesome running at low and moderate engine speeds. Both of these Weber carburetors require rejetting in order to function properly on the B Series engine.

If you are refitting a post-1974 single-carbureted engine with dual 1½" SU carburetors, be aware that the two carburetor types, the 1½" SU HS4 Series and the 1½" SU HIF4 Series, use different intake and exhaust manifolds. The intake manifold for the 1½" SU HS4 Series carburetors has a mounting flange thickness of 9/16" and can be readily modified for provision for distributor vacuum takeoff. In fact, the intake manifold of the SU 1½" HS4 Series-equipped 18GK engine already has this modification. The intake manifold of the 1½" SU HIF4 Series carburetors has a mounting flange thickness of 7/16" (11.1125mm) and also has provision for distributor vacuum takeoff. There are also two different exhaust manifolds with mounting flange thicknesses that are correspondingly paired with these intake manifolds. Should you elect to install a tubular exhaust manifold rather than an Original Equipment cast exhaust manifold, be sure to check the thickness of its flanges before you make your purchase, otherwise you will be likely to find yourself fabricating custom half-moon shims!

Also, be aware that the vacuum advance control capsule of the distributor used with the pre-1971 North American Market 1½” SU HS4 Series carburetors takes its vacuum from a connection on the carburetor, while the vacuum advance control capsule of the distributor used with the North American Market 1½” SU HIF4 Series carburetors takes its vacuum from the intake manifold. These two systems result in highly different initial ignition advance characteristics. Manifold vacuum continuously varies as the throttle is being opened. Only when the throttle is wide open is the vacuum at a minimum, but even then there is still some present because of restrictions in the throat of carburetor and air cleaner housing. It depends on the specification of the vacuum advance control capsule as to when vacuum advance ceases to be applied and can be as low as 3” Hg or as high as 10” Hg, depending on which vacuum advance control capsule it uses. Manifold vacuum distributors have maximum vacuum at idle, and hence have maximum ignition advance at idle because this allows a smaller throttle opening and hence lower emissions for the same idle speed, at the expense of ease of starting and initial throttle response. Carburetor or ported vacuum distributors have no vacuum at idle and hence no ignition advance at idle. However, as the throttle opens the vacuum rapidly increases to become the same as that produced by the gradual fall in vacuum in manifold vacuum distributors. They are the same thereafter. The pre-1971 North American Market 1½” SU HS4 Series carburetor system uses the vacuum that is produced when the throttle opens in order to advance the ignition timing, resulting in easier starting and quicker off-throttle response. The North American Market 1½” SU HIF4 Series carburetor system uses the vacuum of the intake manifold in order to advance the ignition timing while the throttle is closed, resulting in harder starting and slower off-throttle response, but lower exhaust emissions and better fuel economy while idling. The hard starting problem of this latter system can be easily overcome by simply opening the throttle all the way while cranking the engine. Once the throttle opens, the vacuum is the same on both types. If you prefer to use a set of 1½” SU HIF4 Series carburetors while retaining the advantage of the superior off-throttle response of the ported 1½” SU HS4 Series ignition advance system, the UK/European market versions used the ported vacuum of the 1½” SU HS4 Series carburetors and can be ordered from Burlen Fuel Systems. Of course, your distributor’s vacuum advance control capsule will have to be compatible with whichever version of the vacuum system you choose to employ.

Why do performance tuners always recommend a ported vacuum advance control system? Start by picturing what the vacuum advance control system actually does: it rotates the contact breaker plate in order to advance the ignition timing as vacuum is applied. In a manifold vacuum advance control system, there is vacuum applied at idle, hence extra ignition timing advance is applied at idle. Let us assume the ignition advance curve in your distributor is already optimized. When you attempt to push the throttle pedal through the floor for a fast take-off (or literally any start from a stop) there is too much ignition timing advance for the engine to be able to respond properly. The vacuum level in the fuel induction system decreases after you open the throttle, and the vacuum advance control capsule responds by releasing pressure and rotating the contact breaker plate back to its static position. All this takes time, during which your car is still trying to accelerate with too much ignition timing advance, at a decreasing rate due to the whole procedure of the ignition advance system releasing vacuum. Almost a second passes while the vacuum advance control capsule is retarding the contact breaker plate back to “zero vacuum advance”. During this second, the engine is running with too much ignition timing advance, in effect leaning out the fuel / air mixture (or enrichening it if your fuel / air mixture is way off), and it will take a few more engine revolutions to generate a appropriately clean burning fuel / air mixture. All of this is typically symptomized by “hesitation” in throttle response. By switching to a ported vacuum advance control system, if available, you can alleviate this hesitation and correct your ignition timing and fuel / air mixture to a properly controlled situation from the start. With ported vacuum, a long, slow increase of ignition advance results in a smooth throttle response as vacuum levels rise and drop. However, in the case of manifold vacuum, the extra ignition advance off-idle can cause hesitation, so you need a vacuum advance control capsule that drops all of the ignition advance very rapidly, at a high vacuum level. Since it takes time for the contact breaker plate to rotate back to the point of zero ignition advance, the earlier this process starts, the better! When ignition timing can be off by up to 24º (76-80 MGBs) because of added vacuum advance, getting rid of it under full throttle is imperative! Keep in mind that this has nothing to do with the ignition curve that is controlled by centrifugal advance. It is all about the vacuum levels an engine produces and the type of vacuum source that is employed.

If you have the later SU HS4 Series carburetors that do not have provision for a ported vacuum advance, be aware that they can be modified to provide this feature. You don't have to drill any new holes into the body of the carburetor, which is fortunate because if you try to do so, unless you have uncanny luck, you will most likely damage it.

Upon close examination, you will notice that there is a brass press-fitted plug located behind the throttle disc of the carburetor. You will want to use the rear carburetor, as the factory did. Remove the throttle disc and its shaft so that the plug will be accessible from inside the bore of the carburetor. Make a small tool which resembles a tiny pin-punch that is perfectly flat on both ends. Use a very small "C" clamp and insert the machined locator of the "pin-punch" into the top of the plug. Next, place a 3/8" nut beneath the brass plug. When the brass plug is pushed out by the action of the tightened "C" clamp, it will emerge into the nut. Once it is that far out, it can then be gripped with some vise-grips and pulled the rest of the way out.

Once it has been removed, you will notice the small brass plug is designed to convey air from a hole bored within the carburetor body just behind the jet. This air travels from behind the jet and into the brass plug and thence upwards, exiting at the small "notch" in the early throttle plates. In order to convey this air, the plug is bored perpendicular to its length about in the middle and, of course, is also bored from the its top down. This allows aforementioned air to make the 90^{o} bend upward to the throttle disc.

In order to bore this hole into the body of carburetor, the factory found it necessary to begin at the rear of the carburetor where it mates with the intake manifold. The brass plug is bored on only one side so that when it is inserted, while it allows the air to travel from the area of the jet, it also blocks up the hole in the carburetor body from the plug back.

In order to install a vacuum tube, you not only have to port to the outside of the carburetor, you also have to make certain the hole is there for air to travel from behind the jet. Fortunately, this is fairly easy.

Cut a piece of standard 3/16" brake line approximately 1.5" long. This will fit into the bore in the carburetor from which the brass plug was removed with a nice interference fit...not too snug, yet tight enough to remain in place. This tubing must be pressed in to the exact depth needed by to just barely protrude into the "notch" in the throttle disc. Then, from the back of the carburetor, using the hole bored therein by the factory as a guide, drill half-way through the inserted brake line tube with an appropriate size drill. The tube is then removed and all drilling swarf removed. It is then reinserted 180^{o} from its first position so that the hole just drilled faces forward and the wall of the tube which was not drilled through plugs the rear portion of the factory-bored through-hole.

SU HS vs. SU HIF Carburetors

There has been a great deal of discussion of the relative merits and vices of the 1½” SU HS4 Series carburetor and those of its successor, the 1½” SU HIF4 Series carburetor. Advocates of the 1½” SU HS4 Series carburetors point out the greater ease with which the fuel jet can be changed with the carburetor in place on the engine and the metering advantage of its concentrically mounted fuel-metering needle and fuel jet. Some feel that its remote float bowl design gives it a “Vintage” appearance. However, the 1½” SU HS4 Series carburetors are not without their vices. They have a tendency to run rich or lean under conditions of rapid acceleration and deceleration, during hard cornering, and when on a steep road. The remedy was to fit a spacer between the float lid and float bowl to raise the fuel level held in it. This was satisfactory when running, but at idle and rest fuel would bubble out of the jet, causing bore washing, poor pick-up, and horrendously rich CO2 mixtures at idle! The design of the SU HIF Series carburetors largely addressed these problems by having its float bowl integral with its body, thus allowing the float to surround the fuel jet and hence more consistently meter fuel under high angles of tilt and under conditions of heavy cornering stresses. It also has superior performance potential due to its higher maximum flow rate, a factor that should be considered if you intend to modify the engine for increased power output.

In addition, the 1½” SU HS4 Series carburetor require the removal of their airfilter boxes in order to enable the use of a pair of special short wrenches (Burlen Fuel Systems Part # SUT 2) in order to effect fuel / air mixture adjustment, which results in a somewhat richer fuel / air mixture when the airfilter boxes are refitted, especially if cheap, restrictive paper air filters are fitted inside of them. They also have a tendency to leak fuel from the float bowl junction and from the external base of the fuel jet. The latter feature is the result of the necessity of retracting the fuel jet downward in order to enrich the fuel / air mixture during cold starting conditions, causing wear of the sealing glands. In terms of cold starting, the 1½” SU HS4 Series carburetors use a cable-operated lever that both lowers the fuel jet as much as 7/16” and also opens the throttle disk slightly by means of an attached fast-idle cam in order to prevent low speed stalling under the conditions of an over-rich fuel / air mixture. On the other hand, the design of the 1½” SU HIF4 Series carburetor uses a lower-maintenance separate fueling circuit in order to accomplish this function. Fuel / air mixture enrichment for cold starting is accomplished by means of a rotary valve that has a groove that varies in depth machined into its shaft, ensuring that the degree of enrichment is progressive. The fuel then passes through a separate fuel passage within the body of the carburetor that terminates in the high vacuum area close to the fuel jet aperture. Another, more refined development is that in which in order to provide a well-atomized fuel / air mixture at small throttle openings, a duct adjacent to the jet orifice is connected by a small-bore passage to an outlet point at the edge of the throttle disc. Fuel / air mixture is induced into this passage by the high vacuum at the edge of the throttle disc.

Although more expensive to purchase and a bit more time consuming to set up than the 1½" SU HS4 Series carburetor, the 1½" SU HIF4 Series carburetor is easier to adjust and has superior performance potential. During routine adjustment its fuel / air mixture can be modified from above with nothing more than a simple screwdriver, hence removal of the airfilter boxes is not necessary. In addition, the SU HIF design introduced ball bearing vacuum chambers in an effort to provide ever closer fueling control. A plastic sleeve containing two rows of six small ball bearings was introduced in order to reduce friction in the piston/suction disc/damper rod assembly. The benefit is a more rapid response to varying demands caused by the opening and the closing of the throttle. Its thermosensitive fuel / air mixture control makes for easier cold weather starting. A bi-metal blade is used adjust the height of the fuel jet as needed according to the operating temperature of the engine. This precise fuel-metering control means that once correct fueling is established by appropriate fuel-metering needle selection, the fuel / air mixture is maintained over a very wide range of operating temperatures. The thermally compensated jet also gives a consistent idle speed over a range of temperatures, whereas SU HIF Series carburetors can tend to stall in a long idle in summer where everything heats up, and can need a tweak of richening in winter and leaning (weakening) in summer. Drivability with SU HIF Series carburetors is consequently enhanced and emissions are kept within tighter limits during the cold start and warm-up period. The remedy was to fit a spacer between the float lid and float bowl to raise the fuel level held in it. This was satisfactory when running, but at idle and rest fuel would bubble out of the jet, causing bore washing, poor pick-up, and horrendously rich CO_2 mixtures at idle!

Those who have converted their cars from the 1½" SU HS4 Series carburetors to the 1½" SU HIF4 Series carburetors usually report a 1 to 2 mpg increase in fuel economy. Unfortunately, rejetting requires that it be removed from the intake manifold, plus its thermosensitive fuel / air mixture adjustment control can cause it to run lean if underhood temperatures rise seriously, as in the case of heavy traffic on hot summer days. Consequently, Jet-Hot coating of the exhaust manifold is a worthwhile investment, as is the fabrication of U-shaped heat shields in order to insulate the runners of the intake manifold.

To make your own intake manifold runner heat shields, simply cut out a stainless steel metal strip with tabs projecting up at each end. Drill a hole in each tab for mounting the heat shield onto the manifold mounting studs, and then bend the tabs up at a 90° angle. The metal in front of the tabs should then be bent downwards at a 90° angle to the main body of the heat shield. If you have a vice, then forming the strips for the heat shield into the mandatory "U" shape that will accommodate the bottom curvature of the runners of the intake manifold can be performed using a dowel rod or similar suitable-diameter object.

If you are thinking of replacing a set of worn Original Equipment 1½" SU HIF4 Series carburetors with a set of 1½" SU HS4 Series carburetors because they cost less, think again. It can be done, but it is not the easy bolt-on swap that some presume that it might be. You will need an SU HS4 Series heat shield so that the throttle cable will properly align, distributor, cables, plus the linkages and a lot of other little bits and pieces that are not commercially available anymore, so you will spend a lot of time scrounging around trying to get them. If it is the lower price of the 1½" HS4 Series carburetors that seems attractive, be aware that when you get through buying all of the hardware necessary to do the installation correctly, the total difference in cost will not be anything like what you hoped it would be. Whichever version of the SU carburetor you choose, you will find it helpful to obtain copies of the "SU Reference Catalogue" and "The SU Workshop Manual" from Burlen Fuel Systems.

If you have 1½" SU HS4 Series carburetors or 1¾" SU HS6 Series carburetors, remove the fuel lines from the lids of the float bowls. Using an awl or a sharp nail, scribe a continuous line onto both the upper lip of the bodies of the float bowls and the side of their lids in order to mark their locations so that upon reassembly the fuel lines from the bottom of the float bowls will be properly oriented toward the fuel jets in the bottoms of the bodies of the carburetors. Remove the three retaining screws and their washers from the lid of each the float bowls, clean them thoroughly, and set them aside. Pull upwards on the lids of the float bowls in order to remove them from their float bowls and set them aside. Clean out the overflow vent passages and their overflow tubing completely. If they should ever become blocked, the float will not be able to rise and close the shut-off needle valve, resulting in an excess of fuel running down through the fuel feed line (tube) into the jet and flooding the engine. On engines that are not equipped with an anti-run-on valve, there should be a rubber pipe attached to the overflow vent, followed by a metal pipe attached to it that will then run down to the front engine mount where any overflow of fuel can escape. Clean out the interior of the float bowls with carburetor cleaner and either a clean lint-free cloth (preferably) or a clean paper towel. Next, holding the lids of the float bowls upside down, swivel the floats aside and remove the shut-off needles from their valve bodies. Push out the hinge pin of the float from the end that is opposite to its serrations, and then detach the float. Disconnect the fuel feed line to the fuel jets from the float bowl, and then extract the float needle from its seating. Gently clean all of the fuel passages, as well as the shut-off needles and their seats in their valve bodies with carburetor cleaner.

At this point, you should carefully inspect the .070" shut-off needles and their seats in their valve bodies for any signs of damage. Because both the tip of the shut-off needle and its seat in the valve body can be easily jammed or damaged by debris, an event that is the most common cause of carburetor flooding, you should consider replacing the shut-off needles with a Viton-tipped version (Burlen Fuel Systems Part # WZX 1097). Because they will withstand a fuel system pressure of about 10 psi, these are a distinct improvement over their brass-tipped predecessors which could withstand a fuel system pressure of a mere 6 psi. The shut-off needles should be installed with their pointy ends inside of the valve body, and the flat end with the pin sticking out should be what the float pushes up against. If either of the valve bodies is worn or damaged, it should be replaced. Taking care not to cause any distortion, unscrew it from the lid of the float bowl with an open end wrench and replace it with a new one from Burlen Fuel Systems. Insert the shut-off needles into their valve bodies, swivel the floats back over the shut-off needle valves, and be sure that the floats are 1/8" to 3/16" away from the level of the rim of the lids of the float bowls. Be aware that until recently, the new floats manufactured by Burlen Fuel Systems no longer employed a bendable metal tab for the hinge. Instead, these are of one-piece plastic construction, thus requiring that shims be inserted between the valve body of the shut-off needle valve and the roof of the float bowl chamber. However, their new adjustable height StayUp® floats for the

SU HS2, HS4, HS6, and HS8 carburetors, are resistant to modern ethanol based fuels and has a military specification closed cell construction that makes it puncture proof. The ability to adjust the float arm allows for quick and easy minor alterations to float levels where required. Now, set the lids aside.

Setting the height of the float on an SU HIF Series carburetor likewise needs to be done fairly accurately. When the carburetor is upside down with the bottom cover off, place a straightedge across the face of the carburetor body so that it goes across the middle of the float where its U-shaped indentation is cut out (Note that you need to do this with the jet removed, otherwise the jet will be in the way). When left resting under its own weight upside down with the valve in place, the float should be 1mm below the straightedge. If it is not, then you need to gently bend the little metal tab up or down so that the float is sitting properly.

Upon occasion the jet bearing around the jet of an SU HS-type carburetor can wear, allowing a slow drip of fuel from the bottom of the carburetor onto the heat shield. To remedy this dangerous fault, support the base of the jet and then slacken the screw that retains the choke control lever (jet pick-up link). Relieve the tension of the choke control lever (jet pick-up link) return spring from the screw, and then remove the screw and the brass bush (when fitted). Unscrew the sleeve nut that retains the flexible jet feed line to the float bowl, and then withdraw the jet from the carburetor body. Note the gland, rubber washer, and ferrule at the end of the jet feed line. Remove the jet adjustment nut and screw. Unscrew the jet locking nut and detach both it and the jet bearing. Withdraw the jet bearing from the nut, noting the brass jet centering washer under the shoulder of the bearing. Discard the old jet bearing and replace it with a new one, and then reassemble the jet assembly into the carburetor body.

Next, spray carburetor cleaner down through the fuel jets in order to flush out any debris that might be interfering with their metering function. Spray the vacuum transfer holes in the carburetor bodies, and then clean the passages with a solvent-saturated Q-tip that has been twisted between your fingers so that no lint will remain in the passages. If you choose to spin the Q-tip inside of any of the passages in order to assist in cleaning them, spin it only in the same direction. You do not want any lint remaining in the passages if you can help it. Reinstall the lids of the float bowls using new lid gaskets, as well as antisieze compound on the threads of the retaining screws. This will prevent electrolytic corrosion of the threads in the aluminum alloy bodies of the float bowls and make the retaining screws much easier to remove the next time that you want to clean the float bowls. Take care that the retaining screws are tightened evenly in order to prevent warpage of the lids of the float bowls. Finally, fit the gland washer and the ferrule to the flexible fuel feed line, and then attach the flexible fuel feed line to the float bowl. Tighten the sleeve nut until the neoprene gland is compressed. Be aware that overtightening of the sleeve nut can result in fuel leakage.

Note the location points of the two ends of the return spring of the pick-up lever. Unscrew the lever pivot bolt together with its double-coil spring washer, or spacer. Detach the pick-up lever assembly and the return spring. Note the location of the two ends of the cam lever spring and push out the pivot bolt tube (or tubes), taking care not to lose the spring. Lift off the cam lever, noting the skid washer between the two levers.

Close the throttle and mark the relative positions of the throttle disc and the flange of the carburetor. Unscrew the two retaining screws of the throttle disc from the throttle shaft. Open the throttle and ease out the throttle disc from its slot in the throttle shaft. The throttle disc is actually oval and will jam it its slot if care is not taken. Tap back the tabs of the tab washer that secures the nut of the throttle shaft. Note the location of the lever arm in relation to the throttle shaft and the carburetor body. Remove the throttle shaft and detach the arm.

SU Fuel-Metering Needles

Before you can hope to properly adjust or tune your carburetors, you will need to be sure that they are equipped with the appropriate fuel-metering needles and fuel jets. Unfortunately, some carburetors have been rebuilt in a state of desperation, using parts that have been scavenged from cars in junkyards (autojumbles). Often these parts were acquired on the presumption that all 1½” SU carburetors, regardless of the engine that they may be attached to, use the same needles and fuel jets. Nothing could be further from the truth. As a starting point for this process, it helps to be sure that you have the correct items for your particular carburetors. SU carburetors have a small aluminum tag with the model identification code that is to be found under the screw that attaches the vacuum chamber (dashpot) to the body of the carburetor. The code numbers that are found at the base of the fuel-metering needle can correctly identify the fuel-metering needles.

The Original Equipment dual 1½" SU carburetors were equipped as follows:

Production Period	**Market**	**Type Model**	**Fuel-Metering Needle**	**Front Fuel Jet**	**Rear Fuel Jet**	**Throttle Disk**
1962-1963 18G Engines	All	1½" HS4 AUD 52	Rich- 6 Standard- MB Lean- 21 (Fixed Needle)	AUD 9141	AUD 9142	WZX 1323 (Plain Throttle Disk)
1963-1964 Competition Engines	All	1½" HS4 AUD 129	Rich- N/A Standard- UVD Lean- N/A (Fixed Needle)	AUC 5186	AUC 5186	WZX 1373 (Plain Throttle Disk)
1963-1964 Competition Engines	All	1½" HS4 AUD 279	Rich- N/A Standard- UVD Lean- N/A (Fixed Needle)	CUD 2706	CUD 2707	WZX 1373 (Plain Throttle Disk)
1964-1967 18GA, 18GB Engines	All	1½" HS4 AUD 135	Rich- 6 Standard- 5 Lean- 21 (Fixed Needle)	AUD 9141	AUD 9142	WZX 1323 (Plain Throttle Disk)

1967-1968 18GD Engines	All	1½" HS4 AUD 278	Rich- 5 Standard- FX Lean- G2 (Fixed Needle)	AUD 9141	AUD 9142	WZX 1323 (Plain Throttle Disk)
1968-1968 18GF Engines	North America	1½" HS4 AUD 265	Rich- N/A Standard- FX Lean- N/A (Fixed Needle)	AUD 9141	AUD 9142	WZX 1323 (Plain Throttle Disk)
1968-1969 18GH Engines	North America	1½" HS4 AUD 326	Rich- N/A Standard- AAE Lean- N/A (Fixed Needle)	AUD 9141	AUD 9142	WZX 1323 (Plain Throttle Disk)
1969 Special Tuning Engines	All	1 ½" HS4 AUD 505	Rich- N/A Standard- SY Lean- N/A (Fixed Needle)	AUD 9148	AUD 9149	WZX 1321 (Plain Throttle Disk)
1969-1971 18GG Engines	UK/Europe	1½" HS4 AUD 325	Rich- 5 Standard- FX Lean- GZ (Fixed Needle)	AUD 9141	AUD 9142	WZX 1323 (Plain Throttle Disk)
1970-1971 18GH Engines	North America	1½" HS4 AUD	Rich- N/A Standard- AAE Lean- N/A	AUD 9141	AUD 9142	WZX 1323 (Plain

		405	(Fixed Needle)			Throttle Disk)
1971-1971 18GK Engines	North America	1½” HS4 AUD 465	Rich- N/A Standard- AAL Lean- N/A (Fixed Needle)	AUD 9141	AUD 9142	WZX 1323 (Plain Throttle Disk)
1972-1972 18V Engines w/ F Prefix	North America	1½” HS4 AUD 492	Rich- N/A Standard- AAU Lean- N/A (Spring-Biased Needle)	AUD 9141	AUD 9142	WZX 1323 (Plain Throttle Disk)
1972-1972 18V Engines w/ Z Prefix	North America	1½” HIF4 AUD 493	Rich- N/A Standard- AAU Lean- N/A (Spring-Biased Needle)	WZX 1454	WZX 1455	WZX 1329 (Poppet Valve Throttle Disc)
1972-1972 18V Engines w/ Y Prefix	UK/Europe	1½” HIF4 AUD 434	Rich- N/A Standard- AAU Lean- N/A (Spring-Biased Needle)	WZX 1454	WZX 1455	WZX 1323 (Plain Throttle Disk)
1972-1974 18V 672/673 Engines	North America	1½” HIF4 AUD	Rich- N/A Standard- ABD Lean-	WZX 1454	WZX 1455	WZX 1329 (Poppet

		550	N/A (Spring-Biased Needle)			Valve Throttle Disk)
1973-1974 **18V 779/780 Engines**	UK/Europe	1½" HIF4 AUD 616	Rich- N/A Standard- ABU Lean- N/A (Spring-Biased Needle)	WZX 1454	WZX 1455	WZX 1329 (Poppet Valve Throttle Disc)
1974-1974 **18V 836/837 Engines**	North America	1½" HIF4 AUD 630	Rich- N/A Standard- ABD Lean- N/A (Spring-Biased Needle)	WZX 1454	WZX 1455	WZX 1329 (Poppet Valve Throttle Disc)
1974-1975 **18V 846/847 Engines**	UK/Europe	1½" HIF4 FCX 1001	Rich- N/A Standard- ACD Lean- N/A (Spring-Biased Needle)	WZX 1454	WZX 1455	WZX 1329 (Poppet Valve Throttle Disc)

Selecting SU Fuel Metering-Needles

Although most people are intimidated by the prospect, the truth is that selecting appropriate fuel-metering needles for your particular enhanced-performance engine is not as difficult as it might seem as long as the matter is approached systematically. Once all of the appropriate modifications have been performed in order to improve the volumetric efficiency of the engine, and the appropriate ignition curve established, the correct fuel-metering needle profiles become identifiable. Using Burlen Fuel Systems' "SU Reference Catalogue", find the section that covers 1800cc four cylinder engines equipped with dual carburetors. Noting the various spring and fuel-metering needle specifications, use the SU Needle Profile Chart to chart the fuel-metering needle sizes from the first diameter to the end of the operating range. In addition, a highly useful interactive chart can be found at that will help you to graphically see the changes of fuel flow rate between various fuel-metering needle profiles. The numbers begin at the biggest diameter and are at 1/8" separations (15 stations) from the base to the tip of the fuel-metering needles. To convert the numbers to inches 890 would be .0890". After selecting a pair of fuel-metering needles that should be slightly lean when compared with your original fuel-metering needles, test their performance in your carburetors at each metering stage of the fuel-metering needles. The first and second fuel-metering stages of all SU fuel-metering needles are for the idling phase of the engine, and for a given size and series SU carburetor are either identical or very similar, so regardless of which fuel-metering needle you have selected, the engine should start and idle reasonably well. The next four fuel-metering stages are for acceleration and cruising, and the next three fuel-metering stages are for high engine speeds. The rest of the fuel-metering stages are for efficiently channeling fuel through the fuel-metering jet without producing cavitation.

You will find that it is easier to determine the correct diameters for the different stages of the fuel-metering needles if you use a Gunson's exhaust gas analyzer. This wonderful piece of equipment can be powered while the car is in motion by using a 12 Volt / 110 Volt power inverter that plugs into the receptacle for the cigar lighter, thus allowing the tuner to accurately analyze the fuel / air mixture of the engine at different loadings and engine speeds. By establishing the carbon dioxide content of the exhaust gases, the fuel / air ratio can be readily determined with the following chart:

% Carbon Dioxide	**Fuel / Air Ratio**		**% Carbon Dioxide**	**Fuel / Air Ratio**		**% Carbon Dioxide**	**Fuel / Air Ratio**
0.1%	14.72 / 1		3.5%	13.20 / 1		6.9%	11.93 / 1

0.2%	14.54 / 1		3.6%	13.15 / 1		7.0%	11.89 / 1
0.3%	14.42 / 1		3.7%	13.12 / 1		7.1%	11.86 / 1
0.4%	14.34 / 1		3.8%	13.08 / 1		7.2%	11.82 / 1
0.5%	14.28 / 1		3.9%	13.03 / 1		7.3%	11.79 / 1
0.6%	14.23 / 1		4.0%	13.00 / 1		7.4%	11.76 / 1
0.7%	14.21 / 1		4.1%	12.96 / 1		7.5%	11.72 / 1
0.8%	14.17 / 1		4.2%	12.93 / 1		7.6%	11.69 / 1
0.9%	14.15 / 1		4.3%	12.90 / 1		7.7%	11.65 / 1
1.0%	14.11 / 1		4.4%	12.86 / 1		7.8%	11.61 / 1
1.1%	14.09 / 1		4.5%	12.83 / 1		7.9%	11.58 / 1
1.2%	14.04 / 1		4.6%	12.80 / 1		8.0%	11.54 / 1
1.3%	14.01 / 1		4.7%	12.75 / 1		8.1%	11.50 / 1
1.4%	13.98 / 1		4.8%	12.70 / 1		8.2%	11.46 / 1
1.5%	13.94 / 1		4.9%	12.66 / 1		8.3%	11.43 / 1
1.6%	13.89 / 1		5.0%	12.64 / 1		8.4%	11.40 / 1

1.7%	13.86 / 1		5.1%	12.59 / 1		8.5%	11.36 / 1
1.8%	13.82 / 1		5.2%	12.54 / 1		8.6%	11.32 / 1
1.9%	13.80 / 1		5.3%	12.51 / 1		8.7%	11.28 / 1
2.0%	13.77 / 1		5.4%	12.46 / 1		8.8%	11.25 / 1
2.1%	13.73 / 1		5.5%	12.43 / 1		8.9%	11.21 / 1
2.2%	13.69 / 1		5.6%	12.40 / 1		9.0%	11.16 / 1
2.3%	13.63 / 1		5.7%	12.37 / 1		9.1%	11.12 / 1
2.4%	13.59 / 1		5.8%	12.30 / 1		9.2%	11.08 / 1
2.5%	13.56 / 1		5.9%	12.25 / 1		9.3%	11.05 / 1
2.6%	13.54 / 1		6.0%	12.25 / 1		9.4%	11.01 / 1
2.7%	13.49 / 1		6.1%	12.22 / 1		9.5%	10.97 / 1
2.8%	13.45 / 1		6.2%	12.18 / 1		9.6%	10.94 / 1
2.9%	13.41 / 1		6.3%	12.13 / 1		9.7%	10.90 / 1
3.0%	13.38 / 1		6.4%	12.10 / 1		9.8%	10.86 / 1
3.1%	13.34 / 1		6.5%	12.09 / 1		9.9%	10.82 / 1

3.2%	13.31 / 1		6.6%	12.03 / 1		10.0%	10.79 / 1
3.3%	13.27 / 1		6.7%	12.00 / 1			
3.4%	13.24 / 1		6.8%	11.96/ 1			

However, if you do not have access to such high-technology tools, you can try an `Ol-Timey-Mechanics seat-of-the-pants method. With the carburetors correctly set for mixture at idle conditions, and with the coolant and the engine oil at normal operating temperatures, perform a number of acceleration tests and part-throttle tests. First, accelerate from about 20 MPH in top gear to about 50 MPH. If the engine displays hesitation or a loss of power, then repeat the test with the choke pulled out approximately 5/8". If you experience an improvement of performance when the engine speed reaches the point at which the hesitation previously occurred, then make a note that the third, fourth, and fifth, as well as possibly the sixth metering stage of the fuel-metering needles need to be a bit smaller in diameter in order to enrichen the fuel /air mixture. Next, drive at a speed of 30, 40, and 50 MPH. If you experience a slight surging of power, then try to cure this by enrichening the mixture control slightly. If you suspect a lean running condition at these running speeds, then the diameters of the third, fourth, and fifth fuel-metering stages of the fuel-metering needles will each need to be decreased in order to enrichen the fuel / air mixture.

Reducing the diameters of the fuel-metering needles is a straightforward affair. With the damper rods removed from their respective vacuum chambers (dashpots), open the throttle very gradually until the butterflies are half-open (fifth or sixth metering stage on 1½" SUs, seventh or eighth metering stage on 1¾" SUs). Note whether or not the engine hesitates while doing so, as well as at which metering station of the fuel-metering needles the hesitation is produced. If it does hesitate, using a drill that is clamped in a vise with its chuck spinning at 200-400 RPM in order to securely hold the fuel-metering needle and #200 Wet and Dry sandpaper to remove material from the chosen stage of the fuel-metering needle, and then #600 Wet and Dry sandpaper for the final polish, polish the appropriate metering stage of the fuel-metering needles which produces the hesitation to a .0005" smaller diameter in order to richen the fuel / air mixture slightly. Bear in mind that the fuel-metering needles are almost always made of brass and thus are very easy to remove material from. Repeat the polishing and testing procedure until the engine no longer has the slightest hesitation when the throttle is very gradually opened.

At this point, you will be ready to do the final sizing of the diameters of the fuel-metering stages of the fuel-metering needle profiles. Taking a half of a second to open the throttle butterflies to the half-throttle position, again note any hesitation. Polish the appropriate metering station of the fuel-metering needles at which the hesitation occurs to a .0005” smaller diameter in order to richen the fuel / air mixture slightly. Repeat the polishing and testing procedure until the engine no longer has the slightest hesitation when the throttle is very rapidly opened. Using the exhaust gas analyzer, aim for a CO^2 rating of 6.7% at each fuel-metering needle metering station. At that point, you will have a correct 12:1 fuel / air mixture. Having accomplished this, you will have noticed that there are more metering stations further down the fuel-metering needle that are never exposed to airflow. These are present in order to effectively channel fuel up the fuel jet. Be sure to incrementally size these remaining metering stations .001” to .002” smaller as they progress toward the tip of the fuel-metering needle.

At this point, you need only to refill the damper tubes of the pistons with oil, reinstall the damper rods, and then you are done!

SU Fuel Jets

Be aware that the fuel jets for the SU HS4 Series carburetors have different feed orientations. If the tubular feed nipple of the fuel jet is pointed to the right side (at about a 2 o’clock position), then it is the fuel jet for the rear carburetor. If the tubular feed nipple of the fuel jet is pointed to the left side (at about a 10 o’clock position), then it is the fuel jet for the front carburetor. Just to make assembly easier, SU was considerate enough to make one in the color white and the other in the color black. In the case of the SU HIF4 Series carburetor, if the bottom cover will not go on, then you have installed the wrong jet. It is just that simple.

In order for the fixed fuel-metering needles that are common to the SU HS Series carburetors to perform their function properly without causing binding of the vacuum pistons or undue wear of the fuel jets, they must be concentrically centered in their fuel jets. Adjust both of the fuel jets and their adjusting nuts to their highest positions, and then place the vacuum pistons with their attached fuel-metering needles and their vacuum chambers (dashpots) onto the carburetor bodies. Lift the vacuum piston of the rear carburetor with its vacuum piston lifting pin on the underside of the vacuum chamber (dashpot) mounting flange and listen for a distinct soft metallic click as the vacuum piston falls onto the fuel jet bridge, and then repeat the process with the front carburetor. If this sound is absent with the fuel jet at its maximum adjustable height, yet is audible with the fuel jet at its lowest adjustable height, the fuel jet bearing and the fuel jet are not concentric with the fuel-metering needle and, this being the case, adjustment must be made. This concentricity is most easily achieved by use of a centering tool that is available from Burlen Fuel Systems. However, if you have SU HS Series carburetors and choose not to purchase this timesaving tool (for reasons that will shortly become obvious, I strongly advise that you do) there is an alternate means of centering the fuel-metering needle and the fuel jet.

On each carburetor, disconnect the interconnecting lever from the head of the fuel jet, and then unscrew the union that holds the fuel feed line of the jet onto the base of the float bowl. Withdraw both the fuel jet and its fuel feed line as a single unit. Next, unscrew and remove the fuel jet adjusting nut, and then remove the lock spring above it. Replace the fuel jet adjusting nut and rotate it to its highest attainable level, and then refit the fuel jet with its fuel feed line. Slacken off the fuel jet adjusting nut until the fuel jet bearing is just free to rotate under finger pressure. Remove the vacuum piston damper rod from the vacuum chamber (dashpot) and gently press the vacuum piston down onto the fuel jet bridge. Tighten the fuel jet locking nut while ensuring that the fuel jet head is properly oriented towards the float bowl. At this point, lift the vacuum piston with the vacuum piston lifting pin and again listen for a distinct soft metallic click as the vacuum piston strikes the fuel jet bridge. If you do not hear this, you must continue repeating the centering procedure until you do or until you break down and purchase the centering tool. Once the fuel-metering needle and the fuel jet are concentric, reinstall the fuel jet operating lever and the fuel jet adjusting nut lock spring. This simple spring is critical to maintaining a chosen adjustment as well as for preventing vibration-induced wear of the threads by means of preloading them.

The centering tool centers the jet so that the fuel-metering needle will enter into it in a concentric manner without dragging on the bore of the jet into a much more simple and straightforward affair. Remove the locking screw in the vacuum piston, and then withdraw the fuel-metering needle. If it cannot be easily withdrawn, then gently tap it into the vacuum piston in order to loosen it, and then pull it out. In doing so, under no circumstances should you attempt to use any force. A bent fuel-metering needle is worthless. Install the centering tool in its place. This is a small round button with a thin shaft on its top and bottom. Assemble the vacuum piston in its normal position, and the centering button will provide a guide for centering the jet. With the jet screw loose, press the vacuum piston down while pressing the jet upwards in order to engage the jet onto the guide pin. Next, with the jet and the guide pin are engaged, tighten the jet screw at the bottom in order to secure the jet bearing assembly in the centered position. Finally, remove the jet centering tool and replace the fuel-metering needle with its shoulder flush with the bottom of the vacuum piston.

Before reinstalling the vacuum chambers (dashpots) with their vacuum pistons, check to be sure that the fuel jets of both carburetors are set at the same height. This is best measured with the depth tail of a vernier caliper. The fuel jets of 1½” SUs should be set .040” below the fuel jet bridge, while those of the 1¾” SUs should be set .050” below the fuel jet bridge. If you do not have a vernier caliper available, adjust the fuel jets so that they are both flush with their fuel jet bridges inside of the bores, and then screw them downwards two complete turns. This setting will probably not be correct, but it should be close enough for the engine to start and idle.

Check to be sure that the plastic plugs are present on the bottom face of both of the vacuum pistons. This plug is incorporated into the design in order to act as a spacer when the vacuum piston is at the bottom of its travel against the fuel jet bridge so that the flow of the fuel / air charge will always be present in order to permit the engine to idle. The gap between the fuel jet bridge and the bottom face of the vacuum piston should always measure between .015” and .018”.

Now, reassemble the carburetors. Make sure that you are putting things back in their original matched sets. When installing the pin on the extension arm of the interconnection clamp into the lost motion lever, be sure that there is a clearance of .120” between the top of the pin and the square notch of the lost motion lever. Do not forget to refill the damper tubes of the vacuum pistons. On dust-proofed carburetors that have a transverse hole drilled into the neck of the vacuum chamber (dashpot), refill the damper tube to ½" (13mm) below the damper tubes of the vacuum pistons with either 30W or 20W/50 engine oil. On non-dustproofed carburetors which instead use a damper rod that has a hole in its damper cap, refill the damper tube to ½" (13mm) above the damper tubes of the vacuum pistons with either 30W or 20W/50 engine oil. Why fill them higher than the damper tubes? If you invert the vacuum chamber (dashpot) and examine it carefully, you will notice a bushing inside of its neck. The damper tube of the vacuum piston is a precision fit inside of this bushing. Because of the limited airflow capacity of the vent hole in the damper cap of the damper rod, a small amount of the oil that is above the damper tube will be pumped down the sides of the damper tube by the air compressing above it slightly when the vacuum piston rises, providing essential lubrication to the bushing inside of the vacuum chamber (dashpot). The maintenance of this fine-tolerance interface is critical to providing the appropriate amount of vacuum so that the vacuum piston will rise to a position in which its attached fuel-metering needle can meter fuel to the correct fuel / air ratio. Should the bushing become badly worn from a lack of lubrication, the vacuum chamber (dashpot) will quickly duct the oil above the damper tube into the intake manifold. In addition, air leaking past the bushing into the vacuum chamber (dashpot) will decrease the pressure differential and thus causing the vacuum piston to rise less than it should. An otherwise unexplainable lean running condition will result. If you find yourself having to constantly refill the damper tube to its correct height, then you either have excessive wear at this interface, or the plug in the bottom of the damper tube is either leaking or absent.

Adjustment of the SU Carburetors

To the uninitiated, adjusting a pair of SU carburetors may seem an imposing task best left to the attentions of a highly trained technician. However, owing to their simplicity of design, it is actually a fairly simple procedure once you know how. Remember that you are working with dual carburetors, so any adjustment that you perform that effects one carburetor must likewise be done on the other carburetor. In order to get satisfactory results, be sure that both the valves and the ignition timing are properly adjusted before you proceed.

It should be noted that most SU HIF carburetors have slotted screws with hexagonally shaped heads, and a lock nut on each that secures the screw in position. Many mechanics find it cumbersome to have to loosen the lock nut in order to turn the screw, and then tighten the lock nut after tuning. As an alternative, put springs under the screws instead of the lock nuts in order to secure them. This will enable you to turn the screw whenever needed without having to tighten down the little nut afterwards, making it much easier to adjust the initial idle and fast-idle settings. The fast idle and choke on each carburetor are adjusted with the same screw, the normal idle with another. Their tops should be slotted as well as hexagonally shaped, but if they lack slots, it is a simple matter to hacksaw a shallow slot into them, enabling you to make easy adjustments with a screwdriver.

Disconnect the throttle cable from its forked actuating arm, and then loosen the actuating arms on the transverse shaft that interconnects the throttle shafts of the carburetors. Disconnect the choke cable and loosen the two pinch bolts in order to free the choke actuating lever. Having accomplished this, check to see that both throttle discs are seating against the carburetor bodies simultaneously. Now, close both throttle disks completely by unscrewing their throttle adjusting screws, and then open each of them one full turn of each adjusting screw. Start the engine and allow it to reach normal operating temperature, then use a vacuum gauge to determine when the throttle disks are synchronized. This is easily accomplished by adjusting the throttle adjusting screws. When the airflow of the carburetors is matched at an engine speed of 1,500 RPM, they are synchronized. Retighten the clamps that secure the interconnection shaft to the throttle shafts of the carburetors so that there is slight play in the linkage and cable before movement of the cable starts to open the butterflies. Ensure that there is 1/32” of endplay (endfloat) on the transverse interconnection rod. Now recheck the airflow balance just off idle, since it is more important to get correct balance off idle than it is at idle, for obvious reasons.

Using a Gunson’s Exhaust Gas Analyzer to measure CO^2 output, the idle fuel / air mixture can now be adjusted. On HS Series carburetors, this is accomplished by turning the adjuster nut at the bottom of the fuel jet one flat at a time, upward to lean out the fuel / air ratio, and downward to richen it. On the SU HIF Series carburetors, this is accomplished by rotating the idle fuel / air mixture adjuster screws on the bodies of the carburetors a quarter-turn at a time. On both SU HS and SU HIF carburetors initial testing of the fuel / air mixture strength can be ascertained by lifting the vacuum piston of the rear carburetor by means of the vacuum piston lifting pin. If the engine speed increases only slightly, then the fuel / air mixture is correct. If the engine speed increases, then the fuel / air mixture is too rich, so on SU HS Series carburetors you will need to rotate the adjuster nuts of the fuel jets of both carburetors upwards in order to lean it out. On the SU HIF Series carburetors you will need to unscrew the adjuster screw outwards in order to lean out the fuel / air mixture. If the engine speed decreases, then the fuel / air mixture is too lean (weak), so you will need to rotate the adjuster nuts of the fuel jets of both carburetors downwards in order to enrichen or to lean (weaken) it. Be aware that the difference on SU HIF carburetors is almost subliminal, whereas with SU HS carburetors it is more marked. If you adjust the fuel / air mixture in one direction just enough to barely detect a weak fuel / air mixture, and just enough the other direction to barely detect a rich fuel / air mixture, then the correct position is halfway between the two. With practice, you should be able to get these two points closer and closer together. Once this balance is achieved, the front carburetor can then be adjusted using the same technique. However, the rear carburetor should be rechecked, as the functioning of the carburetors are interrelated. Once the two seem to be functioning correctly, noting the readout of the exhaust gas analyzer, aim for a CO^2 output of between 2% and 3.5% at the engine's idling speed at normal operating temperature. Once this has been attained, remove both of the vacuum chambers (dashpots) along with their vacuum pistons then remeasure the heights of the fuel jets. They should be within .003" +/- .001" of each other. If this should prove to not be the case, then check for either a worn fuel-metering needle, a worn fuel jet, or leakage caused by either a worn throttle shaft or worn shaft bushings in the body of the carburetor. On SU HIF Series carburetors, there is the additional possibility of worn throttle shaft seals. For the best snap throttle response, adjust the mixture to produce a CO^2 level of 5% at both steady and on trailing throttle. This will make the engine very sharp on throttle changes.

If you do not have an exhaust gas analyzer, a reasonable, though not nearly as accurate, method is available. Start the engine and allow it to run until it reaches operating temperature. Check the strength of the fuel / air mixture by using the vacuum piston lifting pin on the underside of the mounting flange of the rear vacuum chamber (dashpot) in order to lift the vacuum piston, and then make the appropriate fuel / air mixture strength adjustments. As these adjustments are made, the idling speed may be effected. Simply reset the idling speed by means of small changes to the setting of the throttle adjusting screws, and then check the strength of the fuel / air mixture again by using the vacuum piston lifting pins. If necessary, perform minor adjustments to the fuel / air mixture on SU HS Series carburetors by fine-tuning the adjuster nuts of the fuel jets.

On SU HS Series carburetors, hold the adjuster nut of the fuel jet on each carburetor static, and then rotate the fuel jet adjustment restrictor nut until its attached vertical tag contacts the body of the carburetor on the left side. Bend the small end on the top of the tag of the adjustment restrictor until it locks with the flat of the fuel jet adjusting nut and follows its movement. This will prevent the choke (fuel / air mixture control) mechanism from over-richening the fuel / air mixture during cold starting. Once this has been accomplished, it is wise to paint both the lower tag of the fuel jet restrictor and the one adjacent flat on the fuel jet adjusting nut with colored nail polish for future reference.

In order to set the fast idle for cold starting of the engine, loosen the clamping bolts on "W" couplings for interconnecting of the fast idle interconnection shaft, then unscrew the fast idle adjusting screws until the fast idle cam plates rest against their stops. Check to be sure that there is 1/16" of free movement in the choke cable before the choke cable begins to pivot the cam plates. Next, pull out the choke knob on the dashboard until the arrow marked on the cam plate of each carburetor is directly underneath its fast idle adjusting screw. Using the vacuum gauge in order to verify proper synchronization of the carburetor throttle disks, adjust both of the fast idle adjusting screws to set the fast idle speed to its desired level. Next, on the SU HS Series carburetor, tighten the bolts on the "W" couplings of the fast idle interconnection shaft so that when the cable is pulled to the halfway position it is perpendicular to the operating lever on the shaft. Note that the throttle return spring clips, the "W" couplings for interconnecting throttle shafts, as well as the levers of the interconnecting shaft, are all anchored to their respective shafts by means of a pinch bolt. The clips, couplings, and levers all have a recess in order to both retain the head of the pinch bolt and thus prevent it from turning. For this reason, always install the bolt head into the recess with the nut and plain washer on the smooth other side of the fitting. When orienting the fittings on their respective shafts, always have the nut facing upward so that it is readily accessible for making synchronization adjustments on the carburetors.

Too much angularity one way or the other can make the choke control knob hard to pull. The manual choke on the SU HS Series carburetors consists of a cable that pulls a choke lever against the resistance of a jet return spring, as well as moving a fast idle rod against the resistance of a throttle return spring. Friction in the cable and in the choke mechanism, as well as in the fast idle mechanism, plus the added resistance of the two return springs can turn precise setting of the choke into a difficult task. However, if you open the throttle with your foot before attempting to engage the choke (both HIF and HS4). By lifting the fast idle screw off of the cam, much of the drag that is felt at the cable is eliminated. If the choke levers and jets of the HS4 are properly aligned, they should have a fairly smooth action, just a little stiffer than the HIF. You will also eliminate the resistance of the throttle return springs, thus making the task much easier. As before, there should be slight play in the cable so the knob has to be pulled slightly outward before it start to move the cams on their carburetors. Note that on SU HIF carburetors there is an arrow stamped onto the cam. This should be under the screw when you set the fast idle speed to 1,100 / 1,200 RPM. Again, set the fast idle screws such that when in fast-idle both carburetors draw in the same amount of air.

Occasionally some people despair of the fact that they have followed all of the proper procedures for synchronizing their dual SU carburetors, but still have synchronization problems. This is most commonly caused by an air leak interfering with the performance of the system. Fortunately, the source of a leak is normally easy to locate. Simply spray some carburetor cleaner onto each of the bushings in which the carburetor shafts ride and note in each case if there is a change in the running of the engine. If this does not produce results, spray the carburetor cleaner on each gasket where the carburetors mate with the phenolic spacers, then where the phenolic spacers mate with the intake manifold, and then finally where the intake manifold mates with the cylinder head. As before, note in each case if there is a change in the running of the engine. If this does produce a change in the running of the engine, then you have found the source of the air leak.

If this procedure does not produce satisfactory results, then the problem is usually caused by an unequal rise of the vacuum pistons, interfering with the balanced flow of a proper fuel / air charge. Fortunately, this defect is rather easy to diagnose. Remove the damper rod from atop the vacuum chamber (dashpot) and clean the threads on both the damper cap and in the neck of the vacuum chamber (dashpot) in order to ensure a proper airtight seal. If there is air leakage, then the carburetors cannot function properly. Next, remove the three screws securing the vacuum chamber (dashpot) to the top of the front carburetor body and make sure that the aluminum tag with its carburetor specification number does not become lost. If you are removing the vacuum chamber (dashpot) of an SU HIF carburetor, you will need to remove the circlip, otherwise you will not be able to remove the vacuum chamber (dashpot) from the body of the carburetor. Carefully withdraw both the vacuum piston and its spring from the vacuum chamber (dashpot). Examine the vacuum piston assembly for damage to the damper rod as well as to the outside surface of the vacuum piston. The vacuum piston assembly must be scrupulously cleaned only with a good solvent, such as a quality carburetor cleaner. Do the same with the interior of the vacuum chamber (dashpot). Never use any abrasive cleaners on either of these components as they are precision-mated to each other, and each vacuum piston and its vacuum chamber (dashpot) are a factory-matched pair. This being the case, never mix the parts from one pair to another. Make a note of the identification numbers of the fuel-metering needles (they have to be identical), and then check to confirm that each fuel-metering needle is mounted with its shoulder flush with the bottom of the vacuum piston. Be sure that everything is set aside as matched sets.

Once the vacuum chambers (dashpots) and their vacuum pistons are clean, they should be checked for proper fit. Temporarily plug the transfer holes in the bottom of the vacuum piston. Lightly oil the exterior of the piston rod. Next, invert the vacuum chamber (dashpot) in your hand and check the keyway of the vacuum piston for security within the carburetor body, and then install the vacuum piston. Install one of the screws into the bottom of each of the vacuum chambers (dashpots) with a large washer to prevent the vacuum pistons from falling out. Next, screw the caps with their damper rods and sealing washers into the vacuum chambers (dashpots). Turn the vacuum chambers (dashpots) upside down and allow the vacuum piston to settle, then turn it right side up and measure how long it takes for the vacuum piston to descend until it touches the washer. The pistons of both the 1½" SU HS4 Series carburetors and the 1½" SU HIF4 Series carburetors should take between four and six seconds to reach the bottom of their travel, while the pistons of 1¾" SU HS6 Series and 1¾" SU HIF6 Series carburetors should take between five and seven seconds to reach the bottom of their travel. If these times are exceeded, then both the vacuum chamber (dashpot) and its vacuum piston must be replaced with new ones.

When you reassemble the vacuum chambers (dashpots) onto the carburetor bodies, be sure to smear some antisieze compound onto the threads of the screws. This will prevent the corrosion of the aluminum alloy threads that results from the electrolytic interaction between the steel screws and the aluminum alloy, causing them to corrode, and then to seize in place, as well as making the screws much easier to remove the next time that you want to clean the carburetors.

Refit the jet bearing, washer, and locking nut, but do not tighten the locking nut at this time. Refit the jet into its bearing and the fuel feed line (flexible pipe) to the base of the float bowl without either the gland or the washer. Now you will need to centralize the jet.

In order to centralize the jet, remove the jet and the tube, and then refit the spring and the jet adjustment nut. Next, fit the gland washer and the ferrule to the flexible tube. The end of the tube should project a minimum of 3/16" (4.8 mm) beyond the gland. Refit both the jet and the tube. Tighten the sleeve nut until the neoprene gland is compressed. Be aware that overtightening can result in fuel leakage. Refit the damper rod and its sealing washer.

Reassemble the pick-up lever, cam lever, cam lever spring, skid washer, and pivot bolt tube (or tubes) into their original positions. Place the return spring of the pick-up lever into position over its mounting boss and then install the pivot bolt in order to secure the pick-up lever assembly to the carburetor body. Ensure that the double-coil spring washer or spacer fits over the projecting end of the pivot bolt tube. Register the angled end of the return spring in the groove of the pick-up lever, and then hook the other end of the return spring around the molded peg on the carburetor body. Fit the Brass ferrule to the hole in the end of the pick-up link. Relieve the tension of the return spring, and then use the screw to fit the link to the jet. Refit the baffle plate to the nozzle on the lid of the float chamber.

Remove the piston damper and then apply pressure to the top of the piston rod with a pencil. Tighten the jet locking nut, keeping the jet firmly up against jet bearing. Refit the jet locking spring and the adjustment nut. Before reattaching the fuel feed line (pipe) to the lid of the float bowl, fit the rubber sealing washer over the end of the plastic pipe so that at least 3/16" (4.8 mm) of the pipe protrudes. Now, reattach the control cables. Refill the piston dampers with the recommended engine oil.

An Old Myth

At this point, I would like to debunk an old myth about the Original Equipment B Series engine. When equipped with an intake manifold with runners that give the incoming fuel / air mixture a straight shot at the intake ports (such as in the case of the runners of the Original Equipment SU intake manifold), the inner cylinders do not run richer than the outer cylinders. In reality, the exiting pressure waves in the siamesed intake port that result from the 180° throw difference of the crankshaft have a definite influence on the state of fuel / air mixture separation and fuel condensation in the arriving fuel / air charge in the siamesed port, and this is what creates the impression that the inner cylinders run rich. The supposed "rich fuel / air mixture" in the inner cylinders is in reality the consequence of the problem of interplay between the condensed (and thus reduced) atomization of the gasoline and the resulting stuttering flame propagation that are caused by the return pressure wave inside of the runner of the intake manifold. The color striations in the carbon deposited in the combustion chambers that resemble sand ripples on a beach indicate interrupted flame propagation in cylinders #2 and #3, while the combustion chambers of cylinders #1 and #4 are much more evenly colored and grade out from the spark plug to the opposite wall of the combustion chamber. The solution to this problem lies in careful attention to the modification of both the contours of the port and the area around the throat of the port in its approach to the valve seat in order to achieve a corrected airflow.

Weber and Dellorto Carburetors

When it comes to the subject of carburetion, many people tend to opt for items that they perceive as being exotic, such as the Weber DCOE or the Dellorto DHLA. Interestingly, the Dellorto DHLA flows more fuel / air mixture for a given venturi size that the Weber DCOE does. In addition, the Dellorto DHLA atomizes fuel to a much finer degree than the Weber DCOE does, thus allowing lower engine speeds and consequent lower port velocities without the finely atomized fuel droplets dropping out of the stream of air. However, the finely atomized fuel droplets combust more quickly and completely, producing maximum pressure earlier in the combustion stroke. As a consequence, this forces the use of a slightly richer fuel / air mixture in order to slow the combustion process. The larger amount of finely atomized fuel thus takes up a greater volume in the induction tract than the larger, slower-burning droplets produced by the Weber DCOE, thus displacing air and making for an effectively smaller fuel / air charge, and thus less power output than that produced by the Weber DCOE. However, due to its production of a finely atomized fuel charge, combustion of the fuel becomes more efficient, creating less pollution and better fuel economy in proportion to the amount of fuel that is used. Conversely, the less-finely atomized fuel droplets produced by the Weber DCOE design require a more generous supply of oxygen in order to combust properly. This is the explanation for the Weber carburetor's reputation for being sensitive to changes in altitude. However, the slow burn characteristic resulting from these larger fuel droplets do result in maximum cylinder pressure being attained later in the combustion stroke, thus reducing the amount of heat that is lost through the roof of the combustion chamber, and hence enhancing power output at high engine speeds when the amount of time available for combustion is reduced. This is the secret to its marginal superiority in power output at high engine speeds to most other carburetors. All other factors being equal, if you switch from a properly set up Dellorto DHLA carburetor to a properly set up Weber DCOE carburetor, you should then easily get a 5% increase in power output across the entire powerband, with a similar proportion of loss of fuel economy.

Type 1, the first Dellorto DHLA (no letter suffix), boasts superior atomization and excellent tuning ability due to the extra research at Dellorto. This carburetor has an idle jet (which also runs the progression and adds mixture at cruising) that is not connected to the main jet assembly. Instead, the idle jet operates independently from the float bowl. In addition, the idle jet air feed for the emulsion tube is not fixed at a set size like later models. The main jets run on their own from the float bowl, much like the idle jets do. This gives two separate circuits that must be tuned for perfect overlap in order to attain good cruise and acceleration. This has the benefit of giving the ability to make the idle jet progression either long or short. You can activate the main jets early and cut back the progression on the idle jet air feed with different jet holders with different size air holes. A bigger hole will give more air, but less progression. As a result, you can tune this carburetor a million ways for million applications. However, as a downside, they are also extremely hard to tune without a massive box of jets or a dynometer, because they need to be precisely correct. This type of carburetor uses a .1 venturi, which has a bigger pilot hole than the .2 and .3 venturis. This means that a bigger vacuum signal is generated in the venturi tube, in the form of pulsation to the main jet assembly. Consequently, the .1 venturi can be tuned to activate the main jet assemblies immediately from very low engine speeds and are highly responsive to tuning. You have the option of delaying the activation of the main jet assemblies by adjusting the emulsion tubes and air correctors on the main jet assembly so that it will need more vacuum signal in order to activate the main jet assembly, or less vacuum signal when set up in the opposite manner.

Type 2, the Dellorto DHLA E, is the same as the earlier DHLA, having separate circuits, and uses either .1 or .2 venturis. They have extra progression holes for the idle jet so that the progression phase can be increased over an increased throttle plate angle in order to suit a variety of applications. These additional progression holes also benefit fuel economy. They are very smooth runners when set up with lengthy progression. The .2 venturi is the same as a .1 venturi, only with a tiny bit smaller bore size to the signal tube, so they have fitted .2 often in the E variant as the progression phase is thus extended. Using the .2 venturi means that they do not have to delay the activation of the main jet assemblies as much as when using the .1 venturi, thus making jetting easier and more responsive.

Type 3, the Dellorto DHLA F, G, H, N, R, and S models, are completely different to the DCOE science, but many make the mistake of trying to tune them like the earlier Type 1 and Type 2 models. These differ because they have more progression holes and use the idle jet to feed most of the cruise phase and low engine speed / low TP area of the engine whenever the main jet assembly is not in operation. This is accomplished completely automatically with no jetting needed. The idle jet has a very large fixed 2.2mm air feed. This phase of the carburetor cannot be tuned for length like the others, but herein lays the secrets: The idle jet does not feed from the float bowl. Instead, it feeds directly from the main jet assembly. What happens afterwards is what gives these Dellortos the sweetest road behavior that 25 years of research can produce: When the main jet assembly starts to emulsify fuel in the emulsion tube, the idle jet is feeding from it, so the gassy, airy fuel shuts down the idle jet and is drawn backwards through the idle circuit. This happens the moment that the main jet activates, so there is actually no need to tune the length of the progression and idle phase. By design they are automatically calibrated. Simply keep the idle jet above 59 up to 62 and do not make the mistake of fitting numbers suited to an earlier version of the DHLA design.

With this simple technique, you can tune anything without really doing anything. The emulsion tubes in these carburetors are always 8-10-11 and have to stay that way - which are really rich and have a hole straight down with loads of air holes that atomize the fuel to a tremendous degree. They also have to be used because the idle jet assembly will not function correctly while using the DHLA E-type emulsion tubes (1-6-7-5) of the DHLA Type 2. This is due to the fact that the idle jet needs these airy emulsion tubes in order to function and deactivate as designed. Usually people ram the earlier DHLA idle holders with air holes into these carburetors, add the 1-6-5-7 style emulsion tubes, only to end up wondering why the carburetor produces a very large lean spot off idle speed. This is because they have missed the point completely! These carburetors use a .3 venturi, which has a very small signal tube to the main jet assembly. This is because the emulsion tubes are ready to engage from an engine speed of about 1,250 RPM, so the emulsion tubes need inhibiting with a vacuum-signal-killing venturi. These variants are wicked if you want bolt-on power, since they essentially tune themselves!

The early variants are better in respect of punch and tuneability on odd applications but do not really do anything that the later variants will. The later variants are a bit more suited to engines with mild or standard camshaft lobe profiles, and they operate at their best if you are using 30mm-33mm venturis. They hate race engines, mad camshaft lobe profiles, and giant venturis because they are designed mainly for hot production engines with a clear pulse strength to suit the retarded venturi and tubes. This characteristic is of great advantage where silky town driving is paramount and you use the car for commuting or just for Sunday driving. They also give superior fuel economy to the earlier types.

So, while the DCOE is much the same as the DHLA, the DHLA E, F, G, H, N, L, R, and S variants are all further progressive evolutions of a principle. Provided that the best qualities of the carburetor are matched to the application, you have the ability to cater for your tastes and requirements when using a Dellorto DHLA.

Proper carburetor synchronization and idle adjustment of a Dellorto DHLA carburetor requires an exhaust Gas CO Meter, a Tachometer, and a four-column Mercury Manometer. Proceed as follows: Disconnect the throttle control rod and the throttle control lever. Unscrew the idle speed screw until it is out of contact with the lever extension. Next, unscrew the screw of the balance lever until the throttle discs of both front and rear throats are fully closed. Check to be sure that they have light upward pressure on throttle control lever. While maintaining pressure on throttle control lever, tighten the screw of the balance lever until it contacts the tongue of throttle control lever, thus fully closing both of the throttle discs. Tighten the idle speed screw an additional turn after it first contacts the lever extension of the throttle control lever. Remove the anti-tamper seals from the mixture screw housing, and then unscrew each screw five turns from the fully-closed position.

Check that the idle air bypass screws, to which anti-tamper paint has been originally applied, are completely closed. Remove the depression blanking plugs and, using the screw-in adaptors, connect each throat to the four-column mercury monometer. First ensure that any airlocks are removed from the mercury columns which would otherwise result in inaccurate readings. Reconnect the throttle operating rod to the throttle control lever. Start the engine and leave it running to attain its normal operating temperature. Using the balance screw, align the lower mercury column of the front carburetor with that of the lower one of the rear carburetor.

At this point, if necessary, adjust the levels of the other two cylinders to match those of the lowest pair using the bypass screws on these throats. With the # 1 cylinder blow-by pipe temporarily clamped shut, all readings should be brought to the same level. Unclamped for normal running, the mercury level of # 1 cylinder will always be lower than those of the other three. At this time, insert the CO tester probe and connect the tachometer.

Preferably only by using the special Dellorto screwdriver, adjust the running of each cylinder by turning the mixture screw in order to get the most even running with the engine idle speed at the correct level and holding the exhaust CO level well below 4,5%. Remember that by unscrewing these screws, you will increase the CO level and vice versa. The mixture screw of one throat will need to be opened more than the other due to the effect of the blow-by hole on that throat. Remove the vacuum adaptor plugs and refit the blanking plugs in their places. Fit new anti-tamper seals on the mixture screw housings. If all the above-mentioned equipment is not available, it is still possible to obtain the correct idle setting by simply resetting the idle speed screw at the balance adjusting screw.

When starting, fuel at the union passes through the filter and reaches the seat where the needle, which is attached to the float, controls the flow of fuel into the float chamber, thereby maintaining a constant level. The float chamber is vented to the atmosphere through its vent in the float chamber. Upon opening the choke valve, fuel metered through the starter jet passes into the emulsion tube where it is mixed with air from the channel. This mixture then enters the passage, further mixing with air from the vent, and then reaches the valve chamber. From there it is distributed via the two ducts which lead into the main venturis that are located downstream of the throttle discs. Upon closing the choke valve, communication between the main venturis and the starting circuit is broken, as well as communication between the two venturis due to the sealing action of the split bushing.

While idling, fuel from the float chamber is metered through the idle jets and mixes with air from the float bowl through the channels. The mixture through the channels reaches the idle mixture screws and, once regulated by them, reaches the main venturis downstream of the throttle discs.

Upon first opening the throttle discs, that is, when progressing from idle speed to full throttle, the fuel/air mixture also reaches the two ports through the progression holes.

Always adjust the idle speed with the engine at its normal operating temperature, screwing in the idle speed adjustment screw to obtain a slightly higher idle speed than normal. Next, adjust the mixture adjusting screws until you find the most even running. Remember that unscrewing the mixture adjustment screws will result in a richer mixture, and vice versa. Once you have obtained the most even running, steadily unscrew the idle speed adjustment screw until the normal idle speed is obtained.

Adjust the injection quantity of the accelerator pump by fitting the carburetor to the special support with the proper gasket, and then connect the carburetor to a fuel reservoir so that it is continuously supplied with fuel. In order to collect all the fuel pumped out, put the two graduated measuring tubes, each having a capacity of 10 cc, under the drain pipes on the support. Open and close the throttle completely 20 times, with a few seconds between each operation of the throttle, and check that the amount of fuel collected in the tubes corresponds with the correct specification and is the same for both venturis. It not, adjust the fuel delivery of the pump by resetting the nut and locknut fitted on the pump operating rod. Remember that screwing the nuts upward increases fuel delivery and vice versa. Should there be any difference in volume between the two venturis, remove the pump jets and blow through them vigorously in order to correct this. Recheck until the correct setting is obtained, and then ensure that both the nut and its locknut are retightened.

Check that the float, which has the actual weight marked upon it, is undamaged and that it is also tree to rotate on its pivot pin. Hold the float chamber cover vertically so that the float arm is in light contact with the needle and with the spring in the needle uncompressed. In this position, check that both half-floats are at the correct distance measured from the float chamber cover to the top cover gasket that is fitted to it.

In order to keep the carburetor in good condition, especially after operating faults have occurred, proceed as follows: Dismantle the carburetor, wash the components in fuel and blow dry. Special care is needed with the jets, emulsion tubes, needle valve seat, fuel filter and all of the drillings in the carburetor body. Check the condition of all the components before reassembly and replace them wherever necessary, using only new parts. When reassembling the carburetor, always renew all of the gaskets and the O-rings.

When it comes to the process of setting them up, there is a considerable difference between the Weber and the Dellorto on one hand, and the SU on the other. The SU has only one fuel-metering needle and one fuel jet, so you can modify its metering in your driveway. Both the Weber and the Dellorto, conversely, has a multiple choice of replaceable main and auxiliary venturi sizes, six jets (starter air correction jet, starter jet, idle jet, main jet, main air correction jet, and accelerator pump jet), plus two emulsifier tubes and bleed valves! As Peter Burgess rightly points out in his book, carburetors are rarely properly set up as delivered (but people rip a Weber out of its package and slap it on their engines in sheer ignorance of this fact). This multiplicity of jets and venturi sizes does, however, make them almost infinitely adaptable, even to practically any exotic camshaft lobe profile, and this is another reason why the factory racing team elected to use the Weber carburetor. They could more easily tailor the engine's performance characteristics to the type of track that they were about to race on. The stronger part-throttle performance of the variable venturi inherent to the SU design meant little in the realm of racing where full-throttle power was what won races. However, unless you are using a radical camshaft, have access to a dynamometer, and you really understand how a carburetor works, take my advice and use the 1½" SU! Its 130 Cubic Feet per Minute airflow capacity is quite adequate for the majority of streetable small-bore B Series engines, plus it can readily be modified in order to increase its airflow capacity by approximately 30% to 169 Cubic Feet per Minute should the need ever arise.

Weber DCOE carburetors require a fuel pump that can provide a high volume of fuel at a low pressure. The fuel pressure should be regulated to no higher than 3 PSI at low engine speeds and between 1.5 and 2.5 PSI at high engine speeds. Too high a fuel pressure will force fuel past the float valve, causing a rich fuel / air mixture and erratic running.

Be warned that under no circumstances should the Weber DCOE carburetor be bolted solid to intake manifold. Should you decide to use the Weber DCOE carburetor, you would be well advised to use a Soft Mount kit to protect it from the disruptive effects of harmonic vibration (APT Part # SMW-45). At the point when the engine creates its greatest levels of harmonic vibration, fuel can froth in the float bowl, causing the fuel / air mixture to run lean, imperiling your valves and piston crowns. This vibration-induced lean running condition can cause amateur tuners considerable confusion and frustration in setting up the carburetor in their quest for the right combination of jets and settings. When installed, the rubber O-rings incorporated into the Misab plates of the soft mount kit should be compressed between the intake manifold and the carburetor by multicoil Thackerey washers and aircraft-type Nyloc nuts only to the point of providing an airtight seal. Be sure that an O-ring does not slip when you are mounting them, as that would guarantee an air leak. Both the carburetor and its intake manifold should then be held in place by the simple system of a strut, an anchor plate, and an anchor nut that braces it to the side of the engine block. The resulting trapezoidal mounting system was reliable enough for the factory racing tem to adopt it. Be warned that Weber DCOEs prefer to be mounted with a 5° upward angle and should never be mounted at a greater angle that 7° above horizontal. They will not perform properly at a greater angle.

It should be noted that there are two variants the Weber DCOE available that can be used on a streetable B Series engine, the DCOE 40 and the DCOE 45. The maximum airflow capacity for the DCOE 40 is 175 Cubic Feet per Minute while that of the DCOE 45 is 222 Cubic Feet per Minute. The DCOE 40 has a higher airflow capacity than the DCOE 45 when employed with venturis between 24mm and 32mm, while the DCOE 45 has a higher airflow capacity when employed with venturis between 34mm and 40mm. In order for the best ratio of airflow to main fuel jet signal strength to be calibrated accurately, the ratio of the venturi size / airflow capacity needs to be closely matched.

Unfortunately, the Weber's intake manifold for the BMC B Series engine imposes a major drawback: In order to facilitate mounting within the confines of the engine compartment an airfilter that has adequate airflow capabilities, its 95 mm length is short. This shortness forces the use of a very curvaceous path between the carburetor and the intake ports, which in turn causes the fuel in the fuel / air charge to be biased towards the ports for the outer cylinders (#1 and #4) as a result of its own inertia. The consequence is that the outer cylinders (#1 and #4) tend to run richer while the inner cylinders (#2 and # 3) tend to run leaner, the differential between the two increasing with engine speed due to the increasingly greater inertia of the fuel. The Weber 130 mm swan-necked intake manifold, or the similar one offered by Oselli, will reduce this tendency while being more appropriate to camshafts whose designs are oriented toward producing more low-RPM and midrange power at the expense of power output at high engine speeds, but to fit an efficient airfilter you will need to rework the inner body panel with a soft mallet. This was never a problem for the factory race team, but many private owners will take exception to the idea of hammering away with a mallet at the interior side panel of their engine compartments. Consequently, the combined intake manifold, carburetor, and air cleaner assembly should not exceed 13 ¾" in length as this is the maximum allowable dimension for allowing clearance within the engine compartment.

It should be understood that the length of the intake manifold or of the ram pipes merely compliments rather than determines the torque characteristics of an engine. Instead, these are determined by the performance characteristics of the camshaft. The main function of ram pipes is merely to reduce turbulence and contraction in the incoming fuel / air charge. If you look into the backplate of an Original Equipment SU airfilter box, you will then see what is often called a "stub stack". It is there specifically to reduce turbulence at the mouth of the fuel induction system, which will result in contraction (and attendant constraint) of the airflow. The Weber DCOE and the Dellorto DHLA carburetors both have a fair amount of turbulence at their mouths, so a velocity stack needs to be employed in order to reduce it.

It is of significant importance to have the appropriate length of the intake tract In order to take advantage of the performance characteristics of the camshaft. A camshaft lobe profile that produces a powerful low-end torque output functions best with a long intake tract, while a camshaft that produces a powerful horsepower output at high engine speeds functions best with a short intake tract. Either a Weber DCOE or a Dellorto DHLA carburetor can, to a very limited degree, use different length velocity stacks in order to achieve this rather than forcing the racer to spend more money for different length intake manifolds. If the amateur racer is going to drive on a slow, twisting track where low and midrange power output is critical to victory, he can then change his camshaft and tappets, change the metering of his Weber DCOE, and change to a longer ram pipe. If he is going to race on a faster track, he can then change his camshaft and tappets, change the metering of his Weber DCOE, and change to a shorter ram pipe in order to assist in attaining higher output at high engine speeds. There is, however, a major drawback to the use of velocity stacks for this purpose: the carburetion can be very sensitive to small errors in synchronization and / or metering, running rich or lean if the adjustment is off by only a small amount. Due to this decrease in reliability, it is not as good as using a longer or shorter intake manifold, but for an amateur racer it is much more affordable. For professional track racers who do have the optimum length intake manifold for the track that they are racing on, they can fine-tune the intake tract by experimenting with different length short velocity stacks during practice laps. The ready availability of different length and profile velocity stacks is one of the reasons that the Weber DCOE is so popular with racers. However, while these factors tend to make the Weber DCOE carburetor the most popular choice for racing applications, they are largely irrelevant when building a tractable engine for the street. One thing that Weber advertisements always omit is the fact that they are not a constant vacuum device like the SU carburetors are. The effect of this is that whenever you change altitude significantly, or the air pressure changes along with the weather, the fuel / air mixture goes off tune – sometimes to the point that, if set correctly at sea level, it will then go so rich at 3,000 feet that the engine will not run at idle speed! If you live on the plains, then that is fine, but if you drive through any serious mountains, then you will be better off with an SU. Been there, done that!

Be advised that neither the Weber nor the Oselli intake manifolds have a balance tube in order to modulate pressure fluctuations between the two intake tracts, a design feature that is necessary in order to prevent “robbing”. What appears to a takeoff nipple for this purpose on the right side runner of the intake manifold is in fact provided in order to allow the fitting of a vacuum-operated servo mechanism for the braking system. The unmodulated pressure fluctuation, which is aggravated in the siamesed intake ports by the uneven breathing resulting from the 180° opposed throws of the crankshaft, is the reason that these unbalanced intake manifolds have no provision for a suitable vacuum advance takeoff.

While there is a vacuum takeoff fitting (Special Tuning Part # AEH 793) for the Weber DCOE on the factory's Special Tuning intake manifold (Special Tuning Part # AEH 772), it is provided for a power brake servo mechanism, not for a vacuum advance control mechanism. The fluctuating levels of vacuum would cause the contact breaker plate in a vacuum advance distributor to rattle back and forth so violently that consistent ignition timing would be all but impossible to achieve. This in turn forces the use of a pure centrifugal advance distributor. If you decide to use either of these intake manifolds, then expect poor part-throttle response, decreased fuel economy, a ragged idle, higher combustion chamber temperatures and a consequent tendency to burn valves, as well as a tendency to preignition under heavy loads. On the other hand, the Cannon 801 intake manifold has provision for the installation of a primitive balance tube, but its straight runners create fuel / air balance and distribution problems between the outer and the inner cylinders.

Because of their racing heritage, there is a good deal of mystique surrounding the Weber DCOE carburetors, specifically concerning jetting and tuning. However, Weber DCOE and Dellorto DHLA carburetors are not as impossibly complicated as some might imagine. Whereas there is no substitute for a session on a dynometer for correctly tuning them, there is much that you can do in order to initially tune them by selecting the correct venturi sizes and initial jet settings according to a fairly simple set of rules. This should get the engine running to a drivable standard in preparation for definitive tuning on a dynometer. Without being tuned on a dynometer, their performance will be inferior to that of a well-tuned pair of SU carburetors.

Here is how the fuel is delivered: At idle speed the throttle disks are closed and fuel flows into the cylinder from an idle hole that is located behind the throttle disks. As the throttle disks start to open, the top edge of the throttle disks move towards the mouth of the carburetor and uncovers a number of progression circuit holes. These holes provide additional fuel into the increasing airflow. As the edge of the throttle plate passes a progression hole, the vacuum behind the plate draws fuel out of that progression hole. The additional fuel added by each progression hole keeps the cylinder from burning too lean in the engine speeds above idle speed and before the cruise circuit kicks in. If the initial adjustment of the throttle butterflies is open enough to uncover a progression hole, then the engine will suck the fuel from the progression circuit during idle, resulting in a lean off-idle flat spot and poor fuel mileage. If the idle position of the throttle disks is not set correctly, then you will never obtain a good, consistent idle and a smooth off-idle transition. This is where most people go wrong when setting up their Weber DCOE or Dellorto DHLA. One the most frequently experienced problem is a seemingly incurable and very annoying flat spot that rears its ugly head at about 2,200 RPM to 2,800 RPM. This condition is generally caused by one of two things: either the emulsion tube is undersized, causing a rich stumble due to an under-emulsified mixture at that particular engine speed, or the idle circuit is falling off too early to carry the engine up to the point where the main circuit can take over, leaving a "lean hole". In simple terms, the idle circuit goes lean too early. Either condition is easily rectified. If the flat spot is still there even with the correct emulsion tube, then you will need to enrichen the idle circuit. This is sometimes a tricky area, because your first impulse is to install a bigger idle jet. However, sometimes experimenting with air bleeds, mixture screws, or venturi sizes can accomplish the same thing while continuing to use the original jet size. Fuel for the progression holes is supplied by the idle jet, which functions to meter fuel to the progression holes as well as setting the fuel / air ratio of the idle mixture. As the throttle plate rotates past the progression holes, the ones behind the throttle plate supply a mixture of fuel and air. Because the air that is rushing into the mouth of the carburetor compresses in front of the throttle plate, the pressure causes it to enter the progression holes that are in front of and immediately above the throttle plate. This air is drawn through the progression holes that are behind the throttle plate, aerating the fuel just as a combined air jet and emulsion tube would. When the throttle plate rotates beyond the progression holes, the increasing vacuum causes them to gradually cease flowing fuel. By the time that the engine has passed about 2,800 RPM, the top edge of the throttle plate has passed all the progression holes and has opened wide enough to cause the vacuum to drop to a point where it is drawing gradually less fuel out of the progression holes. This is when the auxiliary venturi needs to start delivering fuel so that the increasing level of fuel delivered by the main jets will supplement the decreasing level of fuel supplied by the progression holes. The size of the auxiliary venturi determines when the main jets will start delivering fuel. If it kicks in too early (too small of an auxiliary venturi size), you will then get an over-rich condition and the engine will either bog or stumble during the progression (or just waste fuel with no noticeable symptoms). If the auxiliary venturi is too large, then there can be a lean area where the progression openings are not delivering enough fuel and the main cruise circuit has not yet kicked in. During acceleration this leanness is often masked by the accelerator pump. This condition would be evinced as a leanness in a narrow band of engine speed while in a constant low engine speed cruise (this is where an onboard CO^2 monitor that can be read during driving comes in handy). The goal in such a case would be to fit the smallest auxiliary venturi that will not cause either an over-rich bog or a stumbling during a slow opening of the throttle disks. This will assure the absence of a lean band of engine speed that might damage the engine over time. Of course, in order for the carburetor to flow air properly in or for such functionings to be diagnosable, it is absolutely necessary for the airstream through the barrels to be devoid of contraction or vortexing. This is accomplished by means of the mandatory use of a set of air horns (ram horns).

When deciding on which Weber DCOE or Dellorto DHLA carburetor to use, some will believe “the bigger, the better”, and then leap to the conclusion that the bigger DCOE 45 or DHLA 45 will automatically give more power. This reveals a basic misunderstanding of the design as well as the principles of operation of these carburetors. It is actually the size of the main venturi, and not the barrel diameter, which will determine the maximum amount of airflow, and therefore the total horsepower potential. Selection of the correct main venturi size is the first step in selecting the carburetor. If you get the venturis wrong, then you will never get it running right. Too large a venturi and you will always have a flat spot that you cannot tune away. Too small a venturi and it will always run rich and not make any power.

While it is easy to assume that bigger is better when selecting a main venturi size, it must be understood that in order to draw in and effectively atomize the fuel, the purpose of the main venturi is to increase the vacuum that acts on the main jet. The smaller the main venturi, the more effective this function is. However a too-small main venturi will restrict maximum airflow. A large main venturi may give a bit more power right at the top end of the powerband, but it will come at the expense of tractability at lower engine speeds. Only a racer on a track will benefit from this sort of choice. On a road car, tractability is much more important. Most of the time a street engine is nowhere near the peak of its horsepower output. However, it is close to the peak of its torque output for perhaps seventy-five percent of the time. It is therefore much more important to compromise and select the main venturi for the best all-around tractability. The basic rule of thumb is to install the smallest main venturi that will give you full power at the power peak established by the camshaft. This smaller main venturi will provide better overall flexibility and useable power. Going to a larger main venturi will allow you to attain absolute maximum power at peak engine speeds, but will reduce low-end torque. Once the optimum main venturi size has been determined, then the appropriate carburetor size can be chosen. The largest main venturi that will work optimally in a DCOE 40 is a 34mm venturi. The smallest main venturi that will work optimally in a DCOE 45 is a 34mm venturi. While larger main venturis are available for the DCOE 40 and smaller ones are available for the DCOE 45, they will not work as efficiently.

Main Jet size (Venturi size x 4)

Air Corrector Jet size (Main jet size + 50)

Using these formulae, a main venturi size of 36mm will require a Weber main jet of 144 and a Weber air corrector of around 194.

Suggested Weber Emulsion Tube Type, for a given single cylinder capacity:

Cylinder Capacity	Suggested Weber Emulsion Tube
250cc - 325cc	F11
275cc - 400cc	F15
350cc - 475cc	F9, F16
450cc - 575cc	F2, F5

Using the above formulae, the theoretical initial settings for a 1812cc B Series engine with its power peak at 6,250 RPM equipped with a Weber DCOE are as follows:

34 mm Venturis

F2 Emulsion Tubes

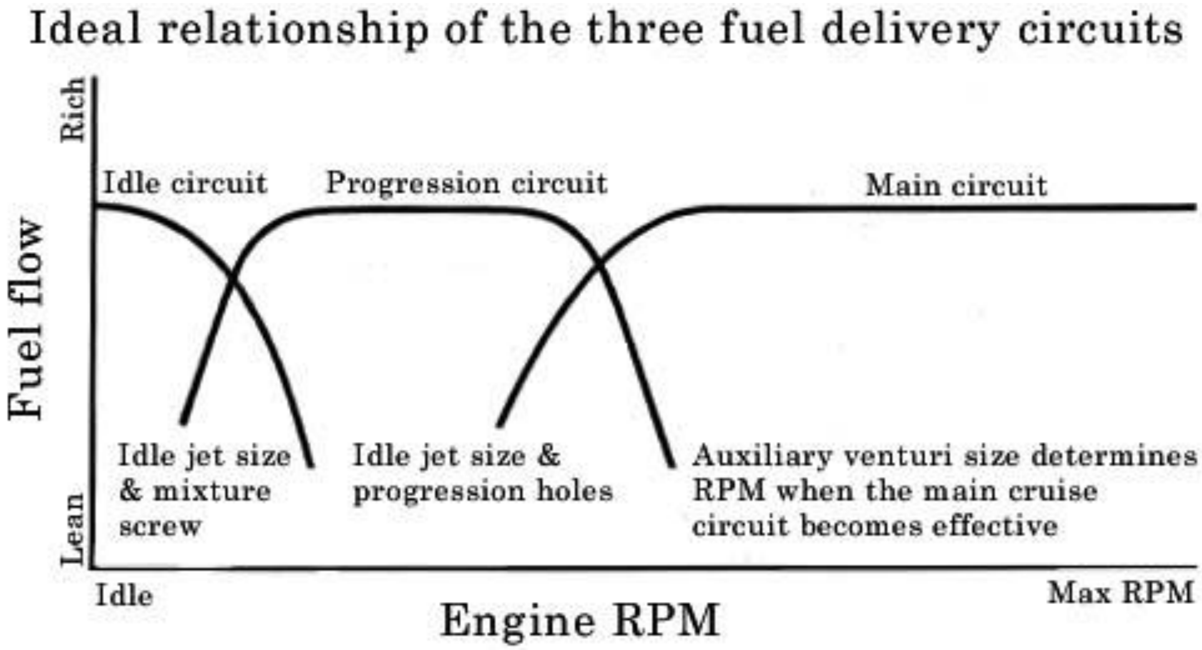
144 or 145 Main Jet

194 Air Corrector Jet

45 or 50 Idle Jet

The size number of the main venturi is stamped on the front so that it will be visible when looking down the barrel. On the little center auxiliary venturi the size is stamped on its outer side and thus is not readable unless the auxiliary venturi has been removed. The little center auxiliary venturi is where fuel from the main jet is drawn into the air flow. Both of the venturis on the Weber DCOE 45 are held in place by 10mm fixing bolts on the underside. The fixing bolts are drilled for tie wiring and can either be kept secure by either a tie wire or by a lock tab. Because of the thinness of the venturis that they hold in place, there is very little torque on these fixing bolts. The lock tabs that are provided by the factory have been known to loosen and allow a fixing bolt to back out and fall off. The result is a lose venturi and a throat that does not operate correctly in the high-2,000 RPM and higher range. The best thing that you can do for reliability once that you have decided upon the correct main venturi and auxiliary venturi combination is to safety wire the fixing bolts together.

The little center auxiliary venturi is a small suspended venturi inside of the throats of the Weber DCOE that delivers fuel from the main cruise circuit into the air stream. The number on the auxiliary venturi refers to the diameter of the cross section area of the delivery port (venturi) and not to the size of the fuel nozzle that delivers fuel into the port. The smaller the diameter, the higher the air velocity through this suspended venturi will be and the sooner the main circuit comes into play. In order to understand the role of the auxiliary venturi you need to understand how fuel is delivered to the engine at different engine speeds. Not including the cold start and the accelerator pump circuits, the Weber DCOE has three different fuel delivery systems that deliver fuel into the throats of the carburetor at different engine speeds. Ideally, you want the fuel / air ratio of each of these be the same during the transition from one circuit to another.



The ideal intersection of the fuel delivery curves should be where the fuel delivery of the circuit fading out plus the fuel delivery of the circuit coming on added together should always equal the total amount of fuel delivered when either circuit is in the middle of its range. The idle and progression circuits are cast and drilled into the carburetor body, making the location of their curves occur at fixed engine speeds. The idle jet provides fuel to both the idle and progression circuits. This jet determines the richness of the progression circuit. The idle mixture adjustment screw fine-tunes the richness of the idle circuit curve. The main jet assembly (main jet, emulsifier tube, and air correction jet) sets the richness of the main circuit. The size of the auxiliary venturi determines the engine speed at which the main circuit curve intersects with the curve of the progression circuit. The main circuit usually comes into play at around 2,800 to 3,000 RPM. Below that engine speed, the engine is fueled by the idle jets. This knowledge should help you in trouble-shooting any problem that occurs exclusively above or below an engine speed of approximately 3,000 RPM.

The float valve assembly regulates the amount of fuel that is allowed into the float chamber and maintains the appropriate fuel level within a very limited range. The float itself opens and closes the float valve. The closed setting regulates the fuel level in the float chamber and in the emulsion tubes. The top stop of the float regulates the level of the fuel, while the lower open stop regulates how far the float valve can open. The diameter of the valve regulates the rate at which fuel that can enter the chamber. There is a range of float valve sizes available for the Weber DCOE and Dellorto DHLA carburetors. The float valve needs to be large enough in order to allow an adequate flow of fuel into the float chamber, but should not be larger than necessary because a too-large valve will quickly let too much fuel enter before it closes, thus causing a pulsing, over-rich condition.

Traditionally, the Weber DCOE came with soldered brass floats. Most, if not all, of the later versions are provided with plastic floats. The float settings are different between metal and plastic floats. Note that some of the additives in newer fuels may attack the plastic floats. If you start having problems that might be caused by incorrect bowl fuel levels and you have plastic floats, then it would be good idea to remove the float chamber cover and inspect the floats. The float setting is dependent upon the carburetor you use. Here is a chart for the brass floats:

Weber DCOE Series	**Float Valve Closed**	**Float Valve Max Open**
DCOE 40 Series 2, 4, 18, 22/23, 24, 27, 28, 31,32	8.5 mm	15 mm
DCOE 40 Series 29/30	5.0 mm	11.5 mm
DCOE 40 Series 44/45	7.0 mm	14 mm
DCOE 40 Series 72/73, 76/77, 80/81	7.5 mm	14 mm
DCOE 45 Series 9	5.0 mm	13.5 mm
DCOE 45 Series 14, 14/18, 17	8.5 mm	15 mm
DCOE 45 Series 38/39, 62/63, 68/69	5.0 mm	14 mm

DCOE 45 Series 72/73, 76/77, 80/81	7.5 mm	14 mm

Both floats need to have identical settings. You may need to bend the arms between the two floats to get them exactly the same closed height. The ideal tool to set the closed float position is a round rod with the precise diameter of the closed float setting. The seam of the float should not be taken into account when measuring the float level so there should be groves cut into the rod to clear the float seams. This will allow you to see both the accuracy of the setting and any variation between the two floats. The open and closed measurements should be taken with the top gasket in place. The closed position should be measured just as the floats close the valve and not with the entire weight of the floats upon the valve. This is done with the top plate tilted a little over vertical.

Make sure that the weight of the float indicates that it is the correct one (26 grams), that the float can freely slide on the axis, and that it does not show any pits. Make sure that the needle valve is tightly screwed into its housing and that the pin ball of the dampening device incorporated in the needle is not jammed. Keep the float chamber cover in a vertical position, since the weight of the float could lower the pin ball fitted on the needle. With float chamber cover in the vertical position and float clip in light contact with the pin ball of the needle, the distance of both half-floats from upper surface of float chamber cover with gasket in place, must measure 8.5mm. After the leveling has been done, check that the stroke of the float is 6.5mm. If necessary, adjust the position of the lug. In case float had not been rightly set, rectify the position of float clip till the required quota is reached, taking care that the clip does not show any pit on the contact surface that could affect the free sliding of the needle. Install the float chamber cover making sure that float can move without any hindrance or friction.

Be aware that Weber DCOE carburetors have a wire screen filter in the inlet. This filter does not have a good reputation for working well over time. Considering the size of some of the jet openings, the built-in screen is not really fine enough to prevent clogging. You should consider installing a high-volume fuel filter between the fuel pump and the fuel regulator. On a street machine, it is a good idea to remove and inspect these screens as part of your 3,000 mile service. New blends of fuel can cause some fuel lines to deteriorate along their interiors. Your first sign of this occurring will be black particles in the filters. If you start to see black particles on the Weber's fuel screens, then it is time to replace all of the rubber fuel lines on the vehicle and to clean the sediment bowl of the fuel pump. (Be sure to use a new rubber seal). A word about steel braided fuel lines: Do not use regular worm gear type hose clamps with steel braided fuel lines. The steel braided fuel lines are specifically designed to not crush. At best, you will end up with a very "iffy" seal.

Be advised that most Weber DCOE jets have tapered ends. The tapered end of these jets sit snugly against seats in the body of the carburetor in order to create effective seals between different areas of the carburetor. These jets are mounted onto holders with a friction fit. As the holder is threaded in, the taper at the end of the jet comes into contact with the passage seat and is pushed back into the holder, thus maintaining a contact seal. If the jet is initially pushed all the way into the holder, then it may not reach all the way to the passage seat. The fit between the jet and holder should always be tight. A loose fit can allow a jet to back away from its seat over time. If the seal is not made, or is broken, then the carburetor will not function properly. The proper method of installing a jet is to fit it only about one eighth an inch into the holder, and then allow the passage seat to push the jet in the correct distance as you screw in the holder. Be careful not over-tighten the jet assemblies. Once the jet is seated, it does not take much torque to hold everything in place. Most jet sizes are in numbers that give their actual diameter in hundredths of a millimeter. Idle jets can also have "F" numbers that indicate their ability to emulsify fuel. Note that this has nothing to do with the flow rate of the hole. Also note that here is no relationship between flow characteristics and the different number designations of emulsion tubes. This is due to the fact that designation of the emulsion tube is by the numerical order in which they were designed.

Idle jets can cause a lot of confusion because their name suggests that they govern the fuel / air ratio at idle speed, but this presumption is incorrect. It is true that the fuel consumed at idle speed is drawn in through the idle jets, but the idle mixture is not metered by these jets, but instead by the idle volume adjustment screws that are mounted above them on top of each barrel. The idle jets control the critical off-idle progression circuit between closed throttle and the engagement of the main jet circuit (somewhere around 2,500 to 2,800 RPM). This part throttle operation is important to a smooth progression between a closed throttle and acceleration, as well as for part-throttle driving. If this circuit is too lean (weak), then the engine will stutter or nosedive when opening the throttle. If it is too rich, the engine will hunt and surge, especially when hot. Idle jets effect the idle and the progression circuits of the Weber DCOE. Proper selection is critical for smooth, economical cruising at low engine speeds. At idle speed, the fuel is mixed into the airflow behind the throttle plate and the quantity of the fuel flow is regulated by the idle volume adjustment screw. There are a series of progression holes, not effected by the idle screw, that become exposed to the airflow behind the throttle disks as the throttle continues to open. Because the air that is rushing into the mouth of the carburetor compresses in front of the throttle plate, the pressure causes it to enter the progression holes that are in front of and immediately above the throttle plate. As the throttle plate top edge moves past each hole, the vacuum behind the plate draws fuel from the idle jet out through that progression hole. This adds progressively more and more fuel to keep the engine running smoothly off of idle speed until the airflow is high enough to draw fuel from the main jet. Since the progression holes are drilled into the body of the carburetor and hence are not adjustable, the idle jet is chosen primarily for the progression circuit.

Idle jets have a fuel hole drilled into the bottom of the jet and an air bleed hole drilled into its side. The fuel hole meters the amount of fuel for both idling and the gradual progression from the idle circuit to the main circuit. The air is drawn out through the progression holes that are behind the throttle plate, aerating the fuel just as a combined air jet and emulsion tube would. When the throttle plate rotates beyond the progression holes, the induced vacuum causes them to cease flowing fuel. The goal of the tuner is to select the smallest hole that will provide a good, smooth progression. The air bleed hole effects the fuel / air ratio of the fuel in both the idle and the progression circuits. A small air bleed hole creates a richer fuel / air mixture ratio, while enlarging the air bleed hole makes the fuel / air mixture leaner (weaker).

Engine Size	**Weber Idle Jet Size**
1600cc	40 / 45
1800cc	45 / 50

2000cc	50 / 55

While establishing the correct idle jet for a given engine is not always easy, usually a close approximation will make the car acceptably drivable. If the progression fuel / air mixture is lean (weak), then the engine will nosedive when it transitions from smaller to larger throttle openings. A certain amount of change (richer / leaner) to the progression can be achieved by changing the air jet size on the idle jet. Changing the air jet size alters the amount of air that is emulsified with the fuel that is drawn in through the idle jet. If this does not enrichen the progression fuel / air mixture sufficiently, then the next larger idle jet size should be tried with the same air jet. Below is a small scale that shows the size designations of the most commonly used Weber idle jets, running from lean (weak) to rich. It should be noted that their part number reflects the order in which they were developed, and not any physical attribute.

Lean Normal Rich
F3, F1, F7, F5, F4, F2, F13, F8, F11, F14, F12, F8, F9, F12, F6

The goal is to end up with the leanest mixture that provides correct engine performance throughout the range of progression circuits.

The final selection for idle jets should be based upon how the engine performs over the range of the progression circuit. Prior to making the final testing of the idle jets, make very sure the ignition is properly set up and functioning with properly advanced initial timing and the advance not starting until around 1,200 RPM. With a modified engine, the initial timing will probably be in the neighborhood of 8° to 12° BTDC with at total advance of 32° to 34°. An ignition timing problem can be mistaken for a leanness of the progression circuit. Initial timing that is too retarded for the engine is a source of spitting out through the carburetor throats. Slowly advance the throttle off of idle speed and listen for any hesitation. If there is hesitation, then the mixture is too weak. Adjust the idle flow screws out ½ turn, and then try again. If the hesitation is still there, then the jetting will need to be altered. Try going two steps richer on the air correction hole while leaving the fuel hole size the same. Reset the idle, and then retest. If going to the richest level does not get rid of the hesitation, then go to the next size larger fuel hole and rerun the tests with different air correction holes. When the no-load tests are completed, drive the car and retest under load conditions. The ideal idle jet size provides an idle CO^2 in the 2.5% to 3% range with the idle flow screws adjusted between 7/8ths of a turn and 1½ turn, and does not cause the engine to hesitate during the use of the progression circuit. The next leaner air correction size would cause a hesitation under load conditions. An over-rich jet will not provide top performance, but it will cause increased fuel consumption. If you have the throttle linkage connected at this time, then the rods between the carburetor's throttle arm and the crank need to be identical in length. Different length arms will effect the carburetor synchronization as you move off of idling speed. If your mixture screw is out more than one turn, like 1½ turns, then your idle jet is too lean. If that is the case, then go up one half-size on the idle jet. If your mixture screw is not out one full turn, something like only ½ turn out from its seated position, then your idle jet is too rich. Go down one half-size on the idle jet. This is all based on the important fact that your idle speed screws are not open more than ½ turn. If they are, then that is also an indication that you have a lean Idle circuit. You are cheating by opening the throttle disks and exposing additional progression holes in the transition.

The main jet, the emulsion tube, and the air correction jet combine to form the main jet assembly that provides fuel to the engine once the throttle disks are open beyond the progression holes. The emulsion tube is a long brass tube with openings along the side. The main jet is a friction fit into the bottom of the emulsion tube. The air correction jet is a friction fit into the top of the emulsion tube. Never attempt to separate any of them by twisting or wiggling them. Instead, always pull them apart, otherwise they will be ruined. There is a fuel passage that goes from the float chamber, through the main jet and into the emulsion tube. When the engine is not running, the fuel level inside of the emulsion tube is the height of the level in the float chamber. There is also an air passage from the small hole on the face of the Weber DCOE into the float chamber, through the air correction jet and into the emulsion tube. As the throttle disks move towards full open the progression holes cease to draw enough vacuum to pull fuel from them. At that point air speed, and hence vacuum, through the auxiliary venturi, which is located in front of the primary venturi and is suspended in the center of the air stream, is increased as a result of it location in the front center of the primary venturi where the vacuum is at its greatest. The vacuum starts thus to draw fuel from the auxiliary venturi.

The main jet controls the fuel mixture in the emulsion tube in the mid-RPM range when the cruise circuit is activated. It is stuck into the bottom of the emulsion tube and sits in fuel. As the carburetor begins to work, the main jet meters the amount of fuel allowed to pass through it and up into the "main well" around the emulsion tube. Air enters through the top of the emulsion tube through the air corrector which meters the amount of air to be mixed with the fuel. The air blows out of the emulsion tube through a series of holes along its length and aerates the fuel that is rising up the well around the tube. This emulsified mixture is then drawn out of the main delivery nozzle as the vacuum in the carburetor increases to the point where at it is strong enough to pull it out. This occurs by 2,800 to 3,000 RPM. As the emulsified fuel / air mixture is drawn out of the emulsion tube, fuel flows from the float chamber through the main fuel jet into the emulsion tube to replace the emulsified fuel / air mixture that is being drawn out through the auxiliary venturi. Choosing the correct combination of main jet, emulsion tube, and air correction jet is essential for performance at both cruise and high engine speeds.

As the engine speed increases, the main jet becomes more of a factor and becomes the dominant partner in controlling the mixture at high engine speeds. The main jets are numbered by the diameter of the jet opening and come in size steps of .05 mm. Too lean a main jet can damage the engine through overheating. Too rich a main jet washes the oil off the sides of the cylinder walls and causes rapid wear of the cylinder walls. The basic rule of thumb for picking a main jet size for a street engine operating at sea level is to multiply the venturi size times 4.

At high elevations, the engines is getting less air, so it needs less fuel to maintain the proper air/fuel ratio. Generally, go down one main jet size for every 1,750 to 2,000 feet of rise in elevation. If you normally run a 145 main jet at sea level, then you would drop down to a 135 at 4,000 feet. Something else that decreases as you rise in elevation is horsepower. You can figure on losing about 3% or your power for every 1,000 feet that you go up. At 4,000 feet power output will be reduced by about 12%.

The air correction jet on a main jet only effects the performance of the engine at high engine speeds. The larger the number is on the air correction jet, the larger the air is hole and thus the leaner the main jet will run at higher engine speeds. If the air correction jet is too lean (too large a hole), then the engine will miss near peak engine speed. If the air correction jet is too rich (too small a hole), then the engine will not produce optimum power. For testing purposes, find the largest diameter air correction jet that causes a lean running condition that results in misfire at high engine speed, and then fit a 10 to 20 smaller diameter (richer) air correction jet. The air correction jet number is their hole diameter in hundredths of a millimeter and range in size increments of 5. When testing, the minimum increment of changes should be at least 10, with 20 being the more common increment to notice changes.

As its name implies, the emulsion tube is where air is mixed with fuel to form an air/fuel emulsion (fuel with lots of little air bubbles in suspension). The vacuum formed in the auxiliary venturi draws this emulsion out of the emulsion tube and into the air streaming through the auxiliary choke where it is atomized into the air stream and delivered into the combustion chamber. The emulsion tube effects the acceleration phase as the main jets become activated. If the emulsion tube size is incorrect, then the engine will not accelerate cleanly when the main cruise circuit is operating. The effect of changing emulsion tubes can be very subtle to detect. Emulsion tube operation is very sensitive to the fuel level in the float chamber. As a result, you need the right size float valves and closely set floats for the emulsion tubes to work as intended. Emulsion tubes differ by their internal diameters and the number, size and positions of the side holes. They are complex tubes wherein just the right level of emulsification takes place. Their part number reflects the order in which they were developed and not any physical attribute. The emulsion tube sizes are (in order of rich to lean): F7, F8, F2, F11, F16, F15, F9. There are also additional sizes.

The accelerator circuit of a Weber DCOE consists of a fuel reservoir, a mechanically activated, spring loaded plunger-type pump that flushes the reservoir, a one-way valve that lets fuel into the reservoir from the float chamber while the plunger is purging the reservoir (called an accelerator pump intake / discharge valve), and an accelerator pump jet that both meters the amount of fuel pumped by the acceleration pump (plunger) as well as delivers that fuel directly into the rear of the carburetor throats.

Low- and mid-range hesitation in a Weber DCOE carburetor may be the result of an improperly set up accelerator pump. For example, the Weber DCOE 40 comes in different Series models, each of which has a particular length pump stroke for its accelerator pump. The Series 2, 4, 24, 27, 28, 32, and 33 models have 14mm pump stroke lengths, the Series

18, 22/23, and 29/30 models have 10mm pump stroke lengths, the Series 31, 34/35, 44/45, and 76/77models have 16mm pump stroke lengths, and the Series 72/73 as well as the Series 80/81 models both have 18mm pump stroke lengths. The pump stroke rod (which governs the length of the pump stroke) is interchangeable between the different Series, but must be chosen carefully when combining with the various combinations of accelerator pump jet and accelerator pump bleed valve. Most owners use a blank accelerator pump bleed valve, which, while this is consistent with the recommendations of the Haynes manual, is inconsistent with the official recommendation of the Weber company.

The accelerator pump mechanism of the Dellorto DHLA carburetors is notably different from that of the Weber DCOE system. The accelerator pump is located on the underside of the carburetor body. It is of the diaphragm type. The amount of fuel that it injects is controlled by turning a nut up or down a threaded rod.

The accelerator pump functions during rapid transitions to full throttle. When the throttle is opened quickly, the sudden gust of air into the fuel induction system causes an increase of internal pressure, which can in turn cause atomized fuel droplets to condense and fall to the floor of the intake manifold. This condensed fuel will remain on the floor of the intake manifold unless the intake manifold is heated. (Be aware that this puddle of fuel, incidentally, is often the reason for Weber carburetor fires.) This results in a noticeably flat throttle response. The whole purpose of having an accelerator pump is to fill this lull until the main jet can catch up with the increase in engine speed. The long-stroke accelerator pumps discharge more fuel, but over a longer duration of time than the short-stroke accelerator pumps do. On the other hand, the short-stroke accelerator pumps deliver a shorter, but larger dose of fuel. If the size of the accelerator pump jet is increased, the amount of fuel delivered will increase, but the duration of the fuel delivery will be decreased.

The carburetor is thus designed to harmonize four modifiable variables: the length of the stroke of the accelerator pump, the size of the accelerator jet, the size of the accelerator pump bleed valve, and the pump spring rate, all of which are interdependently involved in regulating both the volume and the duration of fuel delivery between the moment of throttle application and the moment when the main jet can catches up with the increase in engine speed. Of course, the accelerator pump system should also discharge no fuel when the throttle is at a constant setting, or when it is opened slowly.

The ultimate solution to these variables will depend upon the engine modifications, the style of driving, and engine speed. In general, the short-stroke accelerator pumps are well suited to engines having siamesed intake ports or one carburetor that feeds multiple cylinders. This could then explain the Haynes recommendation of using the DCOE 40-2 series model with its 14mm long stroke. However, it seems that in order to compensate for the resulting long duration of discharge, the Haynes manual recommends that the accelerator pump intake/discharge valve be closed.

Blanking off the accelerator pump intake/discharge valve ensures a rapid response, but this also means that the total fuel quantity pumped by the piston differs little whether the accelerator pedal is moved slowly or rapidly, resulting in the mixture being overly rich upon gentle acceleration. Amongst other things, a closed accelerator pump intake/discharge valve will greatly increase fuel consumption. However, the dedicated owner can improve any flat throttle response by experimenting with different open accelerator pump intake/discharge valves. This may well explain the rationale of the Weber company when they suggest a #50 accelerator pump intake/discharge valve with 10mm stroke pumps.

The metering hole in the accelerator pump jet has to be large enough to remove any hesitation or stumble caused by the lean condition that is created by suddenly opening the throttle disks at low engine speeds. A too-large accelerator pump jet will cause a "bogging down" of the engine from too much raw fuel. The accelerator pump jets are numbered for their hole size in hundredths of a millimeter, and are in steps of five hundredths of a millimeter (i.e. 35, 40, 45, 50), i.e., a valve marked 50 has a 0.5mm discharge hole in the side. Use the smallest jet size that will eliminate any hesitation or stalling when the throttle is suddenly opened. The accelerator pump intake/discharge valve can have a discharge hole that fine-tunes the flow of the accelerator pump jet in-between the step increments. The accelerator pump intake/discharge valve is a one-way valve that allows fuel to flow into the accelerator pump reservoir and keeps the fuel from going the wrong way when the accelerator pump is activated. It can also be used to precisely tune the amount of fuel injected into the engine by the accelerator pump jet. This is achieved by selecting a valve with a discharge hole on the side. If there is no discharge hole, then the accelerator pump intake valve acts purely as a one way valve. If there is a hole, then part of the fuel is discharged out the side hole back into the float chamber when the pump is activated, bleeding off any excess fuel that is not required in order to accelerate the car cleanly.

After mounting, linkage going to a common throttle bar should be identical in length. Check to make sure that the throttles of all carburetors are completely closed when the linkage is the closed position. Test the complete throttle linkage for any tendency to bind and not return the throttle disks to the fully closed position. Backup external throttle springs should be considered an important safety feature.

Before setting up the idle, it is important to be very sure there is no throttle shaft bind or over tightened levers. This is best checked for before the carburetors are mounted. Open, and then close the throttle slowly. Next, give the lever a little extra push in the closed direction. If the throttle shaft is binding, then it will not return to fully closed if you let the internal throttle springs close it gently. An accidental drop that strikes the throttle linkage can cause the shaft to bend just a little so that it binds. Always test before you buy.

Some Weber carburetors have a cold start circuit (choke), while others do not. In my experience, it is very easy to flood the engine and wet the spark plugs using the cold start mechanism, as it very crude in operation. The accepted technique for cold starting is as follows:

Allow the float chambers to fill. This should take about five to ten seconds. Fully depress the accelerator rapidly four times, and then applying a light throttle, turn the engine over. If it does not start immediately, repeat the procedure three times. The engine should fire, but may need "nursing" for a minute or two before it will idle properly. Gentle prodding of the accelerator should keep it running long enough for it to warm up. If the engine does not fire within three attempts, then try five or six pumps of the accelerator pedal in order to operate the accelerator pump. If this does not work, then depress the accelerator fully and hold it open while turning the engine over for five to fifteen seconds. Finally, close the accelerator and try again.

Start the engine and then allow it to reach normal operating temperature. This may require adjustment of the idle speed as the engine warms up. Spitting back through the back of the carburetor normally indicates that the mixture is too weak, or the ignition timing is hopelessly retarded. If this happens when the engine is warm and you know that the ignition timing is correct, then the fuel / air mixture will need trimming richer. Set the idle as near as you can to 900 RPM. This may be above 1,000 RPM with a hotter (Piper BP285) camshaft lobe profile.

If you are using a crossflow cylinder head with dual carburetors, then you will need to balance the airflow of the carburetors before setting the final idle. When balancing dual Weber DCOE carburetors, be sure to bring the idle speed of the high carburetor down to that of the low carburetor, and then bring them both up to proper idle speed. Using an airflow meter or carburetor synchronizer, adjust the balance mechanism between the carburetors in order to balance the airflow between them. If the rearmost carburetor is drawing less air than the front, turn the balance screw in a clockwise direction in order to correct this. If it is drawing more air, then turn the balance screw counterclockwise (anti-clockwise). If the idle speed varies at this point, adjust it back to 900 RPM. In order to decrease the idle speed, turn the balance screw in a counterclockwise (anti-clockwise) direction. In order to increase the idle speed, turn the balance screw in a clockwise direction.

When you are sure that the carburetors are both drawing in the same volume of air, adjust each idle mixture screw to identical settings.

Where many people go wrong in setting up a Weber DCOE carburetor is in using the idle lever adjustment screw in order to adjust the idle speed. More often than not, this ends up uncovering the first progression hole at idle. This will cause you to select the wrong idle jet, or if you have the correct one, there will be a lean flat spot right off of idle that you will be unable to compensate for. Rough running and rough idle is normally the result of incorrect idle mixture and balance settings.

Below is a technique for establishing a clean idle and progression. Before adjusting the carburetors in this manner, you must make sure that the following conditions are met:

1) That the engine is at normal operating temperature
2) That the throttle return spring/mechanism is working correctly
3) That the engine has sufficient advance at the idle speed (between 12 ° and 16 °)
4) That an accurate tachometer is connected.
5) That there are no air leaks or electrical faults.

Depending upon the camshaft lobe profile, a reasonable idle speed for a modified engine equipped with a Weber DCOE is between 900 RPM and 1,100 RPM.

First, adjust the idle jets for a smooth idle, and then set the idle speed. Turn the idle mixture adjustment screws inward until they bottom out, and then out 2.5 turns. Turn the idle mixture screw counter clockwise (richening) in small increments (quarter of a turn), allowing a good five to ten seconds for the engine to settle after each adjustment. Note whether engine speed increases or decreases. If it increases, then continue turning in that direction and checking for engine speed, then the moment that engine speed starts to fall, back off a quarter of a turn. If the engine idle speed goes well over 1,000 RPM, then trim it down using the idle speed screw, and re-adjust the idle mixture screw. If engine speed decreases, then turn the idle mixture screw clockwise (weakening) in small increments. If the engine speed continues to rise, then continue in that direction, then the moment it starts to fall, back off a quarter a turn. The mixture is correct when a quarter of a turn in either direction causes the engine speed to fall. If that barrel is spitting back, then the mixture is too weak, so start turning the idle mixture screw in a counterclockwise (anti-clockwise) direction in order to richen the fuel / air mixture. During this procedure, the idle speed may become unacceptably high, so re-adjust it and repeat the procedure for each barrel. If you are taking a CO^2 reading, then 2.5% CO^2 at idle is ideal for a street engine. Higher than 3.5% is unusable and just provides poor fuel consumption without any gains. If you cannot obtain a correct idle by adjusting the idle mixture screw, then the throttle disks are not passing enough air and you need to increase the amount of air going past the throttle disks.

On newer Weber DCOEs that have a venturi balance adjustment screw, the venturi balance circuit is an air path that bypasses the throttle plate. The amount of air that bypasses the plates is controlled by the adjustment screw. This allows additional air to go past the throttle plate at the idle position when the throttle plate does not pass enough air for the engine to idle properly (remember that you do not adjust the idle adjust screw after it is set at ½ turn). Adjusted properly, it will allow enough air past the throttle disks in order to allow the engine to idle properly. This adjustment is very sensitive, so a small amount of turn can make a big difference in idle.

If the engine will not idle with the throttle disks closed, then set the idle lever adjustment screw to one-half turn beyond initial contact, the idle adjustment screws set at one turn up from fully seated. If the carburetors idle and both venturi balance adjustment screws fully seated, then adjust the venturi balance adjustment screws 1/16th turn off of being seated, and then restart the engine and see if it idles. If not, then adjust both venturi balance adjustment screws out in 1/16th turn increments until a reasonable idle speed is achieved, and then zero in the idle using the idle mixture adjustment screws. When you are satisfied that the venturi balance adjustment screws are correct, tighten down on the lock nuts and replace the cover over the adjustments as they should never need to be adjusted again.

On older Weber DCOEs that have no venturi balance adjustment screw, the throttle disks need to be modified in order to allow additional airflow in the idle position. This is done by drilling holes in the throttle plate near the bottom edge. Drill a ½ mm diameter hole in each plate on each carburetor, refit the carburetors, and then try to get the engine to idle. If there is not enough airflow, then enlarge the holes to 1mm and then try again. If you still do not have enough air, then drill a second ½ mm hole near the first, refit the carburetors and try again. Some people file the bottom of the throttle disks in order to get extra airflow. I do not recommend this as there is no way to assure you have filed the exact same amount off of each throttle plate.

After both the mixture screws have been set, the idle should be fairly even with no discernible “rocking” of the engine. If the engine is pulsing, spitting, or hunting, then the mixture screws will need further adjustment. However, be aware that no amount of adjustment will give a good idle if the throttle shafts are bent, leaking air, or the linkages are loose on their shafts.

Cylinder Head Castings

The North American Market MGB engine used five different cylinder head castings over the course of its career, all of which used variations of the same 1.344” diameter exhaust valve size. This Original Equipment exhaust valve borders on being overlarge. This was a deliberate design feature made for the benefit of the factory race team so that there would not be any problems with the homologation rules of racing associations.

The first version of the cylinder head (BMC Part # 48G 318) was used on the 18G, 18 GA, and 18GB engines, used an 89.2 gram 1.565” diameter intake valve (BMC Part # 12H 435), and can be identified by its cylinder head casting number of 12H 906.

The second version was used on the 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines and can be identified by its casting number of 12H 2389. Like its predecessor, it also used a 89.2 gram 1.565" diameter intake valve (BMC Part # 12H 435), but switched to a newer version of the intake valve (BMC Part # 2115) that made use of a new valve spring cup (BMC Part # 12H 3309) and cotters (BMC Part # 2117), dispensing with the use of the earlier retaining circlip (BMC Part # 1K 372) of the earlier valve. It should be noted that both of these cylinder head castings had an identical combustion chamber height of .425" and combustion chamber volume of 43cc. Both this cylinder head casting and its predecessor are of a Weslake heart-shaped "Closed" design configuration, featuring a large promontory between the valves for ducting of the incoming fuel / air charge away from the hot exhaust valve as well as for reinforcing the thin roof of the combustion chamber in order to prevent cracks from forming on the valve seats. However, in the second version of the cylinder head the promontory was reduced in order to prevent it from becoming a hot spot which could trigger preignition. It was also given a slightly improved intake port design as well as mounting bosses on the spark plug side of the cylinder head for the mounting of air injectors. In addition, it introduced oil drain grooves that connect the valve spring recesses to the aperture for the pushrod that operates the intake valve, thus increasing lubrication down the bore of the tappet below. This feature was deemed advantageous due to the fact that in order to simplify assembly on a mass-production basis, the same valve springs were used on both the intake and the exhaust valves. Because the intake valve was heavier than the exhaust valve, the greater side thrust loadings on their tappets that were generated by the angular deflection of the pushrods resulted in accelerated wear of the bores for these tappets. The simple expedient of channeling the oil that drained from the valve assemblies to these tappets largely solved this problem. A later change to longer pushrods and shorter, bucket-type tappets in order to reduce deflection and its attendant side loads eliminated it.

The third version of the cylinder head, introduced on the earliest 18V engines, used larger 94.8 gram 1.625" diameter intake valves (BMC Part # 12H 2520) that gave a 8% increase of valve area, as well as revised ports in order to produce more power at high engine speeds, although at the expense of a very small loss of torque at very low engine speeds. It can be readily identified by its casting number 12H 2923, and is commonly found on engines with the engine numbers 18V-584-Z-L, 18V-585-Z-L, 18V-672-Z-L, and 18V-673-Z-L, all of which it was Original Equipment for. Development of this cylinder head was originally started in response to a request from the factory race team for a more open-type combustion chamber that would reduce valve shrouding so that larger intake valves would become more practical on their racing engines, but it was found to offer the advantage of maintaining the power output of the North American Market cars when both the valve timing was revised and the compression ratio reduced in order to meet government-imposed air pollution standards. Because of their more "open" design, these heads have a reduced squish area. In the previous Weslake kidney-shaped combustion chamber the greater amount of squish causes the fuel/air charge to rebound around inside of the combustion chamber, creating eddies and flows, the characters of which are dictated by the force with which they are created and thus are RPM-dependent, making for very real preignition issues where ignition timing is concerned. The later "open" design of the combustion chamber with its reduced squish area reduces this problem. The 12H 2923 cylinder head castings have thicker walls around their intake ports that allow the diameter of the intake ports to be more easily and safely increased in size, plus relatively large bowl areas in their ports, thus making them the most sought-after cylinder head casting for oversize valve installations. If fact, the throats of their bowls are actually 1/8" (.125") larger than those of the earlier cylinder heads, and are actually larger than the 1.625" diameter intake valve for which they were intended, thus revealing the designer's original intention that they were to have potential for racing purposes with oversize intake valves. It should be noted that this 1.625" diameter intake valve is actually larger in diameter than the bore of the 1½" SU HS4 Series and 1½" SU HIF4 Series carburetors that were Original Equipment on the mass production MGBs, making them a good companion for use with 1½" SU HS4 Series and 1½" SU HIF4 Series carburetors that have been modified for increased flow capacity, as well as for use with oversize carburetors such as the 1¾" SU HS6 Series and the 1¾" SU HIF6 Series. These heads also introduced a larger-diameter port in its side for the heater valve, a feature that was continued on all later heads.

The fourth version of the cylinder head was the 12H 4736 cylinder head casting (BMC Part # 12H 4735) that was first introduced on the Austin/Morris Marina and also used on the 18V-846-F-H and 18V-847-F-H UK/European market MGB engines, all of which it was Original Equipment for. It reverted to the original smaller 89.2 gram 1.565" diameter intake valve size that was used on the first two cylinder head castings, but in redesigned form (BMC Part # 12H 4211), and had a somewhat improved intake port design that produced a 4% increase in flow at maximum valve lift, as well as offset oil feed in the rear rocker shaft pedestal (BMC Part # 12H 4737) in order to accommodate coolant passages that were redesigned in order to assist in preventing overheating of the exhaust valve of the rear cylinder. This redesign necessitated the relocation of the oil passage in the cylinder head as well as in rear rocker shaft pedestal, which means that if you should choose to install it onto an earlier engine block you are going to need the later rear rocker shaft pedestal (BMC Part # 12H 4737) that accommodates the offset oil port. It should be noted that for the North American Market this particular cylinder head was used solely on the 18V-836-Z-L and 18V-837-Z-L engines that were produced from September of 1974 through December of 1974 for the so-called "1974 ½" models.

The fifth version of the cylinder head (BMC Part # BHM 1062) was used exclusively for the North American Market can be readily identified by its casting number CAM 1106 and is commonly found on engine numbers 18V-797-AE-L, 18V-798-AE-L, 18V-801-AE-L, 18V-802-AE-L, 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L, and 18V-891-AE-L, all of which it was Original Equipment for. This cylinder head casting always had a special drilled port located on its top rear end for ducting hot coolant to a thermo-controlled automatic choke mechanism. It is sometimes referred to as the "lead-free" cylinder head casting as its valve seats were induction hardened in order to withstand the higher combustion temperatures of lead free fuel. This was made possible by adding 1% nickel to the molten iron prior to casting. This led to an issue wherein the iron that came into contact with the mould suffered an altered molecular structure. In order to deal with this issue, an additional depth of material was produced in specially modified molds that was machined away prior to the induction hardening process being applied to the valve seat area. This was a complex process and applied only to these cylinder head castings for the North American Market engines. However, one should be aware that once the valve seats are remachined, the valve seats will be no more tolerant of the higher combustion temperatures of lead-free fuel than those of the earlier cylinder head castings, and that lead-free fuel tolerant valve seat inserts should be installed.

All three of the later type cylinder head castings have a lower combustion chamber height of .375" and a combustion chamber volume of 39cc. Being of kidney-shaped "Open" design and featuring a larger squish (quench) area, as well as a reduced promontory between the valves, they are a considerable refinement of the earlier heart-shaped "closed" design. These cylinder heads can be identified by their casting numbers that are to be found on the top deck of the cylinder head, underneath the rocker arm cover. With the exception of the

12H 2923 cylinder head casting, these new cylinder head castings not only had larger coolant passages at the rear of the cylinder head, they also introduced redesigned coolant passages between the intake and exhaust valves with greater surface areas in order to assist in dealing with the higher combustion chamber temperatures that resulted from efforts to reduce exhaust emissions. In addition, the extra material provided created both the indentations behind the spark plug holes as well as the mounting bosses provided for the installation of air injectors for the exhaust ports on these cylinder head castings, along with a shelf on the edge of the casting on the same side, which had both the additional benefit of making them more resistant to cracking between combustion chambers #2 and #3, and the blowing of cylinder head gaskets due to warpage.

In view of the fact that many engines have been previously rebuilt and have had their cylinder heads replaced with another that is not of the original type, the following table should prove useful in identifying the specifications of your cylinder head:

Engine	Cylinder Head Part Number	Cylinder Head Casting Number	Combustion Chamber Volume	Intake Valve Diameter	Exhaust Valve Diameter
18G	48G 318	12H 906	43cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18GA	48G 318	12H 906	43cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18GB	48G 318	12H 906	43cc	1.5625" (39.6875mm)	1.344" (34.1376mm)

18GD	12H 905	12H 1326	43cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18GF	12H 905	12H 2389	43cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18GG	48G 318	12H 1326	43cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18GH	48G 538	12H 2389	43cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18GH	48G 538	12H 2389	43 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18GJ	48G 538	12H 2389	43 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18GK	48G 538	12H 2389	43 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 581	12H 2708	12H 2709	39 cc	1.625" (41.275mm)	1.344" (34.1376mm)
18V 582	12H 2708	12H 2709	39 cc	1.625" (41.275mm)	1.344" (34.1376mm)

18V 583	12H 2708	12H 2709	39 cc	1.625” (41.275mm)	1.344” (34.1376mm)
18V 584	48G 644	12H 2923	39 cc	1.625” (41.275mm)	1.344” (34.1376mm)
18V 585	48G 644	12H 2923	39 cc	1.625” (41.275mm)	1.344” (34.1376mm)
18V 672	48G 644	12H 2923	39 cc	1.625” (41.275mm)	1.344” (34.1376mm)
18V 673	48G 644	12H 2923	39 cc	1.625” (41.275mm)	1.344” (34.1376mm)
18V 779	12H 2708	12H 2709	39 cc	1.625” (41.275mm)	1.344” (34.1376mm)
18V 780	12H 2708	12H 2709	39 cc	1.625” (41.275mm)	1.344” (34.1376mm)
18V 797	BHM 1062	CAM 1106	39 cc	1.5625” (39.6875mm)	1.344” (34.1376mm)
18V 798	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344” (34.1376mm)

18V 801	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 802	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 836	12H 4735	12H 4736	39 cc	1.625" (41.275mm)	1.344" (34.1376mm)
18V 837	12H 4735	12H 4736	39 cc	1.625" (41.275mm)	1.344" (34.1376mm)
18V 846	12H 4735	12H 4736	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 847	12H 4735	12H 4736	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 883	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 884	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 890	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)

18V 891	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 892	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)
18V 893	BHM 1062	CAM 1106	39 cc	1.5625" (39.6875mm)	1.344" (34.1376mm)

If you are going to have professional headwork done, specify the earlier large intake valve size (1.625" / 41.275mm diameter) and use either the 12H 4735 or the CAM1106 cylinder head casting of the later 18V engines, as it is preferable due to its revised coolant passages giving cooler running characteristics under conditions of high power output. These cooler running characteristics assist in maintaining concentricity between the valve guide in the cylinder head and the head of the valve on its valve seat, making it the most likely candidate for a three-angle valve seat with a 30º sealing configuration. It will be necessary to fabricate a blanking plate in order to seal off the outlet for the water choke fitting exclusive to the CAM1106 cylinder head casting.

Be aware that due to their shallower combustion chambers, if either of these two later cylinder heads are fitted onto the engine block of any pre-18V engine (18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines), then it will be necessary to machine counterbores into the deck of the engine block in order to prevent the exhaust valves from hitting the engine block, just as on the 18V engine blocks. This is also the case if a high lift camshaft lobe profile is to be employed. These deeper counterbores will need to be cut with a 1 17/32" (1.53125") diameter end mill. They need to have a 1/16" (.0625" / 1.59mm) radius at their edges and be recessed in order to provide at a minimum a 1/16" (.0625" / 1.59mm) clearance when the valve is at full lift. When calculating the needed depth of the counterbores, be aware of the amount of valve lift produced by your chosen camshaft, and the fact that that the height of the combustion chambers employed in the 12H 906 and 12H

2389 cylinder head castings was .425” (10.795mm), while the height of the combustion chambers employed in the 12H 2923, 12H 4736, and CAM 1106 cylinder head castings was .375” (9.525mm). The depth of the counterbores in the deck of the engine block should not exceed .200” (5.08mm) below the original height of the deck; otherwise, the combustion flame will be directed onto the top ring of the Original Equipment style pistons, resulting in severe ring damage and ring land breakage on the pistons.

43cc Combustion Chamber			
Camshaft	**Valve Lift Intake / Exhaust**	**Combustion Chamber Depth**	**Required Valve Counterbore***
Piper BP255	.405” / .389” 10.287mm / 9.8806mm	.425” (10.795mm)	.0425” (1.0795mm)
Piper BP270	.405” / .403” 10.287mm / 10.2362mm	.425” (10.795mm)	.0425” (1.0795mm)
Piper BP285	.445” / .445” 11.303mm / 11.303mm	.425” (10.795mm)	.0825” (2.0955mm)

* Presumes an Original Equipment combustion chamber height position of .425” (10.795mm) of a cylinder head that has not been skimmed and that the face of valve head is flush with the roof of the combustion chamber.

39cc Combustion Chamber			
Camshaft	**Valve Lift**	**Combustion**	**Required Valve**

	Intake / Exhaust	Chamber Depth	Counterbore**
Piper BP255	.405" / .389" 10.287mm / 9.8806mm	.375" (9.525mm)	.0925" (2.3495mm)
Piper BP270	.405" / .403" 10.287mm / 10.2362mm	.375" (9.525mm)	.0925" (2.3495mm)
Piper BP285	.445" / .445" 11.303mm / 11.303mm	.375" (9.525mm)	.1325" (3.3655mm)

** Presumes an Original Equipment combustion chamber height position of .375" (9.525mm) of a cylinder head that has not been skimmed and that the face of valve head is flush with the roof of the combustion chamber.

In the case of all of these cylinder head designs, the two coolant holes at the rear should be checked for size and, if necessary, enlarged to a diameter of 9/16" (.5625") in order to maximize coolant flow past the rear cylinder and through the cylinder head, the two corresponding holes at the rear of the deck of the engine block on 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines being correspondingly enlarged to match (an old MG Factory Race Team trick). This modification is incorporated into the design of both the engine block and cylinder heads of the 18V engines. Although the coolant passages located in the deck of the engine block are of a smaller diameter than those that mate to them in the cylinder head, do not be tempted to enlarge them. They are larger in order to compensate for a possible off-center mounting resulting from the extra play allowed by the cylinder head stud passages in the cylinder head. If they are enlarged, the coolant system will be short-circuited, the rear cylinders receiving inadequate coolant flow.

Aside from matching the weights of the reciprocating components and independent dynamic balancing of both the crankshaft and the flywheel, perhaps one of the best ways to create a smooth engine is to equalize the compression and thus the power impulses occurring inside of each cylinder. Once the throws of the crankshaft have been indexed, and the lengths of its throws and the lengths of the connecting rods have been properly matched, this can be accomplished by making sure that the combustion chambers are of equal volume so that the compression ratio inside of each of the cylinders will be the same. After the cylinder head has been skimmed flat, the volume of each combustion chamber can be measured by using a clear piece of sheet plastic with a small hole drilled in it. Simply put a bead of chilled grease around the edge of a combustion chamber and press the plastic down onto it so that the grease forms a seal. Using a syringe or an eyedropper with a scale of measurement on it, carefully fill each combustion chamber with light oil, keeping a record of how much is necessary to fill each one. A useful pictorial description of this procedure can be found at http://www.mintylamb.co.uk/?page=measurecc.htm .

Next, use a Dremel tool to gently remove small amounts of metal from the smallest combustion chamber. Work slowly, making repeated checks. For small-bore engines (+.040 or smaller), the walls of the combustion chamber should be kept perpendicular to its roof in order to ensure the best squish (quench) characteristics. The roof of the combustion chamber should be flush with the valve seat and reasonably flat in order to ensure the best airflow characteristics. Finishing can be done with a sanding disc, care being taken to not undercut or groove the base of the wall where it adjoins the roof. This juncture should have a generous radius in order to both permit smooth airflow and to discourage the formation of carbon deposits that can lead to preignition. Because the roof of the combustion chamber is very thin, having coolant passages above it, remove no more material than is absolutely necessary in order to achieve your goal. Do not polish either the walls or the roof of the combustion chamber in an attempt to discourage carbon buildup as this will lead to condensation of the fuel / air charge both as it enters the cylinder and as it is being compressed. A glass-beaded finish will produce sufficient border turbulence to do nicely in terms of discouraging not only this problem, but that of the development of surface cracks as well.

Equalizing the volumes of the combustion chambers has a long-term benefit as well. When performing a compression test, there should be no more than a ten percent variation between the compression figures of the cylinder with the highest compression and the cylinder with the lowest compression. By eliminating the variable of differing combustion chamber volumes, the compression figures can be considered to be a more reliable indicator of the state of the affairs that relate to compression conditions.

Upon examination, there is always a carbon-free area on the chamber walls around the intake valve. This signifies that fuel has condensed in that area and has literally "washed down" the walls of the combustion chamber during the intake stroke. This absence of carbon also evinces a lack of combustion in that area of the combustion chamber. The solution to this problem is modification of the combustion chamber in order to unshroud the intake valve.

Unshrouding the valves of the B Series engine is a tricky business, bordering on being a black art. Theoretically, the adjacent combustion chamber wall should be about 50% of the radius of the intake valve distant from the edge of the intake valve and 40% of the radius of the exhaust valve distant from the edge of the exhaust valve. However, due to the close proximity of the combustion chambers to one another and the undersquare configuration of the cylinders, this theoretical ideal cannot be attained. As a result, some rather artful techniques have to be employed in order to attain the desired airflow.

Depending on the size of the intake valve, bore diameter, camshaft lobe specifications, and intended use of the engine, sections of the adjacent walls of the combustion chamber, particularly in the vicinity of the intake valve, may require an angle progressively increasing from 7^o to 14^o. Their juncture with the roof of the combustion chamber must be properly radiused. Although effecting squish (quench) characteristics, this angled orientation of the walls of the combustion chamber is advantageous because when the valve is near its valve seat, the close base of the wall of the combustion chamber does not cause airflow restriction. A further increase of the volume of the combustion chamber would both not only decrease the compression ratio, but it would also become a contributing factor to preignition by reducing squish (quench) turbulence. In addition, a further increase of the volume of the combustion chamber would also present a grave risk of accidentally breaking through into a coolant passage. This angled wall modification having been done, it is only when the valve opens further that the close proximity of the wall of the combustion chamber interferes with airflow. Care must be taken that in any attempt to unshroud the intake valve you do not attempt to remove too much material from the combustion chamber wall nearest to it as this can lead to breaking into a coolant passage.

Unshrouding is obviously best left to experts who are well familiar with the hazards involved, and what benefits can be expected from which modifications. Instead, confine your work and simply remove material evenly from the roof of the combustion chamber. As you remove material, measure the volume of the combustion chamber repeatedly until it matches that of the largest one. Repeat this process on all of the combustion chambers until their volumes are matched. You should now have equal compression on all four cylinders, making for a smoother engine.

Glass beading to relieve stress works on machined surfaces to reduce subsurface stresses that result from the machining process, as in the case of the fillets of crankshaft journals, but will do nothing deep inside of a casting, which is essentially a bunch of holes held together by metal. Glass beading a casting essentially simply compacts the surface, creating a density differential that will prevent subsurface cracks from merging with the surface. The cracks will still be there inside of the casting, but they are less likely to get through to the surface. This is the reason why so many race engine builders glass bead their combustion chambers, crankshaft fillets, and connecting rods.

Custom Headwork

One crucial bit of advice about Do-It-Yourself cylinder heads: Be Careful! Once you remove metal, you cannot put it back. To use a Dremel tool with a flap sander attachment to smooth the existing contours is one thing, but to alter the contours with a grinding stone or a rotary file is entirely something else. Peter Burgess gives some crude drawings and simple instructions in his book “How to Power Tune MGB 4-Cylinder Engines” and says that you can do it yourself, but a highly practiced Master of the Art often forgets how difficult it is for a rank beginner. He gives a much fuller and more detailed description of what is actually involved in his later book “How To Build, Modify, and Power Tune Cylinder Heads” which should be carefully read and fully understood prior to deciding to set out on such a venture. Remember, the B Series cylinder head is special. Siamesed ports are an antiquity in this modern era of crossflow cylinder heads with separate ports, and there are very few people who truly understand the subtleties of them. This is no Ford or Chevrolet V8 cylinder head we are talking about here! Serious work on these cylinder heads entails specialized knowledge and skills. Just the process of removing the valve guide bosses is very tricky due to the fact that the difference between removing just enough metal and breaking into one of the coolant passages is very, very small. If you do not have genuine blueprints of the ports and the coolant passages that are inside of the particular cylinder head casting that you are working on (there were five cylinder head castings that were used on the North American Market engines alone), complete with dimensions, radiuses, etc., and the appropriate precision measuring tools, then you are taking a big gamble with all of the odds stacked heavily against you. You will need a flowbench, too. This machine equipped with sensing probes draws ambient temperature air in through the intake ports and blows combustion temperature air out through the exhaust ports. It is a must-have for getting the airflow rates of the ports individually matched at all points of valve lift at all engine speeds within the engine's operating range. If the airflow rates for each of the ports are not equal under all circumstances, the result will be differing fueling requirements for the cylinders. This is a serious problem for an engine in which two pairs of cylinders must each share the same fuel metering device. In addition, with differing quantities of air entering into each cylinder, the Effective Compression Ratio (ECR) of each cylinder will also correspondingly differ, making for rough running.

Many well-intentioned local Good 'Ol Boy Hot Rod Engine Builders (the ones that the local pimply Hot Rodders call "experts") have reduced MGB cylinder heads to scrap metal. Once this happens you will spend at least as much money buying another cylinder head and getting the needed parts for it as you would have spent shipping the cylinder head to a qualified professional, having him do the work, and then shipping it back again, complete with insurance. The one thing that you cannot cheapo your way through on an engine is the headwork. Without access to a flowbench, blueprints, precision measuring instruments, a good working knowledge of the mysteries of siamesed ports, and the specialized manual skills, the likelihood of an amateur doing it correctly on a first attempt is so small that it makes me shudder. How do I know? About twenty-nine years ago I worked for Rockwell International making valves for use in nuclear power plants. The valves had to be flowed on a bench in order to be government certified for use in a nuclear installation. This meant custom contouring work on their ports, all done by hand with a die-grinder-type Dremel tool. It took about three years of prior experience and a practiced eye to be able to do it right every time, and this was working daily for eight to ten hours with a flowbench, repeatedly making small contour corrections on every individual port! Recontour siamesed ports on my garage workbench? Hey, my name is not Peter Burgess! Ship the cylinder head to Peter or purchase one from him outright, you will be glad you did. After all, you would not try to bore your cylinders in the garage with a file, would you?

Peter offers multiple levels of headwork suitable for an easily streetable engine: Standard Leadfree, Econotune, Fast Road, and Fast Road Big Valve. In all of his cylinder heads the exhaust valve is the Original Equipment 1.343" (34.1122mm) size and all of the valve seats are cut using three angles with a 45° valve sealing area.

The first and simplest is his Standard Leadfree specification which features manganese silicone bronze valve guides to aid in heat removal, Austenitic 214N stainless steel exhaust valves, EN52B alloy intake valves, lead-free fuel compatible exhaust valve seat inserts, and "top hat" style intake valve stem oil seals.

The second level is his Econotune specification that adds his own custom intake valve guides that have been tapered (bulleted), plus the combustion chambers and valve throats are all modified to enhance the flow of the fuel / air charge and smooth combustion. Neither the valve nor port sizes are increased, thus the resulting high port and valve seat velocities produce a broad spread of very useable power from idle to a maximum power output peak at around 4,800 RPM. This results in an increase in power output of approximately 30% at 3,000 RPM, and maximum power is increased by approximately 18% at 4,800 RPM.

The third level is the Fast Road specification in which the cylinder head is fully reworked prior to the lead-free fuel compatible valve seats and Peter's custom tapered (bulleted) manganese silicone bronze valve guides being fitted. The intake and exhaust ports are modified to enhance the flow of the fuel / air charge without increasing the port sizes to any great extent. This keeps the port velocities high and aids in the production of low-end torque. The increase in power output from idle with a gain of approximately 25% at 3,000 RPM, as well as a maximum increase of power output of approximately 30% at 5,200 RPM when equipped with an Original Equipment camshaft and less restrictive K&N air filters. Beyond that point, the power will fall off much more gradually than with a Original Equipment specification cylinder head, so you can say good-bye to that frustrating "after-that-the-engine-seemed-to-run-into-a-wall" experience. If you add a less restrictive Peco exhaust system, it will extend the peak further (to about 5,500 RPM) with yet more power which afterward will decline much less precipitously afterwards. The Fast Road cylinder head also takes beautifully to a Piper BP270 camshaft, the combination sacrificing a little power down very low in the powerband where you rarely go anyway (below 2,000 RPM) and singing merrily all the way to 6,000 RPM. As you can see, the Fast Road cylinder head should be considered to be the jumping-off point when it comes to a quest for really serious power. It is the essential foundation that everything else is built on. To do it last is putting the horse behind the cart. This specification of cylinder head performs well with an Original Equipment camshaft, and shows even more impressive gains with either the Piper BP270 or the Piper BP285 camshaft. While the cylinder head works extremely well with the standard twin 1½" SU's, it will also show worthwhile gains at high engine speeds with either modified 1½" SU's or twin 1¾" SU's. The Piper BP285 camshaft is recommended to compliment this increase in carburetion.

The fourth level, the Fast Road Big Valve cylinder head, features larger 1.67" (42.418mm) diameter intake valves and is ideally suited to a Fast Road camshaft such as the Piper BP285. The increased breathing capacity of the cylinder head will show good returns with either a set of twin 1¾" SUs or a Weber 45 DCOE. When complimented with a Big Bore engine conversion the cylinder head is well suited to restore the engine speed at which peak horsepower is achieved to its original position in the powerband. The increase in power output is approximately 25% at 3,000 RPM and 35% at 5,300 RPM when used with an Original Equipment camshaft and K&N air filters.

Peter offers a fifth option, which, although not falling into the "easily streetable" category, is mentioned here only for the sake of completeness. The Fast Road Plus cylinder head is fully modified and is fitted with one piece 214N Austenitic stainless steel tuftrided 1.72" (43.688mm) diameter intake valves. The combustion chamber walls are dressed back in order to unshroud the valves, thus increasing the flow of the fuel / air charge, and the intake ports are very slightly increased in size in order to allow the engine to rev out more. The cylinder head has been developed for very fast road use with "hairy" camshafts and larger venturi carburetion. Although not really suitable for Original Equipment / mild camshaft use, the increase in power output is approximately 25% at 3,000 RPM and approximately 38% at 5,300 RPM when used with an Original Equipment specification camshaft and K&N air filters. It is highly appropriate for meeting the needs of high-power-output Big Bore engines.

Peter offers these cylinder heads on both an exchange basis wherein you ship your cylinder head to him for modification, as well as an outright purchase wherein you purchase an already finished cylinder head without shipping yours to him. For those faced with transatlantic shipping charges or for those whose cylinder heads are irreclaimable, the latter is often the most economical approach.

Unlike some individuals who tune cylinder heads, Peter warranties every one that he modifies. Peter stands proudly by his work and appreciates the difference between goodwill and guarantee.....and his goodwill extends a long way. So much so that if you supply the cylinder head casting and the cylinder head cracks within two years of his modification, then he will modify another one for you free of charge, even including free delivery, if you will supply another cylinder head casting for him to modify. On the other hand, if he supplies the cylinder head from his own stock and it cracks within two years of his modification, then he will modify another one from his own stock for you free of charge, including free delivery.

If you do not want the extra expense of professional porting, remove your old valve guides, and then spray the ports with machinist's bluing so that you will be able to see what you are actually removing. Use a Dremel tool with a #80 grit, and then a #100 grit flap sander in order to gently smooth the existing port contours, removing the typical turbulence-inducing lumps and bumps that are a result of the casting process. Do not be surprised if progress on the exhaust ports takes longer than it does on the intake ports. Because the exhaust ports only see hot exhaust temperature gas, whereas the intake ports are cooled by the incoming fuel / air charge, they will have become carbon-case-hardened. A mirror finish can be advantageous in reducing future carbon build-up in the exhaust ports. However, a mirror finish on the intake ports and the combustion chambers is not only unnecessary, but is actually undesirable because it will eliminate border turbulence along these surfaces, thus leading to fuel condensation and a consequent loss of power. Be sure to carefully blend the port to the valve seat in order to remove any steps. Do not yield to the temptation to knife-edge the port divider thinking that such a modification will improve the flow of the fuel / air charge. The opposite will be the result. It is a common misconception that the port divider exists to channel the flow of the incoming fuel / air charge into two streams that each flow to its own intake valve. However, most of the time only one intake valve is open, thus no such channeling is occurring. A knife-edge at the nose of the port divider would actually serve to inhibit the transfer flow of any residual fuel / air charge carried by inertia into the port of the closed valve on into the port of the open valve. Air does not like to flow around sharp angles, and thus flow-inhibiting turbulence is created.

That is why a three-angle valve flows better than a single-angle one does. A smooth radius at the nose of the divider will actually improve its transfer flow performance.

Have installed at equal depth lead-free fuel compatible three-angle valve seats and three-angle 214N alloy Austenitic stainless steel valves with chrome plated stems and stellite tips (do not panic, they may sound exotic, but they are easily obtained and not very expensive), tapered (bulleted) valve guides, a set of the highly superior Fel-Pro Teflon-lined valve stem seals on the intake valves, 6” diameter X 3¼” deep K&N air filters, 1½” SU carburetors with richer fuel-metering needles, and a 1¾” Peco exhaust system. This will get you started with a relatively small investment and you will be both surprised and impressed at the improvement. The richer fuel-metering needles are needed as the Original Equipment airfilter housings restrict airflow enough to create a pressure drop downstream of the air filters. This pressure drop results in a pressure differential between the atmosphere above the fuel jet and the ambient pressure inside of the float bowl that causes more fuel to flow through the main fuel jet. The greater vacuum inside of the vacuum chamber (dashpot) also results in the vacuum piston rising to a higher level, thus the fuel-metering needle will be at a higher, richer stage. When the restrictive airfilter housings are eliminated, you will have a smaller pressure differential and thus less fuel flowing from the fuel jet to mix with the increased airflow, hence the need to introduce more fuel into the carburetor in order to maintain the proper fuel / air ratio.

Aluminum Alloy Cylinder Heads

An item that has gained some acceptance amongst the racing crowd is the aluminum alloy cylinder head. These expensive items shave about twenty pounds off of the weight of the engine and tend to run cooler under the high stresses of racing. Their rapid transference of heat also helps in preventing the development of “hot spots” which can cause preignition under the thermal conditions generated by heavy loads. Because the cast iron of the engine block and the aluminum alloy of the cylinder head have different expansion rates, the use of a high quality resin type cylinder head gasket should be considered to be mandatory. Whenever an engine exceeds its normal operating range and overheats, the elevated temperatures can cause extreme stress in the cylinder head that may result in a cylinder head gasket failure. This is especially true in the case of an aluminum alloy cylinder head mounted onto cast iron engine block because aluminum alloy has a coefficient of expansion that is about two to three times greater than that of cast iron. When combined with the added stresses induced by overheating, this difference in thermal expansion rates between an aluminum alloy cylinder head and that of a cast iron engine block can cause the cylinder head to warp. This, in turn, may lead to a loss of clamping force in critical areas and allow the cylinder head gasket to leak. In addition, since the thermal expansion rate of aluminum and cast iron are greatly different, and aluminum expands much more, it makes the role of the cylinder head studs very critical in maintaining correct clamping pressures on the cylinder head gasket. In view of the fact that aluminum alloy expands and contracts much more than cast iron during thermal cycling, stress on the cylinder head studs is correspondingly greater, shortening their effective lifespan and thus requiring their replacement every time that the engine is rebuilt. They also require that washer-like steel shims or steel valve spring collars be placed under the springs in order to protect the aluminum alloy material of the cylinder head from being galled by the springs. Be aware that aluminum alloy cylinder heads should be torqued only when cold, and to no more than 38 to 40 Ft-lbs.

Note that whenever you install spark plugs into an aluminum alloy cylinder head, you will want to be sure to smear some antisieze compound onto the threads of the spark plugs. This will prevent the corrosion of the aluminum alloy threads that results from the electrolytic interaction between the steel of the spark plug threads and the aluminum alloy threads of the cylinder head, which can cause them to seize in place, as well as make the spark plugs much easier to remove.

Unfortunately, the poor thermal efficiency of aluminum alloy forces the use of an increased compression ratio of about one point in order to produce the same amount of power. This, coupled with the obsolete kidney-shaped combustion chambers in turn creates a problem with preignition when running on the gasoline available at a gas station unless frequent and considerable attention is paid to the maintenance of precisely correct ignition and carburetor settings. This is due to the fact that the power stroke takes place at the same speed as the compression stroke. Put crudely, an “Octane” rating is merely an indication of the fuel’s resistance to combustion. This means that it can be compressed to a higher ratio within the same timespan without the heat generated by compression resulting in the initiation of combustion (preignition). The more rapid heat transfer capability of aluminum alloy allows a faster rise in pressure without preignition occurring because some of the heat resulting from compression is conducted away from the combustion chamber by the more rapid heat conductivity factor of aluminum alloy. The caloric value of the fuel remains the same, so power output at a given compression ratio is greater with a cast iron cylinder head than with an aluminum alloy cylinder head due to cast iron's reduced heat loss through the roof of the combustion chamber. This is why it is necessary to boost the compression ratio by about one point in order to attain the same power output when substituting an aluminum alloy cylinder head for a cast iron cylinder head. That is, a Geometric Compression Ratio (GCR) of 9:1 when using a cast iron cylinder head must be increased to 10:1 when using an aluminum alloy cylinder head, but the needed octane rating of the fuel can remain the same. A beneficial side effect of this increase in compression ratio will be a minor improvement in fuel economy. In addition, careful attention to the coolant system is a necessity as an aluminum alloy cylinder head has a severe tendency to warp should it overheat. If this should occur, the temper of the alloy will have been ruined and thus the torque settings of the nuts of the cylinder head studs will not hold, the cylinder head having become just so much scrap metal. You do not get something for nothing!

Aluminum alloy cylinder heads come in two basic types: an aluminum alloy version of the Original Equipment cast iron cylinder head, and a seven-port crossflow cylinder head design. Unless you are replacing a cracked cylinder head or building an engine that employs a hot camshaft such as the Piper BP285 and requires the use of high compression pistons, there is little practical advantage to the extra expense of using the aluminum alloy five-port design in a street application. At present, there are two versions of this design. One is the American-made cylinder head offered by Pierce through multiple aftermarket suppliers such as Brit Tek, Moss Motors, and Victoria British. The other is the UK-made George Edney cylinder head available exclusively from Brown & Gammons, the latter having a superior port design that has significantly more tuning potential.

Crossflow Heads

With its independent intake port design, the crossflow cylinder head has greater performance potential, but will require the additional expense of special tuning by a professional, a pair of either highly modified 1½” SU HS4 Series or 1½” SU HIF4 Series carburetors, or a pair of larger 1¾" SU HS6 or 1¾" SU HIF6 carburetors, a larger-diameter custom intake manifold, or, preferably, either dual Weber DCOE or Dellorto DHLA carburetors as well as a pair of suitable intake manifolds designed for them (they both use the same intake manifolds), as well as a pair of K&N air cleaners and their K&N custom housings in order to fully exploit that potential. Due to the extra stresses on load-bearing surfaces resulting from the increased power output that is possible with this modification, the installation of a higher-flow oil pump is advisable in order to help cool and protect the bearings from the increased pounding. Be aware that the carburetors will overhang the distributor, so the fuel lines must be leak-free and conversion to solid-state ignition triggering in order to eliminate the maintenance involved with the contact breaker points is highly advisable. In addition, the removal of an Original Equipment oil filter will be transformed into a memorable experience, requiring removal of both the air filters and their boxes, and possibly the carburetors and their intake manifolds as well. While this might prompt thoughts of converting to a downward-hanging oil filter stand (BMC Part # 12H 4405) from a Morris Marina, unfortunately, that stubby oil filter (BMC Part # 12H 4405GFE148) just does not have the oil flow capacity required for such an engine. In a truly fine-straining oil filter the reduced surface area of its filtration element would actually restrict the oil flow. MG tried installing this item on its B Series engines for the MGB from December of 1973 to February of 1974, only to discover the problem and find that they had to switch back to the previous inverted oil filter stand. Instead, a remote oil filter similar to that used in an MGB GT V8 would be a quite adequate solution. This conversion will require the installation of a simple spin-on bypass cover with ½" NPT threads (Summit Racing Part # TRD-1013) in place of the oil filter and a single remote oil filter bracket with ½" NPT threads (Summit Racing Part # TRD-1045). This will allow you to conveniently continue to use either your choice of standard MGB oil filters or larger capacity units. Summit Racing has a website that can be found at http://www.summitracing.com/ . Special-length oil lines will have to be custom-fabricated with ½"-14 BSP threads on the fittings for both the oil outlet at the rear of the engine block and at the oil cooler. Be aware that these are not compatible with any American threading system as they are based on the British Whitworth 55° thread. Just to confuse matters a little more, the current designation for a BSP thread is a "G" thread!

Currently there are two crossflow cylinder head designs available: The Pierce MSX and the HRG Derrington. Contrary to popular belief, The Pierce MSX cylinder head is not a true replica of the Derrington cylinder head, the production rights of which currently belong to its present manufacturer, George Edney. Perhaps it would be best to describe the Pierce MSX cylinder head as a modified copy of the Derrington design that has been altered enough to get around the patent laws in order to avoid a lawsuit. Unfortunately, the design changes result in a product whose performance potential is inferior to that of the original Derrington design. The Pierce MSX cylinder head is supplied with hardened lead-free compatible intake valve seats that are appropriately sized for 1.5625" (39.751mm) diameter intake valves. Unfortunately, their older-version cardioid-shaped combustion chambers will not allow the use of intake valve sizes larger than 1.625" (41.275mm) diameter. Due to shrouding problems within the heart-shaped "closed" combustion chamber, the 1.625" (41.275mm) diameter intake valve size is not recommended. In order to make use of a 1.625" (41.275mm) diameter or larger intake valve it would be necessary to both enlarge the port throats and modify the combustion chambers in order to accomplish the necessary unshrouding of the intake valves. This would result in the valve seats becoming so thin that they would lack structural stiffness, warpage thus becoming a perilous risk. Should warpage occur without the valve seat separating from its recess in the cylinder head, the roof of the combustion chamber would in all likelihood crack, the cylinder head then becoming just so much scrap metal. If a valve seat were to separate from its recess in the cylinder head, then the results would be even more severe. If the intake valve happened to be of one-piece construction, both the valve and the cylinder head would be ruined. If the valve happens to be of two-piece construction, it is likely that the head of the valve would separate from its stem and both it and its valve seat would then drop into the cylinder, destroying both the cylinder head and the piston, as well as possibly gouging the wall of the cylinder and bending the connecting rod. Damage to the crankshaft would then consequentially become a very real possibility. Obviously, if a 1.625" (41.275mm) diameter intake valve were to be employed, it would be necessary to machine out the existing hardened valve seats, machine new valve seat counterbores, and then install the necessary larger-diameter hardened valve seats. The exhaust valves are Original Equipment 1.344" (34.1375mm) diameter. The combustion chambers of the Pierce MSX cylinder head are the same 39cc Weslake design as the Original Equipment five-port cast iron cylinder heads of the North American Market version of the 18V engines whose design was configured to direct the flow of the incoming fuel / air charge toward the spark plug. Unfortunately, the incoming fuel / air charge of the Pierce MSX cylinder head is entering from the opposite direction than in the original

Weslake design without any modifications included in order to compensate for this fact. It is important that upon installation of the Pierce cylinder head that either the later MGB cylinder head gasket (BMC Part # GEG377) or, better yet, a Payen cylinder head gasket be used. Do not use the original Clough & Wood style cylinder head gasket (BMC Part # GEG366) as it is inappropriate for use with an aluminum alloy cylinder hear / cast iron block combination..

The original First Generation MK I Derrington cylinder head was patterned loosely on the original 1500cc B Series cylinder head used in the MGA, making use of the same 33cc combustion chamber. For the Mk II version, more modern combustion chambers and larger ports were used to accommodate larger valves that were increased to the size as those used in the 1622cc B Series engine (1.5625” (39.751mm) diameter Intake, 1.344” (34.1375mm) diameter Exhaust). These valve sizes were continued for the MK IV version intended for the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK 1800cc B Series engines and used their 43cc Weslake combustion chambers. This version is often referred to as “The Late Derrington Cylinder head” by Derrington aficionados. The cylinder head stayed like this for years until Peter Burgess was brought in to complete the development of what has come to be known as the Second Generation Derrington cylinder head. This new design outwardly resembled the old Mk I First Generation Derrington cylinder head (as we call it today), but internally it was radically different. The coolant passages were enlarged and the port contours took advantage of the latest understanding of airflow and the technology surrounding it. In addition, the design of the combustion chamber was a complete departure from what had been used before.

Amongst other differences, the ports of the Derrington cylinder head are of notably better design than those of the Pierce MSX, having a more generous bottom radius where they curve into the throats of the intake ports, thus making them easier to modify for optimizing the flow of the fuel / air charge. Although the combustion chambers of the Derrington cylinder head appear similar at first glance, they incorporate certain features of the “bathtub” type combustion chamber design in order to compensate for the relocated position of the four independent crossflow intake ports, as well as to create better charge turbulence and more efficient combustion. The curvature of the spark plug side of the combustion chamber is recontoured outwards to decrease squish (quench) on that side of the combustion chamber while the opposite wall is relocated closer to the valves in order to increase squish (quench) on the opposite side, the consequent change of balance resulting in the direction of the compressed fuel / air charge toward the spark plugs, thus producing more efficient flame propagation. This relocation also allows the promontory of the opposing wall to be minimized, thus reducing its shrouding effect on the valves and in turn benefiting the flow of the fuel / air charge. The spark plugs are relocated closer to the exhaust valve in order to prevent the advancing pressure wave generated by the combusting fuel / air charge from compressing the unburned fuel / air charge into the vicinity of the hot exhaust valve at maximum velocity and thus causing detonation. The slope added to the spark plug side of the combustion chamber also deflects the mass of the pressure wave towards both the piston crown and the cooler intake valve, as well as deflecting the vertically swirling fuel / air charge towards the spark plug prior to ignition. As a side benefit of this redesign, the larger 43cc combustion chambers are also deeper, thus permitting them to better accommodate higher-lift camshafts. The quality of the casting is obviously superior in the Derrington cylinder head. The heater valve's hot water takeoff port of the Derrington cylinder head is in the same position as that used in the Original Equipment cylinder head, while that of the MSX cylinder head is at the rear. There are other minor detail differences, but these are the most significant ones.

Obviously, because of the provision of a separate intake port for #3 cylinder, neither of them has provision for mounting the Original Equipment Heater Valve. However, this drawback can be overcome by simply installing a threaded "L" tube into the water port in the cylinder head beneath #3 Intake Port on the HRG Derrington cylinder head or in the water port on the rear of the MSX cylinder head, and running a hose (flexible pipe) to a 1968 model MGC heater valve (Victoria British Part #2-378) mounted with its bracket (Victoria British Part # 12-4808) on the heater box, just like a 1968 MGC. The right hand spigot (as seen from the front) of the heater core will need to be shortened to mount the heater valve onto the heater box. An MGC heater cable (Victoria British Part # 6-7985) can then be run from the dashboard control knob. The MSX cylinder head is available with a custom heater valve. Logically, for the most efficient coolant flow, the position of the intake port for the heater matrix should be higher than that of its outlet port so that natural convection (the temperature of the coolant reduces as the heater matrix transfers the heat to the air) will aid the pumped flow, but in fact it is the reverse, i.e., heated coolant comes out of the heater valve on the cylinder head, into the intake port of the lower end of the heater matrix, then flows upwards through the heater matrix, and out of the outlet port of the top end of the heater matrix to the lower radiator hose (flexible pipe) in order to be circulated through the engine by the coolant pump. This method of circulation is employed in order to reduce the problem of an air pocket forming inside of the heater matrix as it forces out any air in the heater system.

All other factors being equal, an unmodified Second Generation HRG Derrington cylinder head produces 15% more power @ 3,000 RPM than that of an unmodified cast iron five-port early 18V cylinder head with a 1.625" (41.275mm) diameter intake valve. In as-cast condition, its airflow at high engine speeds is roughly equal to that of a fully reworked five-port cylinder head. Reworking of the cylinder head by a professional yields proportional improvements in power output directly comparable to those of an Original Equipment cast iron cylinder head reworked in the same manner without sacrificing any of the Derrington cylinder head's advantage in midrange torque production. Once the cylinder head has been fully modified, the current version of the HRG Derrington crossflow cylinder head produces 40% more power @ 3,000 RPM than that of an unmodified Original Equipment cast iron five-port early 18V cylinder head with a 1.625" (41.275mm) diameter intake valve. Even a professionally reworked five-port cylinder head cannot match this without the forced induction supplied by a supercharger. Dual Weber DCOE 45 carburetors seem to meet its needs best, probably as a result of the plethora of fuel metering jets that are available for them. Interestingly, both the five-port cylinder head and the HRG Derrington cylinder head will out flow the early versions of the MSX cylinder head when fully reworked, due to the MSX cylinder head utilizing the smaller valves of the 1622cc MGA engine!

Yet another item that has become popular with the racing crowd is the cast aluminum alloy rocker arm cover. These are normally deeper than the Original Equipment rocker arm cover in order to better accommodate high-ratio rocker arms, and thus they require longer mounting studs. Available in differing finishes, they are often advertised as having the advantage of reducing engine temperature. However, outside of racing applications it is doubtful that the difference in this area would be significant. However, there are two practical advantages to their use on a street engine: one is that, being a machined die casting, their bottom mating surface is truly flat, making for better sealing, and the other is that some of them, such as the ones produced by Kimble Engineering and Oselli, do reduce the noise level of valve clatter. One problem that they commonly present is the difficulty of remounting the original heater pipe in its original position above the engine block. Another is that if the mating surface of the cylinder head is warped, there exists the danger of the rocker arm cover cracking if overtightened, and leakage if it is installed loosely in order to avoid this possibility. Thus, it is advisable to have the upper mating surface of the cylinder head skimmed flat if you desire to install one. Due to the fact that all engines used in UK/European market MGBs were produced without an evaporative loss antipollution system, they thus had vented oil filler caps. As a result, these UK-made alloy rocker arm covers also have vented oil filler caps and no provision for a restrictor tube that would enable the use of the North American Market MGB's evaporative loss system and its anti-run-on valve. The aperture in the oil filler cap is oversize in order to accommodate the greater airflow requirements necessitated by the higher engine speeds attained in a race engine and as such is inappropriate for a street engine that normally operates at considerably lower engine speeds, so even if the evaporative loss system is not used, a reduction of the diameter of the aperture in the vented oil filler cap to 5/64" will usually be required in order to maintain a partial vacuum inside of the crankcase. If the evaporative loss system is to be retained in order to facilitate the use of an anti-run-on valve, then the elimination of the aperture in the oil filler cap plus the custom-fabrication of a restrictor tube and modification of the rocker arm cover to accept it will be required in order to permit it to be connected to the adsorption canister. Fortunately, Victoria British offers two cast aluminum alloy rocker arm covers that have provision for the needed restrictor tube, both in either polished aluminum alloy* or black powder-coated finishes. One is for the 18G and 18GA engines (Victoria British Part #s 17-714* and 17-715) and the other is for all of the 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, 18GK, and 18V five-main-bearing engines (Victoria British Part #s 17-716* and 17-726). Both use a non-vented oil filler cap (Victoria British Part # 17-696) that that is not interchangeable with any of the Original Equipment oil filler caps.

Be advised that not all rocker arm cover gaskets are equal. They differ in both design and in the quality of materials. Irregularities in some cork gaskets can provide a ready path for oil leaks. Compare the cork-rubber gasket from Fel-Pro (Fel-Pro Part # VS21509-1) to those of other manufacturers, and you will discover that there is a notable difference in both their grain size and uniformity, the result of Fel-Pro making use of both quality materials and carefully controlled production processes. As a result, the uniform, consistent grain pattern in Fel-Pro gaskets makes for superior leakage resistance. In addition, in order to increase its crush resistance, their rocker arm cover gasket is reinforced with a binder of

Viton, a synthetic rubber-like material that is more commonly used in seals resists and temperatures of up to 450º Fahrenheit (232.2º Celsius).

For those who seek a more advanced gasket material than antiquated cork, there are two choices: Nitrile and Silicone Rubber. Nitrile Rubber has excellent oil and chemical resistance, but its various compounds have upper temperature limits of only 200º Fahrenheit (93.3º Celsius). This is just too close to the operating temperatures of most engines, plus it tends to harden and crack with age. Silicone rubber offers a better choice. Silicone rubber has a temperature limit of 450º Fahrenheit (232.2º Celsius), and rebounds to nearly its original shape even after years of installation. Such gaskets resist hardening and cracking from hot engine operation, as well as from cold winter storage. Best of all, silicone rubber is highly resistant to drying out, or becoming saturated with oil and moisture. These are available from Roadster Gaskets (Roadster Gaskets Part # BMC B-VC). They have a website at http://www.roadstergaskets.com/index.html . Roadster Gaskets recommends sticking the gasket to either the rocker arm cover or cylinder head surface by coating one surface with Permatex Ultra 'Copper' RTV and curing for 24 hours.

Camshaft Bearings

Prior to installing new camshaft bearings, check to be sure that a 30º lead-in chamfer is present on the faces of their mounting bosses inside of the engine block in order to prevent galling which will distort their Internal Diameter. When installing them, take care to assure that the oil feed holes are properly aligned.

Also, make sure that of the rear bearing is properly aligned with the oil passage in the engine block that feeds oil to the rear rocker shaft pedestal, otherwise the rocker arm bushings will be starved of lubrication.

Avoid the camshaft bearings as sold by Moss, et al, as they are formed from a flat strip and rolled into shape. Consequently, they require that they be simultaneously reamed with a special factory tool after installation. Instead, use a set in which each bearing is manufactured as a single piece and requires minimal fitting. The best of this type are the Dura-Bond High Performance camshaft bearings (Dura-Bond Part # DA-2).

The Dura-Bond camshaft bimetal bearings are constructed from a seamless steel tube with a thin layer of Babbitt material. Their seamless construction makes for easy installation, eliminating breakage and bearing surface interruptions. These high performance camshaft bearings offer more than double the fatigue strength of conventional bearings, withstand increased valve spring loads better, while maintaining the excellent surface characteristics of Babbitt. Babbitt's superior embedability, conformability, and anti-seizure characteristics reduce camshaft failures that harder bearing materials can cause. Babbitt material will deform under overload conditions, sacrificing itself rather than damaging an expensive crankshaft. A very thin layer of Micro-Babbitt lining reduces the microscopic deflections that occur in a heavily loaded bearing and thus increases fatigue life, making these bearings ideal for supporting a high lift camshaft at their higher operating speeds. Rapid cooling of the Babbitt during the casting process creates a very fine grain structure. By leaving the structure as cast, tensile strength is almost doubled over that of an Original Equipment bearing. The resulting hard and high strength condition provides the "toughness" needed for high performance applications. Micro-fissures that can lead to fatigue failure are eliminated by cold working of the surface during the burnishing process. Their tolerances are held closer in order to control installed oil clearances, which reduces their required minimum operating pressure. These camshaft bearings are available with a Fluoropolymer composite coating that actually penetrates the surface. The primary advantage is that bearings with this coating retain oil on their surfaces, even under extreme heat and pressure conditions. Being a lubricant itself, the coating provides back-up lubrication in the event that momentary oil starvation occurs. This characteristic is especially important during startup because oil does not reach all critical components immediately.

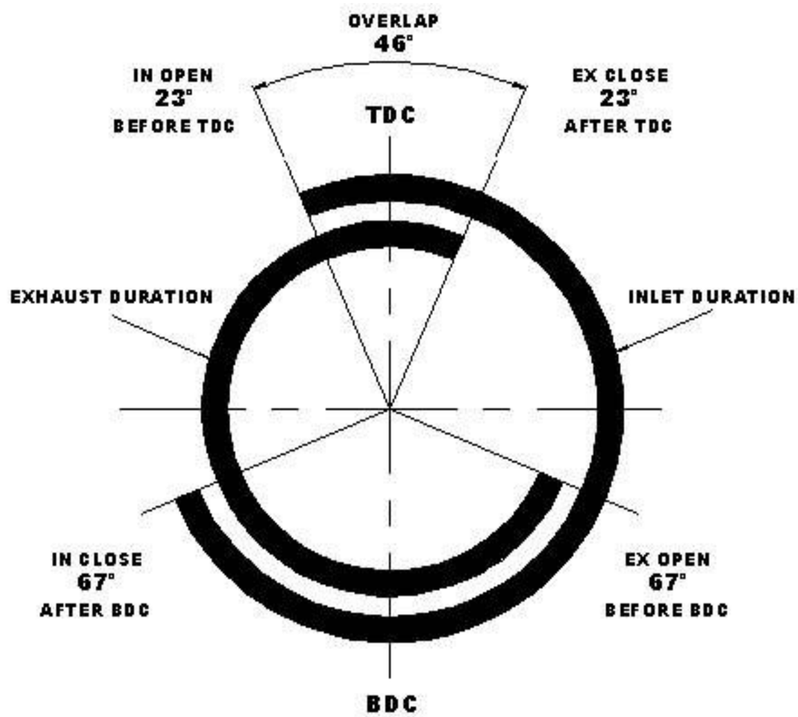
Should you elect to install Dura-Bond camshaft bearings, remember that under no circumstances should you attempt to hone the Internal Diameter of camshaft bearings, as the honing process will impregnate their surfaces with grit. A bearing diameter of .618" (15.6972mm) with an optimum Diametrical Clearance range of .003" to .004" (.0762mm to .1016mm) for high performance applications should be accomplished by reaming. Afterwards, the camshaft bearing journals should be polished to a surface finish of 10 microinches Ra in the same direction as that in which they rotate under service conditions. The journal diameters of an Original Equipment camshaft are- front: 1.789" +/- .00025" (45.4406mm +/- .00635mm), center: 1.729" +/- .00025" (43.9166mm +/- .00635mm), and rear: 1.623” +/- .00025” (41.2242mm +/- .00635mm). The endplay (endfloat) of the installed camshaft should be .004” +/- .002” (1016mm +/- .0508mm).

Camshaft Terminologies

Prior to delving into the mysteries of camshafts, there are certain basic terminologies that you should be aware of:

“Camshaft Lobe” is that part of the camshaft that is eccentric to the shaft beyond its base circle and transmits a lifting motion through the valvetrain in order to control the operation of the valves.

“Camshaft Lobe Profile” is the actual shape of the camshaft lobe. The design of the camshaft lobe profile determines the performance characteristics of the camshaft.



"Duration" is a term that signifies the length of time, measured in crankshaft degrees, that the valve is off of its seat. In order to calculate Duration, add together the timing numbers of a valve event and add 180^{o}. For example, a camshaft with timing figures of 23^{o} / 67^{o} added together totals 90^{o}, plus 180^{o}, gives 270^{o} of Duration.

"Overlap" is a term that signifies the number of crankshaft degrees wherein both the intake valve and the exhaust valve are simultaneously open. In order to calculate Overlap, add the opening number in degrees of the intake lobe to the closing number in degrees of the exhaust lobe, i.e., the first and last numbers of the valve timing. Using as an example a 23^{o} / 67^{o} intake valve timing and a 67^{o} / 23^{o} exhaust valve timing (usually referred to in data charts as (23^{o} / 67^{o} – 67^{o} / 23^{o}), add together the first and last numbers (23^{o} and 23^{o}) and the total (46^{o}) is the amount of the Overlap. In general terms, the larger this number is, the hotter the camshaft is generally considered to be.

"Cam Timing" is a term that signifies the position of the camshaft relative to that of the crankshaft. This is expressed as the number of degrees that full lift occurs After Top Dead Center (ATDC) in the case of the intake valve, and Before Top Dead Center in the case of the exhaust valve. In order to calculate this, take the Duration figure and divide it by 2. EXAMPLE: With an intake lobe profile of 23^{o} / 67^{o}, the Duration is the sum of these two numbers plus 180^{o}, thus equaling 270^{o}. Next, divide the sum by 2, which in this case results in 135^{o}. Deduct the number of degrees Before Top Dead Center at which the valve begins to open, i.e., 23^{o}, and the result is 112^{o}. Therefore. the valve will be correctly timed when its full lift is attained at 112^{o} After Top Dead Center.

"Lobe Separation Angle" is a term that signifies the included angle between the peak of the nose of the intake lobe and the peak of the nose of the exhaust lobe for a given cylinder, measured in degrees.

"Clearance Ramp" is a term that signifies that part of the camshaft lobe profile whose purpose is to gradually take up both the valve clearance and the slack in the valve train prior to the valve being lifted off of its seat. Its reverse side also gently rests the valve back down onto its seat as the tappet descends down the closing flank. Profiles employed with mechanical tappets employ a much larger ramp than those employed with hydraulic tappets, as the hydraulic tappet should be in contact with the lobe at all times. The height of the ramp dictates what measurement the valve clearances should be set to. A descending ramp begins when the valve touches the valve seat and ends when the tappet returns to the base circle. Ramp designs have a tremendous effect on power output and valve train reliability.

"Flank" is a term that signifies that part of the lobe profile that is located between the end of its ramp and the beginning of its nose. In terms of power production, it is the most significant part of the design of the profile. The design of the flank determines both the acceleration and the velocity of the valve train. The acceleration / deceleration rate must be within the working limits of the valve spring. If this working limit is exceeded, then valve float with result. Generally, high acceleration and high velocity figures are beneficial to engine performance, although they do increase stress and accelerate wear of the components of the valvetrain.

"Nose radius" is a term that signifies that part of the lobe profile where the valve is fully opened. The best designs utilize the largest nose radius possible in order to keep stresses to a minimum.

"Base Circle Diameter" is a term that signifies the distance across the lobe, calculated by measuring the overall height and subtracting the cam lift.

"Lift" can be either the amount of lift that is produced at the lobe of the camshaft, or it can be the amount of lift at the valve, the latter being equal to the amount of lift at the camshaft lobe multiplied by the rocker ratio.

"Valve Timing" is a term that signifies the opening and closing positions of both the intake valve and the exhaust valve of a given cylinder in relation to the crankshaft, expressed as figures of Before Top Dead Center and After Bottom Dead Center.

"Dwell" is a term that signifies the period wherein the valve reaches full lift and stops moving for a few degrees of camshaft rotation before starting to descend back towards its seat. When checking the camshaft timing using the full lift figure method, the mid-point of the dwell should be taken as exact full Lift.

"Rocker Arm Ratio" is a term that signifies the ratio of valve motion vs. tappet motion. Pushrod engines typically make use of a ratio that is somewhere between 1.1:1 to 2.0:1.

"Overall height" is a term that signifies the distance from the nose of the lobe to the bottom of the base circle, measured in a straight line through the rotational axis of the lobe.

Original Equipment Camshafts

It is not commonly understood that the B Series engine underwent changes to its valve timing during the course of its career under the bonnet of the MGB. Originally, the engine used a duplex (double row) roller-type camshaft drive chain (BMC Part # 2H 4905) and duplex sprockets (BMC Part #'s 12A 1553 and 11G 203). This camshaft lobe profile was used in 18G, 18GA (BMC Part # 88G 252, County Part # C612A), 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, 18GK, 18V-581, 18V-582, and 18V-583 engines (BMC Part # 88G 303, County Part # C612). The specifications of this camshaft were:

	Opens @	**Closes @**	**Duration**	**Lobe Center Angle**	**Tappet Lift**	**Valve Lift***
Intake Valve	16° BTDC	56° ABDC	252°	110° ATDC	.2567" (6.5202mm) @ 110° ATDC	.3645" (9.2583mm) @ 110° ATDC
Exhaust Valve	51° BBDC	21° ATDC	252°	105° BTDC	.2567" (6.5202mm) @ 110° ATDC	.3645" (9.2583mm) @ 110° ATDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The LSA (Lobe Separation Angle) was 107.5°. Overlap was 37°. The camshaft was retarded @ 2.5° ATDC.

In August of 1971 a new camshaft was introduced for the North American Market 18V-584-Z-L and 18V-585-Z-L engines. While the camshaft lobe profile design was slightly altered, the keyway of the camshaft driven sprocket was advanced another 2.25° in order to increase midrange torque output. It should be noted that due to the larger 1.625" diameter intake valves of the 12H 2923 cylinder head casting with which this valve timing was used, the power output at high engine speeds remained unaltered. In October of 1972 the camshaft drive chain and both the drive and the driven sprockets were changed from a Duplex (dual-row) camshaft drive chain system to a Simplex (single row) camshaft drive chain system (Drive Sprocket BMC Part # 12H 4201; Driven Sprocket BMC Part # 12H 4200, Drive Chain 3H 2127) on the North American Market 18V-672-Z-L, and 18V-673-Z-L engines.

	Opens @	**Closes @**	**Duration**	**Lobe Center Angle**	**Tappet Lift**	**Valve Lift***
Intake Valve	18.25° BTDC	58.25° ABDC	256.5°	112.25° ATDC	.2567" (6.5202mm) @ 112.5° ATDC	.3645" (9.2583mm) @ 112.25° ATDC
Exhaust Valve	53.25° BBDC	23.25° ATDC	256.5°	112.25° BTDC	.2567" (6.5202mm) @ 112.5° ATDC	.3645" (9.2583mm) @ 112.25° BTDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The LSA (Lobe Separation Angle) remained 107.5°. Overlap was increased by 4.5° to 41.5°.

In December of 1974, Rubber Bumper cars for the North American Market with 18V-797-AE-L, 18V-798-AE-L, 18V-801AE, and 18V-802AE engines received a new camshaft (BMC Part # CAM 1156, County Part # C6012B). The specifications of this camshaft were:

	Opens @	Closes @	Duration	Lobe Center Angle	Tappet Lift	Valve Lift*
Intake Valve	8º BTDC	42º ABDC	230º	107º ATDC	.2567" (6.5202mm) @ 107º ATDC	.3645" (9.2583mm) @ 107º ATDC
Exhaust Valve	54º BBDC	18º ATDC	252º	108º BTDC	.2567" (6.5202mm) @ 108º BTDC	.3645" (9.2583mm) @ 108º BTDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

Revised, shorter-duration camshaft. The LSA (Lobe Separation Angle) remained 107.5º. Overlap was reduced by 15.5º to 26º. The camshaft was advanced to .5º BTDC.

Starting during the month of June of 1977 cars with 18V-846-F-H and 18V-847-F-H engines for the UK Home and Export markets (not North America) received a new camshaft driven sprocket (BMC Part # ?) that further advanced the camshaft timing by another 1º. The specifications of this camshaft were:

	Opens @	Closes @	Duration	Lobe Center Angle	Tappet Lift	Valve Lift*
Intake Valve	20º BTDC	52º ABDC	252º	106º ATDC	.2567" (6.5202mm) @ 106º ATDC	.3645" (9.2583mm) @ 106º ATDC

Exhaust Valve	55º BBDC	17º ATDC	252º	109º BTDC	.2567" (6.5202mm) @ 109º BTDC	.3645" (9.2583mm) @ 109º BTDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The LSA (Lobe Separation Angle) remained 107.5º. Overlap was increased by 11º to 37º. The camshaft was advanced to 1.5º BTDC.

If it were my engine, I would not spend any money on a non-Original Equipment camshaft except as a final modification performed in order to compliment the headwork, and then only if I was not satisfied with the results of the sum total of the previous modifications. Changing the camshaft before doing the headwork is putting the cart before the horse. The specifications of the Original Equipment pre-1975 factory camshaft are hard to improve upon for general duty use, producing a smooth idle and ample low-end and midrange power. Also, realize that changing the camshaft to one with lift that is more radical and / or more duration will increase wear on the tappets, camshaft lobes, valve guides, and valve stems by means of the increased side thrust loads. Should you simply wish to relocate the existing power curve in order to suit your driving style, you might consider retarding or advancing the timing of the Original Equipment specification camshaft a very few degrees (4º maximum, beyond that point the gains are increasingly small while the losses become increasingly excessive) in order to respectively move the power up or down the scale as much as 400 RPM. This is the approach taken by the engineers at the MG factory when they introduced the 18V version of the B Series engine, so such a modification does not involve going into uncharted territory. If you move the power downward, with professional headwork you should still have at least as much power at high engine speeds as an Original Equipment specification engine, and vice-versa. Be advised that you will need to change the fuel-metering needles in your carburetors if you choose to pursue this course.

It should be noted that in the case of the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines, it is necessary to remove both the oil pump drive gear and the distributor drive gear in order to remove or install the camshaft. However, on 18V engines only the distributor drive gear need be removed. Upon close inspection, you will find that there is a very distinct wear pattern on the teeth of the distributor drive gear and those of the oil pump drive gear. The teeth of gearsets tend to “mate” as they wear in. Because the teeth of both the distributor drive gear and those of the oil pump drive gear have mated to the teeth of the drive gear on the old camshaft, it should be noted that if they are reinstalled with a new camshaft, wear of all of the associated gear teeth will be accelerated. A worn distributor drive gear will cause the ignition timing to vary, making precise setting impossible. If you can see any wear, replacement is called for. Replacing only one gear in a mated set with a new one is asking for trouble in the future. Thus, in the interests of long-term durability, replacement of both a worn distributor drive gear and a worn oil pump spindle drive gear with new drive gears is strongly recommended whenever a new camshaft is installed. If anything is worth doing, it is worth doing right.

The distributor drive shaft housing (BMC Part # 1G 2285) serves two functions: it both sleeves the hole in the engine block down to the size of the lower portion of the distributor body, as well as secures the distributor drive shaft (BMC Part # 1G 2062) into the engine. Without removing the distributor drive shaft housing, it is impossible to remove the distributor drive gear from the drive gear of the camshaft. The distributor drive shaft housing is a tight slip-fit into the engine block and is held in place by a single ¼”-28 machine screw that is awkward to reach when the engine is in the car and, all too frequently, is frozen in place. Often the distributor drive shaft housing must be moved in a rotational manner before the bond between the distributor drive shaft housing and engine block releases sufficiently to allow it to be withdrawn, thus allowing the distributor drive shaft housing to be removed. If necessary, this can be done with the engine still in the car by using a ½”-width stubby screwdriver in order to remove the ¼”-28 machine screw that secures the flange of the distributor dive shaft housing, removing the distributor, then pulling the distributor drive spindle out with a 5/16”- 24 UNF machine bolt from the airfilter housing. As the camshaft rotates, the side thrust from the helically cut gears forces the distributor drive shaft upwards and against the distributor housing. While the engine is turning, the distributor drive shaft cannot move downwards as long as the gear on the camshaft is rotating it. The principle involved is simple: the greater the clearance between the gear of the distributor drive shaft and the distributor housing, the greater the alteration of the ignition timing will be. If the distributor dog is allowed to move too much, then it will tend to batter the distributor housing, causing end-shake and, in extreme circumstances, could even allow the distributor drive to disengage and thus quit rotating the distributor.

Damage to both the camshaft gear and the distributor drive gear would be probable during such an occurrence. The desired range of movement is between .010" to .015" (.254mm to .381mm) of the distributor drive. When reinstalling the distributor drive gear, turn the engine until # 1 piston is at Top Dead Center on its compression stroke. This will require one complete rotation of the crankshaft from the position it was in when the camshaft drive chain was installed, thus causing the camshaft to rotate 180°. When the helically cut gear teeth engage the mating gear on the camshaft, the slot should be pointing to approximately the 2:00 o'clock position.

Should you choose to reuse your Original Equipment camshaft rather than install a higher-performance item, be sure to minutely examine all of its friction surfaces for signs of wear. The lobe should measure 1.370" (34.798mm) from the peak of its lobe to the bottom of its base circle. Should it appear to pass inspection, remember that you can reuse the original tappets with their original camshaft only if they are installed in their original order, and only if the camshaft to which they originally mated is installed with the same amount of endfloat as that which you found before you removed it, otherwise you will most likely wipe out the lobes of the camshaft, as well as the domed ends of the tappets. New tappets are always a much surer approach to achieving a happy union of these components. However, if you do want to reuse your old tappets with their original camshaft, use a micrometer to ascertain that each of their diameters is consistent around its circumference. If the diameter shows a variation of more than .0005" (.0127mm), then that tappet should be scrapped. The domed end of each tappet should be examined for pitting. If you see any pitting, then the hardened domed end has been worn away and you are looking at the soft, shock-absorbing inner core of the tappet. With its domed end worn to this point, the tappet will not rotate effectively in its bore, thus resulting in accelerated wear of the lobe of the camshaft. Such a tappet should also be scrapped.

High Performance Camshafts

If you absolutely insist upon changing your camshaft, use new tappets (always!) and regardless of the camshaft that you select, be aware that normal engine assembly lubricants such as engine oil, STP, and Lubriplate white lithium grease are unsuitable for camshaft installation. The cause of premature tappet and camshaft lobe failure is metal to metal contact at the interface of the tappet and camshaft lobe. Should this contact occur due to either the lack of proper lubrication or excessively high pressure, two sequential events will follow. First, shearing of the oil film, and then galling, will take place. When this occurs, metal is transferred between the domed end of the tappet to the camshaft lobe in a process comparable to welding. Microscopic high spots, which are present on all machined parts, become overheated due to the combined effects of friction and pressure, and then bond together, tearing sections loose from either the tappet or the lobe, or both. These pieces of metal remain attached and create further localized overheating during the following revolutions of the camshaft and lead to the ultimate failure of the effected components. Only Molybdenum Disulphide (MoS_2) Extreme Pressure assembly lubricants such as CRC Sta-Lub Extreme Pressure Lubricant can be safely used for prelubrication of the camshaft. All of the camshaft lobes must be coated with this lubricant, otherwise premature lobe and tappet failure is highly likely to occur.

When installing the camshaft, turn the engine block onto its rear end. This will allow you to carefully lower the camshaft into position with a lessened risk of damaging its bearings. Although some will point out that the camshaft will turn freely with an end clearance of .001” (.0254mm), it is best to install it to the factory-recommended end clearance of .004” (.1016mm) when measured between the thrust face of the camshaft and the camshaft retainer plate. The available amount of lash allows greater variation of the point of interface between the domed ends of the tappets and the lobes of the camshaft, thus reducing wear of the lobes. Be aware that because the distributor is gear-driven directly from the camshaft, and thusly any excessive endplay (endfloat) will result in a minor ignition timing wander that may not be detectable with an inexpensive strobe-type timing light. In turn, this may result in a misdiagnosis of subtle running problems, such as thinking that the cause could be attributed to either a worn distributor shaft and shaft bushing, or an unevenly worn camshaft drive chain, or even possibly to a defective camshaft drive chain tensioner. Torque the three machine bolts for the camshaft locating / lock plate to 10 Ft-lbs and the camshaft nut to 60-70 Ft-lbs. Do not neglect to use the internal star washers.

While the vast majority of camshaft manufacturers rarely encounter problems with producing accurate lobe profiles and precise lobe center angles, there is always something worth checking whenever you install a modified camshaft. What can occasionally happen is that the camshaft was not properly centered when set up for grinding the cam profile onto the lobes. This results in the base circle not being concentric with the journals of the camshaft, i.e., the base circle is eccentric when the camshaft is rotating in its journals. This base circle run-out, as it is known, can be as much as .005" to .010" on some aftermarket camshafts. In the best of these circumstances, the valve clearances can be very difficult to set, and in bad cases, it is impossible to get the clearance take-up to occur on the opening ramp and the closing ramp as it should. You are forced to make a choice between one or the other, even if you go to the extent of plotting each lobe from ramp to ramp to see where the ramps are.

This manufacturing defect is easy to detect with a dial indicator. Set the gauge up on the closed valve on a rule of nine pair with its mate fully open. Rotate the crankshaft clockwise one half of a revolution. The dial indicator measures the height of the base circle and theoretically should not move for most of the half turn. Rotate the crankshaft counterclockwise a full revolution, so that the rule of 9 mate goes back to the fully open and then half a revolution past that. Again, the reading on the dial indicator should remain constant until about fully open. Any variation during this base circle measurement is a measure of the base circle run-out.

Note that if there is base circle run-out, it will be different for each lobe depending on where the lobe centre points in relation to the run-out of the camshaft. .002" or .003" is not usually too problematic as the variation is small enough to be accommodated by the length of the opening ramp. More than this will cause problems in accurate setting of the valve clearance, and obviously the worse the run-out, the bigger the problem will become - how can anyone hope to accurately set a valve clearance of .015" if the base circle is varying by .010"? In this case, I would reject the camshaft and start again with a good one.

The use of reground increased-lift camshafts should be avoided whenever possible. Although notably less expensive than a camshaft that is made from a new blank, the additional expense of custom-length pushrods are likely to become necessary due to the diameter of the base circle of the camshaft lobe being decreased out of necessity in order to achieve the desired amount of valve lift, the additional consequence of which is steeper lobe ramps that increase pressure loadings on the tappets. If a camshaft lobe profile is reground, whatever is removed from the base circle will also have to be removed from the nose radius of the lobe as well. Given that the nose radius of the Original Equipment camshaft lobe is only 2mm to 3mm and that Hertzian contact stress is inversely proportional to the square root of the nose radius, it becomes obvious that the undersized lobes of a reground camshaft are a bad idea. Rapid pitting is usually the result of excessive Hertzian contact stress, i.e., the nose radius being too small and / or the spring loads being too high. These Hertzian contact stresses are cyclic in nature and over time lead to sub-surface fatigue cracks that precede and accelerate pitting. Performance camshafts made on new blanks that retain the original base circle can probably contend with the increased stress as they tend to extend the period when they hold the valves open by at least 12º, thus making it easier to obtain the needed larger nose radius. In addition, as the amount of valve lift increases, the more that custom-length lightweight high-strength pushrods become advisable in order to reduce Hertzian contact stress. Obviously, such pushrods turn out to be less inexpensive than they might initially appear to be, especially when compared to the cost of a replacement camshaft made from a new blank, along with the mandatory new set of tappets.

In addition, the reduced diameter of the base circle of the lobe of a reground camshaft results in a reduction of the rotational speed of the tappet. Because the rotation of the tappet draws oil into the thrust area of the tappet bore, this reduction in rotational speed reduces needed lubrication and increases wear of both the tappet and that of the tappet bore. If barrel tappets with a lubricating passage are employed, the reduced rotational speed minimizes the centrifugal force that draws oil through the tappet into the tappet bore, partially negating the advantage of such a modified tappet design.

One should not quickly dismiss the first version of the Original Equipment camshaft as a poor performer. In reality, it is a surprisingly capable performer. Its modest period of overlap offers a smooth idle while its conservative duration offers a broad powerband, allowing it to produce most of its useable power in the range of engine speed wherein most drivers commonly operate in day-to-day driving situations. Its relatively modest valve timing permits a long life expectancy for the components of the valve train as well.

Original Equipment Camshaft (BMC Part # 88G 303)					
Opens @	**Closes @**	**Duratio**	**Lobe Center**	**Tappet**	**Valve**

			n	Angle	Lift	Lift*
Intake Valve	16° BTDC	56° ABDC	110° ATDC	252°	.2567" (6.520mm) @ 110° ATDC	.3645" (9.258mm) @ 110° ATDC
Exhaust Valve	51° BBDC	21° ATDC	105° BTDC	252°	.2567" (6.520mm) @ 105° BTDC	.3645" (9.258mm) @ 105° BTDC

* Presumes an Original Equipment rocker ratio of 1.42:1

The Lobe Separation Angle was 107.5°. Overlap was 37°. The camshaft was retarded @ 2.5° ATDC.

The most versatile aftermarket camshaft presently available is the Piper BP270 camshaft that Peter Burgess recommends. Unlike some camshafts that have long duration timing, it does not require large amounts of compression in order to compensate for poor volumetric efficiency at low engine speeds. This being the case, it is less apt to produce detonation and is less sensitive to small variations in ignition timing than designs that employ long duration timing. Its high volumetric efficiency at low engine speeds will give a smooth idle with excellent throttle response and power, even right up to 6,000 RPM with an Original Equipment configuration cylinder head and up to 6,400 RPM and yet more power across the entire powerband when used with fully modified Fast Road configuration cylinder heads. Best of all, because it uses only 12% more valve lift than an Original Equipment specification camshaft, you can avoid the worst of the excessive side thrust loads on the tappets as well as on the lobes of the camshaft that are produced by more radical camshaft lobe profiles. You will, however, need to either use stronger valve springs in order to handle the greater inertia loads resulting from the more rapid openings of the valves or, preferably, lighten the reciprocating mass of the valvetrain by means of tubular chrome-moly pushrods and the lighter bucket tappets in order to reduce the inertia loads.

Although it may seem that simply fitting stronger valve springs produces an inexpensive solution to the problem of increased inertia loads in the valvetrain, it will be at the expense of the tappets hammering the camshaft lobe when closing and greater pressure loads upon the ramps and nose of the camshaft, thus accelerating wear of both the tappets and the lobes of the camshaft.

In order to understand the effects of stronger valve springs upon the pressure loadings involved with the Original Equipment valve train, the following table is enlightening:

Tappet Lift	**Valve Lift**	**Inner Valve Spring Rate** **BMC Part #**	**Outer Valve Spring Rate** **BMC Part #**	**Total Spring Load**	**Valve Clash Occurs @**	**Load Increase**
.2567" (6.520mm)	.3645" (9.258mm)	50 Ft-lbs 1H 723	117 Ft-lbs 1H 1111	167 Ft-lbs	6,230 RPM*	Original Equipment Standard
.2567" (6.520mm)	.3645" (9.258mm)	57 Ft-lbs C-1H 1112	117 Ft-lb 1H 1111	174 Ft-lbs	6,360 RPM*	4.2%
.2567" (6.520mm)	.3645" (9.258mm)	50 Ft-lbs 1H 723	140 Ft-lbs C-AHH 7264	190 Ft-lbs	6,480 RPM*	13.8%
.2567" (6.520mm)	.3645" (9.258mm)	57 Ft-lbs C-1H 1112	140 Ft-lbs C-AHH 7264	197 Ft-lbs	6,600 RPM*	18.0%

.2567” (6.520mm)	.3645” (9.258mm)	70 Ft-lbs C-AHH 7265	140 Ft-lbs C-AHH 7264	210 Ft-lbs	6,680 RPM*	25.7%

* Note: These engine speed limits are applicable only when used with an Original Equipment 89.2 gr. 1.565” (39.751mm) diameter intake valve, an Original Equipment 76.3 gr. 1.345” (34.162mm) exhaust valve, the Original Equipment 1.42:1 ratio rocker arms (BMC Part # 12H 3377), the Original Equipment 79.7 gram long barrel tappet (BMC Part # 1H 822), and the Original Equipment 72 gram short pushrod (BMC Part # 11G 241).

It should be noted that the maximum permissible total spring load is 230 Ft-lbs, a spring load increase of 37.7%, allowing a maximum permissible engine speed of greater than 7,000 RPM before valve clash occurs. However, be aware that these total spring load figures are pertinent solely for an Original Equipment camshaft with its camshaft lobe lift of .2567” (6.520mm), producing a valve lift of a mere .3645” (9.2583mm) with the smaller-diameter, lighter Original Equipment 89.2 gr. 1.565” (39.751mm) diameter intake valve. When using Original-Equipment valvetrain components exclusively, higher-lift camshafts require stronger spring loads in order to prevent valve clash at high engine speeds. As a consequence, more rapid wear of both the domed ends of the tappets and the lobes of the camshaft, as well as the ends of the pushrods, are the result. In order to prevent catastrophic failure, strict quality control of the involved components and careful breaking-in of the tappets becomes mandatory. In the interests of long-term durability, decreasing the reciprocating mass of the valvetrain is the preferred approach to the problem of controlling valve clash.

In determining which valve springs you will require, it should be noted that, along with duration and ramp profile, the amount of valve lift produced by the camshaft lobe profile is also a determining factor. Even at 100% volumetric efficiency, the optimum airflow around any valve occurs when it has been lifted to a height that is 25% of its diameter because its curtain area then equals its diametric area. For practical examples, a 1.565” (39.75mm) diameter intake valve will require a lift of no more than .39125” (9.93775mm) in order to reach its optimum airflow capability, a 1.625” (41.275mm) diameter intake valve will require a lift of no more than .40625” (10.3188mm) in order to reach its optimum airflow capability, a 1.68” (42.672mm) diameter intake valve will require a lift of no more than .420”

(10.668mm) in order to reach its optimum airflow capability, and a 1.72" (43.688mm) diameter intake valve will require a lift of no more than .430" (10.922mm) in order to reach its optimum airflow capability. Whichever valve springs you choose to employ, carefully inspect their end coils. They should always be ground square for optimum alignment and reduced solid height.

Be aware that if you choose to employ a camshaft that produces more lift, then it would be best to replace all of your rocker arms with new ones. This is due to the fact that the thrust pads of the original rocker arms will have a groove with a "lip" formed on its end at the furthest point of contact with the tip of the valve stem. This "lip" will impinge on the tip of the valve stem when the rocker arm is lifted beyond the point at which the previous lower-lift camshaft reached it maximum travel, causing rapid wear of the tip of the valve stem. The additional side loads created by this impingement will also force the valve stem to tilt in the bore of the valve guide, resulting in rapid wear of its bore.

Peter Burgess offers both the Piper BP270 and Piper BP285 camshafts with additional lubricating passages in their helically cut drive gears for providing supplemental lubrication in order to reduce the wear that is attendant to the increased loads that are a consequence of more radical camshaft lobe profiles. These specially modified camshafts are not regrinds. They are made from new billets, thus providing a larger heel diameter than that of a regrind. While this feature allows the use of gentler ramp angles that will reduce stress at the interface of the domed end of the tappet with that of the lobe of the camshaft, the rotational speed of the tappet is increased beyond that resulting from a reground camshaft, thus making an oiling hole in the sidewall of the tappet a wise precaution against accelerated wear. They also have provision for the mechanical drive mechanism of the tachometer of the early three main bearing 18G, and 18GA B Series engines of the MGB, as well as the B Series engines of its predecessor, the MGA.

It should be noted that Piper periodically makes improvements to its line of camshafts. The HP Series of camshafts was succeeded by the HR (Magnum) Series, and the current series is the BP Series, which has recently undergone a refinement of the specifications used. Piper is now stating that the BP255 will produce an increase power output of 8 BHP, the BP270 an increase of 12 BHP, and the BP285 an increase of 18 BHP over that of the Original

Equipment camshaft. Make sure that you look at the specification sheet that comes with the camshaft. The specifications have changed as follows:

The old version of the Piper BP255 was:

	Opens @	**Closes @**	**Duration**	**Lobe Center Angle**	**Tappet Lift**	**Valve Lift***
Intake Valve	24º BTDC	60º ABDC	264º	108º ATDC	.278" (7.066mm) @ 108º ATDC	.395" (10.033mm) @ 108º ATDC
Exhaust Valve	60º BBDC	24º ATDC	264º	108º BTDC	.278" (7.066mm) @ 108º BTDC	.395" (10.033mm) @ 108º ATDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The Lobe Separation Angle was 108º. Overlap was 48º. The powerband spans from 1,000 RPM to 6,000 RPM.

The new version of the Piper BP255 is now:

	Opens @	**Closes @**	**Duration**	**Lobe Center Angle**	**Tappet Lift**	**Valve Lift***

Intake Valve	27º BTDC	63º ABDC	270º	108º ATDC	.285" (7.239mm) @ 108º ATDC	.405" (10.287mm) @ 108º ATDC
Exhaust Valve	64º BBDC	28º ATDC	272º	108º BTDC	.274" 6.960mm) @ 108º BTDC	.389" (9.8806mm) @ 108º BTDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The Lobe Separation Angle remains 108º. Overlap has been increased by 7º to 55º. The duration has been increased by 6º for the intake and 8º for the exhaust, with maximum valve lift increased by .010" (.254mm) for the intake valve and reduced by .006" (.1524mm) for the exhaust valve. The period of overlap has been increased by 7º. Valve lift at Top Dead Center is .072" (1.8288mm) for the intake valve and .059" (1.4986mm) for the exhaust valve. The valve clearances are set at .012" (.30mm) for the intake valve and .014" (.35mm) for the exhaust valve. The powerband spans from 1,000 RPM to 6,000 RPM.

The old version of the Piper BP270 was:

	Opens @	**Closes @**	**Duration**	**Lobe Center Angle**	**Tappet Lift**	**Valve Lift***
Intake Valve	38º BTDC	74º ABDC	292º	108º ATDC	.282" (7.163mm) @ 108º ATDC	.400" (10.160mm) @ 108º ATDC
Exhaust	74º	38º		108º	.282" (7.163mm)	.400" (10.160mm)

Valve	BBDC	ATDC	292°	BTDC	@ 108° BTDC	@ 108° BTDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The Lobe Separation Angle was 108°. Overlap was 76°. The powerband spans from 1,500 RPM to 6,500 RPM.

The new version of the Piper BP270 is now:

	Opens @	**Closes @**	**Duration**	**Lobe Center Angle**	**Tappet Lift**	**Valve Lift***
Intake Valve	31° BTDC	65° ABDC	276°	107° ATDC	.285" (7.239mm) @ 107° ATDC	.405" (10.287mm) @ 107° ATDC
Exhaust Valve	65° BBDC	31° ATDC	276°	107° BTDC	.284" (7.214mm) @ 107° BTDC	.403" (10.236mm) @ 107° BTDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The Lobe Separation Angle is reduced by 1° to 107°. Overlap has been reduced by 14° to 62° The duration has been reduced by 16°, with maximum valve lift increased by .005" (.127mm) for the intake valve and increased by .003 (.0762mm) for the exhaust valve. The period of overlap for both the intake and the exhaust has been decreased by 14°. Valve lift at Top Dead Center is .085" (2.159mm) for the intake valve and .075" (1.905mm) for the exhaust valve. The valve clearances are set at .014" (.35mm) for the intake valve and .016" (.40mm) for the exhaust valve. The powerband spans from 1,500 RPM to 6,500 RPM.

The old version of the Piper BP285 was:

	Opens @	Closes @	Duration	Lobe Center Angle	Tappet Lift	Valve Lift*
Intake Valve	37º BTDC	75º ABDC	292º	109º ATDC	.285" (7.239mm) @ 109º ATDC	.405" (10.287mm) @ 109º ATDC
Exhaust Valve	71º BBDC	41º ATDC	292º	109º BTDC	.285" (7.239mm) @ 109º BTDC	.405" (10.287mm) @ 109º ATDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The Lobe Separation Angle was 109º. Overlap was 78º. The powerband spans from 2,000 RPM to 6,750 RPM.

The new version of the Piper BP285 is now:

	Opens @	Closes @	Duration	Lobe Center Angle	Tappet Lift	Valve Lift*
Intake Valve	32º BTDC	66º ABDC	278º	107º ATDC	.313" (7.950mm) @ 107º ATDC	.445" (11.303mm) @ 107º ATDC

Exhaust Valve	66° BBDC	32° ATDC	278°	107° BTDC	.313" (7.950mm) @ 107° BTDC	.445" (11.303mm) @ 107° BTDC

*Presumes an Original Equipment rocker arm ratio of 1.42:1.

The Lobe Separation Angle has been reduced by 2° to 107°. Overlap has been reduced by 14° to 64°. The Duration has been reduced by 14°, with maximum valve lift increased by .040" (1.016mm). The period of overlap for both the intake and the exhaust has been decreased by 14°. Valve lift at Top Dead Center is .097" (2.4638mm) for the intake valve and .097" (2.4638mm) for the exhaust valve. The valve clearances are set at .014" (.35mm) for both the intake valve and the exhaust valve. The powerband spans from 2,000 RPM to 6,750 RPM.

In the cases of the Piper BP270 and BP285 camshafts, the reduced duration has resulted in less overlap, decreasing reversion problems and making for smoother idling and improved torque at low engine speeds when compared to the earlier versions of both camshafts. In addition, in both cases the amount of valve lift has been increased, the airflow characteristics of the Heron-type sidedraft cylinder head configuration responding well to the increased valve lift. However, it should be noted that this new configuration has resulted in the lobes having steeper ramp angles, thus creating a mandatory requirement of top-quality tappets.

The more radical variety of camshafts will produce more power at notably higher engine speeds, but, because of reduced volumetric efficiency at low engine speeds, will also sacrifice so much low-end torque that the loss of tractability at low engine speeds can make normal driving in heavy traffic difficult. In addition, the lower volumetric efficiency at low engine speeds forces the use of a higher compression ratio, which in turn reduces the required ignition advance due to the quicker burn time of the fuel / air charge. If you choose to follow this path, expect a lumpy, vibrating idle and, in order to fully exploit the advantages of the camshaft, a change to both a pair of 1¾" SU carburetors with their larger intake manifold, as well as a switch to an pure centrifugal advance Lucas 45 distributor (Aldon Part # 101BR1). Starting the engine in extreme cold weather will also become something approaching an art and occasionally an exercise in frustration. Piper has a website that can be found at http://www.pipercams.co.uk/ .

If you choose to install a more radical camshaft than the Piper BP270, you should first read “How To Build And Power Tune Distributor Type Ignition Systems”, “How To Build And Power Tune SU Carburetors”, and “How To Choose Camshafts and Time Them For Maximum Power”, all of which are available from Veloce Publishing through their website at http://www.veloce.co.uk/newtitle.htm so that you will be properly prepared to make the most out of your choice without an undue compromise in reliability.

It should be noted that dual pattern camshafts are irrelevant in a B Series engine that is not intended to routinely operate at very high engine speeds (above 6,800 RPM), nor has a Geometric Compression Ratio (GCR) less than 9.5:1, the practical limit for a street engine that is equipped with a cast iron cylinder head. Scatter pattern camshafts are ineffective in the B Series engine when their duration is less than 300°, and any camshaft with a duration greater than 290° is simply too radical for a street engine as it will provoke chronic ‘robbing’ of the intake charge in the siamesed intake ports at low engine speeds.

Note that much of the performance increase that can be gained by going this route could be achieved at a far lesser expense and with much better streetability simply by having quality headwork done by a professional such as Peter Burgess. While the early Original Equipment rocker arms with internal oiling passages and their rocker adjustment screws with oiling passages in their ball ends will give superior oiling of both the pushrods and the tappets and are of sufficient strength for use with any of the Original Equipment camshafts, the later, solid rocker arms and solid adjusting screws are more desirable for use with higher-lift camshafts such as the Piper BP285 due to their greater strength, making them less prone to breakage.

Be aware that if you choose a camshaft lobe profile that extends the powerband into or beyond the 6,500 RPM range (such as the Piper BP285), then in the interests of long-term reliability it would be wise to have the shop both cross-drill and center groove your #2 and #4 crankshaft journals, and then cross-drill the journals for the connecting rods 110° back from Top Dead Center with the drilled passage intersecting the original oil passage in order to prevent lubrication failure resulting from centrifugal forces at high engine speeds. Interestingly, this cross-drilling was done on all production crankshafts in order to meet racing homologation rules until the factory racing team was discontinued, whereupon it was dropped as a cost-cutting measure. The drilled holes should then be chamfered in order to eliminate stress risers and the bearing journals reground. Due to the fact that centrifugal force imparts so much inertia to the oil flow at high engine speeds that it tends to push most of the oil outward through the passages to the crankpins, resulting in starvation of the oil flow to the passages that are closer to the central rotational axis of the crankshaft, it is prudent to install oil flow restrictors into these oiling passages. This simple precaution will thus guarantee that there will be an adequate amount of oil to flow through the passages that are closer to the central rotational axis of the crankshaft in order to feed both the main bearings, the low-pressure gallery, and the camshaft drive chain.

It would also be wise to Nitride-harden your crankshaft, as well as to modify your oil pump, oil filter head, and oil feed passage to the center main bearing of the crankshaft. The use of a high performance ignition system, a 2" Big Bore exhaust system, recessed valve spring seats, lightweight valve spring retainer caps (cups), custom valve springs and valve guides, lightweight tubular chrome-moly pushrods and the lighter 18V bucket tappets, solid rocker adjustment screws (the ones without the oiling passage), and solid rocker arms will both be needed for their additional strength, stronger outer rocker shaft pedestals that support the outer rocker arms from both sides of the rocker shaft, plus a high velocity camshaft drive chain and sprocket set (Kent Part # HV19) will all become desirable at this point as well. The last item is sometimes referred to as a 'silent chain'. Its greater tooth-to-roller contact area reduces the wear that causes the camshaft to gradually go out of phase with the crankshaft, and it is in fact nearly silent. As the speed of the engine achieves 6,500 RPM, the velocity of the camshaft drive chain becomes 110.59 feet per second, making such a high velocity camshaft drive chain highly desirable for sustaining long-term reliability. In order to maximize lubrication of the camshaft drive chain, it is best to install the later version of the oil thrower from the 18GH and later engines. The earlier dished type (like a dinner plate) oil thrower (BMC Part # 12H 775) of the three main bearing engines is fitted with its dish facing forward so that the edge overlaps the inside of the seat of the felt seal inside of the earlier timing chain cover (BMC Part # 12H 3317), i.e., cupped around the seal, thus keeping sprayed oil off of the seal and helping to prevent leakage. Note that the later oil thrower (BMC Part # 12H 1740) should be installed with the center portion having the embossed 'F' facing forwards, further outwards than its outer rim. If installed otherwise, it will rub against the cover. This version of the oil thrower will require the use of one of its later model companion timing chain covers as well (BMC Part # 12H 3510, or BMC Part # CAM 1393).

Timing The Camshaft

Although often viewed as a mysterious, highly technical process best performed only by the most highly initiated engine gurus, the timing of a camshaft is a relatively straightforward matter. Mount a degree wheel onto the front of the crankshaft. Use a piece of stiff wire in order to fashion a pointer that can be attached under one of the nearby machine bolts. Make sure that the pointer is firmly attached and be careful throughout the process to not disturb it.

Now you need to find true Top Dead Center. This may or may not be where the mark is located on the crankshaft pulley wheel on the harmonic balancer (harmonic damper), although the reading should be close. Set up a dial indicator on the deck of the engine block so that its finger touches the crown of the piston in #1 cylinder. Rotate the degree wheel so that its reading is near 0^{o} when the piston approaches Top Dead Center. After the degree wheel is set, record the number on the dial indicator gauge when you are at approximately Top Dead Center. Slowly rotate the engine clockwise as viewed from front until the dial indicator gauge reads .050" (1.27mm) from the Top Dead Center measurement. Record both this number and the degree wheel reading on a piece of paper. Continue rotating the crankshaft until you pass Top Dead Center and the piston begins to descend. Watch the degree wheel until its reading is the same as when you recorded your first measurement, and then record the degree wheel position. True Top Dead Center lies at a point on the degree wheel midway between these two points. Rotate the crankshaft clockwise to the calculated midway point. You will now be at true Top Dead Center. Without moving the crankshaft, either rotate the degree wheel or bend the pointer to indicate zero on the degree wheel at the previously calculated midway point. I would strongly recommend doing this more than once so that you have repeatability in order to ensure that you are at true Top Dead Center.

Rotate the crankshaft and remember that as you pass Bottom Dead Center the engine begins its exhaust stoke. As the piston returns to near Top Dead Center, the intake valve would theoretically begin to open. Keep rotating past Top Dead Center until the degree wheel indicates the Lobe Centerline Angle (usually 104º – 110º, depending upon the camshaft specification). The piston is now at the point that the intake valve on #1 cylinder would be fully open and where the camshaft manufacturer recommends that you set the camshaft.

At this point, rotate the camshaft so that the intake lobe of #1 cylinder (the second lobe from the front) is at full lift. With the crankshaft set at the Lobe Centerline Angle, you can install both the drive and the driven sprockets and their camshaft drive chain. When doing this, make sure that the keyways in the sprockets and shafts are very close to being lined up on their shafts. Use an Allen wrench in order to release the camshaft drive chain tensioner so that the camshaft drive chain is properly tensioned.

In order to position the lobe of the camshaft accurately in phase with the crankshaft, you need to go through a process of finding the actual top of the lobe just as you did when finding true Top Dead Center. Install a lubricated tappet and an oiled pushrod into the intake hole for the #1 cylinder (second hole from the front on all B Series engines). Set up the magnetic stand with the dial gauge on the deck of the engine block so that you can measure the pushrod lift of #1 intake lobe. Using the cupped end at the top of the pushrod as your measurement point, follow the same process that you used in finding Top Dead Center in order to find peak pushrod lift. Rotate the camshaft so that peak pushrod lift on #1 cylinder occurs at precisely this point.

You can either use a vernier camshaft driven sprocket or offset Woodruff keys in order to make the sprockets fit into this proper position.

Lock everything down and recheck everything in order to verify that everything is the same as before. You can then take open, closed, and lift measurements for each valve in order to be sure that the camshaft is not of defective manufacture.

Timing Chain Covers

Be aware that three different timing chain covers were used on the MGB engine. The first (BMC Part # 12H 3317) had a leakage-prone felt oil seal (BMC Part # 88G 561) and its timing plate with ignition timing marks on the bottom, requiring that one crawl under the car in order to check the ignition timing with its pulley wheel on the harmonic balancer (harmonic damper) (BMC Part # 12H 963). This timing chain cover is found an 18G, 18GA, 18GB, 18GD, and 18GF engines. The second and third timing chain covers (BMC Part # 12H 3510, and BMC Part # CAM 1393, respectively) both use a modern lip type oil seal that has an Outside Diameter (OD) of 2.312", an Inside Diameter (ID) of 1.562", and a Width (W) of 0.375". These have their timing pointer plate with its ignition timing marks on top. In order for the timing marks on the pulley wheel on the harmonic balancer (harmonic damper) to align properly with the timing pointer plate, you must use the 5" (127mm) diameter pulley wheel on the harmonic balancer (harmonic damper) (BMC Part # 12H 3515) with the second timing chain cover (BMC Part # 12H 3510) found on 18GH, 18GJ, 18GK, 18V584, 18V585, 18V672, and 18V673 engines. The 6 ½" (165.1mm) diameter pulley wheel on the harmonic balancer (harmonic damper) (BMC Part # 12H 3516) whose timing marks align with the pointers on the later timing chain cover (BMC Part # CAM 1393) is found on 18V836, 18V837, 18V797, 18V798, 18V18V801, 18V802, 18V883, 18V884, 18V890, and 18V891 engines. The timing pointer plate is located further out on this later timing chain cover in order to compliment the larger-diameter pulley wheel of the harmonic balancer (harmonic damper).

The OE front crankshaft oil seal (Moss Motors Part # 120-000) for the timing chain cover tends to leak engine oil because there is not a built-in device for ensuring that the oil seal is properly centered with the crankshaft centerline after the oil seal is replaced. A special factory service tool that is slipped onto the crankshaft and into the oil seal after the timing chain cover is installed but before the timing chain cover machine bolts are tightened is supposed to accomplish this. However, I have never seen this tool offered for sale in the United States. However, there is an effective substitute: the crankshaft drive sprocket for the MGB simplex (single row) camshaft drive chain (BMC Part # 12H 4201, Moss Part # 460-425). When its tapered end is slipped onto the crankshaft and into the oil seal, it will center the oil seal perfectly while the timing chain cover is being torqued to its specified settings of 6 Ft-lbs for the ¼" machine bolts and 14 Ft-lbs for the 5/16" machine bolts. Be aware that the flange should have significant flutes pressed in between the holes, and the area around the holes is also shaped in a particular way. Check to be sure that the sealing flange is flat, and not bowed. If yours is an engine that is equipped with one of the earlier timing chain covers that has a flat faced flange (BMC Part #'s 12H 3317 and 12H 3510), be sure to not lose the elliptical washers (BMC Part #'s 2K 5197 and 2K 7440) that are peculiar to it. These are necessary for spreading the sealing load evenly across the face of the flange of the timing chain cover. Keep them paired with their respective timing chain cover machine bolts as they have become very hard to obtain in the USA. However, they are still available from the MG Owner's Club in the UK (1/4" washer, MGOC Part # 2K 5197; 5/16" washer, MGOC Part # 2K 7440). The later timing chain cover (BMC Part # CAM 1393) have reinforcing ribs on their flange, permitting the use of simple, easy-to-obtain round machine washers. Be careful to not exceed the recommended torque values as warpage of the sealing flange of the timing chain cover can be a very possible result, with leakage as a consequence.

I would also suggest that you retain the use of the later type of crankshaft oil thrower (BMC Part # 12H 1740) that is common to all five-main-bearing engines, as well as one of its three matching timing chain covers (BMC Part #'s 12H 3510 and CAM 1393) that all use the press-fitted synthetic rubber oil seal (BMC Part # 12H 1740) rather than the earlier timing chain cover (BMC Part # 12H 3317) with its leak-prone felt oil seal (BMC Part # 88G 561). However, be aware that the Original Equipment synthetic rubber front crankshaft oil seal is technologically obsolete and should be replaced with a higher performance National brand oil seal (National Part # 1873) as a long-term solution in order to preclude leakage. This elastomeric oil seal is internally reinforced by a stainless steel band instead of an injection-molded plastic ring, offers complete compatibility with either petroleum-based or synthetic oils and can withstand service temperatures up to 450° Fahrenheit (232.2° Celsius). The PTFE material's low coefficient of friction ensures performance for shaft speeds up to 4,500 surface feet per minute while its lay-down sealing lip design ensures long life in spite of shaft / bore misalignment and excessive shaft thrust or movement. Installed into a freshly rebuilt engine, it should easily outlast the bearings, making oil leakage from the front crankshaft oil seal of the timing chain cover into something for others to be bothered with.

Tappets

Junk (soft) tappets have become a very real danger nowadays. Many people end up with ruined camshafts as a result of improperly hardened tappets and think that they purchased a faulty, improperly hardened camshaft or somehow blew the running-in process, even though they went to great care both prior to and during it. In order to help offset the wearing effects of the higher pressures on the tappet bores that result from the use of higher lift camshafts, a lighter version of the bucket tappet used in the 18V engine variants has been developed by Arrow Precision that has provision for additional lubrication. These are a worthwhile addition to any engine as they reduce wear on both the camshaft lobes and the faces of the tappets, as well as on the tappet bores and the load bearing lower end of the pushrods. While an Original Equipment bucket tappet weighs 47.5 grams, this high performance version weighs only 39.7 grams, adding an extra 50 RPM to the maximum safe speed of the engine. The wall thickness of the tappet has been wisely left at .050" (1.27mm) in order to preclude breakage resulting from the high side thrust loadings incurred with high lift and / or long-duration camshaft lobe profiles. They are made of carburized low carbon steel, which is an ideal tappet material, providing a shock resistant inner core with an external 'skin' that is much harder than normal in order to resist wear. This is the same technology used in the Original Equipment rocker arms. The hardening of the tappet is the last step in the manufacturing process, so testing for Rockwell hardness can only be done on the finished product. The oiling hole is drilled in the shank of the tappet after lathe turning, but before reaming and centerless grinding, and then the tappet is heat-treated in order to eliminate any stresses in the surface of the material that have resulted from the machining process. The tappet then emerges hardened as a result of the heat-treating process. Testing for Rockwell hardness is done by impacting the surface with a slender diamond-tipped punch-like bit, and then the resulting indentation is measured for depth with a needle-like gauge. Once all of the machining procedures are completed, the tappet is given a black phosphate coating. Do not remove this black phosphate coating. This coating helps the bedding-in procedure by aiding oil retention on the surface of the tappet.

While chilled iron (white iron) tappets have a greater Rockwell Hardness, they lack the additional lubrication provided by the side drainage bucket tappets and are more appropriate for high engine speeds with radical camshaft lobe profiles such as that of the Piper 300 that are intended for exclusive use on a racetrack. Iron has carbon levels of up to around 4%, as opposed to steel which has up to around 1%. When molten iron cools slowly, some of this carbon comes out of solution in the form of graphite flakes, with a lower-carbon matrix of a relatively soft structure (pearlite and ferrite). This results in the grey iron that is commonly used for engine blocks. However, if the molten iron is cooled very quickly, i.e., chilled, the carbon is trapped while it is still in solution resulting in a very high carbon, hard, and brittle structure called cementite. As the rapid cooling will be most severe at the surface of the casting and become more gradual as it progresses into the casting, the molecular structure will also change across its cross section. There will be an extremely hard surface layer of high-carbon cementite, through a mottled layer, then finally a softer and tougher grey iron core. This is somewhat analogous to a carburised or case-hardened surface in steel. If the casting is fractured, the cementite area appears white, hence the term 'white iron'. It is used for applications which require a very hard, wear-resistant surface with a reasonably tough, shock-resistant interior core, such as tappets. It should also be noted that chilled iron (white iron) tappets are compatible only with steel camshaft lobes.

Whichever tappet design you elect to use, make sure that each one has been individually tested in order to be sure that it has a Rockwell Hardness of at least 55 HRc, or, preferably, 60 HRc. Otherwise, you may be faced with a ruined camshaft.

Most people think that the purpose of the cavity within the old long barrel tappets is to keep reciprocating mass to a minimum. This is only part of the truth. The engineers at the factory designed an elliptically shaped aperture canted at a 60° angle into the sides of the barrel tappets in order to maximize its scraper length. The oil on the floor of the tappet chest, which was allowed to drain into the tappet, would then be spun outwards in order to lubricate the tappet bore as the tappet was lifted by the camshaft. While the rate of gravity flow into the tappet bore is fairly constant, oil enters the cavity within the tappet and momentarily collects there. While the tappet is at the bottom of its travel, the lack of spring-loaded downward pressure from the valvetrain allows it to glide along on the base circle of the camshaft, thus its rotational speed is at its slowest. As the tappet rises, its upward acceleration causes the oil to puddle in the floor of the tappet. While the rotational speed of the camshaft is constant, because the surface area of the camshaft is greater on its lobe than on its base circle, the rotational speed of the tappet increases as it rises, and with the increase in rotational speed, the centrifugal force acting upon the oil puddled in the cavity of the tappet also increases. As the rate of rotation of the tappet increases, centrifugal force combines with capillary action to cause the oil within the tappet to move outwards through the aperture, both lubricating the tappet bore and, in the case of the bucket tappet, draining away the current puddle of oil and thus assuring its replacement with a fresh supply of lubricant to the load-bearing lower end of the pushrod along with its seat inside of the tappet. Thus, the bore receives a small increase in lubrication at the moment of its highest sidethrust loading. The higher the engine speed, the faster the rate of tappet rotation and the greater the sidethrust acting on the side of the tappet becomes, hence the greater the need for lubrication. As a side effect, the faster the rate of rotation, the greater the centrifugal force will be, hence the greater the aid to lubrication. The design of the new side drainage 18V bucket tappet thus combines the best design features of both the older long barrel tappet and the lighter 18V bucket tappet. In addition, the oil drain hole of these tappets has the additional small benefit of preventing a few grams of oil from collecting inside of the bore of the tappet and adding to its reciprocating mass, thus inducing unnecessary wear.

There is a current myth concerning the drainage from the top section of the tappet being for the purpose of reducing the additional weight of excess oil and thus permitting higher engine speeds without incurring valve float. If you could measure the effect of the weight of the oil trapped in the bucket tappet, it would only apply during the period in which the tappet is being lifted, and the effect of any addition inertia would be quite small. At the top of its travel, its upward inertia would immediately cause the oil to simply fly out of the tappet. Thus when the point of maximum travel is reached, the tappet decelerates under the resistance provided by the valve springs, and then begins to descend and the valve begins to close, the weight of the few grams of oil in the tappet no longer having any effect upon the limiting factor of valve float.

Tappet failure is more likely in the case of camshaft lobe profiles that produce a radical amount of lift or with high ratio rocker arms, either of which can produce excessively high pressure loadings at the camshaft lobe / tappet interface. Due to their reduced base circle forcing the use of steep ramps and also restricting curvature design at full lift, this risk increases when a reground camshaft lobe profile is employed. Reducing reciprocating mass in the valvetrain in order to allow the use of lower pressure valve springs can only partially compensate for the greater pressures resulting from these increased loadings. While it is possible to harden the metal of a conventional tappet to the point that it can deal with these increased pressure loadings, the effects of such an extreme degree of hardening cannot be isolated to its friction surface alone. Unfortunately, it will also extend into the inner core of the tappet, consequently depriving it of its resilient shock-absorbing capability. This loss of shock absorbing capacity can result in the tappet hammering the lobe of the camshaft into oblivion.

In order to deal with this risk, Arrow Precision now offers a coating that both offers the required hardness while leaving the shock absorbing qualities of the core of the tappet unaltered. This coating, Diamond-Like, is a single layer, pure carbon coating, producing a very low coefficient of friction and good running-in qualities. It has a friction coefficient that is less than 25% of that of a standard nitride-hardened steel tappet. More importantly, in a tappet application it offers greatly reduced sliding load in comparison to nitride-hardened tappets. It has excellent resistance to wear, and can cope with the higher sliding speeds common to radical camshaft lobe profiles. Tappet / lobe interface pressures that would lead to seizure or cold welding under normal conditions are tolerable with this advanced coating, and even a momentary total starvation of lubricant will not result in the failure of a Diamond-Like coated tappet. High-lift performance cams exert much greater friction forces on the sliding face of a follower. Lowering the sliding friction while retaining oil for lubrication during the running-in period is one of the main problems facing the designer. Diamond coatings and Superfinishing offer just one solution to this problem. A ground finish will give surface readings of about 0.4 - 0.3RA, a lapped finish will give 0.2RA, while a Superfinish will give an amazing 0.05RA. Add to this a Diamond coating to negate the need for oil retention and a large part of the running-in problem is solved. You can contact Arrow Precision by e-mailing to enquiries@arrowprecision.co.uk .

The limiting factor for camshaft lobe design is the maximum acceleration rate of the valvetrain. Should the acceleration rate be fixed by limiting factors of either the rocker arm ratio or tappet diameter, then a design point is reached at which increases in tappet lift at critical piston velocities can only be achieved by means of the employment of a camshaft lobe profile that results in an increase in a duration of valve timing. This is due to the geometries involved. Opening the valve further at any given point in the rotation of the crankshaft will require that the opening point will also have to occur at an earlier point in the rotation of the camshaft. Conversely, it will also have to close at a later point in the rotation of the camshaft. This is the reason for high lift racing camshafts for the MGB having such long duration valve timing phases (300° to 320°). Unfortunately, this has a tendency to result in both overlap and intake valve closing points that will produce a very narrow, peaky power curve with little in the way of usable low-end torque. The dilemma is that although the desired amount of valve lift at the critical periods of high piston velocity is attained, it is achieved at the expense of the valve also being open at times when it is detrimental to the flexible performance that is desired in a street engine. The solution is either the use of a larger-diameter tappet, a roller camshaft lobe profile and roller tappet, or a rocker arm with an increased lift ratio which will hold the rate of valvetrain acceleration to a minimum.

Remachining the tappet bores in the engine block in order to permit the installation of larger-diameter (.9345" / 23.7363mm), longer (1.752" / 44.5008mm) Original Equipment specification barrel tappets designed for use in the "New C" Series engine of the MGC is not the simple solution that it may initially seem to be. Because of the larger diameter of the MGC tappet, it is necessary to machine material from the journals of the camshaft, its bushings, and occasionally the engine block in order to provide sufficient clearance for the tappet to engage the lobe of the camshaft. In practice, the amount to be removed from the camshaft is typically not very much, amounting to nothing more than taking the bevels off of the edges of the journals. The tappet bores have to be carefully bored from below in order to establish the proper tappet axis. This boring of the floor of the tappet chest does not usually involve the journal support for the camshaft, although some engine block castings can vary slightly as a result of core shifting during the casting process. Do not bore completely through the floor of the tappet chest, as this will unnecessarily weaken it, rendering it prone to cracking. No more material should be removed than is necessary in order to give sufficient clearance for the tappet at maximum lift. Do not be tempted into substituting Triumph tappets for MGC tappets by the fact that they are both of the same diameter (.9345" / 23.7363mm). The Triumph tappets weigh 69 grams, making them 11% heavier. As if that is not bad enough, they are actually .203" (5.1582mm) longer than the 1.752" (44.5008mm) length of the MGC tappet. While the cup end of the pushrod and the rocker arm ball adjusters are both the same size (11/32") as those found in the BMC B Series engine, and the 5mm base thickness of the MGC tappet is the same as that of the 18V bucket tappet, the seat inside of the tappet for the ball end of the pushrod is of a different design in order to accommodate the ½" ball end of the MGC pushrod instead of the domed end of the MGB pushrod. In addition, the MGC pushrod also is .199" (5.0546mm) shorter in length that that of the 18V pushrod (10.591" vs. 10.790", 269.011mm vs. 274.066mm), so the approach of using the longer Triumph tappets has the dual disadvantage of requiring custom-length pushrods and, due to the 11% greater weight of the Triumph tappet, the installation of stronger valve springs. The resulting additional pressure of these stronger valve springs will result in faster wear of the lobes of the camshaft, the rocker shaft, the rocker arm bushings, and the thrust face of the rocker arm. Racing engines are disassembled and inspected several times during a racing season, but this is obviously not a practical solution for the streetable engine that is the goal of this article.

However, a lightweight version of the MGC tappet intended for use in the B Series engine is available from Cambridge Motorsport. Because an MGC tappet is larger in diameter (.9345" / 23.7363mm) than an Original Equipment MGB tappet (.8125" / 20.6375mm), its heel engages the ramp of the camshaft lobe a little earlier in the stroke and disengages a little later, thus the valve consequently both opens a little earlier and closes a little later, plus valve lift is increased at most points in the stroke. Not surprisingly, maximum valve lift remains the same, but by beginning and ending that process both earlier and later than would otherwise be possible for a camshaft with such a small radius to its base circle, valvetrain acceleration becomes more gradual, thus reducing valvetrain inertia at the expense of a small increase of duration. This increased duration in turn gives rise to a problem with robbing effects amongst the cylinders at low engine speeds and decreased fuel economy. They also allow the use of a camshaft lobe with more a more radical lift ramp without the lobe running off the edge of the tappet, with the resultant effect of gouging. This in turn, combined with the greater side surface area created by the larger diameter of the MGC tappet, reduces side thrust loading on the tappet by virtue of its 65% greater load bearing surface area and thus permits it to rotate freely under the heavy loadings generated at very high engine speeds, thus preventing failure. This is an old racer's trick. However, because both the duration and overlap of the valve open periods are increased, they will require a faster idling speed and the powerband will narrow somewhat, although maximum power output will be enhanced. In short, the engine will become more "cammy" and the idle will be rougher than it would be with the same camshaft when used in conjunction with Original Equipment specification tappets. In addition, due to the horizontal port configuration inherent to the Heron-type cylinder head, in terms of airflow there is no worthwhile advantage to a valve lift of more than .455" (11.557mm). Because most of these improvements in power output can be achieved by simply substituting a different camshaft, and the side thrust loadings on Original Equipment-diameter MGB tappets would still not be excessive on an engine with a streetable camshaft, this expensive and radical approach would be of little value to anything other than a race engine equipped with oversize intake valves and a radical camshaft that is intended to operate at very high engine speeds.

The fitting of roller tappets would require machining away substantial material from the bridge section in which the tappets are mounted in order to accommodate their greater length, thus reducing the bearing area for the shanks of the tappets which in turn would require fabricating and press fitting custom-made tubular sleeve extensions into the tappet bridge in order to provide adequate bearing area. However, removing such a radical amount of material would critically weaken the floor of the tappet chest. It would also require custom length pushrods and the development of a custom camshaft lobe profile, as well as increasing valvetrain inertia by virtue of the greater weight of the roller bearings, so this option is also both undesirable as well as impractical.

Big Bore Engines

Much advertising has been done over the years by vendors of overbore kits. Claims of massive increases in power output that spark mental images of acceleration powerful enough to rotate the earth are really just so much propaganda. The truth is that these kits seem to most commonly come in two sizes: 1868cc and 1950cc. An 1868cc kit uses +.060" oversize pistons, which means that when the day comes that you need new pistons you will most likely need to either have the engine block sleeved (Cost: $500 and up), obtain another engine block (cheaper if you can find a good one), or bore it out to the point that it enters into the 1900cc-1950cc Big Bore category and incur the trouble and expense of retuning the engine all over again, including the cost of larger venturi carburetors and a complementary intake manifold. The additional displacement of the 1868cc engine works out to slightly more than an additional 4 cubic inches, which in and of itself just does not give enough additional displacement (about 4.5%) to rationally justify any of these long-term hassles. If the cylinder head is Original Equipment specification, the additional displacement will result in high fuel / air charge velocities occurring earlier in the powerband. You will have a bit more torque at low engine speeds, but the engine will also attain its peak power output earlier in the powerband. With a Fast Road cylinder head, the engine can continue to wind out nicely after the point where the power output of an unmodified cylinder head would seem to run into a wall.

In short, unless you have already reached the factory's specified maximum overbore size of +.040" (which will give a displacement of 1840cc) and are willing to perform other power enhancing modifications, do not bother spending the extra money for trick oversize pistons unless you are prepared to meet the considerable financial expense that will be necessary in order to fully exploit the increased power output potential that they give. However, if you are willing to do this, the best pistons for an 1868cc application are the flat-topped Accralite +.060" oversize 3.2201" (81.79mm) forged pistons (Accralite Part # 1195xc8179). Accralite's heat treatment procedure is based on T6 international specifications that give reduced stress, maximum hardness, and a greater working life span. Constructed of RR58 alloy, commonly referred to as Rolls Royce 58 (2618A), they make use of 13/16" (20.638mm) wrist (gudgeon) pins that will conveniently fit the small ends of the connecting rods that are used in the five-main-bearing versions of the B Series engine, have a compression height of 1.6449" (41.78mm), and a 7mm crown thickness that together will enable the custom-tailoring of both the compression ratio and squish (quench) area, plus they weigh a very respectably light weight of 356 grams. Accralite also takes the time to match the weight of all of their pistons to within .1 gram so that you will not have to bother with this rather tricky and time-consuming weight matching task. Accralite also goes to the trouble of producing their own rings for this piston (Accralite Part # A81790/1-12-28), as well as their own lightweight wrist (gudgeon) pins (Accralite Part # 5001).

The 1950cc kits do produce more low-end torque when used in combination with Original Equipment specification cylinder heads, camshaft, and modified Original Equipment carburetors, but the potential of the increased displacement cannot be fulfilled without spending the money required for professional headwork with oversize intake valves such as in Peter Burgess' Fast Road Plus specification, a Big Bore tubular exhaust manifold and exhaust system, and either a Maniflow or a Special Tuning intake manifold in order to accommodate the larger displacement, plus either 1¾" SU HS6 or 1¾" SU HIF6 carburetors. This nominal 8% increase in engine capacity can produce around 15% more power in the midrange of the powerband and between 10% to 12% more power at peak power engine speed using an otherwise standard engine. However, when paired with an Original Equipment camshaft and unmodified cylinder heads peak horsepower is developed at around 4,650 RPM and peak torque sees a similar drop in engine speed (to about 2,250 RPM), the overall effect of which is a power curve that is not entirely dissimilar to that of a diesel engine. This does point to a need for additional modifications as a desirable part of this conversion.

With a Peter Burgess Fast Road cylinder head such an engine really starts to deliver impressive power as it is able to not only deliver the added torque, but also the lost enthusiasm to rev is largely regained. The added breathing potential ensures that with an Original Equipment camshaft the engine does not start to fade until 5,000 RPM, still a lower engine speed than seen with a standard bore engine, but the gains in both horsepower and torque output are usually masked in normal use. Midrange power is the main benefactor but the upper reaches of the powerband will still seem to be capped, especially when you have prior experience with standard bore engines with the same cylinder head modifications. This capped effect is the result of the way that increased displacement softens the power output characteristics of the camshaft. To restore the original character of the engine, a change to a camshaft lobe profile that has a greater duration of about 280º is necessary, instead of the Original Equipment camshaft's 252º of duration. With the increase of displacement capacity softening the power surge effect that results from the longer duration, the effect will be to restore the peak torque and peak power points to much the same locations in the powerband as they are with the standard MGB engine, assisting in maintaining the appropriateness of the Original Equipment gear ratios, plus, of course, the added benefit of tapping into the airflow gains of the modified cylinder head.

Due to the variances in cylinder wall thickness that are the result of a less-than-optimum casting process, it is necessary to torque a reinforcing plate to the engine block prior to overboring to prevent the finished bore from being distorted. Fitting 1950cc Big Bore pistons requires boring the cylinders out so far that the side thrust loading of the piston against the thin cylinder walls of some engine blocks can in some cases cause the bore to distort, the consequent loss of compression and high oil consumption becoming a headache. Either sonic testing or X-raying of the engine block in order to determine cylinder wall thickness prior to boring becomes advisable at this point. The absolute minimum cylinder wall thickness after boring should be considered to be .100" (2.54mm). Even so, a certain amount of cylinder wall flexure is to be expected, so state-of-the-art pistons with very thin rings are likely to become a necessity. When boring to such an extreme diameter in a B Series engine block, it is not uncommon to encounter porosity, in which case the installation of sleeves will become a necessity. Such sleeves (liners) are available from Westwood

Trading (Westwood Trading part # WCL 22). They have a website at http://www.westwoodtrading.co.uk/ . These have a wall thickness of .105" (2.667mm), are 6.060" (153.924mm) long, and have an external diameter of 3.265" (83.931mm), making them appropriate for engines in the +.020" to +.060" oversize category. In addition, sleeves can also be obtained for the B Series engine from County (County Part # CL1950). These have a wall thickness of .130" (3.302mm), are 6.060" (153.924mm) long, and have an external diameter of 3.380" (85.852mm), making them appropriate for engines in the 1869cc-1948cc category. Sleeves have the additional advantage of being made of spun cast iron that is of better quality than the "block-type" cast iron. Their advantage is better wear characteristics and superior oil control, which are the reasons that Peter Burgess uses them in every large-displacement engine that he builds! If the sleeves are shrink-fitted and silver-soldered into place, then the heat distribution should be as good as that of a normal cylinder of equal wall thickness, although the ultimate rigidity at the cylinder / engine block interface will be slightly lessened. In addition, the larger the bore becomes, the less the sealing area remaining between the cylinders for the cylinder head gasket on the deck of the engine block becomes, so the torque settings of the cylinder head will have to be scrupulously maintained in order to avoid either pressure leakage between the cylinders or a blown cylinder head gasket. Another downside is that the future reboring and fitting of oversize pistons cannot be done as the cylinder walls will be too thin. However, both of these drawbacks can be overcome by offset boring of the engine block and fitting oversize sleeves with adequate wall thickness. This involves offset-boring the cylinders toward their respective ends of the engine block in order to maintain sufficient clearance between the cylinders, prevent the blowing of cylinder head gaskets, and the development of "hot spots" that can cause cylinder distortion. Offset connecting rods would consequently become necessary as well, although they would cause uneven loading of the big end bearings of the connecting rods and consequent accelerated wear. An oiling system with modified feed passages can assist in protecting the bearings from the additional pounding of the increased power output. Of course, this implies that the engine would have to be built as oil-tight as possible, but all of these challenges have been met before, so dealing with these issues would hardly involve blazing new trails into uncharted territory. All this, of course, is not to mention the problems of the excessive heat that would be produced with such an uprated power output, which in turn will require modifications to the coolant system. For anything other than use on a racetrack, a fully developed Big Bore engine is likely to prove to be financially impractical. A compromise displacement of 1900cc-1926cc is probably the practical limit for a fully developed street engine. No matter what you do, both the ignition timing and the carburetion have to be scrupulously maintained, otherwise you will have running problems with a Big Bore B Series engine.

Most 1950cc kits commonly on the market today make use of Hepolite +.040” oversize 83.57mm domed Lotus Twin Cam pistons in order to produce an additional 8.2% (9 cubic inches) of displacement beyond that of an Original Equipment specification engine. These pistons do not have offset wrist (gudgeon) pins as the Original Equipment Hepolite pistons do, so the bores must be offset in order to compensate for this difference. They use a standard thickness set of rings that lack the flexibility to compensate for flexure of the cylinder wall. They also have tops that are approximately .090” (2.286mm) closer to their wrist (gudgeon) pins than standard MGB pistons, thus it is necessary to end mill the deck of the engine block .100” (2.54mm) in order to achieve a reasonable Geometric Compression Ratio (GCR) of 9:1 with the 39cc combustion chamber of the cylinder heads used on the 18V engine. This will place the deck of the engine block very close to the top of the coolant jacket, the consequent loss of rigidity resulting in a risk of cracking in some engine blocks. Because this reduction of the thickness of the deck of the engine block will also consequently decrease the number of threads available for the cylinder head studs, the depth of the threads will need to be carefully examined prior to redecking in order to determine that they will still be able to offer sufficient grip to the threads of the cylinder head studs without incurring the risk of cracking and / or distorting the deck of the engine block when the cylinder head is torqued. In addition, the use of these pistons also require the use of the horizontally-split connecting rods of the 18GG, 18GH, 18GJ, and 18GK engines (BMC Part # 12H 2445) that have bushed small ends in order to accommodate the use of floating pistons. As an alternative, either of the later connecting rods of the 18V engines that have small balance pads (BMC Part # 12H 3596) or no balance pads (BMC Part # CAM 1588) can have their small ends suitably modified in order for the wrist (gudgeon) pins and their bushings to fit properly. These later, lighter connecting rods would also help to compensate for the dynamic effects of the greater reciprocating mass of the larger pistons. Unfortunately, these pistons use failure-prone wire-type circlips in order to retain their wrist (gudgeon) pins, so considerable skill and care must be taken when installing them.

Unfortunately, the .060" (1.524mm) domes of the Hepolite Lotus Twin Cam pistons, combined with their recesses for the valve heads of the Lotus engine, interfere with both the flow of the fuel / air charge and the needed combustion characteristics when paired with a flat-topped combustion chamber as is found in the Heron-type cylinder head of the B Series engine. Whenever the bore of an engine is increased radically, its squish (quench) area is also increased. This factor is aggravated by the excessively wide .125" (3.175mm) squish (quench) band of the Hepolite Lotus Twin Cam pistons, and so flame propagation becomes a problem, especially if domed pistons are used. Let's face it: A domed piston design and the flat-topped Weslake kidney-shaped combustion chamber design are not exactly in harmony with each other. Domed pistons present enough problems in a hemispherical combustion chamber, but in a Weslake flat-topped kidney-shaped combustion chamber, they are bad news.

County makes a cast piston that is, while not being quite as strong as the Original Equipment Hepolite pistons, adequate for a mildly tuned road engine. Developed specifically for increasing the displacement of the B Series engine, its piston crown is of the same crown-to-wrist pin (gudgeon pin) dimension as that of an Original equipment MGB piston, thus eliminating the need for radical shortening of the deck of the engine block. Its lower .010" dome presents less of an obstruction than that of the .060" (1.524mm) dome of the Lotus Twin Cam piston, plus its narrower squish (quench) band is quite appropriate to the 43cc combustion chamber of the 18G through 18GK engines. However, if you are building an engine that is intended to make use of high engine speeds in order to produce maximum power output, then a forged piston that is both lighter and has thin piston rings that will cope with bore flexure better, giving both better oil and compression control, is the preferred way to go.

One of the better pistons for this application is the flat-topped Accralite BGT oversize 3.2874" (83.57mm) pistons (Accralite Part # 1196xc835). They make use of the Original Equipment 13/16" (20.638mm) diameter wrist (gudgeon) pins, have a compression height of 1.6417" (41.7mm), as well as a .243" (6mm) crown thickness, which together will enable the custom tailoring of both the compression ratio and squish (quench) area, plus they weigh a very respectably light 341.5 grams. Like their 1868cc cousins, they are matched for weight to within .1 grams so that you will not have to bother with this rather tricky and time-consuming task. Accralite also goes to the trouble of producing their own lightweight wrist (gudgeon) pins (Accralite Part # 5001) for this piston, as well as their own rings (Accralite Part # A83500/1-12-28) which do a superior job of coping with bore flexure, thus giving better oil and compression control.

The combustion chamber volume of a Big Bore engine is relatively smaller in relation to the cylinder volume on a Big Bore engine than it is on a 1868cc engine, so the pressure rise within it is correspondingly faster than on the smaller bore 1868cc engine, leading to an increased risk of detonation. In addition, this increase in bore size results in a larger squish (quench) area that induces too much turbulence and consequent fuel condensation for flame propagation to be smooth and even unless the combustion chamber is opened up. It also inhibits flame propagation in the areas near the roof of the combustion chamber, a factor aggravated by both the excessively wide .125" (3.175mm) squish (quench) band and the dome of the Lotus TC piston. Due to the positional relationship between the circular cylinder and the kidney-shaped combustion chamber, the increased squish (quench) area increases the velocity of the turbulence in the direction of the spark plug. Due to the direction of the moving fuel / air charge being biased toward the spark plug, this thus guarantees that the turbulence around the valves will be at its lowest when ignition takes place. The position of the spark plug also plays a big part in the detonation problem. As the flame travels outwards towards the lobes of the kidney-shaped combustion chamber, it creates a pressure wave at the advancing periphery of the combusting fuel / air charge. This particular combustion effect is more severe with unleaded fuel as it is more volatile than slower-burning leaded fuel. As the pressure wave advances, the unburned fuel / air charge trapped in front of it is increasingly compressed against the roof of the combustion chamber. When the pressure wave arrives in the vicinity of the hot exhaust valve, its velocity and pressure is at its greatest just as the remaining volume available for the unburned fuel / air mixture is decreasing at its fastest rate. Because the area around the exhaust valve is the hottest region of the combustion chamber, its environmental conditions are best for producing preignition and detonation, and the arrival of the pressure wave compressing the unburned fuel / air charge against it triggers the event. While opening up the combustion chamber in order to decrease the squish (quench) area will help to alleviate the squish (quench) problem, the resultant increase in combustion chamber volume can increase the likelihood of preignition in the vicinity of the hot exhaust valve at the expense of a lower compression ratio, which in turn will prevent the maximum power output potential of the engine from being attained. Obviously, it is difficult to reach a happy medium, so the combination of both the clearance dimension between the piston crown and the cylinder head and the width of the squish (quench) band adjacent to the dish of the piston crown is critical to producing the correct amount of squish (quench) turbulence. It would seem that the most practical cold clearance dimension is .012" (.3048mm), with a 10mm (.401") to 3/8" (.375" / 9.525mm) squish (quench) band around the periphery of the piston, the combination of which seems to be adequate for producing this desired effect. However, it should be noted that this in turn may force a compromise when selecting a high-lift camshaft that is intended to be combined with oversize valves, some of them produce so much lift that it becomes necessary to relieve the deck of the engine block to a valve clearance depth that is greater than that of the piston / cylinder head clearance. The edge of the compression ring may be directly exposed to the heat of combustion, in turn leading to premature ring failure and piston land breakage.

These problems could be minimized by using less ignition advance, a lower compression ratio, and a mild camshaft such as the Piper BP270, but this solution would in turn result in the engine reaching its peak output at a less-than-optimum engine speed. Due to the increased displacement, higher port velocities occur at lower engine speeds, resulting in a flatter power curve that reaches its peak at substantially lower engine speeds. What is really needed is either a Piper BP285 camshaft or a Piper BP270 camshaft that has been coupled with a 1.69" (42.926mm) diameter intake valve in order for the engine to fulfill its power output potential and keep the power peak where it should be in order to retain the Original Equipment transmission ratios, as well as a Geometric Compression Ratio (GCR) of 10.5:1 in order to keep the power output at an efficient level.

Of course, the combination of a high compression of 10.5:1 and the reduced sealing area between the cylinders of a Big Bore engine will consequently mean that stress upon the sealing properties of the cylinder head gasket will be rather extreme. This being the case, a special means of ensuring that the seal of the cylinder head gasket will not be blown by these extreme stresses must be employed. A purpose-made Big Bore Payen cylinder head gasket will normally suffice, but for severe service applications, O-ringing by the placement of a 20 gauge copper wire around the inner periphery of the cylinder head gasket will accomplish this task. This wire should protrude .008" (.2032mm) above the surface of the deck of the engine block so that it will compress and effectively seal the combustion chamber.

So, as you can see, there is still a problem that remains to be solved: That of finding a way to use a Geometric Compression Ratio (GCR) of 10.5:1 and still enable the engine to run reliably using the 93 Octane Oxygenated fuel. This can be accomplished by using a set of customized forged pistons.

Compression Ratio and Fuel

A desired compression ratio is often attained by end milling the deck of the engine block to the appropriate clearance height, as in the case of using Lotus TC pistons. Of course, that automatically implies that the pushrods will have to be shortened in order to maintain proper rocker arm / valve stem geometry, but Crane Camshafts offers that service too, so that is not a problem, although this solution is a bit costly.

It should be noted that Geometric Compression Ratio (GCR) figures are somewhat misleading. To most people they imply that the cylinders are completely filled with fuel / air mixture when compression begins, and that therefore the effective amount of compression is the same as that of the Geometric Compression Ratio (GCR). However, this is not the case. Many factors influence the Volumetric Efficiency of an engine, among them the connecting rod / stroke ratio, valve size, valve timing, the length of the runners of the intake manifold, and the optimum flow rate of the ports. This volumetric inefficiency that is inherent to all normally aspirated engines in turn results in an Effective Compression Ratio (ECR) that is altogether quite different from an engine's Geometric Compression Ratio (GCR). It is the Effective Compression Ratio (ECR) that reflects how much fuel / air mixture is actually entering into the engine and being compressed, and thus is the true determining factor of the Octane requirements of the engine.

A 7.0:1 Effective Compression Ratio (ECR) requires 92-Octane fuel.

A 7.5:1 Effective Compression Ratio (ECR) requires frequent attention to ignition timing and carburetion tuning in order to run on 92-Octane fuel.

Over 7.5:1 Effective Compression Ratio (ECR) requires the above, as well as that the Octane level be boosted with additives.

Over 8.0:1 Effective Compression Ratio (ECR) requires racing fuel.

Upon the initial running of your new engine, you still might run into problems with either preignition, or that great destroyer of engines (God forbid!), detonation. If that should occur, do not become frustrated or fall into a state of despair. Instead, run down to your friendly local paint store and buy a big can of Toluene. Toluene is a common ingredient in Octane Boosters in a can. Pure toluene has a RON octane rating of 121 and a MON Octane Rating of 107, leading to a (R+M) / 2 octane rating of 114 octane. The (R+M) / 2 formula is how ordinary road-use fuels are octane-rated here in the USA. Note that Toluene has a sensitivity rating of (121-107) = 14. This compares favorably over alcohols that have sensitivities in the 20-30 range. The more sensitive a fuel is, the more its performance degrades under load. Toluene's low sensitivity makes it an excellent fuel additive for an engine that may have to face heavy loadings, which is the reason that racers favor it as an additive for raising the Octane Rating of their fuel. If you mix it with 93 octane Premium gasoline you will get the following results:

90% 93 (R+M) / 2 Octane Gasoline + 10% Toluene = 95.2 (R+M) / 2 Octane

80% 93 (R+M) / 2 Octane Gasoline + 20% Toluene = 97.4 (R+M) / 2 Octane

70% 93 (R+M) / 2 Octane Gasoline + 30% Toluene = 99.6 (R+M) / 2 Octane

Another useful octane booster is xylene, which has a RON octane rating of 118 and a MON Octane Rating of 115, leading to a (R+M) / 2 octane rating of 116.5 Octane. Note that Xylene has a sensitivity rating of (118-115) = 3. It is similar to Toluene, usually being mixed with Toluene and advertised as *race formula*. If you mix it with 93 octane Premium gasoline you will get the following results:

90% 93 (R+M) / 2 Octane Gasoline + 10% Xylene = 95.3 (R+M) / 2 Octane

80% 93 (R+M) / 2 Octane Gasoline + 20% Xylene = 97.8 (R+M) / 2 Octane

70% 93 (R+M) / 2 Octane Gasoline + 30% Xylene = 100.2 (R+M) / 2 Octane

However, you should be aware that the low volatility of xylene results in decreased throttle responsiveness during use in cold weather.

Compared to gasoline's specific-gravity of 0.751-g/cc, toluene is 0.881-g/cc and xylene (most likely a mixture of m-xylene; o-xylene; p-xylene) is around 0.871-g/cc. This means that they have more hydrocarbons per gallon to combust with the Oxygen in the air that is being pumped through the engine. The result of using large-percentage mixtures of these aromatics in your fuel is a richer mixture than previously with just pure pump gasoline, so adjustment of the carburetion becomes necessary.

Once you have determined the Octane requirements of your engine after some progressive experimentation, you will have a very good idea of what volume of the dish should be machined into your piston crowns in order to obtain the optimum compression ratio. You will need to disassemble the engine and purchase a new set of piston rings, plus possibly rehone the cylinders, but this is a far better (and cheaper!) approach than the alternative of purchasing new oversize pistons and starting all over again by disassembling the engine and reboring all of the cylinders.

On the subject of fuels, be advised that the gasoline that we purchase for our cars is in the process of a major reformulation. Ethanol, an alcohol derived primarily from corn, is both a domestically produced and renewable fuel. E-10 (10% ethanol) may become a critical component of our fuel infrastructure, being phased into the fuel supply across the country. The problem is that all alcohols are hygroscopic. That means that it attracts and couples with the moisture in the air, resulting in greater amounts of ethanol / water mixture collecting in the fuel tank. Since water is heavier than gasoline, the ethanol / water mixture settles to the bottom of the fuel tank. This is referred to as "phase separation", and eventually the ethanol / water phase is drawn into the fuel delivery system. If there is no water separator in the fuel line, the separated water goes into the carburetor and, consequently, the engine does not run properly. In extreme cases, the engine will stop running. An additional problem that owners will face lies in the reduction of octane, which is critical to the engine's performance. Ethanol is rated at over 100+ octane, and provides the fuel with much of its octane rating. Once water reaches about a .5% level, it will phase separate. With the ethanol settling to the bottom along with the water, the octane of the fuel will be reduced, and this can cause a loss of performance, including preignition which can damage the engine. It also severely worsens fuel economy and power output. There are other problems with E-10 fuel. Ethanol is a powerful solvent that readily breaks up both the tars and the organic sediments that are commonly found in many fuel tanks. The ethanol /

water mix also makes a potent stripping agent for old varnish and gum accumulated from years of gasoline sitting in the tank. These organic contaminants, once loosened from tank walls, can plug fuel filters and carburetor jets quickly, disabling the engine. In cold weather, the ethanol / water phase can also freeze, turning into a syrupy mix that plugs fuel filters. Alcohol also has a lower caloric content than gasoline, thus its heat output, and hence its power output, is inherently less. This results in turn with the driver operating his engine at a wider throttle opening in an attempt to compensate for the decreased power output, with the attendant consequence of decreased fuel economy. Obviously. if you can avoid purchasing E-10 fuel, you would be wise to do so.

Reduction of the Cold Clearance Depth from the Original Equipment specification of .040" to .012" (from 1.016mm to .3048mm) in order to attain optimum squish (quench) characteristics will alter the Clearance Volume, as well as increase both the Geometric Compression Ratio (GCR) and the Effective Compression Ratio (ECR), thus having the following direct effects upon these Compression Ratios:

43cc Combustion Chamber with 16.2cc-dish Piston @ Original Equipment Specification .040" (1.016mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	5.2 cc	5.2 cc	5.3 cc	5.3 cc
Cylinder Head Volume	43 cc	43 cc	43 cc	43 cc
Dish Volume	16.2 cc	16.2 cc	16.2 cc	16.2 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**6.7 : 1**	**6.7 : 1**	**6.8 : 1**	**6.9 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	5.9 : 1	5.9 : 1	6.0 : 1	6.0 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	6.6 : 1	6.6 : 1	6.6 : 1	6.7 : 1
Effective Compression Ratio (ECR) (Piper BP255)	5.5 : 1	5.6 : 1	5.6 : 1	5.6 : 1
Effective Compression Ratio (ECR) (Piper BP270)	5.4 : 1	5.5 : 1	5.5 : 1	5.5 : 1
Effective Compression Ratio (ECR) (Piper BP285)	5.4 : 1	5.4 : 1	5.4 : 1	5.5 : 1

43cc Combustion Chamber with 16.2cc-dish Piston @ .012" (.3048mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	1.6 cc	1.6 cc	1.6 cc	1.6 cc
Cylinder Head Volume	43 cc	43 cc	43 cc	43 cc
Dish Volume	16.2 cc	16.2 cc	16.2 cc	16.2 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**7.0 : 1**	**7.1 : 1**	**7.2 : 1**	**7.3 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	6.2 : 1	6.2 : 1	6.3 : 1	6.3 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	6.9 : 1	6.9 : 1	6.9 : 1	7.0 : 1
Effective Compression Ratio (ECR) (Piper BP255)	5.8 : 1	5.8 : 1	5.9 : 1	5.9 : 1
Effective Compression Ratio (ECR) (Piper BP270)	5.7 : 1	5.7 : 1	5.7 : 1	5.8 : 1
Effective Compression Ratio (ECR) (Piper BP285)	5.6 : 1	5.7 : 1	5.7 : 1	5.7 : 1

43cc Combustion Chamber with 6.5cc-dish Piston @ Original Equipment Specification .040" (1.016mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	5.2 cc	5.2 cc	5.3 cc	5.3 cc
Cylinder Head Volume	43 cc	43 cc	43 cc	43 cc
Dish Volume	6.5 cc	6.5 cc	6.5 cc	6.5 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**7.8 : 1**	**7.9 : 1**	**7.9 : 1**	**8.0 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	6.7 : 1	6.8 : 1	6.8 : 1	6.9 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	7.5 : 1	7.5 : 1	7.6 : 1	7.6 : 1
Effective Compression Ratio (ECR) (Piper BP255)	6.7 : 1	6.8 : 1	6.8 : 1	6.9 : 1
Effective Compression Ratio (ECR) (Piper BP270)	6.2:1	6.2 : 1	6.2 : 1	6.3 : 1
Effective Compression Ratio	6.1 : 1	6.1 : 1	6.2 : 1	6.2 : 1

(ECR) (Piper BP285)				

43cc Combustion Chamber with 6.5cc-dish Piston @ Original Equipment Specification .012" (.3048mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	1.6 cc	1.6 cc	1.6 cc	1.6 cc
Cylinder Head Volume	43 cc	43 cc	43 cc	43 cc
Dish Volume	6.5 cc	6.5 cc	6.5 cc	6.5 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**9.2 : 1**	**9.2 : 1**	**9.3 : 1**	**9.4 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	7.1 : 1	7.2 : 1	7.2 : 1	7.3 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	7.9 : 1	8.0 : 1	8.0 : 1	8.1 : 1
Effective Compression Ratio (ECR) (Piper BP255)	6.7 : 1	6.7 : 1	6.7 : 1	6.8 : 1
Effective Compression Ratio (ECR) (Piper BP270)	6.5 : 1	6.6 : 1	6.6 : 1	6.7 : 1
Effective Compression Ratio	6.5 : 1	6.5 : 1	6.5 : 1	6.6 : 1

(ECR) (Piper BP285)				

43cc Combustion Chamber with Flat-Top Piston @ Original Equipment Specification .040" (1.016mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Cold Clearance Volume	5.2 cc	5.2 cc	5.3 cc	5.3 cc
Cylinder Head Volume	43 cc	43 cc	43 cc	43 cc
Dish Volume	0 cc	0 cc	0 cc	0 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**8.8 : 1**	**8.8 : 1**	**8.9 : 1**	**9.0 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	7.5 : 1	7.5 : 1	7.6 : 1	7.6 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	8.3 : 1	8.3 : 1	8.4 : 1	8.5 : 1
Effective Compression Ratio (ECR) (Piper BP255)	7.0 : 1	7.0 : 1	7.0 : 1	7.1 : 1
Effective Compression Ratio (ECR) (Piper BP270)	6.8 : 1	6.9 : 1	6.9 : 1	7.0 : 1
Effective Compression Ratio (ECR) (Piper BP285)	6.8 : 1	6.8 : 1	6.8 : 1	6.9 : 1

43cc Combustion Chamber with Flat-Top Piston @ .012" (.3048mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	1.6 cc	1.6 cc	1.6 cc	1.6 cc
Cylinder Head Volume	43 cc	43 cc	43 cc	43 cc
Dish Volume	0 cc	0 cc	0 cc	0 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**9.4 : 1**	**9.5 : 1**	**9.6 : 1**	**9.7 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	8.0 : 1	8.0 : 1	8.1 : 1	8.1 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	8.9 : 1	8.9 : 1	9.0 : 1	9.0 : 1
Effective Compression Ratio (ECR) (Piper BP255)	7.4 : 1	7.5 : 1	7.5 : 1	7.6 : 1
Effective Compression Ratio (ECR) (Piper BP270)	7.3 : 1	7.3 : 1	7.4 : 1	7.4 : 1
Effective Compression Ratio	7.2 : 1	7.2 : 1	7.3 : 1	7.4 : 1

(ECR) (Piper BP285)				

39cc Combustion Chamber with 16.2cc-dish Piston @ Original Equipment Specification .040" (1.016mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	5.2 cc	5.2 cc	5.3 cc	5.3 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	16.2 cc	16.2 cc	16.2 cc	16.2 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**7.1 : 1**	**7.2 : 1**	**7.2 : 1**	**7.3 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	6.2 : 1	6.2 : 1	6.3 : 1	6.3 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	6.9 : 1	6.9 : 1	6.9 : 1	7.0 : 1
Effective Compression Ratio (ECR) (Piper BP255)	5.8 : 1	5.9 : 1	5.9 : 1	5.8 : 1
Effective Compression Ratio (ECR) (Piper BP270)	5.7 : 1	5.7 : 1	5.8 : 1	5.8 : 1
Effective Compression Ratio (ECR) (Piper BP285)	5.7 : 1	5.7 : 1	5.7 : 1	5.9 : 1

39cc Combustion Chamber with 16.2cc-dish Piston @ .012" (.3048mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	1.6 cc	1.6 cc	1.6 cc	1.6 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	16.2 cc	16.2 cc	16.2 cc	16.2 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**7.5 : 1**	**7.6 : 1**	**7.6 : 1**	**7.7 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	6.5 : 1	6.5 : 1	6.6 : 1	6.6 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	7.2 : 1	7.3 : 1	7.3 : 1	7.4 : 1
Effective Compression Ratio (ECR) (Piper BP255)	6.1 : 1	6.2 : 1	6.2 : 1	6.2 : 1
Effective Compression Ratio (ECR) (Piper BP270)	6.0 : 1	6.1 : 1	6.1 : 1	6.1 : 1
Effective Compression Ratio	5.9 : 1	6.1 : 1	6.0 : 1	6.1 : 1

(ECR) (Piper BP285)				

39cc Combustion Chamber with 6.5cc-dish Piston @ Original Equipment Specification .040" (1.016mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	5.2 cc	5.2 cc	5.3 cc	5.3 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	6.5 cc	6.5 cc	6.5 cc	6.5 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	. 4 cc	.4 cc
Geometric Compression Ratio (GCR)	**8.4 : 1**	**8.4 : 1**	**8.5 : 1**	**8.6 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	7.2 : 1	7.2 : 1	7.2 : 1	7.3 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	8.0 : 1	8.0 : 1	8.0 : 1	8.1 : 1
Effective Compression Ratio (ECR) (Piper BP255)	6.7 : 1	6.7 : 1	6.8 : 1	6.8 : 1
Effective Compression Ratio (ECR) (Piper BP270)	6.6 : 1	6.6 : 1	6.6 : 1	6.7 : 1
Effective Compression Ratio (ECR) (Piper BP285)	6.5 : 1	6.5 : 1	6.6 : 1	6.6 : 1

39cc Combustion Chamber with 6.5cc-dish Piston @ .012" (.3048mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	1.6 cc	1.6 cc	1.6 cc	1.6 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	6.5 cc	6.5 cc	6.5 cc	6.5 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**9.0 : 1**	**9.0 : 1**	**9.1 : 1**	**9.2 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	7.6 : 1	7.6 : 1	7.7 : 1	7.8 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	8.5 : 1	8.5 : 1	8.6 : 1	8.7 : 1
Effective Compression Ratio (ECR) (Piper BP255)	7.3 : 1	7.2 : 1	7.2 : 1	7.3 : 1
Effective Compression Ratio (ECR) (Piper BP270)	7.0 : 1	7.0 : 1	7.0 : 1	7.1 : 1
Effective Compression Ratio (ECR) (Piper BP285)	6.9 : 1	6.9 : 1	7.0 : 1	7.0 : 1

39cc Combustion Chamber with Flat-Top Piston @ Original Equipment Specification .040" (1.016mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	5.2 cc	5.2 cc	5.3 cc	5.3 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	0 cc	0 cc	0 cc	0 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**9.5 : 1**	**9.6 : 1**	**9.6 : 1**	**9.7 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	8.0 : 1	8.0 : 1	8.1 : 1	8.2 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	8.9 : 1	9.0 : 1	9.9 : 1	9.1 : 1
Effective Compression Ratio (ECR) (Piper BP255)	7.5 : 1	7.5 : 1	7.6 : 1	7.6 : 1
Effective Compression Ratio (ECR) (Piper BP270)	7.3 : 1	7.4 : 1	7.4 : 1	7.5 : 1
Effective Compression Ratio (ECR) (Piper BP285)	7.2 : 1	7.3 : 1	7.3 : 1	7.4 : 1

39cc Combustion Chamber with Flat-Top Piston @ .012" (.3048mm) Cold Clearance Depth:

Total Displacement	**1812 cc**	**1822 cc**	**1844 cc**	**1868 cc**
Bore Oversize	+.020"	+.030"	+.040"	+.060"
Swept Volume	453 cc	458.4 cc	461.3 cc	466.9 cc
Clearance Volume	1.6 cc	1.6 cc	1.6 cc	1.6 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	0 cc	0 cc	0 cc	0 cc
Gasket Volume	3.2 cc	3.2 cc	3.2 cc	3.2 cc
Counterbore Volume	.4 cc	.4 cc	.4 cc	.4 cc
Geometric Compression Ratio (GCR)	**10.3 : 1**	**10.4 : 1**	**10.4 : 1**	**10.6 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	8.6 : 1	8.6 : 1	8.7 : 1	8.8 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	9.6 : 1	9.6 : 1	9.7 : 1	9.8 : 1
Effective Compression Ratio (ECR) (Piper BP255)	8.0 : 1	8.0 : 1	8.1 : 1	8.2 : 1
Effective Compression Ratio (ECR) (Piper BP270)	7.9 : 1	7.9 : 1	7.9 : 1	8.0 : 1
Effective Compression Ratio (ECR) (Piper BP285)	7.8 : 1	7.8 : 1	7.8 : 1	7.9 : 1

Big bore pistons present a somewhat different set of requirements. Due to the fact that the optimal squish (quench) cold clearance of the B Series engine is .012”, it becomes necessary to custom-tailor the volume of the dish in the piston crown in order to achieve the desired compression ratio:

43cc Combustion Chamber with Flat-Top Piston @ .012 (.3048mm) Cold Clearance Depth:

Total Displacement	**1892 cc**	**1915 cc**	**1924 cc**	**1948 cc**
Bore Oversize	+.080”	+.100”	83mm (3.268”)	83.57mm (3.29”)
Swept Volume	472.9 cc	478.7 cc	481 cc	486.8 cc
Clearance Volume	5.5 cc	5.5 cc	5.5 cc	5.6 cc
Cylinder Head Volume	43 cc	43 cc	43 cc	43 cc
Dish Volume	0 cc	0 cc	0 cc	0 cc
Gasket Volume	4.5 cc	4.5 cc	4.5 cc	4.5 cc
Counterbore Volume	.3 cc	.3 cc	.3 cc	.3 cc
Geometric Compression Ratio (GCR)	**8.9 : 1**	**9.0 : 1**	**9.0 : 1**	**9.1 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	7.5 : 1	7.6 : 1	7.6 : 1	7.6 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	8.4 : 1	8.4 : 1	8.4 : 1	8.4 : 1
Effective Compression Ratio (ECR) (Piper BP255)	7.1 : 1	7.1 : 1	7.1 : 1	7.1 : 1

Effective Compression Ratio (ECR) (Piper BP270)	7.0 : 1	6.9 : 1	6.9 : 1	6.9 : 1
Effective Compression Ratio (ECR) (Piper BP285)	6.8 : 1	6.9 : 1	6.8 : 1	6.8 : 1

39cc Combustion Chamber with Custom-Dished Piston @ .012" (.3048mm) Cold Clearance Depth:

Total Displacement	**1892 cc**	**1915 cc**	**1924 cc**	**1948 cc**
Bore Oversize	+.080"	+.100"	83mm (3.268")	83.57mm (3.29")
Swept Volume	472.9 cc	478.7 cc	481 cc	486.8 cc
Clearance Volume	5.5 cc	5.5 cc	5.5 cc	5.6 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	3.3 cc	3.7 cc	3.7 cc	4.9 cc
Gasket Volume	4.5 cc	4.5 cc	4.5 cc	4.5 cc
Counterbore Volume	.4 cc	. 3 cc	.3 cc	.3 cc
Geometric Compression Ratio (GCR)	**9.0 : 1**	**9.0 : 1**	**9.0 : 1**	**9.0 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	6.8 : 1	6.8 : 1	6.8 : 1	6.8 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	7.6 : 1	7.6 : 1	7.6 : 1	7.6 : 1
Effective Compression Ratio (ECR) (Piper BP255)	6.4 :1	6.4 : 1	6.4 : 1	6.4 : 1

Effective Compression Ratio (ECR) (Piper BP270)	6.3 : 1	6.3 : 1	6.3 : 1	6.3 : 1
Effective Compression Ratio (ECR) (Piper BP285)	6.2 : 1	6.2 : 1	6.2 : 1	6.2 : 1

39cc Combustion Chamber with Custom-Dished Piston @ .012" (.3048mm) Cold Clearance Depth:

Total Displacement	**1892 cc**	**1915 cc**	**1924 cc**	**1948 cc**
Bore Oversize	+.080"	+.100"	83mm (3.268")	83.57mm (3.29")
Swept Volume	472.9 cc	478.7 cc	481 cc	486.8 cc
Clearance Volume	5.5 cc	5.5 cc	5.5 cc	5.6 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	.6 cc	.7 cc	1.2 cc	1.8 cc
Gasket Volume	4.5 cc	4.5 cc	4.5 cc	4.5 cc
Counterbore Volume	.3 cc	. 3 cc	.3 cc	.3 cc
Geometric Compression Ratio (GCR)	**9.5 : 1**	**9.5 : 1**	**9.5 : 1**	**9.5 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	7.2 : 1	7.2 : 1	7.2 : 1	7.2 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	8.0 : 1	8.0 : 1	8.0 : 1	8.0 : 1
Effective Compression Ratio (ECR) (Piper BP255)	6.7 : 1	6.8 : 1	6.7 : 1	6.7 : 1
Effective Compression Ratio (ECR) (Piper BP270)	6.6 : 1	6.6 : 1	6.6 : 1	6.6 : 1
Effective Compression Ratio (ECR) (Piper BP285)	6.5 : 1	6.5 : 1	6.5 : 1	6.5 : 1

39cc Combustion Chamber with Flat-Top Piston @ .012 (.3048mm) Cold Clearance Depth:

Total Displacement	**1892 cc**	**1915 cc**	**1924 cc**	**1948 cc**
Bore Oversize	+.080"	+.100"	83mm (3.268")	83.57mm (3.29")
Swept Volume	472.9 cc	478.7 cc	481 cc	486.8 cc
Clearance Volume	5.5 cc	5.5 cc	5.5 cc	5.6 cc
Cylinder Head Volume	39 cc	39 cc	39 cc	39 cc
Dish Volume	0 cc	0 cc	0 cc	0 cc
Gasket Volume	4.5 cc	4.5 cc	4.5 cc	4.5 cc
Counterbore Volume	.3 cc	.3 cc	.3 cc	.3 cc
Geometric Compression Ratio (GCR)	**9.6 : 1**	**9.7 : 1**	**9.8 : 1**	**9.9 : 1**
Effective Compression Ratio (ECR) (Original Equipment 88G 303)	7.5 : 1	7.6 : 1	7.6 : 1	7.6 : 1
Effective Compression Ratio (ECR) (Original Equipment CAM 1156)	8.4 : 1	8.4 : 1	8.4 : 1	8.4 : 1
Effective Compression Ratio (ECR) (Piper BP255)	7.1 : 1	7.1 : 1	7.1 : 1	7.1 : 1
Effective Compression Ratio (ECR) (Piper BP270)	7.0 : 1	6.9 : 1	6.9 : 1	6.9 : 1
Effective Compression Ratio (ECR) (Piper BP285)	6.8 : 1	6.9 : 1	6.8 : 1	6.8 : 1

Exotica

For those who truly lust after power, an aluminum alloy engine block Rover 3.9L V8 conversion would be much better (200 to 260BHP), but that is a subject for another article. If this thought tickles your fancy, Roger Parker has an excellent website on how to perform this conversion that can be found at http://www.mgcars.org.uk/v8_conversions/rogv8.html . A British website for purchasing the Rover V8 engine itself in different displacements and various states of tune can be found at http://www.rpiv8.com/ .

Another, even more dubious possibility, is that of a "Stroker" engine. Increasing the stroke of an existing engine results in the shortening of the connecting rod / stroke ratio. Although the side thrust loadings on both the pistons and cylinder walls, as well as piston inertia loadings and crankshaft inertia loadings are all increased, this also results in the piston accelerating faster down the bore, thus increasing the atmospheric pressure differential between that of the outside of and the inside of the cylinder. This increased difference in atmospheric pressures also occurs earlier in the stroke, resulting in higher velocities of the incoming fuel / air charge, which in turn results in a larger volume of fuel / air charge filling the cylinder. However, accomplishing this on an existing engine requires a shorter distance from the axis of the wrist (gudgeon) pin to the piston crown in order to avoid hitting the roof of the combustion chamber, as well as a shorter distance from the axis of the wrist (gudgeon) pin to the bottom of the piston skirt in order to avoid hitting the crankshaft.

It should be understood that the power-producing advantage achieved by the increased angularity achieved by the shorter connecting rod to stroke ratio is partially offset by the attendant frictional losses, so the looked-for increase in power output turns out to be a disappointment. In extreme cases, The longer stroke also forces the shortening of the skirt of the piston in order to provide adequate clearance with the connecting rod when the piston is at the mid-point of its travel. The resulting decrease in the load-bearing surface area of the now-shorter piston, combined with the increased sidethrust, can result in an increased rate of wear. In all cases, the end result of this shortening of the piston will be an increased tendency towards and piston scuffing. Just to complicate matters further, the longer stroke results in the rings of the piston descending below the bottom of the coolant jacket. Such an engine would obviously be harder on its oil and crankshaft bearings as well, although a slight lateral offsetting of the bores or of the wrist pins (gudgeon pins) would help somewhat to avoid the worst of the crankshaft bearing loads.

In addition, due to the altered atmospheric pressure differentials, a different camshaft lobe profile would still have to be custom-developed. I doubt that it would be possible to offset the bore of a B Series engine enough to make this approach worthwhile, even if a satisfactorily modified camshaft lobe profile could be developed to accommodate the altered breathing characteristics produced by the shorter connecting rod to stroke ratio of the engine. In addition, the shaft of the camshaft would have to be of minimal diameter in order to provide adequate clearance for the connecting rod assembly. Reducing the diameter of an Original Equipment camshaft would be a poor idea, as this would weaken it to the point that both flexure and breakage would be likely, especially if a stress-producing high lift camshaft lobe profile were to be employed. To accomplish this would require the use of a steel alloy that would have a high chromium content (for hardness and rigidity), molybdenum (to avoid molecular shear), and vanadium (to control distortion), plus it would have to be heat treated to a hardness that might cause its small diameter to snap under the stress of high engine speeds. In order to maintain compatibility of the materials used so as to avoid premature wear, the use of chilled iron (white iron) tappets would be mandatory.

The maximum permissible engine speed would have to be reduced due to maximum permissible piston speed being attained at lower engine speeds, and a greater order of precision in balancing would become an important issue unless you are willing to tolerate some of the additional power being dissipated in the form of increased vibration. To go from a displacement of 1.8 Liters to a displacement of 2.1 Liters by increasing the stroke alone would require an increase in stroke of 16% to 17%, which is not possible without relocating the axis of the camshaft, as well as those of the tappets, and fabricating custom-length pushrods. As a consequence of the increased deflection of the pushrods, side thrust loads on the tappets would also be increased. To obtain such a displacement while retaining the original camshaft position would require a radical overbore, sleeving the cylinders to withstand the increased side thrust loads, and a set of oversize pistons. The small increase in stroke would not result in a sufficient increase in power output to justify the hassles and the expense. Obviously, the stroker approach to obtaining more power would be expen$ive. A well-developed 1.8L engine would be far less expensive and would live far, far longer. The only rational justification for a stroker 2.1L B Series engine would be in the eyes of those who want the ultimate in B Series power for use in autocross or on a dragstrip. In short, if you want maxipower for the street, fit a Rover V8 instead.

The Cooling System

Of course, an engine that produces more power also makes more heat. This is where your coolant system becomes crucial. An engine can reliably produce no more power than its coolant system can cope with.

When a localized hot spot forms, it causes the surrounding metal in the cylinder head and engine block to expand excessively. This, in turn, can damage and crush gaskets, causing leakage. Hot spots also create added stress in the cylinder head and engine block itself, which may cause warpage and / or cracking. One of the most common causes of the formation of localized hot spots is air pockets in the coolant system. These can occur when the coolant system is being refilled, or when other engine repairs are being made (valve job, replacing a coolant pump, etc.). As coolant enters the engine, the closed thermostat often blocks the venting of air from the engine, leaving air trapped in the upper portion of the engine block and / or the cylinder head. Some thermostats have a small bleed hole in order to prevent this blockage from happening, but many do not. If the air trapped within the coolant system is not removed, it will expand as it heats, creating overpressurization, as well as formation of localized hot spots and steam pockets when the engine reaches operating temperature, in turn causing localized overheating within the engine. A symptom of air trapped within the coolant system would be little or no heat output from a normally functional heater when the engine is warm.

Thermostats

The function of the thermostat is to maintain a stable engine temperature, thus keeping the running tolerances of the engine constant, and by doing so, prolong the lifespan of the engine. Too much coolant flow can force the coolant through the coolant passages too quickly, causing it to not accomplish maximum heat transfer. This condition can also prevent the coolant from having enough time inside of the radiator in order to allow efficient heat transfer. On the other hand, inadequate coolant flow can overheat the coolant before it gets a chance to release the heat energy that is stored in its mass into the radiator. If coolant flows too rapidly, as in the case of no thermostat being present, the coolant can leave air pockets inside of the coolant passages of the cylinder head, thereby "superheating" the trapped air into an expanding gas, which forces water out of the overflow even though the water leaving the engine appears to be well below its boiling point. This is a dangerous situation that leads to serious engine damage resulting from cracked heads and cylinder blocks, often without warning. Tests have proven that by simply opening the thermostat precisely at the same temperature over and over again, the average temperature of the engine remains cooler. Engines can warm up faster to normal operating temperature, and cool down quicker once the engine exceeds the set point. This results in slower cylinder wear, and consequently longer engine life. By opening the thermostat "on time", every time, the temperature swing is reduced, allowing for more consistent cooling. Opening the thermostat as quickly and as far as possible allows the warm coolant to exit quickly, thereby delivering it to the radiator to begin its cooling cycle as quickly as possible. As a result, getting the coolant into the engine and holding it there extracts as much heat from the engine as efficiently possible, therefore enabling to radiator to work at its maximum potential as well.

It is not commonly understood that a thermostat starts to open at its rated temperature but does not become fully open until 20° Fahrenheit (6.7° Celsius) later. This being the case, a 165° Fahrenheit (73.9° Celsius) thermostat will start to open at 165° Fahrenheit (73.9° Celsius) but will not be fully open until the coolant temperature reaches 185° Fahrenheit (85° Celsius). A winter thermostat such as the 195° Fahrenheit (90.6° Celsius) thermostat will begin to open at 195° Fahrenheit (90.6° Celsius) but will not be fully open until 215° Fahrenheit (101.7° Celsius), which is 3° Fahrenheit (1.61° Celsius) more than the boiling point of pure water. It should be noted that the thermostatic sensor that is incorporated into its fan control switch (BMC Part# URP 1126, Moss Motors 542-215) inside of the radiator header tank of the Rubber Bumper MGBs is calibrated at 194° Fahrenheit (90.0° Celsius) / 180° Fahrenheit (82.2° Celsius). Consequently, the fan switch will close when the temperature of the coolant that is entering the radiator is measured as being 194° Fahrenheit (90.0° Celsius), and open when the temperature decreases to about 180° Fahrenheit (82.2° Celsius). The popular aftermarket Hayden fan has a thermostatic sensor that is incorporated into its fan control switch that is calibrated to open at 185º Fahrenheit (85º Celsius). This being the case, a either a 185º Fahrenheit (85º Celsius) or a 195º Fahrenheit (90.6º Celsius) thermostat will cause either of these electric fans to run almost continuously.

It is a widely known fact that that at atmospheric pressure at sea level, pure water will boil at a temperature of 212º Fahrenheit (100º Celsius). Ascend to a higher altitude and the boiling point will occur at a lower temperature (approximately 4.5º Fahrenheit (15.3º Celsius)lower for each 1,000 feet in altitude). Fortunately, the boiling point of the coolant is raised by both the addition of antifreeze and by the radiator cap of the radiator, which raises the boiling point by 3º Fahrenheit (1.61º Celsius) for every PSI of pressure. You would be well advised to use the "fail-safe" type of thermostat that locks in the full-open position should it fail in order to preclude overheating in the middle of nowhere. Moss Motors sells a 180º Fahrenheit (82.2º Celsius) "fail-safe" type general-purpose thermostat (Moss Motors Part # 434-205).

Be aware that a thermostat cannot prevent overheating. It can only prevent overcooling. A thermostat can only be the cause of overheating if it is defective and does not open as it should. In selecting a thermostat, be aware that the B Series engine tolerates high operating temperatures quite well. Whenever a thermostat is changed for one with a different operating temperature, it will be necessary to adjust the fuel / air mixture of the carburetion, richer for a cooler thermostat and leaner for a hotter one. This is due to the fact that the hotter the intake ports become, the increased heat is transferred into the incoming fuel / air charge, expanding the air and thus effecting the fuel / ratio. At an operating temperature of 190º Fahrenheit (87.8º Celsius) or higher, it will normally run best with a fuel / air ratio of 12:1. Happily, this is the ratio at which both power output and fuel economy are maximized. Unfortunately, some owners go to great lengths in order to keep the engine temperature down to 180º Fahrenheit (82.2º Celsius). Although the engine does not overheat, they do not realize that they are diverting energy in the form of heat into the coolant system that should be used in order to produce pressure on the piston. Operating the engine at 180º Fahrenheit (82.2º Celsius) will result in a reduction of power by from 2 to 3%.

Decades ago when control of air pollution was not a priority for engine designers, engines typically employed cooler 180º (82.2º Celsius) or even 165º (73.9º Celsius) thermostats not for the purpose of keeping the engine's operating temperatures lower, but rather for the purpose of allowing the oil to run cooler in order to prevent it from breaking down. The oils of that era would break down at relatively low temperatures, so engines were run at as low an operating temperature as possible in order to preclude this problem. However, today's modern oil formulas are designed to withstand much higher operating temperatures. Also, before the days of today's ethylene glycol antifreeze with its boiling point of 386° Fahrenheit (197° Celsius), the antifreeze most commonly employed was ethyl alcohol. Ethyl alcohol has a much lower boiling point of 172.4° Fahrenheit (79° Celsius) than that of water at 212° Fahrenheit (100° Celsius), so by keeping the coolant temperature as low as possible, the alcohol was not driven out of the system as quickly. This was more important during the winter, obviously, so cars typically had winter thermostats that were, while warmer than their summer thermostats, still colder than those used in the modern cars of today.

The Original Equipment Smiths bellows-type thermostat originally developed for use in the B Series engine was a rather interesting design. It not only had an orthodox (for its time) bellows valve for controlling coolant flow, it also had a vertically reciprocating sleeve that surrounded its bellows unit. Whenever the temperature of the engine was below its optimum level, the sleeve remained in its bottom position, leaving a bypass passage in the cylinder head open. This bypass passage was intended to allow coolant from the warming engine to recirculate in a closed circuit back through the cylinder head and thence onward to the engine block so that the engine could warm up as quickly as possible prior to the thermostat opening, thus permitting heated coolant so that the cylinder head would warm up more quickly. As the temperature of the engine approached its optimum level, the sleeve rose and blocked off the recirculation port, thus allowing the heated coolant to bypass and circulate exclusively into the radiator matrix instead into the cylinder head where it could only contribute to higher temperatures in the head than are considered to be appropriate, resulting in hot-spots and possible localized boiling. However, such thermostats are unavailable new, often being regarded when encountered as quaint curiosities from a bygone day. When the Original Equipment Smiths bellows-type thermostats ceased to be produced, the wax pellet type becoming its substitute, a key feature was lost : the sleeve which controlled the bypass passage in the cylinder head.

The modern wax pellet style thermostats of today, lacking the reciprocating sleeve, have no provision for exclusive recirculation. As a result, hot coolant can not only move off into the radiator via the opening in the thermostat, but a certain amount can also recirculate back into the head, diluting and warming the coolant that was returning from the radiator, thus causing the head to run hotter than was originally intended. In addition, because of the routing of the coolant through the system, this also increases the amount of heat retained in the system in general. This can lead to overheating under the most adverse running conditions of ambient heat and heavy loadings, resulting in preignition. The most practical way to deal with these problems when they arise is to install a blanking sleeve in addition to a modern balanced-type thermostat. The blanking sleeve will permanently cover the bypass port , and you still need some circulation of coolant while the engine is warming up. In order to fulfill this need, drill a 3/16" diameter hole in the outer flange of the thermostat in order to allow a coolant to pass through to the radiator during the warm-up period. If the thermostat that you are using has a small air-relief poppet valve in the top, you can just remove the poppet valve and leave the hole open. This small amount of coolant circulating through the radiator will result in a somewhat slower temperature rise during the warm-up period. Once the engine attains operating temperature and the thermostat begins to open, it will then effectively regulate the minimum operating temperature in the same manner as originally intended.

Although the modern wax pellet type thermostat is a non-rebuildable item, it can be helpful to understand how it works. Contained inside of a copper cup is a specially-formulated combination of thermosensitive powdered metal and wax that forms a pellet. The upper section of the copper cup forms a poppet valve that seats against a valve seat that is formed by a canelure in the upper bridge section. A coil sealing spring surrounds the copper cup in order to maintain the sealing pressure against the valve seat. Attached to the bridge section is a rod that projects down into the copper cup, attached to which is a piston that is sealed inside of the copper cup. When the wax pellet is exposed to heat, it melts and expands, overcoming the resistance of the sealing spring and forcing the copper cup and its upper section away from the piston and consequently away from the valve seat, thus opening the valve. Over time, the sealing edges of the piston and the inner wall of the copper cup wear to the point that the wax compound leaks out and the thermostat ceases to function. Fortunately, this is an item that is easy and inexpensive to replace.

However, you do not have to resign yourself to the use of a conventional wax pellet thermostat. Prestone markets an updated version of what is called the “balanced” thermostat that was originally designed and marketed by Robert Shaw Controls (Prestone Part# 330-195 for 195° Fahrenheit, Prestone Part# 330-180 for 180° Fahrenheit). The 3-port construction equalizes the coolant pressure from above the valve (radiator side) to the higher, pump pressure side, hence the use of the term “balanced”. In a cooling system that is equipped with a conventional thermostat, there is always a relatively high pressure difference between the intake and the outlet of the coolant pump, especially when the thermostat is partially closed. This is due to the fact that the coolant pump struggles to draw the coolant through its intake while at the engine outlet the conventional thermostat reduces the outgoing coolant flow. As the purpose of the coolant pump should be to supply flow and not pressure, some of its work, and thus power, is wasted. In addition, due to the thermal inertia of the thermostat bulb, every time that there is a quick variation in temperature of the coolant that is returning from the radiator, a relevant part of this variation is increased by the loading on the pump. When a conventional thermostat is either closed or partially open, the coolant flow inside of the engine is low and its pressure is high. This also leads to a gradient between the pump and the engine outlet. Also, when the speed of the coolant pump increases, its output pressure then increases. However, a pressure release thermostat, in addition to opening in response to temperature, will also open in a manner that is related to the pressure within the engine that is created by the coolant pump. This opening of the pressure release thermostat is accomplished by a simple comparison of these pressures, which is achieved mechanically by its pressure balancing spring which is designed to operate at a trigger pressure drop that is determined by the aforementioned pressure balancing spring. This allows the thermostat to open effortlessly and accurately no matter what the rate of coolant flow or the speed of the coolant pump happens to be at that particular time. The inherently inefficient hydrodynamic shape of the poppet-type valve of the non-balanced designs is prone to being forced closed when the pressure resulting from increased coolant flow abruptly increases, such as during sudden increases in coolant pump speed after downshifting. The balanced design is such that it is not influenced by variations of coolant pressures as engine speed increases and decreases, and that means that it is better able to more accurately control the operating temperature of the engine than the simple wax pellet type thermostats. While its effectiveness is not immediately obvious when first installed, its superiority does become more obvious while driving at sustained high speeds (70+ MPH) with a power-enhanced engine. A *balanced thermostat* maintains the temperature of the coolant to within +/- 2º Fahrenheit (.67º Celsius) compared with temperature fluctuations of up to 20º Fahrenheit (6.7º Celsius) with a conventional wax pellet thermostat. Consequently, the operating temperature is more constant at highway speeds, and when under the strain of heavy loads, it takes longer for the inevitable rise in temperature to occur. In conventional wax pellet thermostats, the small area of the poppet valve requires that the piston must make a long stroke in order to open the thermostat far enough for adequate coolant flow. Unfortunately, the long stroke compromises durability. In the case of the balanced thermostat, the triangulated strut design, being inherently stronger and more stable than the single-span bridge design of conventional wax pellet thermostats, provides superior strength, thus permitting a larger aperture area for coolant flow. This in turn allowed the achievement of a shorter stroke by means of a uniquely-designed flange and a larger-diameter sleeve-type valve. This design feature increases the longevity of the thermostat, yet still allows adequate coolant circulation. The Robert Shaw design is also far less prone to failure. In other thermostat designs, the stem of the bypass valve is welded on. The weld tends to fail under stress. In order to eliminate this problem, in the Robert Shaw design the entire copper cup and the bypass stem are manufactured from a single piece of metal. Fabricating the triangulated strut assembly from brass instead of steel provides another benefit: brass, being more malleable than steel, can be precisely formed in into a more efficient hydrodynamic profile in order to maximize coolant flow. Most manufacturers use a one-piece rubber diaphragm in order to seal the charge and drive the piston. Should the rubber seal rupture, the thermostat then fails. The Robert Shaw design uses two separate parts: a diaphragm to seal the wax, and a stem seat or plug that drives the piston. The rubber material for each part is formulated especially to meet the unique requirements of each part. Consequently, wear or minor damage to the stem seat will still permit the thermostat to operate in a satisfactory manner. The piston itself is activated by a temperature-sensitive mixture of metallic powder and wax. Some wax pellet thermostats use an all-wax charge which reacts slowly to temperature changes. Other designs mix copper powder with the wax for faster response, but the copper quickly separates from the wax. The Robert Shaw design uses a process to maintain suspension of the copper powder in the wax so that its rapid response to temperature changes will not deteriorate over time and so that the thermostat will not "stick-open", thus causing the engine to run cool.

Happily, it is also a “fail-safe” design that will remain open should it ever fail, thus preventing overheating.

The only advantage to the sole use of a blanking plate in lieu of a thermostat is that there is no thermostat to stick in the closed position and thus cause the engine to overheat. However, it should be understood that a blanking plate is normally intended for racing use. On a street machine, installing a blanking plate without including a thermostat while leaving the pulleys the original diameter usually results in hotter running, as well as much longer warm up periods. This is due to the fact that without the flow restriction caused by the presence of a thermostat, the coolant circulates so rapidly that it does not have time to absorb heat from the engine, nor does it have time to release heat into the cooling matrix of the radiator. This will lower the coolant temperature, only to leave the heat inside of the engine where you don't want it. Thus, if you have chosen a camshaft which causes the engine to be normally operated at a higher average engine speed (such as a Piper BP285), then it would be wise to install a larger-diameter pulley wheel onto the coolant pump in order to reduce the pumping speed of its impeller. This will assure that the coolant has sufficient time to absorb heat from the engine and release it into the radiator matrix.

Radiator Caps

Be aware that there are two slightly different radiator caps in common use today. The standard MGB cap as found on the pre-1975 cars that were not equipped with an overflow tank has one seal at its base that is spring loaded to 7lbs. A closed system cap employs a second seal at the top of the cap where it rests on the lip of the neck for the radiator. This second seal allows coolant to be drawn back into the radiator from the overflow container instead of drawing in air from the atmosphere.

Here are the different radiator caps that were used on the BMC B Series engines that were originally installed in the MGB:

Years	**1963-1967**	**1968-1975**	**1976**	**1977-1980**

Poundage	7 Lbs	10 Lbs	13 Lbs	15 Lbs
BMC Part #	GRC102	GRC109	GRC111	GRC110

In the days before product safety liability lawsuits against manufacturers became rife, radiator pressure vent caps were commonly available with a small, lever-actuated valve to permit the easy ingress of air when refilling a cold coolant system, consequently eliminating the vacuum inside of the sealed system and thus making removal of the radiator cap an easy affair. Unfortunately, some would mistakenly believe that its purpose was to release pressure from within a hot coolant system prior to adding coolant to a hot radiator, unwittingly open the valve while the engine was hot, and be scalded by the ensuing gush of hot coolant. As a result, finding one of these convenient items today often requires some persistent hunting.

Pulley Wheels

In racing, the sizes of both the engine pulley wheels on the harmonic balancer (harmonic damper) and the coolant pump pulley wheels are often reduced in order to lower the pump speed to engine speed ratio so that the pump will turn more slowly and thus allow the coolant sufficient time to absorb heat from the engine block and release it through the radiator matrix. This is also occasionally done in the case of street engines that use camshafts that produce their best power at higher engine speeds. Should you decide to experiment with changing engine pulley wheels in order to alter the speed of your coolant pump, you will achieve the following results:

Engine Harmonic Balancer Pulley Wheel Part #	Speed Change From / To	5.125” 130mm	6.000” 152mm
BMC Part # 12H 963 (1964-1970) BMC Part # 12H 3515 (1971-1974)	**5.125” 130mm**	0%	+37%

BMC Part # 12H 3516 (1975 -1980)	**6.000” 152mm**	-23%	0%

In addition, the pulley wheels used on the coolant pumps can also be changed:

Coolant Pump Pulley Wheel Part #	**Speed Change From / To**	**110mm**	**130mm**	**142mm**
BMC Part #’s 12B 174, 12H 2452, 12H 3696, 12H 3700	**110mm**	0%	+18%	+29%
BMC Part # 12H ?	**130mm**	-15%	0%	9%
BMC Part # 12H ?	**142mm**	-23%	-17%	0%

Coolant

All Original Equipment B Series engine blocks are of the “Wet Liner” type in which the grey iron cylinders are exposed directly to flowing coolant contained within a coolant jacket. The early three main bearing B Series engines had coolant passages between all of the cylinders, but when the engine was redesigned into its five-main bearing version the coolant passages between cylinders #1 and #2 and #3 and #4 were deleted in order to enhance the rigidity of the engine block. The coolant passages within the engine block extend to just below the position of the piston rings when the piston is at Bottom Dead Center. The coolant pump flows the coolant into the engine block to first cool the cylinders, then through the cylinder head in order to cool the valves and the combustion chambers, and thence out through the thermostat housing into the top tank of the Morris downflow radiator. The coolant then releases the heat that it contains into the three-staggered-row radiator matrix as it drains down to the bottom tank from whence it subsequently flows to the coolant pump in order to repeat the process.

Never use plain water as a coolant medium in the coolant system as it will react with the iron of the cooling surfaces inside of the engine to rust. Rust acts as a heat insulator, reducing heat transfer inside of the engine. Instead, use a mixture of antifreeze and distilled water. Aside from protecting the coolant mixture from freezing, antifreeze has three other important functions: First, it contains a coolant pump lubricant. Second, it contains flow modifiers that reduce cavitation. Third, it contains corrosion inhibiting ingredients that inhibit the formation of insulators such as rust inside of the coolant passages of the system. On a related note, if you have installed an aluminum alloy cylinder head, be sure that your antifreeze is specifically formulated for use in engines that have aluminum engine blocks and / or cylinder heads.

As a coolant medium, water has an assigned cooling value of 1.0, while unadulterated antifreeze has a cooling value of .6. Thus, antifreeze is only 60% as efficient at heat transference as water is. A mixture of antifreeze and water is thus not as efficient at heat transference as pure water is. The formula for determining Coolant Efficiency is [(Percentage of Water X 1.0) + (Percentage of Antifreeze X .6)] = Coolant Efficiency. I prefer a mixture of 75% water and 25% antifreeze that results in 90% Coolant Efficiency. [(75% x 1.0 = 75%) + (25% x 0.6= 15%)] = 90% Coolant Efficiency.

Because of the greater cooling efficiency of pure water, racers commonly use distilled water in their engines and add “Water Wetter”, a product formulated to reduce the cohesion of water and thus reduce cavitation at high engine speeds, allowing the coolant to flow more efficiently. If you add a bottle of Water Wetter to a coolant mixture of 25% antifreeze and 75% distilled water, you will have an excellent coolant suitable for use in a high performance engine that will not sacrifice adequate protection against corrosion, coolant pump lubrication, or coolant flow at high coolant pump speeds.

Make sure that the system is refilled with a mixture of a good ten-year antifreeze and distilled water. Why use distilled water? Because it will not coat the interior of your coolant system with mineral scale. Why the more expensive ten-year antifreeze? Because it has special additives that will extend the life of your coolant pump and because you do not really want to do all this all over again next year, do you? You do not have to take this extra step, of course. When your coolant system fails due to a lack of proper care, you can always send Moss Motors $229.95 for a new radiator and $94.95 for a new coolant pump, plus the additional cost of shipping.

Be aware that the published figures quoted for the volume of the coolant system are always for dry systems. When simply draining out an old coolant / water mixture and refilling with new this is never the case, and if you pre-mix and fill with that, then you will end up with a weaker concentration of coolant to water. Put some distilled water in, then all of the pure coolant required, the required volume of pure coolant having been previously calculated using the dry volume, then finish filling up the coolant system with as much distilled water as is necessary. This will produce the desired coolant / water ratio. You might need to go through two or three heating / cooling cycles in order to get all of the air out of the cooling system. While doing this, it is best to add pure distilled water for topping-up of the coolant system. After that, only ever top-up using the correct concentration, i.e., pre-mix it.

So, what types of antifreeze are available? Basically, there are four-

Traditional ethylene glycol antifreeze is toxic but highly effective antifreeze that contains silicates as an inhibitor to help prevent corrosion in an engine with mixed metals in its make-up. Be aware that there are also low or no-silicate ethylene glycol formulations (usually red) available which may not be suitable for all engines.

Propylene glycol antifreeze is another well-known and less toxic antifreeze formulation and usually contains silicates. Polypropylene glycol needs to be periodically changed or subjected to "regular monitoring of freeze protection, pH, specific gravity, inhibitor level, color, and biological contamination", as once bacterial slime starts to grow within it, the corrosion rate increases. Another down side of this product is that if it is overheated, it will then turn very corrosive very quickly. Thus the need for troublesome regular monitoring of its pH level.

Both of the above products use inorganic additive technology (IAT) with iron and steel corrosion prevention provided by nitrites and aluminum protection by silicates and phosphates and molybdates providing high lead solder protection, with borates helping to control acidity.

Recently problems have been reported concerning the use of antifreeze mixtures using Organic Acid Technology (OAT) antifreeze. This was introduced in the mid-1990s and the products are biodegradable, recyclable and do not contain either silicates or phosphates and are designed to be longer lasting. Be warned that Organic Acid Technology antifreeze, when mixed with ethylene glycol, results in the formation of sludge. Some newer OAT products are said to be compatible with all OAT and glycol types and are green or yellow. Also, HOAT (Hybrid OAT), both are claimed to have a life of five years or 150,000km. However, these products do seem to cause problems in older engines; over and above the ability of it to find the smallest crevice and leak. OAT antifreezes have been accused of destroying seals and gaskets and causing a great deal of damage in older engines. For this reason, the manufacturers of OAT antifreeze products do not recommend their use in historic and classic vehicles.

The final category of antifreeze is HOAT. These products use Hybrid Organic Acid Technology in an ethylene glycol base with some silicates in the formulation alongside the organic corrosion inhibitors. The product is not recommended for use in historic vehicles.

So, what are we MGB owners to conclude?

• Only use IAT antifreeze in historic vehicles.

• Never mix different types of antifreeze without thoroughly flushing out the system.

• Always replace the coolant within the time scale specified by the antifreeze manufacturer as the corrosion inhibitors break down over time.

As a general rule of thumb, anti-freeze can remain in the cooling system for two years provided that the specific gravity of the coolant is checked periodically and fresh compatible antifreeze is added as necessary. After the second year the system should be drained and flushed by inserting a hose in the filling orifice and allowing water to flow through until clean. Refill with the appropriate antifreeze solution and add a quarter of a pint (0.15 liter) of neat anti-freeze to the expansion tank.

Coolant Tanks

Of course, even the most commonplace coolant system needs a tank in order to handle overflow. The relative density of water at 39.2° Fahrenheit (4° Celsius) is 1, and at 212° Fahrenheit (100° Celsius) its relative density is 0.97. The Law of Conservation of Mass says that the mass of water inside of the coolant system must always be the same unless you physically lose some out of the coolant system. Since the mass of coolant inside of the coolant system must stay the same, but its volume increases due to heat, coolant being effectively incompressible, the pressure increases against the radiator cap, and thus the extra volume of coolant comes out of the overflow pipe. The term "overflow tank" is commonly used in two different meanings that are very dissimilar. There are actually two different types of systems, each requiring the use of its own type of tank, both of which are commonly referred to as overflow tanks. These two are the true overflow tank and the recovery tank (expansion tank).

A conventional open coolant system uses a radiator matrix and tanks, the header tank of which has a filler neck to which a radiator cap is attached. Such a system should be filled though the filler neck located on the header tank to within about ½" to 1" (12.7mm to 25.4mm) below the bottom of the filler neck. The coolant expands as heat increases, pressurizing the system beyond the holding capability of the radiator's constant-pressure radiator cap (listed on the radiator cap in Pounds per Square Inch or PSI), thus causing excess coolant to be vented out through the overflow vent into an overflow tank. It must be understood that when the system is equipped with a constant-pressure radiator cap, a simple open coolant system overflow tank is under ambient atmospheric pressure while the coolant system is under the same ambient atmospheric pressure, plus the additional pressure generated by the heat-induced expansion of the coolant. As the coolant system begins to cool down, the internal pressure of the system decreases until it drops to lower than the rated pressure of the constant-pressure radiator cap and thus allows the constant-pressure radiator cap to form a seal on the bottom of the filler neck. Thus, the pressure within the coolant system will exceed the ambient atmospheric pressure inside of the overflow tank until the radiator cap again seals and the entire system is completely cool. As can be easily seen, if a constant-pressure type of radiator cap is employed, the flow of excess coolant can be in one direction only—out of the coolant system and into the overflow tank. The coolant in the overflow tank cannot flow back into the coolant system because a liquid cannot flow from a lower pressure to a higher pressure. Thus, when the coolant system is equipped with a simple, inexpensive, constant-pressure type of radiator cap, the overflow tank can serve only as a catchment basin, not as a reservoir of extra coolant that will automatically return to the radiator as the system cools. However, if a closed-system type of radiator cap is employed, the overflow tank can be made to serve as a reservoir of extra coolant. This type of radiator cap has a small, disc-like vacuum valve that is incorporated into the main body of the pressure valve of the radiator cap. As the coolant system pressure increases, its spring-loaded disc valve is held closed. However, as the coolant system pressure decreases to less than ambient pressure, i.e., a partial vacuum, the ambient pressure inside of the coolant recovery tank (expansion tank) forces the contained coolant to flow back past the vacuum valve inside of the closed-system radiator cap and on into the header tank of the radiator.

A recovery tank (expansion tank) was used on later MGBs that were equipped with a sealed coolant system. A sealed coolant system typically normally has no filler neck and is instead filled from a remote source, most frequently through some form of opening on the thermostat housing, as is the case of the engines used in the Rubber Bumper models. In order to handle the problem of coolant expansion, a remotely located recovery tank (expansion tank) is fitted, along with a 15 PSI constant pressure type radiator cap in order to vent any excess pressure buildup. Thus, both the radiator and the engine are full of coolant, and the recovery tank (expansion tank) is typically about half-full of coolant. The empty portion of the tank functions in the same manner that the empty space in a conventional radiator's header tank functions—it allows the air contained inside of the recovery tank (expansion tank) to be compressed in order to allow expanded coolant to be displaced into the recovery tank (expansion tank). With this form of system, as the system pressure is increased, excess coolant is forced out the radiator vent and onward into the recovery tank (expansion tank). As system pressure decreases, coolant is then forced by the trapped pressure from the recovery tank (expansion tank) back into both the radiator, and from there, into the engine.

With this understanding, we can see that an "overflow" tank serves only one purpose—to catch overflow and keep it from being dripped onto the ground. The fitment of an "expansion" tank to a conventional radiator system that has a filler neck and a radiator cap, will actually endanger the structural integrity of the coolant system. The MGB recovery tank (expansion tank) is equipped with a 15 PSI constant pressure type radiator cap while the average conventional radiator uses a 7 PSI radiator cap. Thus, when the 7 PSI radiator cap is unsealed by expanding coolant, the excess coolant will flow into the recovery tank (expansion tank), which will not, due to its 15 PSI radiator cap, relieve the system pressure, but will allow the system pressure to build up to 15 PSI before system pressure is relieved. Thus, it puts an additional pressurized strain on the radiator matrix, one that it was not designed to withstand. Under these conditions, coolant will flow back into the radiator and engine whenever the total system pressure is less than the pressure rating of the radiator cap. At that point, the coolant flow will stop, making the recovery tank (expansion tank) a closed system. This system will have no way to vent pressure, about that of the radiator pressure cap rating, until the pressure cap of the recovery tank (expansion tank) is removed. This can cause some problems with the recovery tank (expansion tank).

The recovery tank (expansion tank) that is found in later model MGBs is made out of formed copper sheeting which has been soldered together to form the recovery tank (expansion tank). Copper has a property called “work hardening” which means that as it expands and contracts, the copper gets harder. When it becomes harder, it also becomes brittle. Thus, it is no longer capable of expanding and contracting without small cracks forming in its surface. If these cracks are above the coolant level, you will only see small lines form in the painted surface. If the cracks are below the coolant level, you will see lines of coolant being forced to the outside of the tank. At this point, the cracks can be sealed by cleaning the area and flowing soft solder into the cracks in order to seal them. However, this is only a temporary measure. The cracks will begin to enlarge and the solder seal will be breached. At this point, the recovery tank (expansion tank) will have to be replaced with a good used tank. Sadly, new Original Equipment recovery tank (expansion tanks) are no longer available.

The Coolant Pump

Engines of the 18G, 18GA, 18GB, 18GD, 18GF, 18GG, 18GH, 18GJ, and 18GK engines use long (4 1/8” / 104.775mm deep) coolant pump (BMC Part # GWP 114 for the 18G and 18GA engines; GWP115 for the 18GB engine) with a much deeper pulley and no spacer with the steel paddle-bladed fan. The 18V engines of the chrome bumper cars use a short (3 ¼” / 82.55mm deep) coolant pump (BMC Part # 12H 2267 for the 18GG, 18GH, 18GJ. and 18GK engines; BMC Part # GWP 117 for the 18V581, 18V582, 18V583, 18V584, 18V585, 18V672 to

L27269, 18V673Z to L3644 engines; BMC Part # 18 GWP 123 for the 18V672/L27270 onwards, 18V673Z/L3645 onwards, 18V779, 18V780 engines; BMC Part # GWP 130 for the 18V801, 18V802, 18V846, and 18V847 engines) with a short pump pulley that mounts on a spacer hub (BMC Part # 8G 724) for transferring power to the fan. All of the coolant pumps use 5/16”-24 UNF machine bolts for mounting to the engine block. On the Rubber Bumper cars, there is no spacer as there is no engine-driven fan. You can use either of the two earlier combinations and the pulleys will properly align. In a few rare cases the distance between the fan and the radiator matrix will be insufficient to permit the mounting of the later, more efficient fan and so the shorter pulleys of the 1972 through 1974 models (BMC Part # 12H 3700) will be necessary in order to provide the needed clearance. A fan shroud will maximize the effectiveness of the fan. If your car is a 1976 or later Rubber Bumper model with its forward mounted diaphragm, it will be necessary to both fabricate a custom-made fan shroud and mount an earlier pulley wheel in order to install the fan. Be sure to make provision for flaps on the sides of the reverse side of the fan shroud to act as ducting valves. In this manner the increased frontal air pressure at road speed can force the flaps open, thus permitting an adequate flow of cooling air to pass through the radiator matrix while reducing the aerodynamic resistance of the car itself by means of a reduction in frontal pressure.

Coolant Pump, Pulley, and Fan Combinations					
Engine	**Coolant Pump BMC Part #**	**Coolant Pump Notes**	**Gasket BMC Part #**	**Coolant Pump Pulley BMC Part #**	**Fan BMC Part #**
18G, 18GA	GWP 115	Long-Nosed	12H 814	12B 174	12H 1058
18GB	12H 2267	Long-Nosed Replaced by and compatible with GWP 114	88G 430	12B 174	12H 1058 No Spacer

18GB	GWP 114	Long-Nosed Clausager indicates a change in June 1966, but this does not appear in the Parts Catalogue	88G 430	12B 174	12H 1058 No Spacer
18GD **18GG**	GWP 114	Long-Nosed	88G 430	12B 174	12H 1058 No Spacer
18GD Rc (Auto)	GWP 114	Long-Nosed	88G 430	12B 174	BHH 1604 No Spacer
18GF **18GH** **18GJ**	GWP 114 w/ 2 spacers	Long-Nosed	88G 430	12H 2452	BHH 1604 No Spacer
18GG Rc (Auto)	GWP114	Long-Nosed	88G 430	12B 174	BHH 1604 No Spacer
18GK	GWP 114 w/ 2 spacers	Long-Nosed	88G 430	12H 2452	12H 4230 No Spacer
18V-581 **18V-582** **18V-583**	GWP 117	Short-Nosed	88G 430	12H 3696	12H 1058 One Spacer
18V-584 **18V-585**	GWP 117 w/ 2 spacers	Short-Nosed GWP 117 &	62H 350	12H 3700	12H 4230 One Spacer

18V-672 #'s 101-27,269 **18V-673 #'s 101-3,644**		GWP 123 are interchangeable			
18V-672 #'s 27,270-on **18V-673 #'s 3,645-on**	GWP 123 w/ 1 long & 1 short spacer	Short-Nosed GWP 117 & GWP 123 are interchangeable	62H 350	12H 3700	12H 4230 One Spacer
18V-779 18V-780	GWP 123	Short-Nosed	62H 350	12H 3696	12H 1058 One Spacer
18V-779 18V-780	GWP 123	Short-Nosed	62H 350	12H 3696	12H 4744 One Spacer
18V-797 18V-798 18V-801 18V-802	GWP 130	Short-Nosed	62H 350	BHH 1864	12H 4744 One Spacer
18V-797 18V-798 18V-801 18V-802	GWP 130	Short-Nosed	62H 350	BHH 1864	12H 1058 One Spacer
18V-846	GWP 130	Short-Nosed	62H 350	CHM 56	12H 4744

18V-847					One Spacer
18V-847	GWP 130	Short-Nosed	62H 350	CHM 56	Electric Fan
18V-883 **18V-884** **18V-890** **18V-891** **18V-892** **18V-893**	GWP 130	Short-Nosed	62H 350	(1) CHM 56 (1) CAM 392	Dual Electric Fans

It should be noted that the original coolant pumps (BMC Part # GWP 114) did in fact have a "lubricating hole". If you look at about the 2 o'clock position on the top of it, you will spot a round plug with a slotted head. If it is not there, then you have the later sealed-type pump (BMC Part # GWP115) that has to be disassembled in order to lubricate the bearings. To do this remove the pulley hub from the shaft, then withdraw the bearing locating wire from the body aperture. Tap the shaft rearwards, and then remove the shaft complete with bearings. Regrease the bearings, and then press the shaft bearing into the body of the pump, leaving a distance from the seal housing shoulder to the rear face of the shaft bearing of .529" to .539" (13.4366mm to 13.6906mm). Fit a new seal, smearing the jointing face of the seal with mineral oil in order to assure a watertight seal, press the impeller onto the shaft, leaving a running clearance with the body of .020" to .030" (.508mm to .762mm). If the interference fit of the fan hub was impaired when the hub was withdrawn from the shaft, then a new nut must be fitted. The fan hub must be fitted with its face flush with the end of the shaft.

Rather than installing the later aluminum alloy-bodied coolant pump with its stamped steel impeller that was originally introduced on the single-carbureted version of the 18V engine used in the Austin Marina, install a coolant pump with the earlier cast iron body as it has the more efficient die cast impeller that has less of a tendency to produce cavitation at high engine speeds, the resultant air bubbles possibly collecting in one location within the coolant passages of the system to create vapor lock. This being the case, avoid the use of any of the coolant pumps sold by Quinton Hazel as they all use a stamped steel impeller and a corrosion-prone thin-wall aluminum alloy body. The iron-bodied coolant pumps with casting #12H 1504 on their body have a larger diameter, deeper cast impeller also have the advantage of a higher potential for coolant flow rate. An iron-bodied coolant pump with casting #12B 172 on its body has a thin, small-diameter impeller with a lower coolant flow rate potential, making them more appropriate for use with camshafts that produce their maximum power output at a significantly higher than normal engine speed. Because the cast aluminum-bodied coolant pumps have a different coefficient of expansion / contraction from that of the cast iron engine block, they have an annoying tendency to weep coolant around their mounting gasket. Not only will the coefficient of expansion / contraction of the cast iron-bodied coolant pump be closer to that of the cast iron engine block, thus helping to maintain the seal of the mounting gasket, the cast iron-bodied coolant pumps also have another advantage over the thin-wall cast aluminum-bodied units: strength. Items attached to its mounting arm should benefit from its extra rigidity and strength that is greater than that of the rather minimally-reinforced cast aluminum-bodied coolant pumps.

In most Original Equipment applications, this latter attribute may not be of much concern. However, when substituting higher output alternators, or installing other types that do not mount in an essentially similar manner as that of the Original Equipment alternators, there is a strong possibility that this may be testing the limits of the lightly constructed cast aluminum-bodied units. In all of such cases that I am aware of in which the mounting arms have broken off in some alternator conversions, virtually all have occurred when cast aluminum-bodied coolant pumps were used. The alternator, regardless of brand, must be supported by both the front and rear brackets. The front bracket is an integral part of the coolant pump. The rear bracket is an “L” shaped bracket bolted onto the engine block. In the case of the Original Equipment Lucas alternators, there is a sleeve in the rear boss of the housing that helps with the fit, when using two machine bolts, in order to ensure that there is not excessive strain on either mounting point. If a single long machine bolt is used, then this design feature may not come into play and either the alternator mounting boss or the coolant pump itself may be damaged. When using a non-Original Equipment alternator, these design issues need to be taken into account. If the rear mounting point does not meet with the rear bracket, then you will need to fabricate a shim to fit between the rear mounting point of the non-Original Equipment alternator and that of the factory bracket. The shim should be of a thickness that is just sufficient to allow it to be inserted when the front mounting boss of the alternator is in proper alignment and fully tightened. If this is not done, it will exert excessive strain on the front mounting boss. When properly shimmed, it should not demonstrate any weakness. One aspect of such conversions that most owners are unaware of is that the addition of any alternator that produces more amperage also draws more torque from the engine whenever the electrical system demands more amperage. That extra torque, coupled with the weakness of the thinner and weaker cast aluminum-bodied coolant pumps, has caused many a broken alternator conversion. Under such circumstances, a coolant pump of cast aluminum construction may crack.

Note that prior to installing either a generator (dynamo) or an alternator, make sure that you loosen the two machine bolts that secure the rear mounting bracket onto the engine block. After installing the unit, adjust the fan belt and tighten all of the pivot bolts first, and only then should you tighten the rear bracket onto the engine block. If you fail to follow this procedure, you may put excess stress on the rear mounting ear of the generator (dynamo) (or alternator, as the case may be), causing it to eventually break off in service.

V-Belts

It should be noted that a toothed V-belt will absorb less power as well as produce less fatigue-inducing heat than an untoothed one. They also have the advantage of reducing parasitic power loss by lessening the amount of compression / decompression of its rubber, which in turn causes changes in the running V angle of the belt. Dayco manufactures an excellent version that has the double advantage of a superior life expectancy (Dayco Part # 15360).

Engine	**BMC Part #**	**Width**	**Length**
18G and 18GA (3-main-bearing) engines, and 18GB five-main-bearing engines	GFB 103	10mm	900mm

18GD to GK engines, 18V engines (except as below) and those for Germany and Switzerland from 1975-on	GFB 176	10mm	900mm
18V 797/798 and 18V 846/847	GFB 205	10mm	950mm

The modern equivalents for the V-belts listed above are the GCB10900 and the GCB10950, which are direct replacements for those in the above table. In the numerical codes the first two digits signify the width (10mm) and the last three digits signify the length (900mm and 950mm respectively). You may well get slightly longer ones to fit, or perhaps even slightly shorter ones. It depends what pulley is fitted to your alternator, which may not be original. A tip on changing is to get the belt over the pulleys of the crankshaft and the alternator first as they wrap furthest round these, and then the water pump lastly, having got it over the mechanical fan blades first, of course! Another tip is that if the belt is only slightly short, then try removing the adjuster bolt altogether, and that should get the alternator a bit closer to the engine block. In any event, any belt you buy as a spare should be trial fitted when you get it, and it should be kept in the car and not in the garage(!), along with the tools to change it. You always *should* be carrying a spare, even if only by the Murphy's Law in order to guarantee that you will never need it!

The Radiator

Many owners neglect one of the most basic parts of radiator maintenance. At least twice per year, the cooling surfaces of the matrix should be thoroughly cleaned with a pressure sprayer in order to remove the accumulated grime that acts as an insulating barrier between the cooling fins and the air. If you do not think that this accumulation can present much of a problem, imagine what your car would look like if you washed it only twice a year! Should you not have a pressure sprayer, this task can be accomplished at a local car wash that has a spray hose (flexible pipe). To prevent damage to the matrix, be sure that it is not hot when you clean it. Be sure to spray it from behind to force out any squashed insects that may be imbedded inside of it.

Should the power output of your engine be so great that it overwhelms your cooling system, have your local radiator shop recore your radiator with an aluminum alloy matrix fully 1" (25.4mm) thicker than Original Equipment (it will still fit without further modifications) and insist upon the highest number of fins per inch available. In order to be sure of the quality of the workmanship involved, tell the shop that the finished radiator must be able to contain the pressure created by a 13-lb rated radiator cap. However, be aware that a closer proximity of the matrix to the cooling fan may interfere with the efficiency of a cooling shroud should you elect to install one. Avoid the use of alternative copper-brass alloy matrixes which some claim are superior performers. The term "copper-brass" is actually misleading. Brass is an alloy of copper and zinc. Brass has higher malleability than either copper or zinc, so it is excellent for forming into the intricate shapes involved in producing a radiator matrix. While it is true that copper in its pure state is a better conductor of heat than aluminum alloy, it must be noted that pure copper is also very soft and highly prone to corrosion. Pure copper work-hardens as it expands and contracts from cycles of heating and cooling and becomes very, very brittle, eventually cracking. It is also just too soft for use in a radiator matrix that will occasionally be struck by road debris such as rocks, etc., so it is alloyed with tin in order to impart hardness and strength, changing its heat conductivity for the worse. Copper also oxidizes quite rapidly, forming copper oxide, a material with considerably less heat conductivity than aluminum alloy. The effect on cooling capacity is not unlike the exterior of the cylinders inside of the coolant jacket of your engine having a coating of rust. While the copper industry is in fact developing new-technology radiator matrixes with super-thin tubes and fins that are intended to be competitive with existing aluminum alloy matrixes, as of yet they are not available outside of a laboratory. How much they will cost when they finally become available to the public is anybody's guess. While copper is the better conductor of heat, it is inferior to aluminum for getting rid of the heat that it has absorbed. Aluminum alloy actually does a better job of dissipating heat, and it is notably lighter as well. For the time being, an aluminum alloy matrix is the way to go. My 1924cc engine has a Derrington crossflow cylinder head and a Piper 280 camshaft, and its downflow four-row aluminum matrix has always managed to keep the coolant temperature stable, no matter how hot the day and how hard it is driven on steep mountain roads. As an additional side benefit, it is lighter than a standard radiator as well.

However, for those who persist in using a copper-brass radiator matrix, the L-type (Louvered Fin) X2000 matrix offered by Modine is an excellent choice. Flat fins as employed in the Original Equipment Morris radiators are technologically obsolete. Louvered fins create greater turbulence in the air passing over the cooling fins of the matrix, thus maximizing its heat-dissipating abilities. Modine's website can be found at http://www.modine.com/ .

By means of using the oil hoses (pipes) and the oil cooler from the 1974 ½-1980 model MGBs (Moss Motors Part # 235-990), it is possible to relocate the oil cooler to a new position behind the front valance will provide unobstructed airflow to the radiator matrix, while mounting the vented front valance from the 1972 to 1974 ½ models along with a venting duct to the oil cooler will, in turn, provide adequate airflow to the oil cooler. As an additional benefit, this vented front valance (BMC Part # HZA 4812) was originally introduced as an aerodynamic improvement in order to reduce the tendency of the front of the car to "lift" at high speeds. Moss Motors here in the USA can supply this later Chrome Bumper front valance with its two front vent holes. (Moss Motors Part # 457-115) for $169.95. Victoria British can also supply it, Part # 9-918, for $109.95. Our British cousins can get it direct from the manufacturer, British Motor Heritage (Part # HZA592), which can supply the later Chrome Bumper Front Valance with its two vent holes for £68.69.

If you choose to continue to use your Original Equipment three-staggered-row Morris downflow radiator (BMC Part #'s ARH 260, ARC 82, NRP 1142), take the car to a competent radiator shop and have the components of the entire system, including the engine, radiator, and heater core, flushed and descaled to remove the 20+ years' accumulation of muck, rust, and mineral deposits that act as insulators that keep heat from being dispelled by the coolant system. If the heat is trapped in the metal of the engine, your water temperature gauge will not tell you. It measures only the heat in the coolant. You will be surprised at how much cooler the engine will run in the summer and how much warmer the heater will be in the winter.

Whichever radiator matrix you elect to use, be sure to protect it from the corrosive effects of electrolysis. Because of the high electrical conductivity of the water in the coolant, any electricity seeking a ground (earth) will pass through the radiator, electrolyzing the metal of the coolant tubes and thus damaging their ability to transfer heat. This will result in the aluminum alloy of the radiator flaking off and settling in unwanted places inside of the coolant system, sometimes creating blockages. Use electrically nonconductive rubber grommets and nylon washers in order to isolate the radiator from the 5/16”-24 UNF machine bolts that attach it to its mounting diaphragm. Never ground (earth) any electrical device on either the radiator or its mounting diaphragm. The engine block should also be well grounded with its ground (earth) strap.

Beware of cheap radiator hoses (pipes). Due to poor wall strength, they can collapse at high pump speeds and restrict the coolant flow to the coolant pump, resulting in overheating. To test the hose (flexible pipe), reach down to the hose (flexible pipe) that supplies the coolant pump and give it a good squeeze. It should be very difficult to compress. A Kevlar reinforced hose (flexible pipe) (available from Victoria British) should not compress or distort any more than is necessary for mounting. Be warned that these hoses (pipes) have been known to split without warning, having no reinforcement at all except for the chopped Kevlar, which does nothing to contain a split. Because these hoses (pipes) have very little flex to them, they can transmit vibration to the neck mounts of the radiator, in some cases resulting in cracking of the base of its neck mounts. Regardless of the type of hose (flexible pipe) that you elect to install, it would be wise to have a radiator shop reinforce the neck mounts of the radiator by brazing on flanged sleeves at the juncture of their bases and the radiator tanks.

Of course, leaks of the coolant outlet elbow housing gasket develop over time as the gasket deteriorates from both heat and its constant contact with coolant, in time becoming a real nuisance, not to mention making a mess on the front of the engine. In attempting to cure this malady, many people find that their coolant outlet elbow housing resists normal efforts to remove it. Spray some penetrating oil onto the three studs and then let it set overnight. Use a propane torch in order to carefully heat the coolant outlet elbow housing around the stud areas. You should then be able to get it free by tapping it with a hammer, using a small block of wood between the hammer and the housing, then sticking a small hammer handle into the open end of the housing (where the coolant hose had been attached), and gently just working it until it comes loose. If you do this, be careful so that you do not break the housing.

Some people will always prefer to use the less expensive cork thermostat gaskets, resigning themselves to the idea that they will have to learn to tolerate the seepage that is common to cork gaskets after they become saturated. However, there is a solution to this problem: First, make certain the bottom mating surface of your water outlet elbow housing is perfectly smooth and flat by lapping it on a piece of emery cloth or garnet paper that is taped to a pane of flat glass. Next, coat the cork gasket with a thin skim coat of Permatex Ultra Black RTV Gasket Maker. This is possibly the best silicone sealant on the market, able to withstand temperatures of up to 500º Fahrenheit (260º Celsius). Be sure to let the gasket cure for several hours before attempting to trim away any excess material using either an Exacto knife or a single edge razor blade or before installing it. This will result in a rubberized gasket that can survive several assemblies and disassemblies, if necessary. Do not use silicone-based Permatex Blue RTV or Permatex Red Ultra RTV sealant on any of the engine gaskets as they are both prone to failure and contamination of the engine oil under hot operating conditions. In addition, they can also substantially contribute to swelling of the seals. Excessive swelling of a shaft seal's elastomeric lip is a good indicator that the lip material and the lubricant in use are not compatible. Swelling of the seal material can be particularly problematic if some materials, such as silicone, come into contact with oil at high temperatures. Under such conditions, softening, swelling, and reversion of the seal material can occur. If material swell becomes an issue, check to be sure that the elastomer in use is compatible with the lubricant and any other fluids coming into contact with the seal, either during cleaning, installation, or operation. This includes any solvents that may be used during teardown. You should also check to be sure that system fluids are not being contaminated in some way; contamination could cause an otherwise acceptable seal material to swell or degrade.

The main reason for seepage and corrosion on BMC coolant outlet elbow housings is that coolant seeps past the underside of the thermostat flange and attacks the mating surface on the underside of the coolant outlet elbow housing. The Original Equipment gasket sets had a thin circular gasket that was to be inserted into the recessed thermostat seat in the cylinder head, prior to seating the thermostat. If in the process of replacing your old thermostat you discover this item and can remove it intact, do not discard it. Since most gasket sets no longer include this gasket, put a thin bead of the Permatex Ultra Black RTV Gasket Maker sealant into the groove before setting the thermostat in place. When you press the thermostat down in order to firmly seat it, some of the sealant will ooze out past it. Wipe away the excess sealant material and let the assembly cure for an hour or more. After it has properly cured, install your cured rubberized cork gasket, then the water outlet elbow housing, and then after coating the threads of the studs with antisieze compound, secure the housing with Nyloc nuts placed over thick 5/16" machine washers. These Nyloc nuts are preferable to split machine lockwashers because you do not have to use excessive torque to compress a set of split machine lockwashers and thus risk crushing the rubberized gasket. Do not make the classic mistake of overtightening the Nyloc nuts until a large portion of the sealant material appears. Simply tighten them until the gasket begins to bulge outward directly adjacent to the studs. Using this system you should never have a problem with leaks, and your coolant outlet elbow housing should come off easily.

However, there will always be those who take exception to the idea of using a homemade rubberized cork gasket. Fortunately, the engineers at Fel-Pro have come up with a solution called the PermaDryPlus® Water Outlet Gasket (Fel-Pro Part # 35562T). Originally developed to deal with leakage-prone, warped, or corroded thermostat housing flanges, they are constructed of edge-molded silicone rubber on a rigid carrier, providing a superior fit, as well as both high heat and pressure resistance. The rigid carrier prevents over-torquing, while the molded rubber assures a secure seal. Because the studs that secure the thermostat housing to the engine block project downwards into the water jacket, it would be prudent to install studs made of stainless steel and coat their threads with a flexible sealer such as Fel-Pro Gray Bolt Prep in order to prevent coolant from rising up the threads and causing corrosion in the future. These studs are made by ARP and are available from Advanced Performance Technology. The mounting nuts of the studs should be torqued to 8 Ft-lbs. Be aware that if while replacing these parts you should happen to accidentally drop a stud or a nut down into the head, then you absolutely must get it out, regardless of the effort or inconvenience involved. If you fail to remove it, it will eventually be drawn into the coolant pump where it will jam the impeller. This will result in breakage of the drive shaft of the impeller, which in turn will result in rapid overheating of the engine.

Refilling the coolant system so that there will be a reduced likelihood of air pockets is easy once you know how: First, fill the radiator and engine block by pouring the coolant in through the aperture for the thermostat and then refit its coolant outlet elbow housing. Next, disconnect the heater hose (flexible pipe) where it connects to the forward part of the pipe that runs along the rocker arm cover. Insert a small funnel into the hose (flexible pipe).

Holding the hose (flexible pipe) above the height of the heater box, pour in the coolant until it flows out of the pipe from the heater box, then remove funnel and reconnect the hose (flexible pipe) to the pipe. This will minimize the amount of air in the system. If your car is equipped with an recovery tank, fill it 2/3 full and check it when the engine cools off after breaking in the camshaft. If your car does not have a recovery tank, it is wise to install one so that the level of coolant within the radiator matrix will remain at its maximum level.

Fans

While an electric fan that is mounted behind the radiator matrix is 10% more efficient when used as a puller fan than it would be when mounted in front of it and used as a pusher fan, in either position it merely inhibits airflow through the radiator matrix at speeds above 35 MPH. Instead, install one of the two versions of the plastic cooling fans for more effective cooling. These molded plastic fans have more efficient aerodynamically shaped blades than the earlier steel three-blade (BMC Part # 12H 1058) and steel six-blade (BMC Part # BHH 1604) fans, thus they move an increased amount of air with less noise and parasitic power loss. The early version (BMC Part # 12H 4230) found on 18GD, 18GF, 18GH and 18GJ engines, was originally introduced on the 18V engine used in the Austin Marina, a single-carburetor engine that was intended to have a lower maximum engine speed. It is of smaller diameter with unreinforced, coarse-pitch blades, which, while producing more noise than the later version, does an excellent job of drawing air through the radiator matrix at low engine speeds, but tends to “stall out” at the higher engine speeds attainable with the dual-carbureted versions of the 18V engine used in the MGB, resulting in little movement of air. The later version (BMC Part #12H 4744) which is commonly found on 18GK, 18V584, 18V585, 18V672, and 18V673 engines, although it was not standardized on North American Market models until November of 1972 on the 18V-672-Z-L and 18V-673-Z-L engines, is of a larger diameter with seven steel-reinforced finer-pitch blades that does a much better job at high engine speeds. Due to its higher aerodynamic efficiency than that of the old paddle-bladed steel fans, this fan draws more air through the radiator matrix rather than expending most of its energy just stirring it around inside of the engine compartment as the older paddle-bladed metal fans did, requires less power to perform its function, and is actually quieter due to the uneven spacing of its seven blades. Because these fans are lighter, they have less inertia and thus absorb slightly less power and put less strain on the pulley belt whenever a change in engine speed occurs, thus prolonging belt life. Note that the blades are not flat, but have a concave and a convex side to them. The fan should always be fitted with its concave side facing the engine in order to perform with maximum efficiency.

The mounting of either of the plastic fans onto an engine that was originally equipped with the earlier paddle-bladed steel fan is a simple matter of removing the fan pulley wheel from the coolant pump and using it as a jig in order to drill four holes through the boss of the plastic fan so that they will align with those of the fan pulley. To install these fans on a MKI model it will necessary to either mount the short-nosed coolant pump of the 18V engines, or to install the Morris downflow radiator of the 1972 through 1975 MKII models along with the complimentary thermostat housing and coolant hoses (flexible pipes) in order to provide proper clearance for the fan. In either case, you will need to mount a shorter pulley in order to maintain proper alignment with the alternator.

Be advised that at highway speed it is primarily air pressure in front of the radiator that forces air through the radiator matrix, not the fan. Air pressure tends to take the path of least resistance, moving through any open spaces in and around the radiator-mounting diaphragm rather than through the radiator matrix. Therefore, if you want the coolant system to function to maximum effect, be sure that all of the spaces around it and above it are well sealed. However, do not seal the circular apertures in the radiator diaphragm as they are present in order to allow needed cooling air to vent into the engine compartment. If the rest of the spaces through which air can flow are properly sealed, the air pressure in front of the radiator diaphragm at highway speeds will force air through these circular apertures fast enough to substantially assist in removing hot air from the engine compartment.

Ignition theory

The primary ignition is so called because it forms the first part of the ignition circuit. This circuit is used to provide the initial stage towards the secondary High Tension (HT) output. The basic origin of this ignition system evolves from the Magnetic Inductive Principle. This principle is based upon an electromagnetic field (or flux) being produced as the ground (earth) circuit of the ignition coil is completed. This ground (earth) circuit is completed either by contact breaker points or by an amplifier that provides the negative (-) terminal of the ignition coil with a path to ground (earth).

When the contact breaker points are closed, the circuit is complete. Current flow through the ignition coil primary increases from zero to a maximum value (determined by circuit resistance) in an exponential manner, rapidly at first, then slowing as the current reaches its maximum value. The rate at which the current rises is determined by the coil inductance and the circuit resistance. This current through the ignition coil builds an electromagnetic field around the ignition coil until it becomes maximized (or saturated, as the term is used). At low engine speeds, the contact breaker points are closed long enough to allow the current to reach a level limited only by the total circuit resistance, i.e., a DC value. This resistance is designed into the ignition coil, in addition to the inherent resistance in the copper windings, in order to limit current through the contact breaker points at idle and low engine speeds. This resistance is not to be confused with the ballast resistor. It serves a completely different function. As stated above, at low engine speeds, the primary coil current can reach a higher value than it can at high engine speeds. If an ignition coil is designed to provide sufficient output at high speeds, the primary coil current can reach excessive values at low speed. Conversely, if the ignition coil is designed to limit primary current at low engine speeds, it may lack sufficient power at high engine speeds. One way to provide for operation at both low and high engine speeds is to provide a "ballast" resistor in series with the primary winding. This resistance consists of an iron wire coil. Iron has the property of increasing resistance with temperature. At low engine speeds, the high current heats up the iron wire, increasing its resistance, and reducing current. At high engine speeds, the current, as described above, is less, so the iron wire resistance does not increase, thus the current is not limited. There are other design techniques available to provide for wide variations in engine speeds, so not all coils will use an internal resistance. When the contact breaker points open, the ground (earth) is removed and thus the current flow through the ignition coil ceases, the electromagnetic field then collapsing across the ignition coil's primary windings as a result, which in turn "induces" a voltage (hence the term "Inductive Ignition System"). This induced voltage will be determined by three factors: the number of turns in the primary winding, the strength of the magnetic flux, which is proportionate to the current in the primary circuit, and the rate of collapse, which is determined by the speed of the switching of the ground (earth) circuit. The collapsing electromagnetic field (flux) tries to maintain the current through the coil. Without the condenser (capacitor), the voltage will rise to a very high value at the contact breaker points, and arcing will occur. The time for the field to collapse will also increase. With the condenser (capacitor), the current provided by the collapsing field will discharge through it, limiting the voltage at the contact breaker points, and the current/field will collapse very rapidly, having a discharge path to ground through the capacitor.

Vacuum Advance vs. Centrifugal Advance

Be aware that the complete elimination of the vacuum advance in some applications is due to the need for more stable ignition timing with no ignition advance at high engine speeds, or based upon the characteristic low vacuum production of certain engine designs, such as racing engines that run with a wide-open throttle. Use of a non-vacuum advance (i.e., pure centrifugal advance only) distributor (Aldon Part # 101BR1) on a street engine is undesirable due to poor part throttle response and the risk of burning the valves, not to mention increased fuel consumption. These running characteristics are the result of the ignition curve being optimized in order to produce maximum power under full throttle at all times, thus making for quicker response to the transition to full throttle. These factors being the case, non-vacuum advance distributors are appropriate only for competition use. Vacuum advance distributors have the advantage of advancing the timing of the ignition spark beyond that attained with a pure centrifugal advance in order to initiate combustion earlier during the compression stroke when the engine is not under full load, thus giving a fuel economy improvement of from 10 to 20 per cent. At part throttle openings, the engine is not inhaling a full complement of fuel/air mixture, thus it is performing at well below its dynamic compression ratio value. In other words, there is a smaller fuel / air charge in the cylinders to squeeze prior to ignition for the power stroke. To this end, higher ignition values can be used to produce better power and economy without detonation becoming a problem. As the distributor is mechanical in nature, it is already tailored to supply the only advance curve it can. The vacuum advance control unit literally pulls the advance curve forward so that the distributor produces the required higher values. Once the throttle is opened wide in order to give more acceleration / power, the vacuum is lost and the distributor returns to its original mechanical advance curve values.

Ignition has to occur at a fairly critical moment in time (hence the term “ignition timing”) during the compression cycle, and the timing of the ignition has to be altered according to what the engine is doing, e.g., starting, cruising, accelerating, low engine speeds, high engine speeds. The distributor has to manage most of this by itself, but usually with a little help from a vacuum advance control mechanism.

Starting is easier when the ignition spark occurs later during the compression stroke, at about 10° Before Top Dead Center (BTDC), the static ignition timing figure. Once the engine starts and is idling, the ignition timing is advanced, typically to 1° to 15° Before Top Dead Center during the compression stroke. This ignition advance (called centrifugal advance) is achieved by flyweights (roller weights) that centrifugal force causes to pivot outwards as the contact breaker plate spins. This movement of the flyweights (roller weights) is employed in order to produce leverage that is used to alter the positional relationship between the contact breaker points cam lobe and its action shaft, thus causing the contact breaker points to cycle earlier. The flyweights (roller weights) are restrained in their outward movement by springs so that they pivot gradually as engine speed increases, maximum ignition advance being achieved at anything from 2,200 RPM to 6,000 RPM, adding as much as 32° to the static ignition timing figure, depending on which version of the distributor you have. Each flyweight (roller weight) has its own spring and the two springs usually have different characteristics. This progressive ignition timing advance is desirable because the fuel / air mix burns at a constant rate irrespective of engine speed. Should the ignition timing not be advanced as engine speed increases, then combustion will commence further and further into the of the descent of the piston, converting less energy into motion and more into wasted heat. This is wasteful of fuel and potentially damaging to the engine.

When the car is accelerating with large throttle openings, a larger fuel / air mix is drawn into the cylinders, and then ignited, thus creating greater compression pressures. There are certain conditions under which the pressure can become so great that when the ignition spark ignites the fuel / air mixture, instead of normal combustion, an explosion (termed “detonation”) occurs. This explosion happens while the piston is still moving upwards and puts great stresses on the engine, and can actually burn holes in the crown of the piston.

This condition is frequently audible as a metallic "pinking" or "pinging" sound when the engine is under load, such as when laboring up a steep hill. If you hear this, then you should immediately reduce the throttle in order to stop it, downshifting if necessary, and investigate the cause as soon as possible. Detonation causes a sharp spike in combustion chamber pressure, which, over time, can overload and crack the cylinder head gasket that surrounds and seals the top of the cylinder. This leads to burn-through and loss of compression. It is often caused by an over-advanced ignition setting or by weak springs in the distributor's centrifugal advance control mechanism, thus allowing premature advance of the ignition. Unfortunately, since standard Lucas distributor springs are not readily available to the general public, the distributor has to be either replaced, or sent away for reconditioning.

The throttle discs are partly closed when the car attains cruising speed, thus vacuum is at its maximum and is used to advance the ignition timing by means of a vacuum advance control capsule which, depending upon which model of the vacuum advance control capsule it is, adds between 14° to 24° in addition to that set by the centrifugal advance control mechanism. There is therefore a greatly reduced likelihood of preignition, and the engine will run more efficiently as well, producing better fuel economy. Note that the intake manifold vacuum brings maximum ignition advance into play at idle, and reduces it as the throttle is opened. Vacuum taken at the carburetor produces no ignition advance at idle, has maximum ignition advance at light throttle, and reduces ignition advance as the throttle is opened further.

The vacuum advance control capsule is a simple, diaphragm-operated mechanism. As the rubber diaphragm is distorted by the vacuum that is produced by the fuel induction system, it advances the ignition timing by means of a wound wire connector that comes out of the vacuum advance control capsule and attaches to the contact breaker plate. As vacuum is applied, it rotates the contact breaker plate in a clockwise direction, thus advancing the ignition timing, resulting in more engine power. Should the diaphragm of the vacuum advance control capsule happen to rupture, something that happens eventually, the vacuum advance control capsule will cease to function. Worse still, air leaks through the ruptured vacuum unit, back down the vacuum pipe, and then into the intake manifold, diluting the fuel / air mixture. Since this additional air is actually bypassing the carburetors, the fuel / air mixture becomes weaker. Symptoms of this operating condition include rough idle, backfiring, loss of part-throttle power, and even overheating.

MGAs and early MGBs, up until about 1967, all used a vacuum ignition advance control capsule that had a threaded metal vacuum fitting which was designed to be used with either a steel or a copper vacuum pipe. Later distributors employ a hard plastic pipe with a push-on rubber connector. Lucas rationalized all of the vacuum advance control capsules several years ago. There is now only one type of vacuum advance control capsule available. It is designed for a push-on vacuum hose (flexible pipe). To use this replacement vacuum advance control capsule on an early distributor, it is a simple matter of cutting the end off of the old pipe and replacing the compression nut and olive with a rubber elbow.

These vacuum advance control capsules are stamped with a three-figure numerical code that indicates its functional characteristics. For example, should the vacuum advance control capsule be stamped 4-12-16, this indicates that vacuum advance will not commence until the diaphragm has 4” of depression (in inches of mercury) and will attain its maximum ignition advance at 12” of depression, at which it will have advanced the ignition timing a total of 16 crankshaft degrees. They are also stamped with a seven or eight digit serial number that indicates its Lucas part number. Should the vacuum advance control capsule on your distributor prove to be faulty, the following newly manufactured vacuum advance capsules for the following distributors are still available-

Lucas Distributor Serial Number	**Vacuum Advance Control Capsule Lucas Part Number**	**Vacuum Ignition Advance Begins @ / Ends @**	**Maximum Ignition Advance in Crankshaft Degrees**	**Vendor Part Number**
40897	54411985	5” Hg / 13” HG	20°	Moss Motors Part # 163-665
41155	54411985	5” Hg / 13” Hg	20°	Moss Motors Part # 163-630

41288	5441270	4" Hg / 12" Hg	16°	Moss Motors Part # 560-150
41290	5441270	4"Hg / 12" Hg	16°	Moss Motors Part # 560-150
41339	54414868	7" Hg / 13" Hg	10°	Moss Motors Part # 163-660
41370	54423989	7" Hg / 13" Hg	6°	Moss Motors Part # 560-530
41491	54425359	10" Hg / 15" Hg	10°	Moss Motors Part # 163-670
41599	54425516	10" Hg / 15" Hg	10°	Victoria British Part # 0-899

Be aware that if you purchase one these vacuum units, you should then check the specifications when you receive them, as many of the part numbers are still listed in their catalogue, but have been superseded with different specifications. Many may also not be suited for your engine if modifications have been made, so you will need to determine which of these vacuum advance control capsules will serve best in your application.

Ignition In Practice

When the ignition system is turned on, voltage is applied to the positive (+) contact of the ignition coil, or the negative (-) contact on positive (+) ground (earth) cars. However, the current cannot flow unless the circuit is completed. The low side of the ignition coil is grounded (earthed), and thus completes the circuit, by the action of the distributor to which it is connected at the contact that is marked "Input from Coil". Inside of the distributor, the electrical input is then connected by means of the input wire through a rivet that passes through an insulator to a bow spring that conducts the current to the Moving Contact of the contact breaker points. The bow spring pushes the Moving Contact toward the Fixed Contact of the contact breaker points, which is electrically connected to ground (earth) via the Contact breaker plate and its grounding wire. However, the insulator section of the Moving Contact has a cam lobe follower that the bow spring forces to follow the profile of the four lobed contact breaker cam. When the cam lobe follower is not riding on the lobe of the contact breaker cam, the contacts are closed and the current from the ignition coil flows to ground (earth). As the engine rotates, the contact breaker cam rotates counterclockwise (anticlockwise) at half engine speed.

The current ceases to flow when the contact breaker cam lobe rotates to a point where its peak forces the contact breaker points open, causing the electromagnetic field in the ignition coil to collapse and a pulse of high voltage current to thus be generated in the ignition coil. Unfortunately, as the contact breaker points open, an electrical spark is generated between them as the current leaps across the gap between them. The result of this electrical spark is to cause an exchange of metallic materials between the contact breaker points so that one becomes pitted while the other accumulates metallic materials (a phenomenon referred to as "piling"), resulting in poor conductivity, as well as a loss of the electrical energy that would otherwise be utilized at the spark plug.

A notch in the shaft of the contact breaker cam serves as a keyway in order to lock it to the electrical contact rotor arm. As each of the four lobes of the contact breaker cam causes the contact breaker points to open, and an electrical spark to be triggered in consequence, the electrical contact rotor arm points towards a different contact inside of the distributor cap, each of which is attached via the High Tension (HT) leads (spark plug leads) to the spark plugs. Note that the electrical contact rotor arm does not actually touch the high-tension contacts inside of the distributor cap. Instead, it passes so close that the current can jump the gap between the electrical contact rotor arm and the high-tension contacts that are located inside of the distributor cap. Because the distributor's contact breaker cam rotates at a rate that is one-half that of the engine speed, it provides a pulse of electricity to each spark plug only once every other revolution of the crankshaft, as per the principle of the four cycle, i.e., "four stroke" engine.

The rate at which the distributor advances between static and maximum is primarily governed by the control springs of its centrifugal advance flyweights (rolling weights). This mechanism is the area of greatest mystery, not to mention misinformation, in the entire engine compartment. However, this need not be the case, since with the application of a small amount of science, almost any ignition advance curve can be designed into a distributor simply by knowing the properties of the centrifugal advance control springs that are to be installed. These properties can be calculated just by measuring a few key properties of springs, namely: the material of the spring (when in doubt, assume standard spring steel), the diameter of the wire of which the spring is made, the diameter of the coils, the number of coils, and the free lengths between the end loops of the spring. These measurements can then be plugged into a standard extension spring force formula, or a convenient program such as the one supplied by Southern Spring that is available through their website at http://www.southernsprings.co.uk/ , in order to calculate the needed properties. These needed key properties of the ignition advance springs, which dictate the contours of the ignition advance curve, are: Primary spring rate (inch-pounds or N/mm), Primary spring initial tension (lbs or N), Secondary spring rate (inch-pounds or N/mm), and Secondary spring free length (uncompressed length).

Below the speed at which the secondary spring comes into play, the primary spring controls the lower portion of the ignition advance curve in three ways. First, by restraining the flyweights (rolling weights) from causing the advance of the contact breaker cam, second, by controlling the linear ignition advance of the contact breaker cam until the secondary spring engages, and third, by returning the ignition advance mechanism to the zero advance position. The secondary spring controls the upper portion of the ignition advance curve by engaging at a predetermined engine speed, and restraining the linear rate of ignition advance until the advance stop is encountered. Its point of engagement is determined by its free length (uncompressed length). Stroboscopic ignition timing of the engine at idle speed is in the steepest portion of the ignition advance curve, so in order to attain any degree of accuracy in the setting, the stroboscopic ignition timing must be done either when the distributor is in a zero advance state (That is, below 300 distributor RPM or 600 crankshaft RPM), or when the flyweight (rolling weight) cam arm has made contact with its ignition advance stop. The ignition advance curve starts to change shape as the action of the secondary spring engages. Because the primary spring is in control of the lower portion of the ignition advance curve, it must be under constant tension under while in static conditions. On the other hand, the secondary spring must be loose in order to allow the primary spring to work independently until the point at which the secondary spring is required to exert its influence on the ignition advance curve in order to produce the characteristic ignition advance curve with two different rates of ignition advance. Of course, the primary spring continues to function as the secondary spring engages, and continues to function right up to the point where the flyweight (rolling weight) cam arm makes contact with its ignition advance stop. Therefore, the shape of the ignition advance curve after the secondary spring engages reflects the interaction of their combined spring rates. For example, if the primary spring rate is 15 inch-pounds and the secondary rate is 210 inch-pounds, then the effective spring rate of the upper portion of the ignition advance curve is 225 inch-pounds. The spring rate dictates the slope of the curve. If the primary spring influences the ignition advance curve 6° from 300 to 700 RPM, which would constitute a slope of 15° / 1,000 RPM, and the secondary spring influences the ignition advance curve a further 6° from 700 to 2,400 RPM, which would constitute a slope of 3.5° / 1,000 RPM. By measuring, the relationship between spring rate, in either lb/inch or in N/mm, and the rate of ignition advance in degrees / RPM can be derived. Since there is a linear relationship between flyweight (rolling weight) -induced contact breaker cam advance and the distance between the spring mount posts, the point of engagement, and thus the advance position, this simple relationship can be determined by measuring the distance between the posts at both extremes. Measure between the spring posts with the contact breaker cam in the zero advance position, and then measure same dimension with the contact breaker cam in the full advance position. The difference between these two values divided by the maximum number of degrees of ignition advance will give the number of degrees of ignition advance per inch.

It should be noted that the Lucas distributor is not exactly a precision-assembled piece of equipment; there is always a small variation in the positioning of spring posts. The weight of the flyweight (rolling weight) will have stamped upon it the number of degrees of maximum centrifugal advance that is it designed to allow. If you should be fortunate enough to have more than one set of them, make a distance measurement between posts on both the cam arm side and the opposite side, and then use whichever pair of weights is most beneficial. Remember that if a flyweight (rolling weight) is either modified or changed in order to give more or less centrifugal advance, then the positions of the posts will not be exactly the same as they were before, so the break point in the ignition advance curve will be at a subtly different position on the ignition advance curve.

After changing anything on the distributor, always check the ignition advance curve. Be aware that using a timing light only at idle and maximum ignition advance will not establish the true ignition advance curve, only its upper and lower limits. There is a grave potential error in blindly putting your trust in such a crude dual-point ignition timing calibration method. The ignition advance curve should always be carefully checked throughout the span of its entire range.

Be aware that contact breaker points bounce causes erratic running of the engine and decreases ignition coil charge time, reducing the available ignition charge power. It should be noted that if the follower of the contact breaker points lifts off of the contact breaker cam lobe at its peak and does not come back down again until after the point at which the contact breaker points normally would have closed had the follower followed the contour of the contact breaker cam lobe, then that is not termed “points bounce” but “overshoot”. However, contact breaker points bounce can occur when overshoot occurs, and is actually the result of the follower of the contact breaker points bouncing off of the contact breaker cam lobe sufficiently to open the contact breaker points a second time. This generates double ignition sparks, and while “overshoot” can reduce ignition coil charge time, “points bounce” can reduce it even further.

The Lucas 25D4 distributor has a symmetrical-profile contact breaker cam lobe that contributes to this problem of “points bounce”, often causing the bouncing of the contact breaker points to start occurring in the engine speed range of 5,500 RPM to 6,000 RPM when used with the standard quick-fit “brown follower” contact breaker points that have a spring tension of 22 ounces. This can be cured by installing a long pin contact breaker plate from the later Lucas 25D4 distributors (Lucas Part # 54412436) which uses a different set of breaker contact points equipped with a spring tension of 32 ounces (Lucas Part # 5441356). This additional pressure will result in quicker wear of the follower block, although frequent attention to its lubrication can somewhat mitigate this problem. On the other hand, the Lucas 45D Series distributor has an asymmetrical-profile contact breaker cam lobe which has a slower closing ramp angle and thus does not produce contact breaker points bounce until 7,500 RPM, an engine speed not seen in B Series street engines. Unfortunately, this asymmetrical-profile contact breaker cam lobe will not interchange onto the action shaft of a Lucas 25D Series distributor.

The Lucas 25D4 distributor was produced in an extensive number of variations, but they were internally identical apart from their cam arms and springs. However, these differences can be significant in terms of their manner of function and performance-

Distributor Specification Number 40897

Cam Arm: 10°

Springs	**Primary**	**Secondary**
Diameter of wire	.020"	.030"
Number of turns	12	9
Spring diameter	.185"	.211"
Coiled length	.260"	.250"
Overall length	.632"	.661"

Distributor Specification Number 40897E

Cam Arm: 10°

Springs	**Primary**	**Secondary**
Diameter of wire	.021"	.039"
Number of turns	12	6
Spring diameter	.186"	.203"
Coiled length	.256"	.255"
Overall length	.625"	.764"

Distributor Specification Number 41400

Cam Arm: 18°

Springs	Primary	Secondary
Diameter of wire	.031"	.033"
Number of turns	5	5
Spring diameter	.244"	.243"
Coiled length	.278"	.285"
Overall length	.695"	.696"

Distributor Specification Number 41491

Cam Arm: not marked

Springs	Primary	Secondary
Diameter of wire	.030"	.030"
Number of turns	7	7
Spring diameter	.204"	.204"
Coiled length	.320"	.323"
Overall length	.735"	.734"

You are not limited to using the Lucas contact breaker points that are readily available for the Lucas 25D4 Series distributors. Standard Blue Streak contact breaker points (Blue Streak Part Number LU-1617XP) are a much higher quality part, albeit at a higher cost, that has a more durable follower block and a felt lubricant strip where grease for the camshaft lobe is applied. The Blue Streak contact breaker points require a seemingly modest spring pressure of 28 ounces to open, but they are of a different design that has a much larger contact area, a reinforced backer at the contact breaker points and a differently-shaped spring design that also combine in order to help prevent contact breaker points bounce at higher engine speeds. It presents a very well thought out redesign. As an alternative, the “brown follower” Lucas contact breaker points, with a spring tension of 37 ounces, are very good at eliminating contact breaker points bounce at the cost of faster wear of its follower, while the “red follower” Lucas contact breaker points (Lucas Part# GCS101) are, at best, marginal in their performance, with bounce occurring as early as 3,500 RPM as a result of their rather meager spring tension of 25 ounces. These are also further handicapped by having a very sloppy fit at their main pivot. In all, this compares poorly with vintage New Old Stock (NOS) Lucas two-piece contact breaker points (Lucas Part# GCS107) with their spring tension of 31 ounces.

Advance Distributors also markets a matching condenser (capacitor) (Blue Streak Part # LU-206) that does not have the failure rate of the standard Lucas item. It appears that the Lucas condensers (capacitors) are failing because of an extremely poor design and equally poor manufacturing quality, resulting in an erratic electrical contact at the can end of the Mylar / foil capsule. Since there is nothing to maintain pressure on the capsule inside of the case, temperature variations and vibration eventually cause this contact to loosen, and the can end begins to arc. This creates additional resistance, and may even cause an intermittent open circuit inside of the condenser (capacitor), which in turn results in a weak spark, burned points, and rapid ignition failure. It is also consistent with reports that new Lucas condensers (capacitors) often seem to work well for a period of time, and then suddenly fail without warning. The poor quality of these components is more than a little disturbing. I view these condensers (capacitors) as being time bombs inside of the ignition system. The chance of one failing, sooner or later, is pretty high. Such sloppy manufacturing and design quality really should not be tolerated.

It should be noted that for the 45D4 distributor there are both “sliding” (Lucas Part# GCS124) contact breaker points and “non-sliding” (Lucas Part# GCS118) contact breaker points variants available. These are not interchangeable as there are significant differences in their appropriate contact breaker points plates. The Lucas GCS118 contact breaker points of the Lucas 45D4 distributor are similar to the Lucas GCS101 contact breaker points of the Lucas 25D4 distributor in that they have a red plastic “cam follower”, the pivot of which fits over a pin that is attached to the contact breaker points plate. However, the adjuster notch is at the pivot end instead of at the connection end, so while the Lucas GCS118 and the Lucas GSC 101 contact breaker points may be interchangeable, an inappropriate combination of distributor and contact breaker points is much more awkward to make fine adjustments to. The "sliding" Lucas GCS124 contact breaker points have a blue cam follower, a brass peg under its pivot that locates into a hole in the contact breaker points plate, and the adjuster notch is located at its connection end. These contact breaker points have a "sliding" contact breaker point that moves across the fixed contact breaker point as well as open and closed as normal. The design incorporates a slotted plastic lever under the pivot that engages with a fixed pin on the distributor. As the contact breaker points plate rotates under changing vacuum, the fixed pin moves the slotted lever back and forth. The lever has a cam on its upper surface and there are pegs on the bottom of the cam follower. As the lever is moved, this causes the moving contact breaker point to move up and down relative to the fixed contact breaker point. Because the contact breaker points are closed approximately half the time there is a 50-50 chance that they will be closed when the moving contact breaker point is moved up or down. This slides the two surfaces of the contact breaker points across each other, and even without sliding, the two contact breaker points will make and break on different parts of their surfaces. Both these effects help keep them clean and free from the spike and pit that so often afflicts fixed contact breaker points.

The Right Distributor

Remember: if you change either the lobe profile or the timing of the camshaft significantly, you will have to alter the ignition curve. My personal favorite distributor for a B Series engine is the Lucas distributor. They are contact breaker points distributors that are perfectly reliable and that can be readily converted to electronic ignition if you decide to eliminate the maintenance of contact breaker points. There is an abundance of parts available, and the maintenance parts are the most affordable of any type of MGB distributor. The centrifugal advance mechanisms are extremely reliable and consistent, and are made from extremely high quality hardened steel components. There are thousands of different replacement centrifugal advance springs as well, and there are also an endless number of choices for vacuum advance capsules for either manifold or ported vacuum, so setting the right ignition advance curve for a custom engine is a matter of routine.

Probably the worst distributor that you can use is the North American Market specification Lucas 45DE4, also known as the infamous OPUS distributor. It is very easily identified by the big black amplifier box on the side of its body. It is this black amplifier box that houses its electronics. Amongst other oddities, it also housed a separate ballast component to produce resistance in addition to the coil ballast wire, which made diagnosing an ignition problem an unduly complex affair. These were the units that typically failed within a couple years of production, helping to give the MGB a bad reputation for unreliability. Because of the fact that it uses so much ignition advance and retard in order to meet US emissions standards, it eliminates any hope of getting real performance from the engine. Of the emissions-era distributors, the sole performance exception is the late model Lucas 45DM4 that offers a total of 10° of centrifugal advance. It utilizes a remote amplifier box that is mounted under the ignition coil. These use a particularly good street performance ignition curve. The Lucas 45DM4 distributor is also known as the CEI (Constant Energy Ignition) distributor, and uses a General Motors HEI ignition module (Delco Part # DM1906, NAPA Part # TP45SB) in its external amplifier box that can be easily replaced from literally any parts store in the US. Higher performance modules are also easy to source from companies such as Pertronix. In order to compensate for a shorter period of available charging time, the dwell on a Constant Energy Ignition system increases as the engine speed increases. The term Constant Energy Ignition refers to the available voltage produced by the ignition coil. This voltage, regardless of engine speed, will remain constant, as opposed to a contact breaker points ignition system wherein an increase in engine speed means that the contact breaker points are closed for a shorter period of time, reducing ignition voltage as engine speed increases. This reduces the effective time that the ignition coil has to fully saturate and maximize the strength of the magnetic flux. Be advised that this GM unit will fail every time if the spark plugs wires are removed and the owner tries to start the engine while checking for spark by holding one of the disconnected spark plug wires in the vicinity of ground (earth).

The most serious problem with using the Original Equipment specification distributors lies in the fact that their original ignition curves are no longer that relevant to today's very different unleaded fuels. The European specification Lucas 45D4, however, is excellent for this purpose, although in its original form it should not be considered to be suitable for a high performance application. This is available from Brit Tek as their Eurospec distributor (Brit Tek Part # ESD-001). Unfortunately, being a new unit, its components are outsourced from India and the quality is not the same as that of a true, British-made Original Equipment Lucas 45D4 distributor. The Eurospec distributors that used to be real Made-in-the-UK Lucas 45D4 distributors had a serious downfall in their ignition curve. It is a really “safe” ignition curve, which I believe was a factory mistake. In order to attain maximum performance, the ignition curve should reach its maximum degree of ignition advance at an engine speed of 3,000 RPM. The Eurospec distributor reaches its maximum degree of ignition advance by a distributor speed of 3,000 RPM, which is an engine speed of 6,000 RPM! That makes for a loss of power due to a low amount of ignition advance when you really need it. Sadly, the supply of real Made-in-the-UK Lucas distributors has dried up. They are now assembled using metric screws, so they are now made in either India or China, although still cast as a Lucas part. Lucas TVS in India actually marks their distributors “Made in India”, so they are obviously not those! It is still a small mystery as to who actually makes them, but it has been narrowed down. The ignition curve is now better since they have changed manufacturing sources, but the distributor makes use of lower quality shaft bushing materials that probably will not last more than about 50,000 miles, the center pivot on the contact breaker plate is plastic and will wear quickly, the vacuum advance control capsule is rarely the correct specification for the engine that it is designed to be used in, and the ignition curve is never optimal. In fact, the new 45D4s distributors are much like snowflakes – no two are ever the same (at least in terms of the ignition curve). Another issue with some of them is the points cam: they can offer up to .010” (.254mm) different lift at each of the 4 lobes, causing a major problem with setting the dwell angle and getting the distributor to run properly. Fortunately, this problem can be remedied by eliminating the need for contact breaker points through the installation of an electronic ignition system. An original Made-in-the-UK Lucas distributor that has been rebuilt and recurved by either Advance Distributors or by Aldon Automotive is even better still. The Aldon distributor is available in the USA from Brit Tek as their Stage II distributor (Brit Tek Part # SSD001).

Aldon Automotive in the UK makes an entire range of recurved Lucas distributors for different specifications of the BMC B Series engine. The significant difference between the different models of Aldon distributors and the Original Equipment distributors is that their ignition advance curves have been customized for different specifications. While the same can also be said for the units produced by Advanced Distributors, theirs are machined to tolerances much tighter than Aldon and hand-fit every distributor, matching components for the best possible performance in its intended application.

The Original Equipment distributors have a very conservative ignition advance curve that the engineers at the factory designed for long-term reliability. That is, if the ignition timing is somewhat out of phase with the crankshaft, or the gap of the contact breaker points is a bit undersized, then the engine will still be reasonably reliable. A factory-specification engine can probably go about 6,000 miles before the engine will run so poorly that you will be forced to readjust the ignition timing, which is about as long as a set of ignition contact breaker points will last when used with the Original Equipment 20 Kilovolt Lucas HA 12 unballasted ignition coil (BMC Part # GCL 101). Hence, the entire ignition system "tune up" can be comprehensively done all at the same time. Although a different ignition advance curve would have given better performance, the engineers at the factory were instructed that owner convenience and long-term reliability were of a higher order of priority than maximum performance, so an Original Equipment specification ignition advance curve was developed that was deemed appropriate for most owners. The Aldon distributors make use of an ignition advance curve that is calculated to give more power and a crisper throttle response. As such, they are more appropriate for an engine that is being modified for a higher level of performance. However, when installed on an otherwise Original Equipment specification engine, the different ignition advance curve will result in an increase in midrange torque. One model is intended for use on Original Equipment specification engines equipped with the HS4 Series carburetor with ported vacuum advance (Aldon Part # 101BY1). Another model is intended for use on Original Equipment specification engines equipped with the HIF4 Series carburetor with intake manifold vacuum advance (Aldon Part # 101BY2). Yet another model (Aldon Part # 101BR2) is intended for use on engines that have been fitted with a Piper BP270 or BP285 camshaft. Aldon also markets both optically and magnetically triggered contact breaker points replacement systems for Lucas distributors under the Petronix brand name. Aldon Automotive has a website that can be found at http://www.aldonauto.co.uk/

In recent years a relatively new, all-electronic distributor called the 123 has appeared on the market. Its only moving part is the action shaft. Flipping dipswitches under the bottom of the distributor selects the available ignition advance curves. There are 10 different ignition advance curves to chose from, and that is the downfall of this distributor. The ignition curves are all Original Equipment specification ignition curves, designed decades ago to work with fuels that had different combustion characteristics from those of the fuels of today. Unfortunately, it has a more integral issue. As the engine speed rises, the ignition advance should come on smoothly and evenly, but instead it appears to jump 1° at a time, like the second hand of an old mechanical clock. Tick, tick, tick. That is not going to give the crisp throttle response of a well-tuned old-fashioned Lucas distributor! This distributor uses both a Bosch cap and a Bosch electrical contact rotor arm, either of which are not widely available, but they can be obtained with a bit of diligent searching.

Professional Ignition Service

For those who do not relish the idea of paying the cost for transatlantic shipping and insurance in order to obtain the services of Aldon Automotive, there is an alternative source for a distributor expert who can customize the ignition curve of a Lucas distributor. Jeff Schlemmer, of Shakopee, Minnesota, an enthusiast-turned-entrepreneur, has been working on Lucas distributors for over eight years, and on several other brands for over 20 years. Jeff only recently opened his Advanced Distributors business on a commercial basis about five years ago when his "hobby" got beyond the time that he had available and the demand for his work became apparent. It is truly a passion for him. So much so that Jeff has gone to the trouble of having distributor caps with brass contacts as well as special red high-performance electrical contact rotor arms especially made for all of the four-cylinder Lucas 22D, 23D, 25D, and DM2 distributors, as well as for the six-cylinder versions, because the commonly-available ones are so unsatisfactory. Recently manufactured rotor arms have been failing due to the typical "mix" used in the injection molding containing more carbon blacking, making it more conductive. Still more importantly, the rivet on top of the rotor arm inside the base circle of the rotor which holds the brass inlay into the molding is slightly longer than the original, which puts it very close to the distributor shaft and too close to the spring clip on the underside. The high tension current, averaging 30,000 volts, is always seeking the easiest route to ground (earth), and shorts out from the tip of the over length rivet, through the reduced thickness of more conductive plastic and the spring clip on the underside of the rotor arm, to ground (earth) out down the distributor shaft. The result - no sparks at the spark plugs. The situation sometimes rectifies itself on cooling, but then reoccurs with increasing frequency until the rotor permanently short circuits. Jeff's rotors do not have this problem.

Jeff offers full distributor rebuilding with either Original Equipment or customized ignition curves for anything that you can dream up that is British. Lucas 22D, 23D, 25D, 45D, DM2, DM6, even through all of the old T Series distributors, even the Opus units, as well as the CEI units. His rebuilding procedure starts with a thorough cleaning of every component. I am not talking about simply running them through a parts washer or just degreasing them - they must appear as new in order to perform as new. Jeff knows from hard-won experience that it is only at this point that accurate measurements can be taken in order to perform all of the necessary precision machine work. At that point, all of the parts are machined as necessary in order to eliminate play and slop that results from wear, until precision components are produced that can be fitted back together. If necessary, he can even repair the common forms of damage that are often found at the canelure for the distributor clamping plate of many 25D distributors. He machines away the broken portion, and then press fits a laser-cut steel ring in its place. The steel ring is then bonded to the housing, positively engaged with internal tabs in order to prevent rotation, the resulting structure being absolutely crush proof, no matter how bent your distributor clamping plate may be. The distributor body is cleaned, glass-bead blasted, and then satin polished using a proprietary process in order to seal its pores and thus make it more corrosion- and stain-resistant. Afterwards, if the customer should choose the option, it is then polished to a high gloss finish. The polishing process is also used for other critical parts, such as the shafts and the centrifugal advance flyweights (roller weights) where consistent finishes are a basis to reliability and repeatability. Afterwards, the parts are dried and clear-coated should the possibility of corrosion could prove to be an issue. Some parts are zinc plated as needed for long-term durability, such as cap clips, breaker plates, and clamps. In the case of breaker plates, they are repolished after plating for an ultra-slick, low friction surface in order to maximize both functional precision and reliability. This is obviously not a typical rebuild wherein parts are simply cleaned and then reassembled. The normal factory tolerances may have to go out the window. Every distributor that he rebuilds is a hand-assembled masterwork that is specifically assembled for its intended vehicle. The last step of his rebuilding procedure is the choosing of appropriate centrifugal advance control springs and then testing the distributor on a Sun machine. The distributor is fitted with its fully rebuilt breaker plate, and then the triggering device of choice is installed for testing, whether it be contact breaker points, or either a Pertronix or a Crane breakerless conversion system.

Perhaps one of the most notable services that Jeff offers lies in his ability to fabricate centrifugal advance control springs, even for one-off custom applications. A unique service that is offered by Advanced Distributors is the rebuilding of your old vacuum ignition advance control capsule. In order to replace the failed rubber diaphragm with a modern, fuel-resistant diaphragm, he has designed a way to open the vacuum ignition advance control capsule and recrimp it in the manner that the factory used. This will permit you to keep the original look of your engine without sacrificing performance, should you ever elect to enter a concourse event. Advanced Distributors offers the simplicity of a "no part number" system for placing orders– simply visit their website and print out a copy of their Ignition Curve Worksheet:
http://advanceddistributors.com/Curve%20Worksheet%2007.htm and fill in the pertinent details of your engine and car. Unlike other companies, that makes the selection of the appropriate ignition advance curve characteristics the responsibility of Advanced Distributors, rather than forcing you to take the responsibility to make an uninformed choice. Perhaps Jeff describes his business attitude best: "My goal is really to help folks get their cars running better. That means 100% customer satisfaction..... I do what I would expect as my own customer. I'll always be my own biggest critic!!" Advanced Distributors has a website at http://advanceddistributors.com and Jeff Schlemmer himself can be Emailed at jeff@advanceddistributors.com .

Custom Ignition Curves

In developing a custom ignition curve, the objective is to achieve maximum combustion pressure at the optimum point in the stroke. Although engine speed can vary, the fuel / air mixture combusts at a fixed rate. Therefore, the fuel / air mixture has to be ignited progressively earlier in the compression stroke as engine speed increases. However, if ignition occurs prematurely, then the pressure wave inside of the combustion chamber will reach the piston crown while the thrust axis of the connecting rod is aligned with throw of the crankshaft, overcoming the pressure of the oil in the bearing, thus causing engine knock, and resulting in damaged bearings, journals, and even a collapsed or broken piston crown.

However, should ignition occur later than the optimum moment, the pressure wave generated by combustion will end up chasing the piston crown down the cylinder, reaching the piston crown too late to achieve maximum pressure against it and thus exert maximum force against it. The result is decreased power output. If the ignition timing is correctly advanced, much of the heat energy is absorbed by the piston crown because that is where the greatest effect of the pressure wave is. However, it must be understood that the heat energy has to actually go somewhere. If the ignition timing is retarded, even though power output is lessened partly by achieving relatively less pressure on the piston during the downstroke (and relatively more on the upstroke), a relatively larger proportion of the heat produced has nowhere to go except into the roof of the combustion chamber. Hence, hotter running is the inevitable result.

There are a few basic rules to be taken into consideration when trying to figure out what king of ignition advance curve is appropriate for your particular engine specification. For example, the longer the duration figure of a camshaft is, the more ignition advance it will need at low engine speed, and less it will need at high engine speeds.

The higher the compression ratio, the less the total ignition advance that is required. This is due to the fact that as the fuel / air mixture is forced into a smaller volume, the shorter the distance will be that the flame front will be required to travel across. Likewise it is minimized for flat-topped pistons, and maximized for dished pistons.

As the efficiency of the exhaust system increases and backpressure consequently decreases, the less ignition advance is required when the exhaust extraction effect is working, i.e., as engine speed increases. The exhaust extraction effect helps in riding the engine of exhaust gases that would retard the combustion process.

As induction efficiency increases, the less ignition advance is required. This is due to the fact that the engine receives a larger volume of fuel / air charge, and thus the compression ratio is increased.

As the engine's ability to rid itself of residual heat is improved, the need for more ignition advance is increased. This means that aluminum cylinder heads require more ignition advance than cast iron cylinder heads.

As a reasonable starting point, the static setting should be 14° Before Top Dead Center and the maximum centrifugal advance setting should be 20° Before Top Dead Center for a total of 34° of ignition advance. If a very hot camshaft is used, more ignition advance may be necessary in order to obtain the best idle. With these initial settings in place as a starting point, you should be able to develop the optimum ignition advance curve for your engine. Be sure to use no more ignition advance than is necessary to obtain optimum power or you will risk burning the valves. As a result of the higher combustion temperatures involved, the use of Austenitic 214N stainless steel valves is highly recommended.

While the Weslake-designed kidney-shaped combustion chamber renders its best performance when the ignition timing at full advance is set at 34° to 35° Before Top Dead Center, the best ignition timing for setting the idle is dependent upon which camshaft is used. A Piper BP270 camshaft idles best with a total ignition advance ignition setting at 10° to 12° Before Top Dead Center at 600 to 700 RPM while the Piper BP285 camshaft idles best with a total ignition advance ignition setting of 13° to 15° Before Top Dead Center at 950 to 1,150 RPM. Regardless of which camshaft you choose, the ignition should reach full total ignition advance no later than 34° to 35° Before Top Dead Center at approximately 3,500 to 3,700 RPM.

If you seriously want to leave open the option of a customized ignition advance curve in order to meet your individual needs, a distributor that has an adjustable ignition advance curve is desirable, such as the one made by Mallory. It is available with contact breaker points in both vacuum advance and pure centrifugal advance versions (Victoria British Part #'s 17-501 and 17-500, respectively; Mallory Part #'s 2332001 and 2732001, respectively). In both versions, the centrifugal advance control mechanism is adjustable over a range of from 16° to 28° by means of a simple Allen wrench. The vacuum advance curve of the vacuum advance version of the distributor is adjustable by using a 3/32" Allen wrench and inserting it into the hose (flexible pipe) connection nipple and altering the tension value of the diaphragm. Mallory makes this adjustable vacuum advance unit available as a separate replacement part for their distributors (Mallory Part # 29332). An ignition advance curve kit consisting of both an assortment of centrifugal advance weight springs and the Allen wrench is readily available (Moss Motors Part # 143-236). This permits you to start the ignition curve where you want, control the rate of advance as you please, and set the maximum advance where you want it. It also has the advantage of having a dual point ignition triggering system. In this type of system both sets of contact breaker points are joined by a wire so that when the first set of contact breaker points open, nothing happens until the second set of contact breaker points open. The second set of contact breaker points open just as the first set is closing. This quick closing of the circuit (approximately 5°) gives the ignition coil a maximum amount of dwell angle (72°) to charge, thus increasing the voltage of any given ignition coil. This makes the system highly appropriate for engines equipped with a camshaft designed for high engine speed applications. The ignition system of a six-cylinder engine fires 50% more often and an eight-cylinder engine fires twice as often. In such engines equipped with radical camshafts, the increased ignition coil charge time can become critical at high engine speeds. However, other than the ability to have the ignition advance curve custom-tailored to work with almost any camshaft, there is no practical advantage to the increased ignition coil charging time characteristic of the Mallory distributor when used on a four-cylinder engine. Converting the distributor to electromagnetic or electro-optical triggering will eliminate the problem of the notoriously short-lived contact breaker points and will entail no tangible sacrifice. For the MGB with a special camshaft, however, a customized ignition curve can help exploit that last bit of potential power as well as deliver better response to changes of the throttle, while avoiding the dangers of preignition. Both versions are also available as Unilite distributors with solid-state triggering (Victoria British Part #'s 17-503 and 17-502, respectively: Mallory Part #'s 2732001 and 4732001, respectively). Moss Motors has a website that can be found at http://www.mossmotors.com/ , Victoria British has a website that can be found at http://www.victoriabritish.com/ , and Mallory has a website at http://www.malloryracing.com/default.aspx?BrandID=6 . Replacement parts are quite expensive in comparison to Lucas parts, but they last longer and have fewer "issues" than Lucas parts.

However, you do not necessarily have to go such a big bucks route, and you do not have to place your trust in the appropriateness of Aldon's ignition advance curve, either. If your Lucas 25D4 or Lucas 45D4 distributor is in good shape and you do not mind doing things the labor-intensive way, Cambridge Motorsport in the UK offers a package of five advance springs that will enable you to tailor both the primary and the secondary rates of your ignition advance curve. Unless you have an assortment of spare advance plates to work with, however, your total ignition advance will remain the same. Do not bother trying to refine the ignition advance rate of the ignition advance curve by experimenting with different flyweights (rolling weights) for the Lucas 45D4 distributor, as they are all of the same weight and will not interchange with those of the Lucas 25D4 distributor. Due to the fact that the Lucas 45D Series distributor only offers one type of flyweight (rolling weight), the tailoring of the ignition advance curve can be done only by modifying the maximum amount of ignition advance and by changing the ignition advance springs.

However, there are two versions of the flyweights (rolling weights) that can be employed in the Lucas 25D4 distributors, with the latter version being of greater length and of heavier weight. These are found in the Lucas 23D and Lucas 25D distributors and are mechanically interchangeable with those employed in the Lucas 25D4 distributors. Their greater length and heavier weight will result in both a slightly earlier onset and a slightly faster rate of centrifugal advance when employed in a Lucas 25D4 distributor. The inward positioning of the extra weight makes very little alteration of the ignition advance curve, although the position of the flyweights (rolling weights) do make for a small change. However, keep in mind the fact that if you change the flyweights (rolling weights) in a Lucas 25D4 distributor, the two different lengths of flyweights (rolling weights) cannot be intermixed. When the longer, heavier flyweights (rolling weights) are substituted into a Lucas 25D4 distributor, which normally uses the short, lighter flyweights (rolling weights), they will reduce ignition advance by 3º, while the shorter, lighter flyweights (rolling weights) will add 3º of ignition advance when substituted into a Lucas 23D or Lucas 25D distributor, which normally use the longer, heavier flyweights (rolling weights). Because of this, you need to be attentive to the stamped number that designates the number of degrees of maximum ignition advance. For example, a "13" that is normally used with the short, light flyweights (rolling weights) will in effect become a "10" when the longer, heavier flyweights (rolling weights) are installed into a Lucas 25D4 distributor. A "17" that is normally used with the longer, heavier flyweights (rolling weights) will become in effect a "20" when the shorter, lighter flyweights (rolling weights) are installed in the Lucas 23D and Lucas 25D distributors. Be aware that you must either use two of the longer, heavier flyweights (rolling weights) or two of the shorter, lighter flyweights (rolling weights). Mixing the two different types will result in an erratic idle and inferior performance. Be aware that only after the centrifugal advance curve is successfully established can different vacuum advance control capsules be experimented with.

In customizing the ignition advance curve of a modified B Series engine, the easiest and most common tool to use an inductive pickup timing light. The pickup cable attaches around the front spark plug wire of Cylinder #1, and the other two cables go to power (+) and to ground (earth) (-). The black ground (earth) (-) cable can be attached to the alternator at any point that has bare exposed metal. The red wire can be attached to the lowest fuse in the fuse panel, which always has 12 Volts present. Flash the timing light on the timing marks of the harmonic balancer pulley wheel with the engine running. On 18V engines, the timing marks will be on the top passenger side of the harmonic balancer pulley wheel / timing chain cover. On the earlier 18G through 18GK engines, they will be at the bottom center of the harmonic balancer pulley wheel. As you face the front of the engine, the furthest mark clockwise is Top Dead Center (TDC). For the 18V engines that means that the furthest mark to the right is Top Dead Center (TDC), while on the earlier series engines the furthest mark to the left is Top Dead Center. Counter-clockwise from there each pointer represents an additional 5^{o} Before Top Dead Center (BTDC.)

What settings should you use? That is a great question, since most cars no longer have the original distributor, or the distributor is fatigued, rebuilt, or sometimes altered in some way from original. A good, safe starting point is usually somewhere around 10^{o} Before Top Dead Center (BTDC), as measured with the vacuum advance hose disconnected and its connection on the fuel induction system plugged. If you do not have access to a dynometer, you can do a load test on the road in order to see how the car runs both with and without the vacuum advance hooked up.

I recommend performing a load test on the engine by lugging it down while in fourth gear and driving at a wide-open throttle at engine speeds varying between a low engine speed of 1,500 up to 3,500 RPM, the upper point on the ignition advance curve where the ignition should be at full ignition advance. I also recommend that you remove one of the rubber firewall (bulkhead) plugs from either behind the brake box or from the passenger side (Right Hand Drive brake box plug) so that you can more clearly hear what is happening inside of the engine compartment. Sometimes that will allow you to more easily hear preignition that is otherwise muffled by good sound insulation. Preignition is a problem that has many causes. It is best described as an audible “shaking can of marbles” in your engine and is a quite distinct sound. More often than not, over-advanced ignition timing causes preignition. However, there are other causes you need to consider. While retarding the ignition timing can alleviate the problem, vacuum leaks in the fuel induction system can cause a lean fuel / air mixture that causes detonation, which sounds very similar. Making certain that both your carburetor and your intake manifold gaskets in particular are well sealed is a great place to start. The lean condition caused by vacuum leaks may also be symptomized as a surging while maintaining highway speeds. Preignition can also be caused by too rich of a fuel mixture. This is common to hear in high performance engines – especially if running a Weber DCOE or a Dellorto carburetor. When the mixture temporarily becomes rich, a preignition sound will occur. Many times this is the result of having too small of an accelerator pump return jet in a Weber or a Dellorto carburetor. The use of a Gunson's exhaust analyzer powered by a 12 Volt / 110 Volt current inverter that is plugged into the cigar lighter receptacle can offer insight into the question as to whether or not any preignition is being caused by either a lean or a rich fuel / air mixture. If the engine lacks power and there is no preignition noise coming from the engine while under load, then you can advance the ignition timing 2° at a time (higher numerical setting Before Top Dead Center (BTDC). Rotating the distributor clockwise will advance the ignition timing, and rotating it counter-clockwise will retard the ignition timing. Try settings of 10°, 12°, 14°, and 16° Before Top Dead Center (BTDC) until you start to hear preignition under load. At that point, retard the ignition timing until the preignition is gone and record both the engine speed and the ignition timing in your notes so that you can plot the ideal ignition advance curve. In some cases, you will never hear preignition, in which case I recommend not exceeding a total of 34° of ignition advance timing.

How can total ignition advance timing be set? By using a dial-back timing light. There are dozens of affordable options on the market, and they are available in either a digital or an analog (knob type) arrangement. The least expensive type has a knob that you can set to your desired total ignition timing figure (we will presume 34° without vacuum advance). Are the timing marks on the harmonic balancer pulley wheel unsteady? At idle with a timing light on the marks, your timing mark at the harmonic balancer pulley wheel should appear relatively steady. The factory generally considered a mark that stays within about 2° of steady to be perfectly acceptable. A mark that bounces wildly can be caused by fatigued ignition advance springs in the distributor, a loose or worn timing chain, a torn rubber mount inside of the harmonic balancer pulley wheel, or carburetion issues.

With the car parked safely in neutral with the hand brake applied, set the ignition timing by elevating the engine speed to 3,500+ RPM. As you accelerate the engine speed, you will see the ignition timing continue to advance until it finally stops at maximum ignition advance. Record both the engine speed and the ignition timing in your notes so that you can plot the ideal ignition advance curve. At that point, you can adjust the distributor in order to make the pulley mark line up with the Top Dead Center (TDC) mark on the timing chain cover.

For those who use a Lucas 25D4 distributor, there is a small vernier micro-adjuster wheel on the backside of the vacuum advance mechanism with markings just above it that say "A/R". Those markings stand for Advance / Retard so that the vernier micro-adjuster wheel can be used for fine-tuning your ignition timing. Approximately 34 "clicks" of the wheel is one full turn, giving about 2º of ignition advance or ignition retard, dependent upon the direction in which you rotate the adjuster wheel.

If you look on the side of the distributor body, you will find a machined flat surface with some numbers on it. The top number is the Lucas part number, like 40897 (1966 MGB). This part number is the key to finding out which car the distributor is intended to be employed in. The bottom numbers show the month and year in which the distributor was manufactured. In setting up your ignition timing curve, you may find the following reference tables for Original Equipment distributors to be useful as a starting point-

Engine	Distributor Model	Distributor Serial Number	Vacuum Advance Starts / Ends Maximum Advance in Degrees	Contact Breaker Point Gap	Spark Plug Gap w/ Lucas HA12 Ignition Coil	Engine Idles @
18G (Low Compression	25D4	40916	4" Hg / 12" Hg 16º	.015" (.381mm)	.025" (.64mm)	500 RPM

)						
18GA (Low Compression)	25D4	40916	4" Hg / 12" Hg 16°	.015" (.381mm)	.025" (.64mm)	500 RPM
18GB (Low Compression)	25D4	40916	4" Hg / 12" Hg 16°	.015" (.381mm)	.025" (.64mm)	500 RPM
18GD (Low Compression)	25D4	40916	4" Hg / 12" Hg 16°	.015" (.381mm)	.025" (.64mm)	500 RPM
18GG (Low Compression)	25D4	40916	4" Hg / 12" Hg 16°	.015" (.381mm)	.025" (.64mm)	500 RPM
18G (High Compression)	25D4	40897	5" Hg / 13" Hg 20°	.015" (.381mm)	.025" (.64mm)	500 RPM
18GA (High Compression)	25D4	40897	5" Hg / 13" Hg 20°	.015" (.381mm)	.025" (.64mm)	500 RPM
18GB (High	25D4	40897	5" Hg / 13" Hg	.015" (.381mm	.025"	500 RPM

Compression)			20⁰	)	(.64mm)	
18GD (High Compression)	25D4	40897	5" Hg / 13" Hg 20⁰	.015" (.381mm)	.025" (.64mm)	500 RPM
18GF (High Compression)	25D4	40897	5" Hg / 13" Hg 20⁰	.015" (.381mm)	.025" (.64mm)	900 RPM
18GG (High Compression)	25D4	40897	5" Hg / 13" Hg 20⁰	.015" (.381mm)	.025" (.64mm)	500 RPM
18GH (High Compression)	25D4	41264	5" Hg / 11" Hg 16⁰	.015" (.381mm)	.025" (.64mm)	900 RPM
18GJ (High Compression)	25D4	41155	5" Hg / 13" Hg 20⁰	.015" (.381mm)	.025" (.64mm)	900 RPM
18GK (High Compression)	25D4	41339	7" Hg / 13" Hg 10⁰	.015" (.381mm)	.025" (.64mm)	900 RPM
18V 581-F-H 18V 581-Y-H	25D4	41288	5" Hg / 13" Hg	.015" (.381mm	.025"	750 RPM

18V 582-F-H 18V 582-Y-H 18V 583-F-H 18V 583-Y-H			20⁰	)	(.64mm)	to 800 RPM
18V 581-Y-L 18V 582-Y-L	25D4	41290	4" Hg / 12" Hg 16⁰	.015" (.381mm)	.025" (.64mm)	750 RPM to 800 RPM
18V 584-Z-L 18V 585-Z-L	25D4	41370	7" Hg / 13" Hg 6⁰	.015" (.381mm)	.025" (.64mm)	850 RPM
18V 672-Z-L 18V 673-Z-L	25D4	41491	10" Hg / 15"Hg 10⁰	.015" (.381mm)	.025" (.64mm)	850 RPM
18V 779-F-H 18V 780-F-H	25D4	41391	4" Hg / 12" Hg 16⁰	.015" (.381mm)	.025" (.64mm)	850 RPM
18V 779-F-H 18V 780-F-H	25D4	41234	4" Hg / 12" Hg 16⁰	.015" (.381mm)	.025" (.64mm)	850 RPM
18V 797-AE-L 18V 798-AE- L	45DE4	41643	N/A	N/A	None	850 RPM
18V 797-AE-L 18V 798-AE- L	45DM4	41815	N/A	N/A	None	850 RPM

18V 836-Z-L 18V 837-Z-L	45D4	41599	10” Hg / 15”Hg 10°	.015” (.381mm)	.025” (.64mm)	850 RPM
18V 801-AE-L 18V 802-AE-L	45DE4	41600	N/A	N/A	None	850 RPM
18V 846-F-H 18V 847-F-H	45D4	41610	3” Hg / 11” Hg 24°	.015” (.381mm)	.025” (.64mm)	850 RPM
18V 883-AE-L 18V 884-AE-L	45DE4	41693	3” Hg / 11” Hg 24°	N/A	.035” (.9mm)	850 RPM
18V 883-AE-L 18V 884-AE-L 18V 890-AE-L 18V 891-AE-L	45DE4	41813	3” Hg / 11” Hg 24°	N/A	.035” (.9mm)	850 RPM
18V 883-AE-L 18V 884-AE-L 18V 890-AE-L 18V 891-AE-L	45DM4	41814	5” Hg / 11” Hg 14°	N/A	.035” (.9mm)	850 RPM
18V 883-AE-L 18V 884-AE-L 18V 890-AE-L 18V 891-AE-L	45DM4	41851	3” Hg / 11” Hg 24°	N/A	.035” (.9mm)	850 RPM

18V 890-AE-L 18V 891-AE-L	45DE4	41695	5" Hg / 11" Hg 14°	N/A	.035" (.9mm)	850 RPM
18V 892-AE-L 18V 893-AE-L	45D4	41692	N/A 24°	.015" (.381mm)	.025" (.64mm)"	850 RPM

Distributor Serial Number	**Static Ignition Timing Set @**	**Idle Ignition Timing Set @ Engine Speed**	**Centrifugal Ignition timing Reading (Vacuum Advance disconnected)**				
40897	10° BTDC	14° BTDC @ 600 RPM	4° BTDC @ 600 RPM	6° BTDC @ 700 RPM	9° BTDC @ 900 RPM	15° BTDC @ 1,600 RPM	20° BTDC @ 2,200 RPM
40916	8° BTDC	12° BTDC @ 600 RPM	6° BTDC @ 600 RPM	8° BTDC @ 800 RPM	9° BTDC @ 1,000 RPM	18° BTDC @ 3,000 RPM	24° BTDC @ 4,400 RPM
40943	6° BTDC	N/A	0° BTDC @ 400 RPM	4° BTDC @ 550 RPM	8° BTDC @ 700 RPM	12° BTDC @ 850 RPM	16° BTDC @ 1,000 RPM
41032	5° BTDC	15° BTDC @ 1,000 RPM	1.5° BTDC @ 600 RPM	2° BTDC @ 700 RPM	4.5° BTDC @ 900 RPM	12° BTDC @ 1,600 RPM	19° BTDC @ 2,200 RPM

41155	10° BTDC	20° BTDC @ 1,000 RPM	10° BTDC @ 500 RPM	24° BTDC @ 1,625 RPM	30° BTDC @ 3,000 RPM		
41234	6° BTDC	11° BTDC @ 1,000 RPM	.5° BTDC @ 600 RPM	4° BTDC @ 1,200 RPM	12° BTDC @ 2,200 RPM	22° BTDC @ 3,600 RPM	28° BTDC @ 4,500 RPM
41264	N/A	20° BTDC @ 1,000 RPM	0° BTDC @ 700 RPM	3° BTDC @ 1,000 RPM	20° BTDC @ 2,300 RPM	30° BTDC @ 5,200 RPM	
41288	10° BTDC	13° BTDC @ 600 RPM	3° BTDC @ 600 RPM	6.5° BTDC @ 700 RPM	9° BTDC @ 900 RPM	15° BTDC @ 1,600 RPM	20° BTDC @ 2,200 RPM
41290	10° BTDC	13° BTDC @ 600 RPM	3° BTDC @ 600 RPM	8° BTDC @ 800 RPM	9° BTDC @ 1,000 RPM	18° BTDC @ 3,000 RPM	24° BTDC @ 4,000 RPM
41339	10° BTDC	15° BTDC @ 1,500 RPM	10° BTDC @ 1,000 RPM	24° BTDC @ 2,800 RPM	30° BTDC @ 4,600 RPM		
41391	6° BTDC	11° BTDC @ 1,000 RPM	.5° BTDC @ 600 RPM	4° BTDC @ 1,200 RPM	12° BTDC @ 2,200 RPM	22° BTDC @ 3,600 RPM	28° BTDC @ 4,500 RPM
41491	6° BTDC	11° BTDC @ 1,500 RPM	16° BTDC @ 2,025 RPM	32° BTDC @ 3,825 RPM	39° BTDC @ 4,800 RPM		

41599	7° BTDC	13° BTDC @ 1,500 RPM	18° BTDC @ 2,000 RPM	32° BTDC @ 4,000 RPM	36° BTDC @ 5,000 RPM		
41600	7° BTDC	10° BTDC @ 1,500 RPM	15° BTDC @ 2,000 RPM	30° BTDC @ 3,500 RPM	35° BTDC @ 4,500 RPM		
41610	7° BTDC	10° BTDC @ 1,000 RPM	5° BTDC @ 600 RPM	3° BTDC @ 1,600 RPM	8° BTDC @ 2,600 RPM	12° BTDC @ 3,400 RPM	17° BTDC @ 4,400 RPM
41692	7° BTDC	13° BTDC @ 5,000 RPM	18° BTDC @ 2,000 RPM	32° BTDC @ 4,000 RPM	36° BTDC @ 5,000 RPM		
41693	7° BTDC	10° BTDC @ 1,500 RPM	15° BTDC @ 2,000 RPM	30° BTDC @ 3,500 RPM	35° BTDC @ 4,500 RPM		
41695	7° BTDC	10° BTDC @ 1,500 RPM	15° BTDC @ 3,000 RPM	30° BTDC @ 3,500 RPM	35° BTDC @ 4,500 RPM		
41813	N/A	10° BTDC @ 1,500 RPM	15° BTDC @ 2,600 RPM				
41814	N/A	10° BTDC @ 1,500 RPM	15° BTDC @ 2,600				

			RPM
41815	N/A	10^{o} BTDC @ 1,500 RPM	15^{o} BTDC @ 600 RPM
41851	N/A	10^{o} BTDC @ 1,500 RPM	16^{o} BTDC @ 600 RPM
41853	N/A	10^{o} BTDC @ 1,500 RPM	16^{o} BTDC @ 2,600 RPM

Aftermarket Electronic Ignition Systems

Does an aftermarket electronic ignition change the way you need to set the ignition timing? Quite simply, no. The ignition timing will be set using the same procedure no matter if you have a Crane, Pertronix, Lumenition, or Allison ignition installed. Adding an MSD or another external amplifier will not change the procedure either. However, you will have to reset your ignition timing once the system has been installed, since most aftermarket electronic ignitions are not "clocked" in the same positions as the original contact breaker points or original electronics, so upon initial installation an ignition timing adjustment will likely need to be made. Be aware that some electronic ignition modules indicate that as long as the module is powered with a full 12 Volts, in a ballasted ignition system the coil can still be powered via the ballast and hence stay as 6 Volts. Others (oddly enough) say that you must fit a 12 Volt ignition coil (their own, of course) and bypass the ballast, but that can make starting the engine more difficult as the purpose of the 6 Volt ignition coil and ballast is to give a full 12 Volts from the battery to the coil during start up with the purpose of getting a more powerful spark in order to aid starting. Some can be powered from the coil terminal, i.e., the 6 Volt feed.

For this reason the Crane / Allison XR700 distributor conversion is highly recommended as it uses a simple optical trigger and thus eliminates the need for a set of contact breaker points, thus greatly reducing maintenance and permitting the use of the more powerful ballasted Petronix FlameThrower ignition coil (Petronix Part Number H40011) or the Lucas 6 Volt Sport Ignition coil with an external ballast resistor (Lucas Part # DLB 110). It is an electronic Inductive type of system that can switch higher current than a contact breaker points type system, but it does not have current control. As such, it is necessary to run a standard 1.5 Ohm ignition coil coupled with a 1.5 Ohm Ballast Resistor with a sum total of 3.0 Ohms of primary resistance. One of the features of a ballasted system is that a full 12 Volts is applied to the 6 Volt ignition coil while the starter motor is in use, consequently boosting its output and thus counteracting the inevitable reduction in voltage that occurs during starting, even on a car that has both a good battery and clean, sound connections. This ballasted system has the potential to double the current to the spark plug as compared to a contact breaker points type system. Increased current is increased ignition spark heat, and thus it has the potential to make both easier starting and better acceleration possible. This should allow you to open the gap on your spark plugs to at least .038" (.9652mm) and, with the XR700 conversion, have a nice powerful ignition spark of 300 microseconds duration, enough to handle any streetable engine's ignition requirements and make for much easier cold weather starting. Another advantage of this conversion system is that it is produced in versions that can be used on either the Lucas 25D Series (Crane Part # 700-0231) or 45D Series (Crane Part #700-300) distributors.

Installation of the Crane's optical triggering ignition system (Crane Part # 700-0231) is very straightforward-

Attach the mounting arm to the optical trigger with the supplied 4-40 x 3/16" screws. In order to allow for vertical adjustment of optical trigger, do not tighten the screw completely.

Use an existing screw on the breaker plate to install the best-fitting mounting foot. Do not tighten the screw completely as you may need to allow for adjustment.

Select the best fitting shutter with the same number of slots as cylinders. Use a socket to press the shutter down onto the distributor shaft cam until it is firmly seated and level. In order to prevent breaking the shutter, do not press on the slotted rim. Make sure that the flats or springs inside of the shutter line up with the flats on the distributor cam.

Mount the optical trigger and bracket. If clearance is limited, install both the shutter and the optical trigger assembly together. Either side of the mounting arm can face up. Some applications may require attaching the mounting arm directly to the contact breaker points plate, drilling and tapping a new hole in the contact breaker points plate, and cutting or trimming the mounting arm or contact breaker points plate. Attach the mounting arm to the mounting foot with the supplied 6-32 x 3/16" screws. To allow for adjustment, do not tighten the screw completely. Pass the optical trigger cable through the exit hole provided for the wire of the contact breaker points.

Adjust the optical trigger height so that the shutter rim is approximately in the middle of the trigger opening. Check for clearance around all of the parts. Verify that the shutter is level and that it does not rub or touch. If required, readjust it. Reinstall the ignition rotor, and then follow the steps below to install the ignition module:

Mount the XR700 ignition module in any convenient location on the firewall or a fender well. Avoid locations directly exposed to engine or exhaust header heat or where water can splash. Do not mount the XR700 ignition module on the engine. Use the two sheet metal screws provided to mount the XR700 module. Make sure that the wires will reach to the coil and distributor with some slack for engine movement.

Remove all of the original wires from the ignition coil. Connect the original wire that went from the tachometer or from the ignition switch to the XR700 black wire. Connect the XR700 yellow wire and red wire to the ignition coil as shown. Connect the COIL positive (+) terminal to a ground on the chassis.

Insert the three terminals on the end of the cable from the optical trigger assembly into the Molex plug that is supplied with the installation kit. Make sure that the individual wire colors match from side to side. Use a small screwdriver to push the terminals all the way into the Molex plug, and then connect it to the mating Molex plug on the cable from the module. If you must remove a terminal, see Figure 22 and use the provided extraction tool.

If the XR700 runs very hot to the touch after 15 minutes of operation, you may have excessive coil current as a result of having an unballasted ignition system, and as such you must employ a ballast resistor. Use a Chrysler-style two-terminal ballast resistor such as an Echlin ICR23 or a Wells CR107.

A variant of the XR700 system (Crane Part # 700-0309) can also be used in order to eliminate the notoriously short-lived dual contact breaker points in the Mallory distributor as well. Note that if you are using the Crane PS20 or PS40 Ignition coil in conjunction with a ballast resistor, the ignition coil resistance may be too low or too high for certain models of the Smith's tachometer to work properly. However, the Lucas DLB 105 Sports Ignition Coil for straight 12 Volts DC (i.e., with no ballast resistor) should work well.

For those who choose to make use of higher compression ratios, Crane's XR3000 system is the preferred choice (Crane Part # 3000-0231). The XR-3000 system is a true high-energy system. Using the same triggering sensor mechanism as the XR-700 system, but having far more sophisticated and powerful electronics, the XR-3000 can drive a 1.5 Ohm ignition coil directly without the use of a ballast resistor, and has a feedback system in order to adjust both dwell and current as required. When coupled with the low-resistance (1.4 Ohms) Crane PS20 or PS40 ignition coils, an XR-3000 system will reliably produce a strong enough ignition spark down to 6 Volts. This is also an electronic Inductive type system, but, significantly, it has current control. This means that in order to enable faster current charge times for better output characteristics at high engine speeds, you can use an ignition coil without a ballast resistor, such as the 38 Kilovolt Crane PS60, the 38 Kilovolt Crane LX91 (in both cases, breakerless ignition triggering systems only), or the 40 Kilovolt 12 Volt Lucas Sport Ignition Coil (Lucas Part# DLB105), thus permitting it to fire reliably under the conditions inherent with higher compression. However, lacking a ballast resistor, it will be necessary to maintain both the charging system and the battery in good condition in order to assure easy starting.

For those who choose to make use of a camshaft lobe profile that elevates peak power output to higher engine speeds, the Crane HI-6S system is the preferred choice due to it having the considerable advantage of having both a self-diagnosis system and a designed-in engine speed limiter that can be adjusted in 100-RPM increments from 3,100-9,900 RPM. This is also an electronic inductive type system, but has the versatility of being triggered either by contact breaker points or by the output of the Crane 715-0020 optical trigger system (Crane Part # 700-2231 for Lucas distributors, Part # 700-2309 for Mallory distributors), the same optical triggering system that is used in the XR700 and XR3000 systems. Using either the 38 Kilovolt Crane PS60, the 38 Kilovolt Crane LX91, or the 40

Kilovolt Lucas Sport DLB105 unballasted ignition coils, its output characteristics are about the same as that of the XR3000 system.

If you decide to use one of these Crane units, mount the control module under the dashboard in order to keep it away from the heat that accumulates in the engine compartment. Crane has a website that can be found at http://www.cranecams.com/ .

Like the Crane pointless system, the Petronix Ignitor I pointless system for the BMC B Series engine of the MGB is an inductive system, but it employs a completely different approach to triggering the ignition circuit. Instead of using an electro-optical trigger, rotating cobalt magnets mounted upon a trigger ring activate a Hall Effect integrated circuit in order to trigger the ignition circuit. It consists of only two pieces: the trigger ring that fits over the cam under the rotor arm, and a plate that screws to the pre-existing contact breaker plate. On this plate is mounted the pick-up and transistor. The entire system fits self-contained within your existing distributor cap, right where the contact breaker points and condenser used to be. There is no external box to hang on the firewall (bulkhead). It has only two electrical connections to make. One to either side of the coil. The Lucas 25D4 distributors employed on the 1963 thru 1974 engines use the Pertronix Part # LU-142A unit (Moss Motors Part # 222-405) which requires that you retain the grounding wire from the breaker plate to the body of the distributor and that you mount the pickup over the pivot pin of the contact breaker points and anchor it with the old condenser screw. The Lucas 45D4 distributor, which were equipped with contact breaker points, use the Pertronix Part # LU-144 unit (Moss Motors Part # 222-435). This unit requires that you remove the Original Equipment ground (earth) wire from the breaker plate to the body of the distributor, attach the Pertronix unit's own ground (earth) wire to the mounting screw of the contact breaker points, and then use that same screw and its hole to mount the pickup to the contact breaker plate. The Lucas 45DE4 of the 1975-1976 models use the Pertronix Part # LU-141 unit (Moss Motors Part # 222-425) and the Lucas 45DM4 distributors use the Pertronix Part # LU-147 unit (Moss Motors Part # 222-475).

Pertronix's Ignitor II units are available only for the Lucas 45DE4 (Pertronix Part # 9LU-141) and 45DM4 (Pertronix Part # 9LU-147) distributors. They are a more elaborate, Constant Energy Ignition system that adds the capability to sense the coil current level and uses a micro-controller in order to adjust the dwell. This variable dwell feature helps to maintain peak energy throughout the entire range of engine speed. Peak current level is reached just prior to spark for maximum energy, thus greatly reducing disadvantageous heat build-up and increasing both coil and module life. It also adjusts the spark timing at higher engine speeds in order to compensate for inherent electronic delay. It senses startup and then develops more energy in order to enable quicker, easier starting under unfavorable conditions. It also has a designed-in protection against reverse polarity that will sense if the system has been incorrectly wired, as well as sense over-current should the ignition key be left on, and will automatically shut down the system, thus preventing component damage.

Mentioned here only for the purpose of completeness is the Ignitor III system, which adds an engine speed limiter. The engine speed limit can be set by the user and is accurate to within +/- 50 RPM. The system includes a memory safe function for user settings. An LED display provides the user with feedback confirmation when setting the engine speed limit. Unfortunately, Pertronix has no plans to introduce a version of this system for the Lucas distributors at this time.

Note that upon installation, the Pertronix ignition kits are typically 6° to 8° advanced from the Original Equipment contact breaker points position, thus requiring that the ignition timing be reset. Before you remove the distributor from the engine, use a scribe or a nail to make a reference point on the distributor cap, on the distributor body, and on the engine block, of exactly where the firing position for the #1 cylinder is. When you assemble the Pertronix system, make sure that the electrical contact rotor arm is pointing to #1 in the same position, and that firing to the cylinder is occurring at that moment. Be aware that excessive High Tension (HT) voltage can break down any of the High Tension (HT) components. One cause of this problem is that the phasing with an after-market trigger can be significantly different from the original contact breaker points, indicated by having to adjust the timing significantly with the new trigger even if the distributor has not been out of the car. Be warned that in some distributors this can result in the electrical contact rotor arm moving farther away from the contact inside of the distributor cap when the ignition fires, which results in a much larger total gap in the High Tension (HT) circuit, and hence a much higher High Tension (HT) voltage. Eventually the spark will find an easier route through the electrical contact rotor arm or the distributor cap than through the spark plugs! This ruins both the electrical contact rotor arm and the distributor cap, as well as causing weak sparking and misfiring. The centrifugal advance mechanism causes the relationship between the electrical contact rotor arm and the distributor cap to change as the timing changes. This is the reason why the contact area of the electrical contact rotor arm is designed to be as wide as it is and there is an arc of burning and not just a spot, and so even though the relationship may be acceptable at lower engine speeds, it could be too far away at higher engine speeds, or vice-versa. This mistiming of the internal components of the distributor can be rectified by removing and reinstalling the distributor drive shaft into a more synchronous setting. Fortunately, the deeper trench around the central contact and the larger radius between the electrical contacts found in the Lucas 45D Series distributors reduce the probability of tracking and cross-firing when used with higher-output ignition systems.

Be aware that the Lucas 45DE4 and 45DM4 distributors never had their mounting plate, on which the Pertronix unit will be attached, provided with a ground (earth) wire as does the other distributors that originally were installed with contact breaker points. The Lucas 45DE4 and 45DM4 ignition systems did not require that the plate be grounded (earthed) when equipped with their Original Equipment ignition sensor. They do have ground (earth) springs between the moving contact breaker points plate and the fixed base plate, which should mean that a separate ground (earth) wire is not required. In order for the Pertronix ignition system to operate properly, it requires that a secure ground (earth) wire be attached to the mounting plate. Without the ground (earth) wire, the mounting plate will be somewhat attached to ground through the bushing on which it rotates, but, unfortunately, this does not provide an adequate, consistent ground (earth). This inconsistent ground (earth) will cause the needle of the tachometer to swing wildly. Fortunately, the solution is rather easy. Simply attach a flexible wire of approximately 1.5” (38.1mm) in length, with a terminal lug on each end, from one of the screws on the mounting plate to one of the visible screws going into the case of the distributor that attaches the vacuum advance plate assembly to the distributor. A cloth-insulated ground (earth) wire of a Lucas 45D4 distributor should fulfill this purpose quite well. Be sure to leave some slack in the ground (earth) wire so that the vacuum advance mechanism will be able to rotate the contact breaker plate.

Distributor Maintenance

Be aware that installing a pointless conversion kit does not entirely eliminate periodic distributor maintenance. You will still need to periodically replace the distributor cap, the electrical contact rotor arm, lubricate the distributor action shaft and oil the flyweight (rolling weight) cam bearing, as well as oil the pivots for the centrifugal advance counterweights (rolling weights), all on the same required at-a-minimum every 6,000 miles or 6 months maintenance schedule as before. If the system uses an optical triggering device in order to replace the contact breaker points, the light sensor should also be cleaned at the same time with a Q-Tip saturated with alcohol, or, better yet, Kodak lens cleaning fluid. If the system uses a magnetic triggering device, then cleaning with CRC QD Electronic Cleaner will suffice.

Like most tasks associated with the B Series engine of an MGB, this routine distributor maintenance is basically straightforward. Tools required: CRC QD Electronic Cleaner, a large, flat-bladed screwdriver, a very small flat-bladed screwdriver, a posi-drive screwdriver, an 11/16" wrench (in order to hold the flyweight (rolling weight) cam while removing the screw at the bottom of the distributor action shaft), a 3/16" drift and a small hammer. For optional further intensive investigation by the more highly skilled, a micrometer, a dial indicator with a base, and, of course, the all-important Sun distributor tester machine!

First, remove the dipstick and place it out of the way. Remove the distributor cap, and then clean the distributor cap both inside and out with CRC QD Electronic Cleaner. Do not neglect to clean the spring-loaded carbon brush inside of the center of the distributor cap and check to be sure that it moves freely inside of the bore of its boss. This cleanliness is necessary in order to prevent the ignition's electrical current from being lost by being conducted to ground (earth) along any dust or oil film. Inspect the electrical contact rotor arm and the bottoms of the electrodes inside of the distributor cap for signs of arcing, wear, burning, or black carbon buildup. If you see anything suspicious, do not hesitate to replace them (Electrical Contact Rotor arm, Blue Streak Part # LU-300; Distributor cap, Blue Streak Part # LU-421 (Push-In Type) or Blue Streak Part # LU-420 (Horizontal Type). Clean and then check the ignition coil and High Tension (HT) leads (spark plug leads) for any signs of deterioration. Again, if you see anything suspicious, do not hesitate to replace them.

If you are using the Lucas top-entry distributor cap (Lucas Part # DDB106 for Lucas 24D4 distributors, Lucas Part # DDB108 for Lucas 45D4 distributors), then the High

Tension (HT) lead for #1 cylinder should be 15“ (38.1 cm) long, the High Tension (HT) lead for #2 cylinder should be 14“ (35.6 cm) long, the High Tension (HT) lead for #3 cylinder should be 9.5“ (24.2 cm) long, the High Tension (HT) lead for #4 cylinder should be 11.5 “ (29.2 cm) long, and the ignition coil lead (king lead) should be 11.5“ (29.2 cm) long. If you are using the Lucas side-entry distributor cap (Lucas Part # DDB101 for Lucas 24D4-type distributors, Lucas Part # DDB194 for Lucas 45D-type distributors), then the High Tension (HT) lead for #1 cylinder should be 14“ (35.6 cm) long, the High Tension (HT) lead for #2 cylinder should be 11“ (27.9 cm) long, the High Tension (HT) lead for #3 cylinder should be 8.5“ (21.6 cm) long, and the High Tension (HT) lead for #4 cylinder should be 11“ (27.9 cm) long, and the ignition coil lead (king lead) should be 10“ (25.4 cm) long. When reinstalling them into the distributor cap, use a Q-tip in order to smear some dielectric grease inside of the distributor’s mounting bosses for the High Tension (HT) leads (spark plug leads) in order to both ensure maximum electrical conductivity and in order to moisture-proof the terminals, thus preventing corrosion inside of the mounting bosses. Take care to properly seat the boots of their leads onto the terminal bosses.

Disconnect the Low Tension (LT) lead that runs from the ignition coil to the plastic Low Tension (LT) terminal on the body of the distributor. Removing the distributor from the engine by simply loosening the mounting bolts of the distributor mounting plate and withdrawing it from the engine block will make resetting the ignition timing much more difficult after it has been reinstalled into the engine block. Instead, remove the spark plugs and then, noting the timing marks on the crankshaft’s pulley wheel on the harmonic balancer (harmonic damper), rotate the crankshaft until the piston in #1 cylinder is indicated as being at Top Dead Center. After disconnecting the hose (flexible pipe) of the vacuum advance control capsule from the fuel induction system, remove the distributor from the engine by removing the two retaining bolts on the distributor mounting plate that secures the distributor to its drive shaft inside of the engine block. This will make it relatively easy to still have the ignition timing correct enough to start the engine when the distributor is reinstalled. This is the best approach also because frequent loosening and retightening the clamping bolt runs the risk of damaging the groove on the body of the distributor, which in turn can allow it to jump out of the distributor mounting plate while the engine is running.

Remove the vacuum advance hose (flexible pipe) and clean it, then immerse it in a glass of water, leaving its ends outside of the glass. Cover one end of the tube with your fingertip and blow through the other end. If you see bubbles, then the tube has a leak, rendering it useless. Once you have determined that it has no leaks, reconnect the vacuum advance hose (flexible pipe) to the distributor, and then check the function of the vacuum advance control mechanism by sucking on the hose (flexible pipe). You should see the contact breaker plate move when you do this. If it does not, or if you can steadily draw air through the vacuum advance control capsule, then the diaphragm inside of the vacuum advance control capsule is punctured. This will both upset the fuel / air mixture as well as give insufficient ignition advance when cruising, effecting both performance and economy. Obviously, you will need to replace the vacuum advance control capsule. In order to accomplish this, simply unhook the connector of the vacuum advance control capsule from the vacuum control connector post located under the base plate, remove the spring clip of the micro-adjuster, unscrew the vernier adjustment nut from the shaft of the micro-adjuster, and then simply withdraw the vacuum advance control capsule from the body of the distributor. When reconnecting the vacuum hose (flexible pipe) that runs from the vacuum advance control capsule to the source of vacuum, be it on the carburetor or on the intake manifold, be sure to make provision for a small container to serve as a catch tank for any gasoline that may find its way into the hose (flexible pipe). This will help to protect the diaphragm inside of the vacuum advance control capsule. The Original Equipment system had this feature (BMC Part # 12H 733).

As vacuum is applied, the wound wire connector that attaches the vacuum advance control capsule to the contact breaker plate moves the contact breaker plate in a clockwise direction, thus advancing the ignition timing. It is not a spring, even though it appears at first glance to be one. It is not intended to have a spring function and, if the system is clean, lubricated, and in proper working order, does not act as a spring. It is simply a better method of fabricating the connection piece between the contact breaker plate and the vacuum diaphragm. The reason that it is wound during fabrication rather than being produced as a solid rod is so that there can be side-to-side deflection as it operates. Its secondary purpose is to prevent a side load on either the vacuum advance control capsule or the contact breaker plate as the vacuum shaft connector moves linearly, while the contact breaker plate rotates along its arc of travel. In addition, when vacuum advance is applied, the contact breaker plate moves clockwise relative to both the distributor's contact breaker cam and the electrical contact rotor arm, causing the ignition spark to occur at different relative positions of the electrical contact rotor arm and the distributor cap. You can observe the end results of this movement by examining the edge of the electrical contact rotor arm. The arcing will be evident along its range of movement.

When reassembling the vernier adjustment nut and its spring retaining clip onto the micro-adjuster shaft, note that there is a boss on the body of the distributor that is provided with a double-headed arrow marked with an “A” and an “R” on its upper face so that you may easily make minor advance or retard adjustments to the ignition timing. Eleven clicks of the vernier micro-adjuster wheel will equal approximately 1º of movement of the contact breaker plate. However, be wary of this general rule-of-thumb. Over the years, different vernier micro-adjuster wheels were produced and the number of teeth has been changed, so always recheck your adjustments with a timing light. There are Hash marks on the capsule end of the vacuum advance that can be used as a general guideline. Each mark represents approximately 4º. The best way to set up the vacuum advance system is to run the vacuum advance unit as far upwards as it will go on the shaft of the micro-adjuster. Make a mark on the vernier micro-adjuster wheel in reference to some point on the distributor body. Next, while counting the number of turns needed in order to do so, run the vernier micro-adjuster wheel of the micro-adjuster down as far as it will go on the shaft of the vernier micro-adjuster. This will give you the number of turns over the entire length of travel of the vernier micro-adjuster. Then, adjust the wheel of the vernier micro-adjuster so that the vacuum advance control capsule is sitting at one half of the total adjustment range. At that point, the vacuum advance system is centralized and you will have available the maximum adjustment available in each direction. This vacuum advance vernier adjustment is intended for fine-tuning of the basic ignition timing. It does effect the amount of total ignition advance available, which is expressed as [static + centrifugal + vacuum], but it only does that by varying the distance of the vacuum advance control capsule from the body of the distributor, thus altering the position of the wire-connected contact breaker plate in order to alter the starting point of static advance. The amount of additional vacuum advance and where it starts and stops in relation to engine speed is uneffected, likewise the amount of additional centrifugal advance is uneffected as to when it stops and starts in relation to engine speed. Although the ignition advance curve will maintain the same overall profile as previously, it will be accordingly relocated. It is not the primary means of adjusting the position of ignition advance, but rather is a holdover from the days when you might have to slightly advance or retard the ignition timing while traveling. Simply put, rather than forcing the motorist to resort to loosening the securing bolts on the clamping plate of the distributor mounting plate with a wrench and rotating the whole distributor, it is used to conveniently fine-tune the amount of ignition advance while traveling when using a different from standard fuel. Better quality fuels with more consistent octane ratings lead Lucas to eliminate this feature on their later model distributors.

The adjustability of this mechanism, however, may be applied to the quest for finding the ideal ignition curve for your particular engine by making subtle changes to the ignition timing at different engine speeds while road testing the car, and then recording what works best. Likewise, if you have an adjustable timing light, then you can evaluate the ignition advance over a range of engine speeds, chart the ignition curve, and compare the chart that seems to be your best running adjustments to those used by the factory. Once the best ignition curve has been established, the distributor can be modified with different centrifugal advance control springs in order to meet your requirements.

Disassembly of the mysterious internals of the Lucas 25D4 distributor is not difficult if you use a methodical approach. To begin, remove the electrical contact rotor from the distributor action shaft, and then extract the plastic Low Tension (LT) terminal from its slot in the body of the distributor. Next, unscrew the screw that secures the fixed contact breaker points plate to the contact breaker plate beneath it, and then lift off the fixed contact breaker plate, complete with its electronic components. Remove the nut from the mounting post that secures the connector tabs for the capacitor, the Low Tension (LT) lead, and the contact breaker points to the fixed contact breaker plate. Next, remove the upper insulating bushing from the post, and then remove the connector tabs of the capacitor, the Low Tension (LT) circuit, the contact breaker points (if you are using contact breaker points), and the lower insulating bushing. Take care not to misplace the insulating bushings, as their purpose is to prevent the ignition circuit from grounding. Clean them all, along with the fixed contact breaker plate, with CRC QD Electronic Cleaner, then reassemble them in their original order back onto their mounting post.

Be aware that when replacing the contact breaker points on a Lucas 25D4 distributor wherein all of the components are secured by a single nut, it is vital to get the conductor tabs of both the condenser (capacitor) and the ignition coil into the correct position. All of the connections mount between the two insulators in the following order, from the bottom to the top: the baseplate, the insulator (narrow end facing up so that it fits into and locates the spring of the contact breaker points), the spring of the contact breaker points, the conductor tab of the condenser (capacitor), the conductor tab of the ignition coil, then the second insulator (narrow end down so that it will insert into and thus locate the two conductor tabs as well as the spring of the contact breaker points), and then finally the securing nut. If any of the contact breaker points, the spring, the conductor tab of the condenser (capacitor), or the conductor tab of the ignition coil contacts either the baseplate machine bolt or the securing nut, then the ignition current will be grounded out and hence the engine will not run.

The Lucas 45D Series distributors have a much simpler method of location in which the spring of the contact breaker points rests against an insulator that is mounted against a flange on the baseplate, with both the condenser (capacitor) and ignition coil wires connecting to a common conductor tab that slips under a fold at the end of the spring of the contact breaker points. This is a much better system than that employed in the earlier Lucas 25D4 distributor, as its simplicity makes for less likelihood of mislocating the connections. There is, however, a concern with low-quality new contact breaker points not being properly insulated. This can allow the condenser (capacitor) to accidentally contact to ground (earth) on the contact breaker plate. Whenever you install the new contact breaker points and condenser (capacitor), make certain there is ample clearance between the fixed contact breaker points plate and the lower portion of the condenser contact at the contact breaker points. You may bend it slightly to gain more clearance if such is needed.

Now, unscrew the two screws that secure the base plate to the body of the distributor. Note that the two screws that secure the fixed contact breaker plate to the body of the distributor, as well as the condenser (capacitor) securing screw, are not Phillips head screws. They are #1 PoziDriv screws, denoted by the little darts between the slots on the screw heads. However, unless your distributor is like new, most of these screws have been "formed" into Phillips head screws by years of using the wrong tool. Unhook the connector of the vacuum advance control capsule from the vacuum control connector post beneath the base plate, and then simply lift out the entire contact breaker plate assembly. Now, rotate the contact breaker plate counter-clockwise (anti-clockwise) in order to disengage its stud on its bottom from the base plate. Disengage the base plate from the black flat C spring on the contact breaker plate. Clean everything with CRC QD Electronic Cleaner, paying special attention to the two nylon pads on its bottom of the contact breaker plate. Be sure to lubricate them with Mobil 1 synthetic grease so that the rotation of the contact breaker plate will be smooth.

Many people wonder why the factory decided to include the C spring into the design of the plates. Take care to not damage or lose it, as its purpose is to preload the plate mechanism in order to prevent both flotation and rocking of the contact breaker plate at elevated engine speeds, thus allowing for the most accurate ignition timing possible. On the opposing side of the contact breaker plate, the shoulder on the pin that locates the sliding contact breaker plate should engage securely in order to prevent upward movement of the plate and resulting dwell angle variation.

Next, use a small pair of tweezers or needle-nose pliers to gently unhook the centrifugal advance control springs from their posts on both the flyweight (rolling weight) cam assembly and the distributor action shaft, and then set them aside on a clean paper towel. At this point, note the orientation of the ignition advance assembly in relation to the hole punched in the corner of the square advance plate which holds the assembly together. This will be your guide for reassembly. Not noting the location of this hole in relation to the cam position may allow you to assemble the distributor 180° out of synchronization, rendering the engine unable to fire on the correct cylinders. Remove the flyweight (rolling weight) cam assembly retaining screw at the top of the flyweight (rolling weight) cam assembly and then, while holding the flyweights (rolling weights) from underneath with your fingertips, lift the flyweight (rolling weight) cam assembly off of the distributor action shaft. Set the flyweights (rolling weights) aside on a clean paper towel, along with the flyweight (rolling weight) cam assembly retaining screw and the centrifugal advance control springs.

Clean both the distributor action shaft and the flyweight (rolling weight) cam assembly, paying special attention to the mounting pins on the bottom of the flyweight (rolling weight) cam assembly and the top hole of the flyweight (rolling weight) cam assembly. Next, clean the centrifugal advance control springs and the pivot holes of the flyweights (rolling weights) with carburetor cleaner. While dry, check the fit of the flyweights (rolling weights) on their pivot pins. It possible that they will no longer be a proper machined fit. If the holes in the flyweights (rolling weights) are elongated or if you can either see or feel a flat spot on the inside of the pivot pins, then replacement is in order. A diametrical variation of more than .003” (.0762mm) in either the diameter of the pin or the pivot hole of the weight is excessive.

Finally, reassemble the (rolling weight) cam assembly along with its flyweights (rolling weights) and its retaining screw onto the distributor action shaft dry, and then check the endplay (endfloat) of the flyweight (rolling weight) cam assembly on the distributor action shaft. It is often beneficial to reduce the endplay (endfloat) of the (rolling weight) cam assembly to .002 to .005” (.0508mm to .127mm). The brass thrust washer, which bears against the drive dog and the bottom of the aluminum distributor body, will wear over time. The amount of endplay (endfloat) is based upon both the thickness of the brass washer and how far the distributor bushing is sitting up into the distributor body. Moving the distributor action shaft support bushing upwards by a few thousands of an inch will take the endplay (endfloat) out of the system.

Next, reinstall the centrifugal advance control springs onto their posts. Using an eyedropper or a syringe, put a few drops of oil around the flyweight (rolling weight) cam assembly retaining screw and allow the oil to seep in so that the flyweight (rolling weight) cam assembly will move smoothly on the distributor action shaft. The cam should have a small piece of felt beneath its mounting screw in order to retain and allow slow drainage of oil into the cam assembly. If it is missing, thick felt can be obtained from any piano tuning shop. Put a single drop of light oil onto each of the pins of the two flyweights (rolling weights). Taking care to properly align the lug inside of the bore of the electrical contact rotor arm with its slot in the top of the distributor action shaft, reinstall the electrical contact rotor arm. Now, check the functioning of the centrifugal advance control mechanism by twisting the electrical contact rotor arm in its counter-clockwise (anti-clockwise) direction of rotation while holding the drive dog on the bottom of the distributor action shaft. This will operate the centrifugal advance control mechanism and reveal if it moves freely, binds, or if the springs are in a deteriorated condition. A relatively snug operation typically means that it is in good working order. Either binding or a very loose operation for the first few degrees of rotation, or flyweights (rolling weights) that flop freely without any advance rotation, are all signs of weakened springs that should be replaced. When released, the flyweight (rolling weight) cam assembly should smoothly return to its original position.

Using a small drift, drive out the drift pin (parallel pin) that secures the driving dog at the bottom of the distributor action shaft, and then remove the drive dog and its thrust washer. Be sure to closely inspect the O-ring (BMC Part # 27H 6547) as well as the .036" (.9144mm) brass shim washer. Any sign of damage, wear, or deformation immediately qualifies either of these inexpensive items for replacement. Withdraw the distributor action shaft from the body of the distributor and clean it thoroughly.

The primary purpose of the O-ring (BMC Part # 12H 6547) is to maintain the partially sealed state of atmospheric conditions inside of the engine so that the fuel induction system can create a partial vacuum inside of the crankcase. Without this O-ring, the purpose of the restrictor tube in the rocker arm cover would be defeated, the -2 PSI vacuum inside of the engine being decreased by leakage through the engine block at the base of the distributor. Its secondary purpose is to establish the proper alignment of the rotational axis of the distributor action shaft with that of the distributor action shaft, thus both prolonging the lifespan of the distributor action shaft support bushing and minimizing "timing wobble". Be aware that binding of the distributor action shaft against its bushing and consequential excessive wear of the both distributor drive gear and the bushing that supports the distributor action shaft can occur as a consequence of misalignment of the rotational axis of the distributor with that of its drive shaft. This misalignment is due to an incorrect method of tightening the clamping plate of the distributor mounting plate. The clamping bolt should be uppermost and the raised center of the plate should face away from the block. Torquing down the two retaining bolts of the distributor mounting plate first, and then the clamping bolt (or nut, whichever your particular distributor mounting plate may be equipped with) last, can bring about this misalignment of the rotational axis of the distributor action shaft and its drive. The correct method is to leave the two distributor mounting plate retaining bolts only finger-tight, tighten the clamping bolt (or nut) so that the rotational axis of the distributor will be perpendicular to the distributor clamping plate so that when the distributor clamping plate is affixed the engine block its rotational axis will be aligned continuous with that of the distributor action shaft, and then torque the two distributor mounting plate retaining bolts. It is most important that the clamping bolt (or nut) of the distributor mounting plate be accurately torqued to 4.16 Ft-lbs (bolt, nut trapped) or 2.5 Ft-lbs (nut, bolt trapped). If the recommended torque figure is exceeded, then the distributor clamping plate will distort and can fail to prevent the body of the distributor from slipping. It is also quite possible that the die cast body of the distributor will fracture. Fortunately, Jeff Schlemmer of Advanced Distributors can repair this fracture if it occurs, and if not addressed, your distributor can loosen from its mounting plate while driving. The O-ring also has a third, although less significant, purpose: to keep oil mist inside of the engine instead of allowing the pulsating atmospheric pressure within the engine to force oil to ooze out around the base of the distributor and drip onto your garage floor. Oddly, most people seem to illogically believe that this third purpose is its sole purpose.

Once you have the distributor completely disassembled, clean the body of the distributor thoroughly, both inside and out. Although the Service Manual instructs you to merely put a few drops of oil onto the shaft above the bushing in order to provide adequate lubrication, there is a much better way to initially accomplish this task. Stand the cleaned body of the distributor upright in a shallow container and fill it with S.A.E. 30W or S.A.E. 40W oil until the level of the oil is just above the height of the distributor action shaft support bushing. Allow it to sit in the oil for 24 hours so that the sintered bushing will become saturated, just as you would if it was a new, unused bushing. This will more greatly ensure longevity of both the bushing and the distributor action shaft. After completing this process, clean all of the oil from the exterior of the body of the distributor. In order to ensure the most precise ignition timing, I would recommend that this bushing be replaced every 30,000 miles. If you install a replacement distributor action shaft support bushing from Moss Motors, be aware that its length is shorter than that of the Original Equipment Lucas item. This being the case, it will have to be shimmed underneath the ignition advance assembly in order to compensate for its shorter length. It is likely that the bushing will also need to be reamed in order to install the shaft after being pressed into place. Correct length bushings are available from Advanced Distributors, although wear in the nylon thrust washer and the brass washer will likely still require shimming of the shaft.

Reassemble the distributor, making sure that all of the moving parts are properly lubricated. Mobil 1 synthetic grease should be liberally applied to the distributor action shaft in order to ensure minimum friction. Note that the distributor action shaft has both a Major Diameter and a Minor Diameter. This means that the ends are of a greater diameter and the central portion of the shaft is of a lesser diameter. Get as much grease onto the Minor Diameter as you can. Take care that you use anti-seize compound on all of the threads in order to both prevent electrolytic corrosion and to ease future disassembly when you again perform this routine maintenance task.

Installing the Distributor

Be aware that both the driving dog of the distributor and the slots of the distributor drive shaft are offset in order to guarantee proper positioning. Rotate the driving dog on the bottom end of the distributor action shaft until it is properly aligned with the slots in the distributor drive shaft. The hole for the cross pin holding the driving dog to the distributor shaft is also offset slightly so that it can be installed in only one position on the distributor shaft. When attaching the distributor into its clampng plate, be sure to smear some antisieze compound into its mounting groove in order to protect it from electrolytic corrosion. Reinstall the distributor and leave the distributor mounting plate retaining bolts only finger-tight so that the ignition timing can be reset. Do not attempt to set the ignition timing with these two machine bolts loose, as the end thrust of the spiral driving gears will force the distributor outward against the play of the loose retaining bolts and thus alter the ignition timing, resulting in a false setting.

Once the distributor is installed onto the engine block, the first thing that you need to do is get the piston in #1 cylinder up to the Top Dead Center (TDC) position. Of course, I do not know what kind of timing marks you have on your harmonic damper pulley wheel, so I will cover all of the possibilities: Remove the rocker arm cover, and then rotate the engine clockwise (as looking at it from the front), and look for the intake valve on #1 cylinder (second valve from the front) to begin to open. Keep rotating the engine until it starts to close, and then start watching the Top Dead Center (TDC) timing mark on the harmonic damper pulley wheel. Do not go all the way to the Top Dead Center (TDC) timing mark. If you have degrees marked on your harmonic damper pulley wheel, bring it up to the 10° BTDC (Before Top Dead Center) timing mark. That would be the mark that is directly to the right of the Top Dead Center (TDC) timing mark as you are looking from the front. If you do not have degrees marked, but have two marks, align the first mark (the one on the right) with the pointer. On the other hand, if you only have one mark, stop the rotation about 3/8" to ½" (9.5mm to 12.7mm) before the Top Dead Center (TDC) timing mark. This will put you at about the position of the 10° Before Top Dead Center (BTDC) timing mark. You will soon see why you are doing this. Make sure that both of the rocker arms for #1 cylinder wiggle just a little to indicate that they are closed.

Now, remove all of the High Tension (HT) leads (spark plug leads) from the distributor cap, and the ignition coil wire as well. Remove the distributor cap. Loosen the distributor mounting plate and rotate the distributor into position so that it looks as though the vacuum advance control mechanism is pretty much horizontal. Attach a test light or Voltmeter to the ignition coil. Do this on the negative (-) side; the wire there should go to the distributor. If using a Voltmeter, then set it at DC Voltage, and clip the red positive (+) lead from the Voltmeter there. Hook the other end to the negative (-) post on the battery. This goes for the test light or the Voltmeter. Now turn the ignition on, but do not crank the engine over, otherwise you will have to start all over again.

If the test light is on, or the Voltmeter registers, turn the whole distributor counterclockwise until the test light goes off, and then turn it back until it just lights. Tighten down the distributor clamping plate. On the other hand, if the test light is off when you turn on the ignition, turn the distributor clockwise just until the test light goes on or the meter registers, then tighten down the distributor clamping plate. You have just set the ignition timing statically. You can now install the rocker arm cover.

Look to see where the electrical contact rotor arm is pointing, and see which tower of the distributor cap it aligns with. That is cylinder #1. Plug the High Tension (HT) lead (spark plug lead) into that tower on the distributor cap, and then hook it up to the spark plug for #1 cylinder. Continue around the cap in a counterclockwise manner with the rest of the High Tension (HT) leads (spark plug leads). The firing order is 1-3-4-2. Now, put the ignition coil lead (king lead) back on.

Now, start the engine. Cranking the engine for a long time in order to get the car started with mild exhaust or carburetor backfires should tell you that the ignition timing is either much too far retarded or much too far advanced. Once this happens, it is likely that your spark plugs are already fouled or starting to foul, so the less time you spend cranking the engine, the better chance you have to get started without having to change the spark plugs. Keep in mind that any time you attempt to start a car with a freshly installed distributor, it is possible that the ignition timing will not be as you expect. A loud backfire through the exhaust is a clear indicator that the distributor is 180^{o} out of synchronization. You can quickly rotate the plug wires 180^{o} in order to alleviate this or deal with the mechanical issues that caused it. This could be a simple matter of the driving dog on the distributor action shaft being installed 180^o out of phase, or an improperly installed distributor drive shaft being installed 180^o out of phase in the engine.

You can make a very simple Dwell meter by simply connecting an analogue Voltmeter between the negative (-) terminal of the ignition coil or CB terminal and ground (earth), and then switching it to its 12 Volt scale. As the contact breaker points open and close on a running engine the Voltage will switch between 0 Volts and 12 Volts. Because the meter needle cannot respond rapidly, it should settle at an average of the two outputs, which is governed by how long the contact breaker points are closed and how long they are open. The longer they are open, the higher the Voltage will be. For example, the dwell angle for a Lucas 45D4 distributor is 51^o +/- 5^o, call it 50^o. This is relative to 90^o of distributor rotation (being a 4-cylinder engine), and 50^o of that means the contact breaker points are closed for 50/90ths and open for 40/90ths, or 56% and 44% respectively. At 12 Volt system Voltage, being open 44% of the time would cause the meter to show an average of 5.3 Volt Dwell Voltage. However, it all goes wrong beyond this point because the system Voltage can vary anywhere from 12.5 Volts to 14.7 Volts, and you would need to know the instantaneous System Voltage as well as the dwell Voltage. A true dwell meter is designed to ignore Voltage changes.

However, you can use the Voltmeter method in order to indicate whether a varying period of dwell is the result of either the dwell meter or the ignition. This can be achieved by either disconnecting the alternator so that there is a relatively constant system Voltage (you should measure it), or by leaving it connected and only measuring off-idle when the alternator is giving a relatively constant output across a range of engine speed.

You can also use an analogue Ohmmeter to indicate dwell, as it switches between full-scale deflection when the contact is closed and zero deflection when it is open. If you have a 100 Volt or 100 point scale on the meter, then you can read the percentage or time that the contact breaker points are closed directly off of that. However, in the case of an ignition distributor, there will need to be some other device to drive it, such as a variable speed drill. This is due to the fact that because when the contact breaker points are running the engine, the variations of both the Voltage and the current will render the readings from an Ohmmeter unusable.

There are always those who refuse to put their faith in the newer technologies of solid-state breakerless ignition systems. They would much prefer to continue to rely upon a contact breaker point system, despite the Original Equipment system's vulnerability to inconsistent dwell angle, contact breaker points bounce at high engine speeds, and ignition timing scatter of as much as 4º, all as a result of wear of the bushing that supports the distributor action shaft. In extreme cases, the effect of ignition timing scatter can be a reduction of power output of as much as 20%. For those hardy individuals, there is a modification that will eliminate this vulnerability by adapting the distributor to make use of a caged roller bearing in order to support its action shaft (Torrington or Ena Part # NA4901RS). The upper diameter of the distributor action shaft will need to be reduced to .0479" (1.21666mm) by centerless grinding. In addition, a spacer of .070" (1.778mm) thickness with an Outside Diameter (O.D.) of .625" (15.875mm) and an Internal Diameter of .048" (1.2192mm) will need to be fabricated. The spacer should be loctited into its bore and the bearing Loctited onto the distributor action shaft. The nylon thrust bearing and its shim will need to be overbored in order to fit over the inner race of the roller bearing. Finally, the bearing bore of the distributor will need to be lathe-bored twice, first to a depth of .0512" (1.30048mm) and a diameter of .9447" (23.9954mm), then lastly to a depth of .542" (13.7678mm) and a diameter of .635" (16.129mm). The mounting boss at the base of the distributor casting should be drilled and tapped for a lubrication zerk for the bearing. One particular side advantage of this modification is that with the diameter of the distributor action shaft reduced, the distributor action shaft flexes just enough to reduce the stresses upon itself, prolonging bearing life and maintaining greater consistency in dwell angle.

Contrary to popular opinion, the advent of engine speed limiting did not come with the arrival of solid-state electronic systems. The Lucas Company developed and marketed a special engine speed limiting electrical contact rotor arm for use in its Model 23 and Model 25 distributors. Although no longer in production, it is still occasionally to be found amongst the racing set. It is Lucas Part # DRB108 54424982. It is just the thing for those who have equipped their engines with hot camshafts and persist on continuing with a breaker point system.

The Ignition Coil

The amount of power obtainable from any engine is limited to the total caloric value of the fuel / air charge. Once combustion starts efficiently at the optimum moment, the rest is just a matter of the mechanical efficiency of the engine design. Anything more than that will not (and cannot) produce any more power. However, it is theoretically possible that a super-duper megaspark could induce detonation instead of combustion! Of course, a larger and / or denser fuel / air charge requires a stronger ignition spark in order to properly ignite it, and this requirement intensifies along with increases in the compression ratio, so a more powerful ignition coil becomes desirable. 35 to 40 Kilovolts should be fine up to a Geometric Compression Ratio (GCR) of 10.5:1, even when being called upon to start the engine in cold weather. Ideally, this should be paired with its manufacturer's recommended waterproof High Tension (HT) leads (spark plug leads) that will prevent any electromagnetic interference with your stereo system.

Be aware that installing an unballasted ignition coil that produces more than the 20 Kilovolt output of the Original Equipment Lucas HA 12 ignition coil will result in accelerated erosion of the ignition contact breaker points, thus making the pursuit of an uprated ignition system an exercise in frustration unless an alternative breakerless system is installed.

It must be understood that an ignition coil will produce only enough power to create a current that will jump a given spark plug gap. The voltage rise at the output of the coil secondary, although rapid, is not instantaneous. As the voltage rises from zero, as soon as it reaches the value high enough to jump the plug gap, it will. If 20 Kilovolt is required to jump a .024" (.6096mm) spark plug gap, as in the case of the Original Equipment Lucas HA12 ignition coil, a 40 Kilovolt ignition coil will jump the same .024" (.6096mm) spark plug gap as soon as its charge reaches 20 Kilovolt, thus never making use of its higher output potential. A slightly wider gap causes the voltage within the secondary windings of the ignition coil to build up higher before it reaches the point at which the buildup is discharged across the spark plug terminals. Thus, the spark plug gap will have to be widened to the point that the entire capacity of the more powerful ignition coil will be necessary to produce an ignition spark in order to gain any benefit. Within limits, this is obviously a good thing. However, an excessive build-up of the voltage, such as that caused by an inappropriately-wide spark plug gap, can also become a bad thing because it can put more strain on the electrical contact of the electrical contact rotor arm, the electrical contacts of the distributor cap, as well as the ignition coil / High Tension (HT) leads (spark plug leads) than a lower voltage build-up would impart. Consequently, component life may be somewhat shorter than with a lower operating voltage unless the components are upgraded. As a general rule of thumb, the spark plug gap should be decreased by a couple of thousandths of an inch (.002" / .051mm) for every whole ratio that compression ratio is increased beyond a ratio of 10:1.

One of the desirable features of a ballasted system is that a full twelve Volts is applied to the 6 Volt ignition coil when the starter motor is in use, thus boosting its output and so counteracting the inevitable reduction in voltage that occurs during starting, even in a car that has both a good battery and clean, sound connections. A ballasted ignition coil is designed to produce its output with an input of only 6 to 9 Volts, thus making for good reliability under poor conditions. In a ballasted ignition coil system, the starter relay bypasses the ballast resistor while the starter motor is operating, applying twelve Volts to the ignition coil, thus compensating for the reduced battery voltage. Because a ballasted ignition coil is designed to provide its full output with a reduced voltage, the application of the full twelve Volts produces an increased output, assisting in the initiation of combustion inside of the combustion chambers of a cold engine.

One way to determine if your ignition system uses a ballast is to find out if your car has two white wires that have a light green tracer attached to the positive (+) terminal of the coil. If you do, then your ignition system is ballasted, so you should use a 3.0 Ohm ignition coil. Non-ballasted ignition systems normally will have a plain white wire attached to the positive (+) terminal of the coil. The ballasted ignition systems should use a 1.5 Ohm ignition coil such as the Original Equipment 20 Kilovolt Lucas 16C6 ballasted ignition coil (BMC Part #s GCL 110 and GCL 111) which were fitted as standard on cars from 1975 on. This ignition coil should show 1.43 to 1.58 ohms resistance across the low tension terminals (the ones having the white / light green wire going to the positive terminal and the wire going to the distributor. On a pre-1975 model MGB, the ballast resistor is built into the actual lead from the ignition. Without the ballast resistor, the ignition coil will receive twelve Volts, and will draw too much current, burning the contact breaker points. The starter solenoid also has a fourth terminal that connects to the ignition coil live wire, and when the starter is turning, it supplies twelve Volts to the ignition coil, thus giving a better ignition spark for starting. After the starter motor ceases its operation, the starter relay then circuits the power through the resistance wire, which in turn reduces the voltage to the ignition coil. At this point, the output of the ignition coil is the same as that of an unballasted ignition coil of the same output capacity. The purpose for the ballasted-type ignition coil being powered with less than twelve Volts is to prevent overheating damage to the ignition coil and erosion of the ignition contact breaker points.

Lucas Sports Ignition coils are available for ignition systems both with and without the ballast resistor. The original Lucas Sports Ignition coil (Lucas Part # DLB 105) is the 12 Volt sports ignition coil for ignition systems without a ballast resistor. The alternative Lucas Sports Ignition coil (Lucas Part # DLB 110) is for ignition systems with an external ballast resistor. These Sport ignition coils have lower resistance, typically 2.5 ohms for a 12 Volt ignition coil as opposed to 3 ohms for a standard 12 Volt ignition, thus giving about 17% more current through the contact breaker points. Because the High Tension spark energy is increased in sport ignition coils, arcing across the contact breaker points also increases. Most Sports Ignition coils are usually 6 Volts and have a resistor from the ignition live wire to the ignition coil live wire (often called a "ballast" resistor).

The only way you can be sure of whether you have a direct (12 Volt) or a ballasted (6 Volt) feed to the coil is to reconnect the white or white / light-green wires to the positive (+) terminal of the coil, connect a ground (earth) to the negative (-) terminal of the coil, turn on the ignition, and then measure the voltage at the positive (+) terminal of the coil. If you measure battery voltage, i.e., approximately 12 Volts, then there is no ballast in circuit, in which case you must fit a 12 Volt ignition coil. If you measure less than battery voltage, then you probably have a ballast in series, in which case you must fit a 6 Volt ignition coil. If you measure approximately 9 Volts, then it would appear that you have ballast in circuit, but with a 12 Volt ignition coil, which is incorrect. Such a combination will give reduced spark. However, if you have a 12 Volt ignition coil in series with *two* lots of ballast, then you will also measure 6 Volts, but this combination will reduce the High Tension (HT) voltage by approximately 75%(!), which is why you have to measure both the coil resistance and the voltage in order to make sure they are compatible.

The Pertronix Flame Thrower Coil is a high performance, 40,000 Volt unit available in both 1.5 Ohm for non-ballasted systems and 3 Ohm for ballasted systems in oil-filled canisters that keep their low resistance windings secure and cool for reliable long-term performance, with chrome or black finishes, or in epoxy-filled for extra protection from vibration with a black finish only. Be aware that this coil should be used with the original Ignitor system only. With a primary resistance of 1.5 Ohms, it has a 100:1 turns ratio, a secondary resistance of 10,500 Ohms, an inductance of 6.5 mH, a peak current of 7.2 Amps, and a spark duration of 1.5 mS. With a primary resistance of 3.0 Ohms, it has a 75:1 turns ratio, a secondary resistance of 8,500 Ohms, an inductance of 11.6 mH, a peak current of 5 Amps, and a spark duration of 1.5 mS.

The Pertronix Flame Thrower II Coil features a high output of 45,000 Volts. It is available in both 1.5 Ohm for unballasted systems and 3 Ohm for ballasted systems in oil-filled chrome or black finishes, or in epoxy-filled with a black finish only. It has a 100:1 turns ratio, a primary resistance of .6 Ohms, a secondary resistance of 10,000 Ohms, an inductance of 7.2 mH, a peak current of 7.2 Amps, and a spark duration of 1.5 mS.

An unballasted ignition coil is designed to produce its output with an input of 12 Volts. Power is applied to the ignition coil directly from the ignition switch to the resistance wire, and then from there to the ignition coil. When the starter relay operates, power from the battery is routed through directly to the ignition coil. This shorts out the resistor wire by placing 12 Volts onto both ends of it. Because equalized voltage exists at both ends, the current flow bypasses it, placing the 12 Volts into the ignition coil. It should be noted that while unballasted ignition coils have the virtue of rapid charging times, they also release their stored energy faster, producing an ignition spark of shorter duration. For combustion to be triggered efficiently within the combustion chamber, the pressure wave produced by the squish (quench) of the fuel / air charge between the piston and the edges of the combustion chamber must arrive in the vicinity of the spark plug before or as the spark plug fires. Ballasted ignition coils produce a longer-duration ignition spark, so this is rarely a problem and the engine is less sensitive to small errors of ignition timing. However, due to the shorter-duration ignition spark produced by an unballasted ignition coil, the accuracy of the ignition timing is of more critical importance.

One advantageous factor of the 6 Volt ignition coil is that, having half the number of primary windings as a 12 Volt ignition coil, it has half the inductance of a 12 Volt ignition coil, but the condenser value is the same for both. The lower inductance of the 6 Volt coil means that it recharges more quickly when the contact breaker points close again (inductance in a component has the effect of causing the current to build more slowly than in a pure resistance), so it can be used at higher engine speeds without a loss of High Tension output.

Determining just which system any particular car has is essential. The Chrome Bumper model MGBs do not have a ballasted ignition system and need a 12 Volt ignition coil which measures a primary resistance of approximately 3.0 Volts. There should be a single white wire on the positive (+) terminal of the ignition coil of these cars. The Rubber Bumper model MGBs have a ballast inside of the wiring harness and need a 6 Volt ignition coil which has a primary resistance that measures approximately 1.5 Ohms. There should be two white / light-green wires on the positive (+) terminal of the ignition coil on these cars. The difference between 12 Volt and 6 Volt systems lies in where the Pertronix triggering unit draws its operating supply from. In the case of a 12 Volt system, this would be the positive (+) terminal of the coil, but on a 6 Volt system this point is continually switching between 12 Volt and 6 Volt as the Pertronix switch opens and closes, which is why it must obtain a "clean" 12 Volt supply via the white wire from the fusebox in order to function reliably. The problem is that some cars have been molested by D.P.O.s (Dumb Previous Owners). Consequently, you may really have no idea of what you actually do have, especially if an electronic ignition unit has been fitted. Get the wrong combination of ignition coil and ballast / no ballast, and you will have either an overheating ignition coil that burns your contact breaker points, as in the case of a 6 Volt ignition coil without a ballast in circuit, or a weak spark and difficult starting, as in the case of a 12 Volt ignition coil with a ballast. Hence you must measure the primary resistance of any coil that you employ. You must also determine whether or not you have ballasted wiring. However, there is a simple test that you can perform in order to divine what type of system it is that you have.

Both ballasted and non-ballasted ignition system will show 12 Volts at the positive (+) terminal of the ignition coil until ignition coil is drawing current and the contact breaker points or the trigger are closed. Only at that time will ballasted systems see a voltage drop at the positive (+) terminal of the ignition coil, while unballasted systems will still show 12 Volts. This is very difficult to measure with some electronic systems if they are designed only to draw current through the ignition coil when the trigger is being fired by rotation of the distributor. Testing a contact breaker points system is just a matter of turning the engine until they are closed, or simply tapping a known good ground (earth) onto the negative (-) terminal of the ignition coil while you measure the voltage on the positive (+)

terminal of the ignition coil. For an electronic system it is probably be best to remove all the electronics connections from the ignition coil, just leave the ignition feed from the harness to the positive (+) terminal of the ignition coil, and then connect the known good ground (earth) to the negative (-) terminal of the ignition coil as before. You need to know what your ignition coil primary resistance is first. Be aware that in order to prevent any sneak (parallel) paths from changing the reading of the primary resistance of the component under test, one must always disconnect at least one wire from a component when taking resistance measurements. Only if the primary resistance drops to either 6 Volts with a 1.5 Ohm ignition coil, or stays at 12 Volts with a 3 Ohm ignition coil, do you have the correct combination of ignition coil and ballast / non-ballast. The one other thing to be wary of is that some aftermarket electronic systems utilize a brief high-voltage pulse to the ignition coil from the electronics, and these ignition coils can be of a very low resistance, perhaps only a mere .1 Ohms, and these should never be connected direct to 12 Volts and ground (earth).

In order to change from the Original Equipment ballasted ignition coil system to an unballasted one, simply run a wire from the fuse box directly to the positive (+) terminal of the ignition coil. Removal of the resistance wire is unnecessary, as it will then be bypassed. The bypass wire running from the terminal of the starter relay to the ignition coil should then be relocated from the terminal of the starter relay to a fuse box terminal. Note that the ignition coil should always be mounted with its lead on the bottom. There is an air bubble inside of the coil, for expansion. If the oil covers the internal coil connections, there is less chance of internal arcing from high resistance, such as bad spark plug wires. This internal oil insulation is a sort of third order insurance policy, after basic design and voltage limiting internal spark gaps.

The Condenser (Capacitor)

The condenser, also called a “capacitor”, functions much like an electrical spring. It quenches the electrical spark in the way that an air spring stops water from hammering inside of plumbing. The condenser (capacitor) is a compromise device. It effectively keeps the current conduction going for a short period when the contact breaker points open and thus lessens arcing and erosion. If it is of inadequate capacitance, then the contact breaker points will have a very short life. If it is too large, then the collapse of the magnetic field of the ignition coil becomes too slow, and the voltage output is reduced. Electronic ignitions do not require a condenser (capacitor because, having no contact breaker points, they can switch without arcing and have a sufficient, yet small amount of inherent capacitance in order to allow the ignition system to oscillate more rapidly, thus resulting in a higher voltage output than can be produced by a system with a condenser (capacitor). A condenser (capacitor) is connected across the contact breaker points when they are open. Obviously, this component is vital to the ignition system. Despite what many people think, its main function is not to protect the contact breaker points from burning, although it does do this as a secondary function. Its primary function is to cause the coil to generate an intense spark. Because the coil is a transformer it can only generate voltage in its High Tension (HT) circuit (and hence a spark) when the current through the primary circuit is changing, not when it is steady. The faster the current change and the greater the voltage swing in the primary circuit, the higher the output voltage generated. When the contact breaker points are opened, instead of the current immediately ceasing to flow through the coil, it continues momentarily while it charges the condenser (capacitor) with the voltage spike that would otherwise arc across the contact breaker points. It is only when the condenser (capacitor) is charged that the current ceases to flow. Furthermore, the condenser (capacitor) and coil, when the contact breaker points open, are interconnected in such a way as to form a tuned L/C circuit (L = inductor or coil, C = capacitor or condenser) which causes the current in the coil's primary circuit to oscillate rapidly at about 15 thousand times per second with a peak-to-peak voltage swing of about 400 volts. The effect of this is to generate an output pulse, and hence a spark, of about 20 kilovolts that lasts for about ½,000 of a second (2 milliseconds, or 2mS). Not very long, you might think, but at 3,600 RPM, any one cylinder is firing 30 times a second, i.e., every 33mS, so at that engine speed the spark lasts for 22° of distributor rotation, which is 44° of crankshaft rotation! By comparison, the spark duration without a condenser (capacitor) is only about 0.2mS, i.e., one-tenth as long.

A condenser (capacitor) is connected across the contact breaker points when they are open. Obviously, this component is vital to a contact breaker ignition system. Despite what many people think, its main function is not to protect the contact breaker points from burning and prevent coil insulation breakdown by limiting the rate of voltage rise at the contact breaker points, although it does do this as a secondary function. Its primary function is to provide for a rapid decay of the primary coil current, thus causing the ignition coil to generate an intense ignition spark. The condenser (capacitor) also "third-harmonic" tunes the coil, raising the peak output voltage and increasing the secondary voltage rise time. This increases the efficiency and the amount of energy transferred to the spark plugs. If the coil secondary voltage rises too quickly, excessive high frequency energy is then produced. This energy is then lost into the air-waves by electro-magnetic radiation from the ignition wiring instead of going to the spark plugs. Voltage rise time should be more than 10 microseconds; a 50-microsecond rise time is acceptable. Conventional systems have a typical rise time of about 100 microseconds. Because the ignition coil is a transformer, it can only generate voltage in its High Tension (HT) circuit (and hence an ignition spark) when the current through the primary circuit is changing, not when it is steady. The faster the current change and the greater the voltage swing in the primary circuit, the higher the output voltage that is generated. When the contact breaker points are opened, instead of the current immediately ceasing to flow through the ignition coil, it continues momentarily while it charges the condenser (capacitor) with the voltage spike that would otherwise arc across the contact breaker points. It is only when the condenser (capacitor) is fully charged that the current ceases to flow. Furthermore, the condenser (capacitor) and ignition coil, when the contact breaker points open, are interconnected in such a way as to form a tuned L/C circuit (L = inductor or ignition coil, C = condenser or capacitor) which causes the current in the ignition coil's primary circuit to oscillate rapidly at about 15 thousand times per second with a peak-to-peak voltage swing of about 400 Volts. The effect of this is to generate an output pulse, and hence an ignition spark that lasts for about 2 milliseconds, or 2mS). Not very long, you might think, but at 3,600 RPM, any one cylinder is firing 30 times a second, i.e., every 33mS, so at that engine speed the ignition spark lasts for 22° of distributor rotation, which is 44° of crankshaft rotation! By comparison, the ignition spark duration without a condenser (capacitor) is only about 0.2mS, i.e., one-tenth as long.

The secondary purpose of the condenser (capacitor) is to cause the ignition spark to occur at the correct time. With the condenser (capacitor) in circuit, the high-frequency oscillation that occurs immediately when the contact breaker points open means that the output voltage starts just .02ms (20 millionths of a second) after the contact breaker points open. Even at 5,500 RPM, the effect of this delay is less than 1° of crankshaft rotation. This high-frequency oscillation also protects the contact breaker points from arcing because the voltage spike that occurs when the contact breaker points open swiftly decays to zero (as part of its first cycle of high-frequency operation) in about .02ms, and this prevents the contact breaker points from arcing. Without the condenser (capacitor), the ignition spark would cease only when either the voltage dropped sufficiently, or when the contact breaker points opened sufficiently. This would take about 2mS. During this period the contact breaker points would be arcing, which, as well as eroding them, causes spikes and pits. This would signify that some current would still be flowing through the ignition coil during the arcing. This would delay the main collapse of the flux, delaying the output voltage pulse and therefore the ignition spark in the combustion chamber. This delay would vary little with engine speed. This 2mS delay effectively would retard the ignition spark during cranking speed by about 1 crankshaft degree, that is, not very much. However, this delay would increase to about 24 crankshaft degrees at 1,000 RPM, 48 crankshaft degrees at 2,000 RPM, etc., meaning that as well as not only having a very short duration, its ignition timing would also become increasingly retarded even at quite low engine speeds. In other words, if the gap of the contact breaker points is too small, then the contact breaker points would be arcing when they start to open. It would not be not until the arcing ceased that the ignition spark would actually occur. Thus, even though there is a gap when measured statically, too small a contact breaker points gap could mean that you will not get any ignition spark at all.

The Original Equipment Lucas condenser (capacitor) has a value of about 0.18 uF to 0.24 uF. This value is critical for a good High Tension (HT) ignition spark. While experimentally varying the value by quite small amounts will show little variation in either the Low Tension (LT) or High Tension (HT) voltage waveforms or in the visual image of the ignition spark, there is a definite reduction in the strength of the audible “crack” heard at the spark plug. You can test for the symptoms of a weak or failed (open-circuit) condenser (capacitor) by performing this simple test: Remove the distributor cap, remove the ignition coil lead (king lead) from the center of the distributor cap and tape the ignition coil lead (king lead) to a length of wood so that you will have an insulated handle on it. Switch on the ignition, flick the contact breaker points open and closed by hand, and see just how far the ignition spark will jump from the end of the king lead to the engine block. The ignition spark should be able to arc across a gap of at least ¼” (.250” / .635mm), and maybe as much as ½” (.50” / 12.7mm) even with a non-sport ignition coil and make a good “crack” sound. This will show the expected effect of having a condenser (capacitor) in circuit. Now, close the contact breaker points and interrupt the contact breaker points lead somewhere else, such as on the terminal of the ignition coil to show the effect of not having a condenser (capacitor) connected across the break in the circuit. You should find that, as well as much arcing at the terminal of the ignition coil, the spark at the king lead will barely jump a normal plug gap, let alone ¼" or ½". You will also get a very "thin" spark, and it will make very little noise. This is why a bad (i.e., open-circuit) condenser (capacitor) causes poor or non-running as well as burned contact breaker points. Note that a short-circuited condenser (capacitor) will prevent the engine from running at all, as it effectively shorts out the contact breaker points and prevents any ignition spark from being generated.

Distributor Caps

Regardless of what type of ignition triggering system you choose, you will need to decide between the two basic types of distributor cap: first is the original side-entry type, of which there are two versions: the now obsolete DA2 which was often accompanied by a rubber cover, and its successor, the more moisture-proof DA6. The original side-entry type (Lucas Part # DDB101 (DA6) for Lucas 25D-type distributors that were used on the 18G, 18GA, and 18GB engines, Lucas Part # DDB194 for Lucas 45D-type distributors) offers a much neater appearance, allowing more convenient routing of the High Tension (HT) leads, (spark plug leads), plus it also has long-lasting copper alloy terminals. Unfortunately, it can only accept High Tension (HT) leads (spark plug leads) of the 7mm type, thus precluding the installation of most high performance High Tension (HT) leads (spark plug leads) that make use of exotic electroconductive core materials. It also has the disadvantage of requiring that the orifices for the leads be packed with dielectric grease in order to prevent short-circuiting should the heater valve develop a leak. A top-entry type distributor cap (Lucas Part # DC6 for Lucas 25D-type distributors, Lucas Part # GDC136 for Lucas 45D-type distributors) usually has light alloy terminals that have the disadvantage of a shorter service life, but will accept modern 8mm high performance High Tension (HT) leads (spark plug leads). Because the mounting bosses for the High Tension (HT) leads (spark plug leads) can be covered with modern insulating boots in order to keep moisture out, reliability in this area of the distributor is markedly superior to that of the earlier side-entry design.

High Tension (HT) Leads (Spark Plug Leads)

Magneto suppressive type High Tension (HT) leads (Spark plug leads) have a tensile core like fiberglass, usually carbon string or silicone, with a small-diameter wire, usually of nickel alloy, that is coiled around it from end to end. The electromagnetic field generated by current moving along a conductor generates and radiates Radio Frequency Interference, which interferes with communications equipment. This resistance wire is an inexpensive way to reduce this interference. In the magneto suppressive type, the electromagnetic fields around the wound wire interfere with each other, stopping the Radio Frequency Interference, without introducing as much resistance as found in the resistor types. Since they have actual wire, and are in effect a very long spring, they are more robust than the resistor types. Many high-end manufacturers produce them, usually marketing them as their “best”.

Yet another item is currently being manufactured and marketed as being in the “best” category. Pertronix has taken their 8 mm High Tension (HT) lead (Spark plug lead), which is designed to meet the demands of high performance engines, and now offer it with exactly the same features in 7 mm flat black, making it possible to fit state-of-the-art High Tension (HT) leads (Spark plug leads) into the Lucas distributor cap. It has two current paths for reliability and redundancy. Its primary path consists of spiral wound stainless steel alloy, while its secondary path consists of a carbon impregnated fiberglass center core. It has a low 500 Ohm-per-foot resistance. It also has silicone jackets to resist high temperatures, moisture, oil and chemicals, and an EPDM (Ethylene Propylene Diene Monomer) rubber inner insulation for superior heat resistance and prevention of arcing and voltage leaks. Fiberglass reinforcing braid is used for added strength and flexibility.

Spark Plugs

The Heat Range of a spark plug is the function of the spark plug that regulates the temperature of the combustion chamber. This is achieved by its ability to maintain or to remove the heat produced by the combustion process and to transfer excess heat to the cylinder head so that it will then be transferred to the coolant system. In addition, the temperature of the spark plug's firing end must be kept low enough to prevent pre-ignition, but high enough to prevent fouling. This is referred to as its “Thermal Performance” and is determined by the heat range of the spark plug.

Be aware that a high performance modified engine cannot continue using the Original Equipment heat range spark plug, because more power always comes from more heat. When an engine is modified to increase its power, the Original Equipment spark plugs very often produce hotter than necessary running conditions and the fuel / air preignites or, even worse, detonates without the need of an electric ignition spark, creating an uncontrolled explosion that can severely damage the engine.

If the Original Equipment spark plug heat range is colder than necessary in relation to the actual operating conditions, the fuel / air mixture will not burn completely, creating carbon deposits that will not burn and will stick to the spark plug, fouling it. Since carbon is a conductor of electricity, it will deflect the electricity to ground (earth) instead of creating the ignition spark, resulting in engine misfiring and malfunctioning. When this problem exists, the installed spark plugs will show unburned fuel, looking similar to a rich fuel / air mixture. If the heat range of the spark plug is hotter than necessary due to the actual operating conditions, the engine will overheat and the spark plugs will appear similar to a lean fuel / air mixture.

If your cylinder head has Original Equipment specification combustion chambers, do not use a spark plug that is of different length than that of the Original Equipment spark plug. Bear in mind that the insulator nose length is a determining factor in the heat range of a spark plug. The longer the insulator nose, the less heat is absorbed, and the further the heat must travel into the coolant passages of the cylinder head. This means that the spark plug has a higher internal temperature, and is said to be a “Hot” spark plug. A hot spark plug maintains a higher internal operating temperature in order to burn off oil and carbon deposits, and has no relationship to ignition spark quality or intensity. Conversely, a “Cold” spark plug has a shorter insulator nose and absorbs more combustion chamber heat. This heat travels a shorter distance, and allows the spark plug to operate at a lower internal temperature. A colder heat range can be necessary when an engine is modified for performance, subjected to heavy loads, or when it is run at high engine speeds for significant periods of time. The higher cylinder pressures developed by high compression, hotter camshafts, not to mention the engine speed ranges at which such custom-built engines are operated, make colder spark plugs mandatory to eliminate spark plug overheating and engine damage. The colder type spark plug removes heat more quickly, and will reduce the chance of pre-ignition / detonation and burnout of the firing end. Note that engine temperatures can effect the operating temperature of the spark plug, but not the heat range of the spark plug.

The electrode end appearance is dependent upon the temperature of the spark plug tip. There are three basic diagnostic categories for spark plugs: good, fouled, and overheated. The border area between the fouling and optimum operating regions (842° Fahrenheit / 450° Celsius) is referred to as the self-cleaning temperature of the spark plug. This is the temperature point at which any accumulated carbon deposits and combustion residues are automatically burned off. Too low a spark plug temperature can result in spark plug fouling, evidenced by sticky deposits that are found on the spark plug. Too high a spark plug temperature results in pre-ignition and breaking up, with potentially dangerous consequences to the engine should fragments of spark plug insulator material fall into the cylinder bore.

Many custom-built engines no longer have Original Equipment specification cylinder heads. In such cases, the first thing that needs to be done is to figure out the necessary spark plug thread diameter, thread length or "reach," and the type of seat design required by the currently installed modified cylinder head. Failure to select the right spark plug can result in an inappropriate heat range and potential engine damage. When selecting the spark plug "nose" configuration, the simple rule to remember is: The more the spark plug is exposed to the fuel / air mixture, the easier it is to initiate combustion. While many specialized spark plugs have been developed for high-end racecars, for most vehicles the choice typically comes down to either regular-gap (conventional) or projected-nose styles. The regular-gap spark plug is the traditional configuration that was factory-installed on many of the classic muscle cars. For modern high-performance work, it should only be used if there is not enough clearance for a projected-nose spark plug. The latter style "projects" the ignition spark further into the combustion chamber than a standard spark plug does, and will nearly always offer improved performance.

The factory specification engines installed in the MGB use a projected nose spark plug, sometimes referred to as a "Y type" spark plug because of its identifying letter in the old Champion spark plug part numbering system. This spark plug projects about 1/8" (about 3mm) into the combustion chamber. The tip is thus closer to the centre of the fuel / air mixture, aiding the speed of combustion and making the engine more tolerant to variations of the fuel / air mixture. It is also more easily cooled by the incoming fuel. A projected-nose spark plug has a broader heat range than a regular-gap spark plug, and also permits backing off the ignition timing a few degrees for better midrange response and detonation resistance. However, be aware that should detonation occur, the projected tip is more vulnerable to damage than the regular design.

Determining the optimum heat range spark plug for a particular modified engine is primarily a process of trial-and-error. Set up the engine for optimum fuel / air ratio and ignition timing first, and then fine-tune spark plug heat range. Run the car, then “read” the spark plugs by closely inspecting and analyzing the condition of the spark plug tip and insulator. Once you find the correct heat range that prevents fouling without contributing to pre-ignition or detonation, changing to a hotter or colder spark plug will not alter engine performance. Be aware that reading spark plugs on the street is not the same as it is in racing. On the street, as mileage piles up, a properly burning spark plug traditionally has a clearly visible brown or grayish-tan color. Today’s unleaded pump gasoline may use additives that cause a discoloration of the spark plug core nose; they could be pink, purple, or blue. When reading spark plugs, do not consider the presence of this coloration as an indication of heat range.

There is an alternative spark plug design makes use of a finer 0.050" (1.27mm) tip electrode. Electrical discharges tend to occur more readily from both finer and hotter surfaces, and such electrodes exhibit both features. They will fire at lower voltages and, as a result, are often supplied gapped wider than a regular spark plug. However, be aware that the finer tip will erode more quickly than that of a regular spark plug, making for a higher frequency of replacement. Be aware that while platinum tip spark plugs have the advantage of electrode tips that erode more slowly, copper electrodes are more effective at conducting heat, thus reducing the possibility that the hot tip of the electrode will trigger pre-ignition.

If you have a higher than standard energy coil available, then by all means widen the spark plug gap if you can do so while maintaining reasonable geometry of the electrode of the spark plug. It should be remembered that if the gap is widened, then the electrical stress on all the high-tension side of the ignition system is increased, and any deficiencies in that area will quickly exhibit themselves. Unfortunately, most spark plugs do not take well to over-gapping, the ground (earth) terminal becoming non-parallel. With only a single point being correctly gapped, the result will be premature wear-out. Do not over-gap any spark plug by more than .005” (0.127mm).

However, if you choose to retain the Original Equipment coil, there is something that you can do in order to maximize its efficiency. As the spark jumps across the gap between the electrodes, erosion occurs. Material from the center electrode is literally transferred to the side electrode, whereupon the center electrode begins to assume a rounded shape. This creates a problem in that electricity prefers to jump across a gap that is created by sharp edges. In order to deal with this problem, use a fine file to flatten the tip of the center electrode, and then shape the face of the side electrode unit until you have produced a sharp 90^{o} edge that is directly over the middle of the center electrode.

There are myriad spark plug designs, each one claiming some special benefit. The most common improvement is to provide multiple electrodes, by V notching either the centre or the ground (earth) electrode or having up to four ground (earth) electrodes around the periphery. Some spark plugs contain a resistor. A resistor is necessary in order to reduce electromagnetic radiation by effectively damping the ignition system (like a shock absorber) and thus preventing interference to radio and TV reception in the vicinity of the vehicle. Having the resistor in the spark plug allows the use of High Tension (HT) leads (spark plug leads) without the liability of using carbon resistive cores that are generally regarded as being less reliable than copper leads.

Testing the Distributor

If the car seems to be running well, then no further work is needed. However, if you seem to have an ignition miss, you can perform a diagnostic test. This will require an Ohmmeter. An Ohmmeter measures resistance and is a feature normally found on Voltmeters. In fact, most Volt test meters are actually Volt-Ohm Meters (VOM). Inexpensive, yet good quality analog meters may be found at Radio Shack and many other sources. Some dwell / tachometer meters also have both a Volt and an Ohm feature. Many owners prefer to have a separate VOM as it allows them to do tuning using both the dwell / tachometer and the VOM whenever it becomes necessary.

Using an Ohmmeter that has been set to its "Ohms" or "Resistance" function, touch the two probes together and watch to see if the meter's needle swings to zero. If it does, then this shows that there is zero resistance, just as it should. Some of the more expensive meters have a zero function where the probes must be held together and the scale manually adjusted to zero. The less expensive models do not have this feature and it is unnecessary for this type of work. Having confirmed that the meter is working properly, remove the distributor cap from the car, having disconnected the High Tension (HT) leads (spark plug leads) from the spark plugs and the ignition coil wire from the ignition coil. A small piece of masking tape on each High Tension (HT) lead (spark plug lead) with the number of the cylinder that it services makes correct reinstallation easy.

Take one probe and insert it into the spark plug end of the High Tension (HT) lead (spark plug lead). You can probably insert it between the metal terminal and the inside of the rubber boot in order to secure it in place. Next, touch the probe to the terminal inside of the distributor cap. This tests both the distributor cap and the High Tension (HT) lead (spark plug lead) together as a unit. Make a note of the resistance reading, and then check the other High Tension (HT) leads (spark plug leads) using the same technique. Finally, check the ignition coil wire at the end that goes from the ignition coil to the carbon brush inside of the top of the distributor cap. All of the leads should have roughly the same resistance. If one is very much lower or higher than the others are, or if one shows high or infinite resistance, then the lead is bad and the entire set should be considered suspect. How to determine whether it is a High Tension (HT) lead (spark plug lead) problem or a distributor cap problem? Simple: Remove the offending High Tension (HT) lead (spark plug lead) that shows the infinite or high resistance from the distributor cap and test it again. If it now shows resistance similar to others, the problem is in the distributor cap. Firmly seat the High Tension (HT) lead (spark plug lead) into the distributor cap again, making sure that it is fully engaged, and then recheck the resistance reading. If it still shows the problem, then the distributor cap is definitely at fault and should be replaced.

Checking the ignition coil is a very simple task. With the ignition off, use your Ohmmeter in order to check the resistance across the ignition coil terminals by connecting one probe to each of the terminals and then reading the resistance. On a 12 Volt ignition coil, it should read as having between 3.1 and 3.5 Ohms of resistance across the primary circuit. On a 6 Volt ignition coil, it should read as having between 1.43 and 1.58 Ohms of primary resistance across the primary circuit, with the ballast resistance contained within the wiring loom (harness) also measuring about the same in order to result in nominally the same ignition current in the two systems during normal running. The Lucas DLB 105 12 Volt Sports Ignition Coil shows slightly higher primary resistance than the Original Equipment Lucas HA 12 twelve Volt ignition coil, about 5 Ohms of primary resistance. If it reads as having zero resistance, then you have a short in the ignition coil and it is not functioning. If it reads as having infinite resistance, then there is a break in the windings and the ignition coil is not functioning. None of these faults can be repaired, so replace the ignition coil. If the ignition coil passes this test, continue checking the system. There are 6 Volt Sport ignition coils, and these may have a primary resistance even lower than that of the Original Equipment Lucas 16 C6 6 Volt unit (3.1 Ohms of primary resistance). However, the lower the primary resistance, the greater the current, thus the greater the stress will be on the contact breaker points, which is why the contact breaker points burn very rapidly when a 6 Volt ignition coil is used on an unballasted system.

Next, use the Voltmeter in order to test the voltage coming from the ignition coil with the ignition switch on. This should read as being between 6 and 9 Volts, depending on the model of the ignition coil. If it reads as having more voltage than this, then the ignition coil is shorted out internally. If it is less than this, then there is excessive internal resistance. None of these faults can be repaired, so replace it. If the voltage is within limits, turn off the ignition and then use the Ohmmeter in order to test the wire that runs between the distributor and the ignition coil terminal. This terminal is usually marked either "CB" (contact breaker) or "-", depending the vintage of the ignition coil. It should read out as having zero resistance. If it reads out as having infinite resistance, then you have a bad wire. If you show more than a few Ohms of resistance, then you have either a broken wire or one that is going bad. Replace it. Once you have a good wire providing current from the ignition coil to the distributor, you can begin your tests within the distributor.

Turn the ignition switch to the start position, applying power to the system. Check the voltage on the wire coming from the ignition coil to the distributor, and then again at the contact breaker points. The 25D4 distributor has a plastic tab type terminal on the side of the distributor. If the connection is loose or corroded, then you will see a voltage drop between the ignition coil and the contact breaker points. If you have good voltage from the ignition coil wire but low voltage at the contact breaker points, then the wire that goes from the terminal on the distributor to the contact breaker points is bad. Next, with the contact breaker points closed, check the voltage on both sides of the contact breaker point's contacts. A drop of more than one Volt indicates bad contact breaker points. While you are examining this area, make sure that the braided base plate ground (earth) wire is in good condition. This wire runs from the base plate to one side of the distributor and is connected to the distributor by one of the screws that secure the base plate in place. If it is bad, then the grounding of the system is less than optimal and may be the cause of your problem. Although this ground (earth) wire is insulated, there is absolutely no reason for it to be insulated. It should be noted that the exact same wire was used for the terminal blocks, and that that particular wire needed to be insulated. My guess is that it was simply easier and less expensive to inventory and catalogue only one type of wire. The other main ground (earth) for the system is the distributor mounting plate on the engine block. The distributor must be tight in the clamping plate and the mounting plate must be firmly secured to the engine block in order for the system to function properly. After these checks have been completed, you should have discovered any Low Tension (LT) circuit problems and have corrected them. The only part of the system you have not checked is the condenser (capacitor). A bad condenser (capacitor) should not prevent the car from starting and running, it only makes it run poorly. It is rare to find a condenser tester today so the old adage of "replace with a known good unit" applies.

Manufacturer	**Part Number**	**Note:**
AC Delco	42XLS	Standard Iron Electrode
AC Delco	R42XLS	Standard Iron Electrode + Internal Resistor
AC Delco	41-804	Platinum Electrode
Autolite	AP 63	Platinum Electrode

Autolite	APP 63	Platinum, Double Electrode
Bosch	W7DC	Copper Electrode
Bosch	W7DP	Platinum Electrode
Bosch	WR7DP	Platinum Electrode + Internal Resistor
Bosch	WR78	Super 4 (4 Ground (Earth) Contacts)
Champion	N9YC	Standard Iron Electrode
Champion	RN9YC	Standard Iron Electrode + Internal Resistor
Champion	N9YC4	Wide Gap .032" (.8128mm)
Champion	N9YC4	V Tip Electrode, Gold Palladium Tip
NGK	BP6ES	Standard Iron Electrode
NGK	BP6EY	V Tip
NGK	BPR6GP	Platinum Electrode
NGK	BPR6EIX	Iridium Fine Tip, .025" (.635mm) Diameter

Breaking In A New Engine

Prior to starting a newly-rebuilt engine, it is essential to prime the oil pump and the oil circulating passages. Although some commercial garages do not follow this procedure as it is time consuming, failure to do this will risk the result of all of your handiwork being destroyed due to a lack of oil flow and oil pressure. Install a magnetic oil sump plug (Moss Motors Part # 328-282), and then fill the oil sump with the most inexpensive non-detergent 20W/50 oil that you can find, although Castrol HD 30 is an excellent choice for running-in a newly rebuilt engine. Tilt the engine by jacking up the front of the car, remove the threaded plug from the forward end of the rocker shaft, and then pour oil into the rocker shaft in order to lubricate the bushings of rocker arms and allow time for the oil to run down through the oil passage in both the rear rocker shaft pedestal and cylinder head casting to the bushing at the rear end of the camshaft (Now you know why the engineers decided to have a threaded plug instead of a press-fitted plug as at the other end of the rocker shaft). Replace the threaded plug into the forward end of the rocker shaft, then level the car. Next, oil the valve stems, and then pour a tablespoon of oil down the pushrod passages in the cylinder head in order to lubricate the tappets and another tablespoon of oil into each spark plug hole in order to lubricate both the pistons and their rings. Pour oil down the anti-drain tube of the oil filter stand in order to fill the high-pressure oil gallery and supply oil to the main bearings, then install the oil filter. Finally, if your engine is not equipped with an oil cooler, disconnect the large external oil feed line that goes to the back corner of the engine block at the oil filter stand and pour oil into it in order to supply oil to the oil pump. If the engine is equipped with an oil cooler, before installing the oil filter, disconnect the large external oil feed line that goes to the back corner of the engine block from the oil cooler and, holding it above the height of the cylinder head, insert a funnel into the oil feed line and pour oil into it in order to supply oil to the oil pump, then reattach it to the oil cooler. Be sure to complete the filling of the oil cooler by pouring oil into the aperture for the oil cooler return line in the oil filter stand. Next, pour oil down the anti-drain tube of the oil filter stand in order to supply oil to the high pressure gallery for the main bearings of the crankshaft. Next, install the oil filter, rotate the engine the engine 360° counterclockwise with the external oil line connected. That should fill the oil pump and the circuit down to the strainer with oil and also wipe the oil onto the walls of the cylinders. Next, rotate the crankshaft clockwise 360° in order to draw the oil into the oil pump from the sump. Once the pump is primed, disconnect the power supply to the fuel pump, and then use the electric starter to turn the engine until your oil pressure gauge gives a steady reading. The oil cannot drain out of the pump once it is primed because the rotor of the oil pump draws oil from the oil sump up into the top of the pump, and then pumps it out of the top on the other side of the rotor.

At this point it is critical that, in order to avoid ruining the camshaft and its tappets, they be properly bedded in. If you have chosen to use a "hot" camshaft, you can minimize the risk of high spring pressures ruining the tappet / lobe interface during their critical break-in period by using soft valve springs during the bedding-in process. Afterwards, using a fitting with a quick-disconnect screwed into the spark plug holes, attach a hose (flexible pipe) from an air compressor tank to it. With the piston at Top Dead Center, the air pressure will hold the valve closed while the soft springs are replaced with their appropriate service-use items. If you use this method to hold the valves in place while replacing the valve springs, afterwards the pistons and rings should be protected by pouring a tablespoon of oil down the spark plug hole and manually cycling the engine 360° in order to replace the oil that the compressed air displaced.

Before starting the engine, again turn the engine over by hand in order to ensure that it rotates freely. Upon first turning over the engine by means of the electric starter, and with the electrical power to the fuel pump disconnected, observe to see if all of the pushrods are rotating. If one or more of the pushrods are not rotating, then the tappets in which they are seated are not spinning in their bores and must be freed. otherwise both the tappet and the lobes of the camshaft will be quickly ruined. This may be simply corrected by switching tappets into alternate bores. Now you may install the rocker arm cover and its gasket, reconnect the electrical power to the fuel pump, install the spark plugs and connect their high tension leads, and then start the engine.

Do not idle engine during the first twenty minutes of operation. Hold the idle of the engine at 2,500 RPM for twenty minutes, occasionally varying engine speed gently between 2,000 RPM and 2,700 RPM. It is important for new springs to take a heat-set. Never abuse or run the engine at high engine speeds when the springs are new. Upon initial start-up, limit the engine speed until the engine temperature has reached operating levels. Shut off the engine and allow the springs to cool to room temperature. This usually will eliminate early breakage and prolong spring life. After the valve springs have been "broken-in", it is common for them to lose a slight amount of pressure. Once this initial pressure loss occurs, the spring pressure should remain constant unless the engine is abused and the springs become overstressed. Should this occur, the springs must either be replaced or shimmed in order to attain the correct pressure.

After this initial process is completed, the engine will be ready to be broken in on the road. The best bedding in of the piston rings should occur at maximum torque when the highest piston ring pressure will be exerted per cycle. Be aware that damage can occur when the piston rings cannot reject heat fast enough during running in, i.e., when there is too much load at low engine speed and too much of a high engine speed horsepower build-up of heat. Consequently, do not lug the engine or operate it at high engine speeds.

Be aware that the manufacturer of both the Fel-Pro and the Payen resin cylinder head gaskets recommends that they not be retorqued after initial installation. Drive for 100 miles, change both the oil and the oil filter, and then retorque the cylinder head stud compression nuts. When retorquing, use your torque wrench to loosen the cylinder head compression nuts, noting how much torque is required to break the stiction on each. It is only necessary to loosen each of them one at a time to the point that the torque reading is less than the required amount of torque for securing the cylinder head. All of the cylinder head compression nuts should be tightened in the same proper sequence pattern as when the cylinder head was first installed. Once this is accomplished, be sure to inspect the valve lash clearances and adjust them as required. Because retorquing the cylinder head will reduce the valve clearances, after every retorquing of the cylinder head you will need to reset the valve clearances. Torque them again at 500 miles in order to complete the bedding-in of the new camshaft and lifters, let it cool if you using a cast iron cylinder head, and then retorque the cylinder head stud compression nuts using the proper sequence pattern. You will find some cylinder head stud compression nuts almost tight; some may take almost a quarter turn. Run the car for another 100 miles again. You will find that this time the cylinder head stud compression nuts have not lost quite as much torque. Run an additional 500 miles and again retorque the cylinder head stud compression nuts. During this period do not exceed 4,000 RPM or 45 MPH, operate the engine at full throttle, or allow the engine to labor in any gear. Until the next 1,000 miles total has been completed, limit engine speeds to around 4,500 RPM when shifting gears. Cruising on the highway should be limited to no more than 3,500 RPM. Keep varying the throttle opening and engine speed. The secret is to constantly vary the speed and load without creating excess heat through full throttle laboring and high engine speed operation. After 1,000 miles of following this procedure, change the oil and oil filter and refill the oil sump with a quality oil such as Castrol 20W/50. After another 1,000 miles the engine will be properly broken in and ready for service.

High Performance Clutches

At this point, I would like to point out a piece of equipment that does not deal directly with the engine's power output, but plays an essential role in getting it to the rear wheels: the clutch. Yes, a more powerful engine is indeed more demanding on the clutch. The Original Equipment Borg & Beck clutch should be capable of handling the power of a high performance version of the BMC B Series engine, but you may find that its lifespan is compromised more than you would desire. Of course, there are heavy-duty clutches available for the MGB. These are readily identifiable by the thicker coils of their take-up springs in the clutch driven plate that are wound at a shallower angle than those of the take-up springs found in the Original Equipment clutch. However, it should be noted that almost all of them were originally designed for use in Sherpa delivery vans. Yes, this transmission was in fact designed to also be used in delivery vans! It can also easily withstand the 175 Ft-lbs of torque produced by the six-cylinder engine of an MGC. That is why they last so long in our light little cars. These heavy-duty clutches make use of a more powerful diaphragm spring and hence will not only hold the clutch driven plate against the flywheel better in order to handle the increased power output of the engine, but consequently will also increase clutch pedal GRB 106), as well as both the plain shank of the mounting bolt (BMC Part # 11G 3196) and the pivot bushing (BMC Part # 11G 3195) of the clutch actuating fork (BMC Part # 22B 56, Three-synchro transmission; BMC Part # 22H 1056, Four-synchro transmission). The increased resistance of the diaphragm spring makes the installation of a braided steel hydraulic hose (flexible pipe) a wise move in order to contain the increased hydraulic pressure required in order to actuate the clutch without the bulging of the rubber hydraulic hose of the Original Equipment hydraulic hose (flexible pipe). The flat springs in the cover can be thought of as a series of levers with one end attached to the pressure plate, pivoting in the middle on a large circlip that is attached to the cover, and the other end at the carbon clutch release bearing. Due to the MGB weighing less than the Sherpa delivery vans with their one-ton cargo capacity in which these clutches were intended to be employed, plus the take-up coil springs in the clutch driven plate being consequently stiffer, some of these heavy-duty clutches tend to feel “grabby”, many engaging almost like an on/off switch.

There is a better alternative: simply replace the Original Equipment 8” clutch driven plate with the 8 ½” (215.9mm) clutch driven plate used in the Triumph TR7 (Roadster Factory Part # GCP253). Its splines are identical with those of the original MGB clutch; thus it will fit without modification. Having been designed to be used with a more powerful engine, its greater friction surface area will ensure all of the grip that you will need. Because of its compatibility with the Original Equipment pressure plate, it results in a smooth, light clutch possessing good “lockup” and reduced slippage under severe load conditions without putting increased stress on the clutch throw out (release) bearing and the crankshaft thrust bearings as a high performance clutch does. When used in Original Equipment specification engines they tend to last 120,000 miles, which is considerably better than the 80,000 mile life expectancy of the Original Equipment clutch.

Installing The Clutch

There is little mystery to installing a new clutch onto the engine. This is something of a “rite of passage” amongst MG owners. When removing the clutch from the flywheel, take care to loosen the securing bolts no more than one turn at a time in alternating sequence, gradually working your way around the clutch until the pressure of the diaphragm spring is completely released. The machine bolts on the flywheel are each secured by a tab washer that has tabs that bend over the bolt heads in order to lock them in place. If the flywheel has been removed in the past, these tabs may break off if you attempt to reuse the tab washers, so do not hesitate to order new ones. Next, unscrew the three strap bolts that secure the clips to the pressure plate one turn at a time until the diaphragm makes contact with the clutch cover. Now, remove the strap bolts, the clips along with their tab washers, and then remove the pressure plate. Finally, rotate the spring retainers in order to release each end of the carbon throw out (release) bearing and pull it from its clutch actuating fork.

The new clutch driven plate should be checked in order to ensure that it slides freely on the splined input shaft (first motion shaft). Not all of them do, and minor fitting with a fine-toothed file and fine sandpaper is, sometimes, required for smooth operation. When installing the clutch onto the engine, you should first apply a thin smear of Lubriplate white lithium grease into the splines of the input shaft (first motion shaft) in order to prevent rusting.

Installation of a new clutch requires the use of a clutch-centering tool. Fortunately, an inexpensive plastic version of this simple splined tool (Moss Motors Part# 387-210, Three Main Bearing Engines; 387-235, Five Main Bearing Engines) is often included with a new clutch. The clutch driven plate assembly is marked on one side as the flywheel side. Position the clutch driven plate assembly on the flywheel with the large end of its hub facing away from the flywheel. Use the clutch-centering tool to centralize the clutch driven plate by sliding it into the splined hub of the clutch driven plate and over the pivot bearing in the flywheel. Do not remove the centering tool until the following process is complete: Reinstall the pressure plate into the clutch cover with its strap bolts, spring clips, and tab washers, turning the strap bolts one turn at a time. This tightening sequence should be completed by torquing the strap bolts to between 25 to 30 Ft-lbs. Do not forget to bend the tabs of the tab washers onto the flats of the heads of the strap bolts. Finally, fit the clutch assembly onto the dowels of the flywheel, taking care to tighten each of its securing bolts one turn at a time in alternating sequence, working your way around the flywheel. This tightening sequence should be completed by a final torquing the securing bolts to 25 to 30 Ft-lbs. Now you can remove the clutch-centering tool and reinstall the carbon throw out (release) bearing along with its bearing retainer.

At present, other than for racing application, there appears to be no advantage to substituting any of the currently available alternative throw out (release) bearings for the Original Equipment carbon version. This being the case, avoid the temptation to imitate the racers and install a roller type throw out (release) bearing into your clutch mechanism. The basic idea behind the roller type throw out (release) bearing is the reduction of friction so that shifts can be made more quickly and more accurately under extreme conditions, as in racing with a lightened flywheel and crankshaft. On a street machine, a roller throw out (release) bearing has little practical advantage, but has the disadvantage of requiring frequent removal and relubrication in order to prevent it from self-destructing. This is due to the fact that the MGB has a clutch throw out (release) mechanism that has no provision for a pull-off spring to keep the throw out (release) bearing from remaining in constant contact with its pressure surface on the clutch cover. This is a very light contact and the Original Equipment carbon throw out (release) bearing can withstand this constant contact, but as there is no provision in the original design for disengagement, without modification it is sufficient to keep a modern ball bearing spinning constantly which will result in the lubricant drying out, causing premature failure of the ball bearing. In addition, worn bushings or a worn mounting bolt in the clutch actuating fork can allow the roller bearing to misalign relative to the clutch cover, rapidly increasing wear. While this increased maintenance need is not a problem on a racecar, on a street machine it is a big hassle that is just not worth the trouble.

The Clutch Hydraulic System

Once the clutch is in place and the engine and transmission are mated, you will need to install the clutch slave cylinder. If you are going to purchase a new one, be aware that many, if not all, of the aftermarket clutch slave cylinders are not designed to be rebuildable like the Original Equipment Lockheed clutch slave cylinder is. First, prior to installing it, make sure that both the hydraulic hose (flexible pipe) and bleed nipple are in their correct locations. Because both holes use the same thread, it is easy to get them the wrong way around. The clutch slave cylinder often comes assembled in this manner so that it will fit into the box. The bleed screw should be near the high point of the clutch slave cylinder, with the rubber flex line attached to the lower of the two holes. Inspect both the hole in the end of the pushrod as well as that of the clevis pin for signs of wear. It is common for both of them to have to be replaced. The bore should be closely inspected as well. If the bore is pitted, then rebuilding the clutch slave cylinder without sleeving the bore will work for a short time, but it will probably soon be leaking again. Next, inspect both the plain shank of the mounting bolt (BMC Part # 11G 3196) and the pivot bushing (BMC Part # 11G 3195) of the clutch actuating fork (BMC Part # 22B 56, Three-synchro transmission; BMC Part # 22H 1056, Four-synchro transmission) for wear, and replace them if you find any. This will greatly reduce slop in the action of the clutch pedal and make engagement more consistent. In addition, if the pivot bushing and/or the ends of the plain shank of the mounting bolt are unevenly worn, the clutch actuating fork will tilt when under pressure from the slave cylinder, causing both off-center running and uneven engagement of the carbon clutch release bearing. This will cause a grinding sound.

The threads of the mounting bolt for the clutch actuating fork should always be carefully inspected and cleaned with a soft wire brush in order to remove any rust and / or oxidized aluminum prior to reinstallation. Likewise, the threads of its mounting lug on the transmission case should be carefully chased through with a rethreading tap that is designed to clean and straighten the threads. Prior to screwing the mounting bolt into its threaded mounting lug on the aluminum transmission case, its threads should be coated with anti-seize compound in order to prevent electrolytic corrosion which can cause it to seize in place. The clutch actuating fork should always be installed with its open end facing forward. The spring inside of the clutch slave cylinder will take up any free play in the clutch release mechanism. The spring inside of the clutch slave cylinder and the diaphragm spring of the clutch pushrod push in opposite directions. While the diaphragm spring pushes the piston back into the clutch slave cylinder (and the hydraulic fluid back into the clutch master cylinder), the spring inside of the clutch slave cylinder tries to push it back out again. It is the spring inside of the clutch slave cylinder pushing hydraulic fluid back out that takes up the play in the mechanical linkages to the diaphragm and the wear in the friction and drive plates, in order to give a consistent biting point of the clutch. The pedal spring pulls the pedal off of the push-rod of the clutch master cylinder in order to stop it from rattling around and thus causing wear of the linkage, and to give a consistent pedal height. Wear in the mechanical linkages of the pedal mechanism can cause the clutch engagement / disengagement point of the clutch pedal to become closer to the floor. On the other hand, if the engagement / disengagement point of the clutch pedal is becoming higher, then that indicates wear of the clutch driven plate, cover plate and flywheel, or possibly weakening of the diaphragm springs, as all these combine to result in progressively less pressure on the clutch driven plate. Hence, the clutch mechanism will start to slip with a smaller movement of the carbon clutch release bearing. Under these circumstances, it is reasonable to assume that the piston of the clutch master cylinder is not returning back far enough to clear the bypass port, and so pressure resulting from the developing heat can partially release the clutch without the pedal being operated. However, even if wear in the linkages at the pedal end means that the pedal is not pulling the piston back as far as it should, unless the pedal is physically preventing the piston from coming back far enough to clear the bypass port, the return spring in the clutch master cylinder should always push the piston back a sufficient amount to clear the bypass port. This is, of course, presuming that it has been properly assembled with the small end of its return spring towards the piston and its small end in the bottom of the bore of the clutch master cylinder.

Bolt the clutch slave cylinder onto the bellhousing, and then put a little brake cylinder grease on the very end of the pushrod so that it will lubricate the end of the pushrod as it bears against the piston of the slave cylinder. Remove the cap from the clutch master cylinder so that air can escape and you are not trying to compress the air space inside of the clutch master cylinder when you install the slave cylinder pushrod. Insert the slave cylinder pushrod into the bore far enough to allow you to install the clevis pin through both the clutch actuating fork and the pushrod. If you have replaced the original slave cylinder pushrod with a new one and the new pushrod will not go in far enough to allow you to install the clevis pin, remove it and compare its length with that of the original pushrod. Some aftermarket pushrods are ¼" (6.35mm) too long. This has unfortunate consequences. As the clutch driven plate wears and the diaphragm release ring moves back in response, the clutch release bearing and clutch actuating fork also get pushed back, as does the slave cylinder pushrod, into the slave cylinder. If the slave cylinder pushrod is too long, then it will cause the piston to bottom out inside of the slave cylinder. Consequently, the bearing is in permanent under-load contact with the diaphragm release ring. Premature failure of the carbon release bearing follows soon after. Note that the slave cylinder pushrod does not slide in the rubber boot as the rubber boot is designed to move in and out with the slave cylinder pushrod.

Do not omit the rubber boot when you reassemble the clutch release mechanism as its purpose is to protect the seal. Attach the rubber hydraulic hose (flexible pipe) to the metal line first, and then to the clutch slave cylinder prior to bolting the clutch slave cylinder onto the bellhousing. Be sure to use a new copper washer when you attach the rubber hydraulic hose (flexible pipe) to the clutch slave cylinder. If the boot is installed correctly, then neither the clutch actuating fork, nor the clevis pin, should make contact with the rubber boot.

The resistance to the flow of hydraulic fluid in the clutch system is quite marked, and a spring inside the slave piston is continually pushing the linkages at that end of the system together, eliminating any free play. This system, being back-filled by fluid from the reservoir, tends to resist any light movements of piston, release arm and release bearing. However, if there is air inside of the clutch slave cylinder, there is nothing to prevent them from rattling back and forth, thus generating noise and vibration, as well as promoting wear, just as air in a lever arm damper of the front suspension allows rapid movements of the wheel/axle instead of damping the movement. This is just one of several reasons that the clutch hydraulic system must be bled free of any trapped air. Although the factory service manual describes a procedure for refilling and bleeding the clutch slave cylinder after it has been installed onto the bellhousing, it is easiest to bleed it before attaching it to the bellhousing. The clutch hydraulic system is a real pig to bleed because of the long vertical section of pipe with the U-bend at the top. The conventional bleeding technique requires that any air in the pipe must be pushed all the way down the relatively large-bore pipe before it can exit the nipple of the clutch slave cylinder. This is difficult enough with a continuous pressure bleeder connected to the clutch master cylinder, and well-nigh-on impossible when using the old fashioned technique of using the clutch pedal to pump the hydraulic fluid through the system. Failure to get all of the air out of the clutch hydraulic system will result in the clutch having a low biting point, or even failure to fully disengage.

Unless you have either rebuilt the clutch hydraulic system or previously flushed out all the old crud and old fluid with denatured alcohol, do not reverse bleed the system without flushing it out first, otherwise you will push the crud into the clutch master cylinder. Use a C clamp in order to prevent the piston of the clutch slave cylinder from popping out, and then use either a Gunson's EZ Bleed or a Mighty Vac tool to refill the system through the bleeder nipple on the clutch slave cylinder. This method works best because air bubbles tend to rise upwards. After bleeding the system, reach up and push the actuating rod all the way back into the clutch slave cylinder, and then bleed it again. This pushes any air left in the cylinder back into the line that goes up to the clutch master cylinder. Rebleeding then expels this air from the U-bend. Because air automatically rises, it will ultimately emerge into the clutch master cylinder. If you do not have either of these tools, then a more primitive, but by far easier way to bleed the clutch hydraulic system is to cross-connect the nipple of the right front disc brake caliper and clutch slave cylinder bleed nipples (they should be the same size), open both, and use the brake pedal gently to bleed, or even fill, the clutch hydraulic system. This is even easier than using a Gunson's EZ Bleed, and with either technique you should get the full travel of ½" to 5/8" (12.7mm to 15.56mm) of the clutch slave cylinder push-rod. When rebleeding the system make sure that you siphon some fluid out of the clutch master cylinder first, and make sure that you keep an eye on the fluid level inside of the clutch master cylinder and not let it get too low.

There should be a pedal return spring on the clutch pedal. Contrary to what many might think, it is not there simply to return the clutch pedal to a consistent height against its stop. The purpose of the return spring on the clutch pedal is primarily to ensure that the piston of the clutch master cylinder can fully return and thus clear the port into the reservoir. There is a spring inside of the slave cylinder that keeps the carbon clutch release bearing pushed up against the cover plate. This is part of the self-adjusting mechanism, and ensures a consistent biting point over the short term, i.e., that crankshaft endplay (endfloat) does not push the piston further into the bore resulting in a low biting or release point the next time that the clutch is operated.

Another cause of low biting point is wear in the master linkages of the push-rod, the pedal bushing, and the clevis pin. Note that wear in the mechanical linkages at the slave end is rarely an issue as the design of the hydraulic system compensates for such wear. If the biting point is very low, then this will also make gears difficult to engage as the clutch is not being fully disengaged. In this case you will get grinding when selecting reverse. Grinding when selecting reverse usually means problems with the mechanisms of the transmission or of the selector mechanism.

Do not panic if the engagement point of the clutch pedal rises over a short period of time. With a new clutch, there are always surface irregularities in the friction material of the clutch driven plate. These irregularities contact the flywheel / pressure plate surface and begin the engagement with the surfaces relatively far apart and the clutch pedal closer to the floor than normal. As these irregularities wear away, the distance between the surfaces lessens prior to engagement and the clutch pedal assumes a higher position. With the surfaces become true, the pedal should be almost it the top prior to engagement.

A “fierce” clutch refers to a clutch that it is snatching in its engagement, with very little slip. This is different from a low biting point, and can be caused by oil contamination of the clutch driven plate that has been caused by a leaking rear crankshaft oil seal. This is usually revealed by oil dripping out of the hole at the bottom of the bellhousing, which should contain a cotter pin (jiggle pin) to prevent it from becoming blocked. Note that a leaking front oil seal on the transmission will also result in a leak from this location, but is less likely to contaminate the clutch driven plate.

If you feel vibration when the pedal is fully depressed, this can mean that the carbon clutch release bearing is rubbing on the cover plate, and will eventually wear through and break. When new, the carbon ring of the clutch release bearing is 5/8" thick, and in theory all of this is available to wear down before replacement is required. However, that is only the case if the release bearing is co-axial with the cover plate, as all but about 1/4" of the carbon is recessed into its housing. If the two are offset, or if the clutch release arm is loose on its pivot of the clevis pin in the housing, the bearing will start rubbing on the cover plate, and that will wear down, probably accompanied by noise and vibration when the pedal is down, eventually to break up completely. Vibration that occurs as you are operating the clutch pedal could also caused by rough surfaces on the linkages at the clutch master cylinder or the clutch slave cylinder. Vibration when the pedal is partially depressed and held in a fixed location could be problems with the flywheel, the cover-plate and the clutch driven plate surfaces, or the pilot bushing (in the spigot end of the crankshaft), or the input shaft (first motion shaft) of the transmission that fits into the pilot bushing. Judder as it takes up the drive is something else again, but can also be caused by these components.

The Crankshaft Spigot Pilot Bushing

Any time that the engine is separated from the transmission, it is always wise to inspect the pilot bushing located inside of the spigot end of the crankshaft for wear and to replace it if necessary. If it is worn, then it can generate vibration and noise during clutch operation. The simplest way to remove the pilot bushing from the spigot is to use a 7/8-9 UNC tap. Put an old socket into the bushing hole, then screw the tap into the bushing. When the tap bottoms out on the socket it will make the bushing screw out.

The three-main-bearing crankshafts that were initially employed with the three-synchro transmissions were fitted with a 1.500” (38.1mm) long sintered bronze spigot pilot bushing (BMC Part # 1G 765) that had a smaller-diameter bore size in order to accommodate the smaller-diameter .620” (15.748mm) input section of its input shaft (first motion shaft) (BMC Part # 22H 56) of the transmission that was used in conjunction with the three-main-bearing 18G and 18GA engines. The input shaft (first motion shaft) of the three-synchro transmission was subsequently redesigned to have a .850" (21.59mm) diameter input shaft (first motion shaft) (BMC Part # 22H 843) for use in conjunction with the crankshaft of the five-main-bearing 18GB and later engines, and as such required a spigot pilot bushing with a larger Inside Diameter (I.D.). Fortunately for those who wish to modify the transmission for use on either a three-main-bearing or a five-main-bearing engine for which it was not intended, these early and late input shafts (first motion shafts) are interchangeable. It should be noted that while there are two different lengths of sintered bronze spigot pilot bushings, 1.000" (25.4mm) (BMC Part # 12H 1630) and 1.500" (38.1mm) (BMC Part # 22H 1416) used to support the pilot section of the input shaft (first motion shaft) of the five-main-bearing crankshafts, the spigot bore of the crankshaft is of the same diameter and depth on all of the five-main-bearing crankshafts employed in the MGB, regardless of whether they are fitted with the long or the short spigot pilot bushing. The shortening of the spigot pilot bushing was yet another example of cost-cutting by the factory. The longer of the two spigot pilot bushings provides a greater load bearing surface area and thus wears more slowly, providing better long-term support for the input shaft (first motion shaft) of the transmission.

The spigot pilot bushing's Inside Diameter (ID) dimension is about .002" to .004" (.0508 to .1016mm) larger than that of the Outside Diameter (O.D.) of the pilot section of the input shaft (first motion shaft) of the transmission. It is essential that a worn spigot pilot bushing always be replaced. A worn spigot pilot bushing causes the input shaft (first motion shaft) of the transmission to run off-center, resulting in excessive wear to the roller bearings that support it, its bearing surface, and, in extreme cases, cause accelerated wear to the engagement dogs of the of the input shaft (first motion shaft), and to the third and four gear synchronizer hub.

Prior to installing the spigot pilot bushing into the spigot bore of the crankshaft, be sure to slip it over the input shaft (first motion shaft) to determine that its Inside Diameter (I.D.) is not too small. Due to the wicking properties of the saturated bush, the use of sintered bronze for spigot pilot bushings is a standard practice in the automotive industry. It consists of granules of bronze pressed together under high pressure and heat in order to form a rod that is then machined to form a bushing. The pores in the sintered bronze set up a circulation system for the oil while the shaft is rotating. Soak the new spigot pilot bushing in 10W oil for a few days before you install it. The sintered bronze alloy will soak up the oil until it is saturated, ensuring good lubrication. When installing the pilot bushing, you will need to be careful not to use too much force as any lip that you might create on the Inside Diameter (I.D.) will interfere with the free movement of the first motion shaft (mainshaft) and cause it to squeal when it is cold!

Transmissions

The part numbers used to identify each of the various versions of the four-synchro transmission has resulted in a great deal of confusion. This confusion lies in the fact that some individuals believe that part number changes are based solely on the gearsets. There were in fact only four standard production gearsets used in the four-synchro transmissions of the MGB. However, the part numbers were changed not only for a change of the gear ratios, but also for a change of the transmission case types. For example, there was one part number for the early gear ratio set with the top fill case, then another part number for an intermediate gearset with the top fill case, then another part number for an intermediate gearset with the side fill case, then another part number for the late gearset with the side fill case. This then generates 4 different part numbers for combinations of cases and gearsets. Of course, all of the part numbers were again different for the Overdrive-equipped transmissions. That makes for, at the very least, a total of eight different part numbers. I have also found that there are two different types of castings for the side fill transmissions. As yet I do not know what year these castings were changed. The castings have a very different shape for the starter housing, the early one being rounded and the later one being flat. This then generated at least one more part number for each type (four-speed with an Overdrive unit), making for a total of 10 different part numbers. As I mentioned, they made a part number change for any design change, not just for a change of gearsets.

While much has been said about the somewhat eccentric gear ratios in the MGB's four-speed transmission, it is not commonly understood that the MGB transmissions did not all contain identical gear ratios. In the case of the four-synchro transmissions, the second, third, and fourth gear ratios remained unchanged from one year to the next, as was the case of the

LH Series Overdrive ratio, but there was considerable variation in the first gear ratio. Some of the combinations are more appealing for performance-oriented driving than others:

1962-1967 (MKI) (Non-Synchro first gear) transmission, w/ D-type Overdrive:

1st	3.6363 : 1
2nd	2.2143 : 1
3rd	1.3736 : 1
3rd Overdrive	1.101 : 1
4th	1.000 : 1
4th Overdrive	0.802 : 1
Reverse	18.588 :1

These gear ratios create the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.4420	1,271 RPM	1,979 RPM	2,151 RPM	3,349 RPM
2nd-3rd	.8407	1,139 RPM	2,061 RPM	2,088 RPM	3,412 RPM
3rd-3rd O.D.	.9082	845 RPM	2,605 RPM	1,092 RPM	4,408 RPM
3rd-4th	.3736	816 RPM	2,434 RPM	1,504 RPM	3,996 RPM
3rd O.D. -4th	.101	298 RPM	2,952 RPM	505 RPM	4,995RPM
4th-4th O.D.	.198	643 RPM	2,607 RPM	1,089 RPM	4,510 RPM
3rd O.D.-4th O.D.	.299	883 RPM	2,367 RPM	1,494 RPM	4,006 RPM

These gearsets can be found with engines whose engine numbers start with:

18G, 18GA, 18GB

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	7.5 MPH 12.1 KPH	11.3 MPH 18.2 KPH	15.0 MPH 24.1 KPH	18.8 MPH 30.3 KPH	22.5 MPH 36.2 KPH
2nd	12.3 MPH 19.8 KPH	18.4 MPH 29.6 KPH	24.6 MPH 39.6 KPH	30.8 MPH 49.6 KPH	37.0 MPH 59.5 KPH
3rd	19.9 MPH 32.0 KPH	29.8 MPH 48.0 KPH	39.7 MPH 63.9 KPH	49.7 MPH 80.0 KPH	59.6 MPH 95.9 KPH
3rd Overdrive (D)	24.8 MPH 39.9 KPH	37.2 MPH 59.9 KPH	49.6 MPH 79.8 KPH	62.0 MPH 99.8 KPH	74.4 MPH 119.7 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
4th Overdrive (D)	34.0 MPH 54.7 KPH	51.0 MPH 82.1 KPH	68.1 MPH 109.6 KPH	85.1 MPH 136.9 KPH	102.1MPH 164.3 KPH
Reverse	1.5 MPH 2.4 KPH	2.2 MPH 3.5 KPH	2.9 MPH 4.7 KPH	3.7 MPH 5.9 KPH	4.4 MPH 7.1 KPH

*Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles found as Original Equipment in MKI MGB Roadsters use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply

bolted onto the Original Equipment Hardy-Spicer differential cage (BMC Part # BTB 328) that is found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, and with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	7.9 MPH 12.7 KPH	11.8 MPH 19.0 KPH	15.8 MPH 25.4 KPH	19.7 MPH 31.7 KPH	23.7 MPH 38.1 KPH
2nd	13.0 MPH 20.9 KPH	19.4 MPH 31.2 KPH	25.9 MPH 41.7 KPH	32.4MPH 52.1 KPH	38.9 MPH 62.6 KPH
3rd	20.9 MPH 33.6 KPH	31.3 MPH 50.4 KPH	41.8 MPH 67.3 KPH	52.2 MPH 84.0 KPH	62.7 MPH 100.9 KPH
3rd Overdrive (D)	26.1 MPH 42.0 KPH	39.1 MPH 62.9 KPH	52.1 MPH 83.8 KPH	65.1 MPH 104.8 KPH	78.2 MPH 125.8 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
4th Overdrive (D)	35.0 MPH 56.3 KPH	52.5 MPH 84.5 KPH	70.0 MPH 112.6 KPH	87.5 MPH 140.8 KPH	105.0 MPH 169.0 KPH
Reverse	1.6 MPH 2.6 KPH	2.3 MPH 3.7 KPH	3.1 MPH 5.0 KPH	3.9 MPH 6.3 KPH	4.6 MPH 7.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.5 MPH 13.7 KPH	12.7 MPH 20.4 KPH	16.9 MPH 27.2 KPH	21.1 MPH 34.0 KPH	25.4 MPH 40.9 KPH
2nd	13.9 MPH 22.4 KPH	20.8 MPH 33.5 KPH	27.8 MPH 44.8 KPH	34.7 MPH 55.8 KPH	41.6 MPH 66.9 KPH
3rd	22.4 MPH 36.0 KPH	33.6 MPH 54.1 KPH	44.8 MPH 72.1 KPH	56.0 MPH 90.1 KPH	67.1 MPH 108.0 KPH
3rd Overdrive (D)	27.9 MPH 44.9 KPH	40.7 MPH 65.5 KPH	55.8 MPH 89.8 KPH	69.8 MPH 112.3 KPH	83.7 MPH 134.7 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
4th Overdrive (D)	38.3 MPH 61.6 KPH	57.5 MPH 92.5 KPH	76.7 MPH 123.4 KPH	95.8 MPH 154.2 KPH	115.0 MPH 185.1 KPH
Reverse	1.6 MPH 2.6 KPH	2.5 MPH 4.0 KPH	3.3 MPH 5.3 KPH	4.1 MPH 6.6 KPH	5.0 MPH 8.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.9 MPH 14.3 KPH	13.4 MPH 21.6 KPH	17.9 MPH 28.8 KPH	22.4 MPH 36.0 KPH	26.8 MPH 43.1 KPH
2nd	14.7 MPH 23.7 KPH	22.0 MPH 35.4 KPH	29.4 MPH 47.3 KPH	36.7 MPH 59.1 KPH	44.1 MPH 71.0 KPH
3rd	23.7 MPH 38.1 KPH	35.5 MPH 57.1 KPH	47.4 MPH 76.3 KPH	59.2 MPH 95.3 KPH	70.1 MPH 99.9 KPH
3rd Overdrive (D)	29.5 MPH 47.5 KPH	44.3 MPH 71.3 KPH	58.1 MPH 93.5 KPH	73.9 MPH 118.9 KPH	88.6 MPH 142.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
4th Overdrive (D)	39.7 MPH 63.9 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	1.7 MPH 2.7 KPH	2.6 MPH 4.2 KPH	3.5 MPH 5.6 KPH	4.4 MPH 7.1 KPH	5.2 MPH 8.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	9.1 MPH 14.6 KPH	13.7 MPH 22.0 KPH	18.2 MPH 29.3 KPH	22.8 MPH 36.7 KPH	27.3 MPH 43.9 KPH
2nd	15.0 MPH 24.1 KPH	22.4 MPH 36.0 KPH	29.9 MPH 48.1 KPH	37.4 MPH 60.2 KPH	44.9 MPH 72.3 KPH
3rd	19.8 MPH 31.9 KPH	29.7 MPH 47.8 KPH	39.6 MPH 63.7 KPH	49.5 MPH 79.7 KPH	59.4 MPH 95.6 KPH
3rd Overdrive (D)	30.1 MPH 48.4 KPH	45.2 MPH 72.7 KPH	60.2 MPH 96.9 KPH	75.3 MPH 121.2 KPH	90.3 MPH 145.3 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
4th Overdrive (D)	41.3 MPH 66.5 KPH	62.0 MPH 99.8 KPH	82.7 MPH 133.1 KPH	103.3 MPH 166.2 KPH	124.0 MPH 199.6 KPH
Reverse	1.8 MPH 2.9 KPH	2.7 MPH 4.3 KPH	3.6 MPH 5.8 KPH	4.5 MPH 7.2 KPH	5.3 MPH 8.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	9.4 MPH 15.1 KPH	14.1 MPH 22.7 KPH	18.8 MPH 30.2 KPH	23.5 MPH 37.3 KPH	28.1 MPH 45.4 KPH
2nd	15.4 MPH 24.8 KPH	23.1 MPH 37.2 KPH	30.8 MPH 49.6 KPH	38.5 MPH 62.0 KPH	46.2 MPH 74.3 KPH
3rd	24.8 MPH 39.9 KPH	37.3 MPH 60.0 KPH	49.7 MPH 80.0 KPH	62.1 MPH 99.9 KPH	74.5 MPH 119.9 KPH
3rd Overdrive (D)	31.0 MPH 49.9 KPH	46.5 MPH 74.8 KPH	62.0 MPH 99.8 KPH	77.5 MPH 124.7 KPH	93.0 MPH 149.7 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
4th Overdrive (D)	42.5 MPH 68.4 KPH	63.8 MPH 102.7 KPH	85.1 MPH 137.0 KPH	106.3 MPH 171.1 KPH	127.6 MPH 205.3 KPH
Reverse	1.8 MPH 2.9 KPH	2.7 MPH 4.3 KPH	3.7 MPH 5.9 KPH	4.6 MPH 7.4 KPH	5.5 MPH 12.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.8 MPH 15.8 KPH	14.8 MPH 23.8 KPH	19.7 MPH 31.7 KPH	24.6 MPH 39.6 KPH	29.5 MPH 47.5 KPH
2nd	16.2 MPH 26.1 KPH	24.2 MPH 38.9 KPH	32.3 MPH 52.0 KPH	40.4 MPH 65.0 KPH	48.5 MPH 78.0 KPH
3rd	26.0 MPH 41.8 KPH	39.1 MPH 62.9 KPH	52.1 MPH 83.8 KPH	65.1 MPH 104.8 KPH	78.1 MPH 125.7 KPH
3rd Overdrive (D)	32.5 MPH 52.3 KPH	48.7 MPH 78.4 KPH	65.0 MPH 104.6 KPH	80.1 MPH 128.9 KPH	97.5 MPH 156.9 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
4th Overdrive (D)	44.6 MPH 71.8 KPH	66.9 MPH 107.7 KPH	89.2 MPH 143.5 KPH	111.5 MPH 179.4 KPH	133.9 MPH 215.5 KPH
Reverse	1.9 MPH 3.1 KPH	2.9 MPH 4.7 KPH	3.8 MPH 6.1 KPH	4.8 MPH 7.7 KPH	5.8 MPH 9.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset (Autogear Part # CWP033)[1] Hardy-Spicer Banjo-type Rear Axle, (BMC Part # BTB1244)[2] Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.4 MPH 16.7 KPH	15.6 MPH 25.1 KPH	20.8 MPH 33.5 KPH	26.0 MPH 41.8 KPH	31.2 MPH KPH
2nd	17.1 MPH 27.5 KPH	25.6 MPH 41.2 KPH	34.1 MPH 54.9 KPH	42.7 MPH 68.7 KPH	51.2MPH 82.4 KPH
3rd	27.5 MPH 44.3 KPH	41.3 MPH 66.5 KPH	55.0 MPH 88.5 KPH	68.8 MPH 110.7 KPH	82.6 MPH 132.9 KPH
3rd Overdrive (D)	34.3 MPH 55.2 KPH	51.5 MPH 82.9 KPH	68.7 MPH 110.6 KPH	85.8 MPH 138.1 KPH	103.0 MPH 165.8 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
4th Overdrive (D)	47.1 MPH 75.8 KPH	70.7 MPH 113.8 KPH	94.3 MPH 151.8 KPH	117.8 MPH 189.6 KPH	141.4 MPH 227.6 KPH
Reverse	2.0 MPH 3.2 KPH	3.0 MPH 4.8 KPH	4.1 MPH 6.6 KPH	5.1 MPH 8.2 KPH	6.1 MPH 9.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.7 MPH 17.2 KPH	16.1 MPH 25.9 KPH	21.5 MPH 34.6 KPH	26.9 MPH 43.3 KPH	32.2 MPH 51.8 KPH
2nd	17.6 MPH 28.3 KPH	26.5 MPH 42.6 KPH	35.3 MPH 56.8 KPH	44.1 MPH 71.0 KPH	52.9 MPH 85.1 KPH
3rd	28.4 MPH 45.7 KPH	42.7 MPH 68.7 KPH	56.9 MPH 91.6 KPH	71.1 MPH 114.4 KPH	85.3 MPH 137.3 KPH
3rd Overdrive (D)	35.5 MPH 57.1 KPH	53.2 MPH 85.6 KPH	71.0 MPH 114.3 KPH	88.7 MPH 142.7 KPH	106.5 MPH 171.4 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
4th Overdrive (D)	48.7 MPH 78.3 KPH	73.1 MPH 117.6 KPH	97.4 MPH 156.7 KPH	121.8 MPH 196.0 KPH	146.2 MPH 235.3 KPH
Reverse	2.1 MPH 3.4 KPH	3.1 MPH 5.0 KPH	4.2 MPH 6.8 KPH	5.2 MPH 8.4 KPH	6.3 MPH 10.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.9 MPH 17.5 KPH	16.3 MPH 26.2 KPH	21.7 MPH 34.9 KPH	27.6 MPH 44.4 KPH	32.6 MPH 52.5 KPH
2nd	17.8 MPH 28.6 KPH	26.8 MPH 43.1 KPH	35.7 MPH 57.4 KPH	44.6 MPH 71.8 KPH	53.5 MPH 86.1 KPH
3rd	28.8 MPH 46.3 KPH	43.1 MPH 69.4 KPH	57.3 MPH 92.2 KPH	71.9 MPH 115.7 KPH	86.3 MPH 138.9 KPH
3rd Overdrive (D)	35.9 MPH 57.8 KPH	53.8 MPH 86.6 KPH	71.8 MPH 115.5 KPH	89.7 MPH 144.4 KPH	107.7 MPH 173.3 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
4th Overdrive (D)	49.3 MPH 79.3 KPH	73.9 MPH 118.9 KPH	98.5 MPH 158.5 KPH	123.2 MPH 198.3 KPH	147.8 MPH 237.9 KPH
Reverse	2.1 MPH 3.4 KPH	3.2 MPH 5.1 KPH	4.3 MPH 6.9 KPH	5.3 MPH 8.5 KPH	6.4 MPH 10.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900) [1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.6 MPH 18.7 KPH	17.4 MPH 28.0 KPH	23.3 MPH 37.5 KPH	29.1 MPH 46.8 KPH	34.9 MPH 56.2 KPH
2nd	19.5 MPH 31.4 KPH	29.3 MPH 47.1 KPH	39.0 MPH 62.7 KPH	48.8 MPH 78.5 KPH	58.6 MPH 94.3 KPH
3rd	30.8 MPH 49.6 KPH	46.2 MPH 74.3 KPH	61.6 MPH 99.1 KPH	77.0 MPH 123.9 KPH	92.4 MPH 148.7 KPH
3rd Overdrive (D)	38.4 MPH 61.8 KPH	57.6 MPH 92.7 KPH	76.8 MPH 123.6 KPH	96.0 MPH 154.5 KPH	115.2 MPH 185.4 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
4th Overdrive (D)	54.7 MPH 88.0 KPH	79.1 MPH 127.3 KPH	105.5 MPH 169.8 KPH	131.8 MPH 212.1 KPH	158.2 MPH 254.6 KPH
Reverse	2.3 MPH 3.7 KPH	3.4 MPH 5.5 KPH	4.5 MPH 7.2 KPH	5.7 MPH 9.2 KPH	6.8 MPH 10.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.5 MPH 21.1 KPH	18.8 MPH 30.3 KPH	25.1 MPH 40.4 KPH	31.3 MPH 50.4 KPH	37.6 MPH 60.5 KPH
2nd	20.6 MPH 33.1 KPH	30.9 MPH 49.7 KPH	41.1 MPH 66.1 KPH	51.4 MPH 82.7 KPH	61.7 MPH 99.3 KPH
3rd	33.2 MPH 53.4 KPH	49.8 MPH 80.1 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.5 MPH 160.1 KPH
3rd Overdrive (D)	41.4 MPH 66.6 KPH	62.1 MPH 99.9 KPH	82.6 MPH 132.9 KPH	103.5 MPH 166.6 KPH	124.1 MPH 199.7 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
4th Overdrive (D)	56.8 MPH 91.4 KPH	85.2 MPH 137.1 KPH	113.6 MPH 182.8 KPH	142.0 MPH 228.5 KPH	170.0 MPH 273.6 KPH
Reverse	2.4 MPH 3.8 KPH	3.7 MPH 5.9 KPH	4.9 MPH 7.9 KPH	6.1 MPH 9.8 KPH	7.3 MPH 11.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

1968-1974 (Early MKII) (Top Fill Version) Transmission, w/ LH-type Overdrive:

1st	3.440 : 1
2nd	2.167 : 1
3rd	1.382 : 1
3rd Overdrive	1.133 : 1
4th	1.000 : 1
4th Overdrive	0.820 : 1
Reverse	12.098 : 1

These gear ratios create the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.273	1,203 RPM	2,047 RPM	2,035 RPM	3,465 RPM
2nd-3rd	.785	1,087 RPM	2,163 RPM	1,992 RPM	3,508 RPM
3rd-3rd O.D.	.882	586 RPM	2,664 RPM	991 RPM	4,509 RPM
3rd-4th	.382	829 RPM	2,421 RPM	1,520 RPM	3,980 RPM
3rd O.D. -4th	.133	382 RPM	2,868 RPM	646 RPM	4,854 RPM
4th-4th O.D.	.180	585 RPM	2,665 RPM	990 RPM	4,510 RPM
3rd O.D.-4th O.D.	.313	586 RPM	2,664 RPM	991 RPM	4,509 RPM

These gearsets can be found as Original Equipment in transmission assemblies with engines whose engine numbers start with:

18GD, 18GF, 18GG, 18GH, 18GJ, 18GK, 18V/581, 18V/582, 18V/583, 18V/584, 18V585, 18V672, 18V673, 18V779, 18V780

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	7.9 MPH 12.7 KPH	11.9 MPH 19.1 KPH	15.9 MPH 25.6 KPH	19.8 MPH 31.9 KPH	23.8 MPH 38.3 KPH
2nd	12.5 MPH 20.1 KPH	18.8 MPH 30.3 KPH	25.1 MPH 40.4 KPH	31.3 MPH 50.4 KPH	37.6 MPH 60.5 KPH
3rd	19.7 MPH 31.7 KPH	29.6 MPH 47.6 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	56.2 MPH 90.4 KPH
3rd Overdrive (LH)	24.1 MPH 38.8 KPH	36.1 MPH 58.1 KPH	48.2 MPH 77.6 KPH	60.2 MPH 96.9 KPH	72.3 MPH 116.4 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
4th Overdrive (LH)	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	2.3 MPH 3.7 KPH	3.4 MPH 5.5 KPH	4.5 MPH 7.2 KPH	5.6 MPH 9.0 KPH	6.8 MPH 10.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.3 MPH 13.4 KPH	12.5 MPH 20.1 KPH	16.7 MPH 26.9 KPH	20.8 MPH 33.5 KPH	25.0 MPH 40.2 KPH
2nd	13.2 MPH 21.2 KPH	19.9 MPH 32.0 KPH	26.5 MPH 42.6 KPH	33.1 MPH 53.3 KPH	39.7 MPH 63.9 KPH
3rd	20.8 MPH 33.5 KPH	31.1 MPH 50.0 KPH	41.5 MPH 66.8 KPH	51.9 MPH 83.5 KPH	62.3 MPH 100.3 KPH
3rd Overdrive (LH)	25.3 MPH 40.7 KPH	38.0 MPH 61.1 KPH	50.6 MPH 81.4 KPH	63.3 MPH 101.9 KPH	76.0 MPH 122.3 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
4th Overdrive (LH)	32.1MPH 51.7 KPH	48.2 MPH 77.6 KPH	70.0 MPH 112.6 KPH	80.3 MPH 129.2 KPH	96.4 MPH 155.1 KPH
Reverse	2.4 MPH 3.9 KPH	3.6 MPH 5.8 KPH	4.7 MPH 7.6 KPH	5.9 MPH 9.5 KPH	7.1 MPH 11.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.9 MPH 14.3 KPH	13.4 MPH 21.6 KPH	17.9 MPH 28.8 KPH	22.3 MPH 35.9 KPH	26.8 MPH 43.1 KPH
2nd	14.2 MPH 22.8 KPH	21.3 MPH 34.3 KPH	28.4 MPH 45.7 KPH	35.5 MPH 57.1 KPH	42.6MPH 68.6 KPH
3rd	22.2 MPH 35.7 KPH	33.4 MPH 53.7 KPH	44.5 MPH 71.6 KPH	55.6 MPH 89.5 KPH	66.7 MPH 107.3 KPH
3rd Overdrive (LH)	27.1 MPH 43.6 KPH	40.7 MPH 65.5 KPH	54.3 MPH 87.4 KPH	67.8 MPH 109.0 KPH	81.4 MPH 131.0 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
4th Overdrive (LH)	37.5 MPH 60.3 KPH	56.2 MPH 90.4 KPH	75.0 MPH 120.7 KPH	93.7 MPH 150.8 KPH	112.5 MPH 181.0 KPH
Reverse	2.5 MPH 4.0 KPH	3.8 MPH 6.1 KPH	5.1 MPH 8.2 KPH	6.3 MPH 10.1 KPH	7.6 MPH 12.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.5 MPH 15.3 KPH	14.2 MPH 22.8 KPH	18.9 MPH 30.4 KPH	23.6 MPH 38.0 KPH	28.4 MPH 45.7 KPH
2nd	15.0 MPH 24.1 KPH	22.5 MPH 36.2 KPH	30.0 MPH 48.3 KPH	37.5 MPH 60.3 KPH	45.0 MPH 72.4 KPH
3rd	23.5 MPH 37.8 KPH	35.3 MPH 56.8 KPH	47.1 MPH 75.8 KPH	58.8 MPH 94.6 KPH	70.6 MPH 113.6 KPH
3rd Overdrive (LH)	28.7 MPH 46.2 KPH	43.1 MPH 69.4 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
4th Overdrive (LH)	39.7 MPH 63.8 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	2.7 MPH 4.3 KPH	4.0 MPH 6.4 KPH	5.4 MPH 8.7 KPH	6.7 MPH 10.8 KPH	8.1 MPH 13.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	9.6 MPH 15.4 KPH	14.4 MPH 23.2 KPH	19.3 MPH 31.1 KPH	24.1 MPH 38.8 KPH	28.9 MPH 46.5 KPH
2nd	15.3 MPH 24.6 KPH	33.0 MPH 53.1 KPH	30.6 MPH 49.2 KPH	38.2 MPH 61.5 KPH	45.9 MPH 73.9 KPH
3rd	24.0 MPH 38.6 KPH	36.0 MPH 57.9 KPH	48.0MPH 77.2 KPH	60.0 MPH 96.6 KPH	72.0 MPH 115.9 KPH
3rd Overdrive (LH)	29.3 MPH 47.1 KPH	43.9 MPH 70.6 KPH	58.5 MPH 89.8 KPH	73.1 MPH 117.6 KPH	87.8 MPH 141.3 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
4th Overdrive (LH)	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	2.7 MPH 4.3 KPH	4.1 MPH 6.6 KPH	5.5 MPH 8.8 KPH	6.8 MPH 10.9 KPH	8.2 MPH 13.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.9 MPH 15.9 KPH	14.9 MPH 24.0 KPH	19.8 MPH 31.9 KPH	24.8 MPH 39.9 KPH	29.7 MPH 47.8 KPH
2nd	15.7 MPH 25.3 KPH	23.6 MPH 38.0 KPH	31.5 MPH 50.7 KPH	39.4 MPH 63.4 KPH	47.2 MPH 76.0 KPH
3rd	24.7 MPH 39.7 KPH	37.0 MPH 59.5 KPH	49.4 MPH 79.5 KPH	61.7 MPH 99.3 KPH	74.1 MPH 119.2 KPH
3rd Overdrive (LH)	30.1 MPH 48.4 KPH	45.2 MPH 72.7 KPH	60.2 MPH 96.9 KPH	75.2 MPH 121.0 KPH	90.3 MPH 145.3 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
4th Overdrive (LH)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	2.8 MPH 4.5 KPH	4.2 MPH 6.8 KPH	5.6 MPH 9.0 KPH	7.0 MPH 11.3 KPH	8.5 MPH 13.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.4 MPH 16.7 KPH	15.6 MPH 25.1 KPH	20.8 MPH 33.5 KPH	26.0 MPH 41.8 KPH	31.2 MPH 50.2 KPH
2nd	16.5 MPH 26.5 KPH	24.8 MPH 39.9 KPH	33.0 MPH 53.1 KPH	41.3 MPH 66.5 KPH	49.5 MPH 79.7 KPH
3rd	25.9 MPH 41.7 KPH	38.8 MPH 54.4 KPH	51.8 MPH 83.4 KPH	64.7 MPH 104.1 KPH	77.7 MPH 125.0 KPH
3rd Overdrive (LH)	31.6 MPH 50.8 KPH	47.3 MPH 76.1 KPH	63.2 MPH 101.7 KPH	79.0 MPH 127.1 KPH	94.8 MPH 152.6 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
4th Overdrive (LH)	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	3.0 MPH 4.8 KPH	4.4 MPH 7.1 KPH	5.9 MPH 9.5 KPH	7.4 MPH 11.9 KPH	8.9 MPH 14.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset (Autogear Part # CWP033)[1] Hardy-Spicer Banjo-type Rear Axle, (BMC Part # BTB1244)[2] Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.9 MPH 17.5 KPH	16.5 MPH 26.5 KPH	22.0 MPH 35.4 KPH	27.5 MPH 36.2 KPH	33.0 MPH 53.1 KPH
2nd	17.4 MPH 28.0 KPH	26.2 MPH 42.2 KPH	34.9 MPH 56.2 KPH	43.6 MPH 70.2 KPH	52.3 MPH 84.2 KPH
3rd	27.4 MPH 44.1 KPH	41.0 MPH 66.0 KPH	54.7 MPH 88.0 KPH	68.4 MPH 110.1 KPH	82.1 MPH 132.1 KPH
3rd Overdrive (LH)	33.4 MPH 53.7 KPH	50.0 MPH 80.5 KPH	66.7 MPH 107.3 KPH	83.4 MPH 134.2 KPH	100.1 MPH 161.1 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
4th Overdrive (LH)	46.1 MPH 74.2 KPH	69.2 MPH 11.42 KPH	92.2 MPH 148.42 KPH	115.2 MPH 185.42 KPH	138.3 MPH 222.62 KPH
Reverse	3.1 MPH 5.0 KPH	4.7 MPH 7.6 KPH	6.2 MPH 10.0 KPH	7.8 MPH 12.5 KPH	9.4 MPH 15.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.4 MPH 18.3 KPH	17.0 MPH 27.4 KPH	22.7 MPH 36.5 KPH	28.4 MPH 45.7 KPH	34.1 MPH 54.9 KPH
2nd	18.0 MPH 29.0 KPH	27.0 MPH 43.4 KPH	36.1 MPH 58.1 KPH	45.1 MPH 72.6 KPH	54.1 MPH 87.1 KPH
3rd	28.3 MPH 45.5 KPH	42.4 MPH 68.2 KPH	56.5 MPH 90.9 KPH	70.7 MPH 113.8 KPH	84.8 MPH 136.5 KPH
3rd Overdrive (LH)	34.5 MPH 55.5 KPH	51.7 MPH 83.2 KPH	69.0 MPH 111.0 KPH	86.2 MPH 138.7 KPH	103.5 MPH 166.6 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
4th Overdrive (LH)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	3.2 MPH 5.1 KPH	4.8 MPH 7.7 KPH	6.5 MPH 10.5 KPH	8.1 MPH 13.0 KPH	9.7 MPH 15.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	11.5 MPH 18.5 KPH	17.2 MPH 27.7 KPH	23.0 MPH 37.0 KPH	28.7 MPH 46.2 KPH	34.5 MPH 55.5 KPH
2nd	18.2 MPH 29.3 KPH	27.3 MPH 43.9 KPH	36.5 MPH 58.7 KPH	45.6 MPH 73.4 KPH	54.7 MPH 88.0 KPH
3rd	28.6 MPH 46.0 KPH	42.9 MPH 69.0 KPH	57.2 MPH 92.0 KPH	71.5 MPH 115.1 KPH	85.8 MPH 138.1 KPH
3rd Overdrive (LH)	34.9 MPH 56.2 KPH	52.3 MPH 84.2 KPH	69.7 MPH 112.2 KPH	87.2 MPH 140.3 KPH	104.6 MPH 168.3 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
4th Overdrive (LH)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	3.3 MPH 5.3 KPH	4.9 MPH 7.9 KPH	6.5 MPH 10.5 KPH	8.2 MPH 13.2 KPH	9.8 MPH 15.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.5 MPH 20.1 KPH	18.7 MPH 30.1 KPH	24.9 MPH 40.1 KPH	31.1 MPH 50.0 KPH	37.3 MPH 60.0 KPH
2nd	19.5 MPH 31.4 KPH	29.3 MPH 47.1 KPH	39.0 MPH 62.8 KPH	48.8 MPH 78.5 KPH	58.6 MPH 94.3 KPH
3rd	30.1 MPH 48.4 KPH	45.9 MPH 73.9 KPH	61.2 MPH 98.5 KPH	76.5 MPH 123.6 KPH	91.8 MPH 147.7 KPH
3rd Overdrive (LH)	37.3 MPH 60.0 KPH	56.0 MPH 90.1 KPH	74.7 MPH 120.2 KPH	93.3MPH 150.1 KPH	112.0 MPH 180.2 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
4th Overdrive (LH)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	3.5 MPH 5.6 KPH	5.2 MPH 8.4 KPH	7.0 MPH 11.3 KPH	8.7 MPH 14.0 KPH	10.5 MPH 16.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.2 MPH 21.2 KPH	19.9 MPH 32.0 KPH	26.5 MPH 42.6 KPH	33.1 MPH 53.3 KPH	39.7 MPH 63.9 KPH
2nd	21.0 MPH 33.8 KPH	31.5 MPH 50.7 KPH	42.0 MPH 67.6 KPH	52.6 MPH 84.6 KPH	63.1 MPH 101.5 KPH
3rd	33.0 MPH 53.1 KPH	49.4 MPH 79.5 KPH	65.9 MPH 106.1 KPH	82.4 MPH 132.6 KPH	98.9 MPH 159.2 KPH
3rd Overdrive (LH)	40.2 MPH 64.7 KPH	60.3 MPH 97.0 KPH	80.4 MPH 129.4 KPH	100.5 MPH 161.7 KPH	120.6 MPH 194.1 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
4th Overdrive (LH)	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	1.5 MPH 2.4 KPH	2.3 MPH 3.7 KPH	3.1 MPH 5.0 KPH	3.9 MPH 6.3 KPH	4.6 MPH 7.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

1975-1976 (Mid MKII) (Side Fill Version) Transmission, w/ LH-type Overdrive:

1st	3:036 : 1
2nd	2.167 : 1
3rd	1.382 : 1
3rd Overdrive	1.133 : 1
4th	1.000 : 1
4th Overdrive	0.820 : 1
Reverse	12.098 : 1

These gear ratios create the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	.869	859 RPM	2,391 RPM	1,572 RPM	3,928 RPM
2nd-3rd	.785	1,087 RPM	2,163 RPM	1,992 RPM	3,508 RPM
3rd-3rd O>D>	.882	586 RPM	2,664 RPM	991 RPM	4,509 RPM
3rd-4th	.382	829 RPM	2,421 RPM	1,520 RPM	3,980 RPM
3rd O.D.-4th O.D.	1.133	382 RPM	2,868 RPM	646 RPM	4,854 RPM
4th-4th O.D.	.180	585 RPM	2,665 RPM	990 RPM	4,510 RPM
3rd O.D.-4th O.D.	.313	586 RPM	2,664 RPM	991 RPM	4,509 RPM

These gearsets can be found as Original Equipment in transmission assemblies with engines whose engine numbers start with:

18V/797, 18V/798, 18V/801, 18V/802, 18V/836, 18V/837, 18V/846, 18V/847

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	9.0 MPH 14.5 KPH	13.5 MPH 21.7 KPH	18.0 MPH 29.0 KPH	22.5 MPH 36.2 KPH	27.0 MPH 43.4 KPH
2nd	12.5 MPH 20.1 KPH	18.8 MPH 30.3 KPH	25.1 MPH 40.4 KPH	31.3 MPH 50.4 KPH	37.6 MPH 60.5 KPH
3rd	19.7 MPH 31.7 KPH	29.6 MPH 47.6 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	56.2 MPH 90.4 KPH
3rd Overdrive (LH)	24.1 MPH 38.8 KPH	36.1 MPH 58.1 KPH	48.2 MPH 77.6 KPH	60.2 MPH 96.9 KPH	72.3 MPH 116.4 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
4th Overdrive (LH)	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	2.3 MPH 3.7 KPH	3.4 MPH 5.5 KPH	4.5 MPH 7.2 KPH	5.6 MPH 9.0 KPH	6.8 MPH 10.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.5 MPH 15.3 KPH	14.2 MPH 22.8 KPH	18.9 MPH 30.4 KPH	23.6 MPH 38.0 KPH	28.3 MPH 45.5 KPH
2nd	13.2 MPH 21.2 KPH	19.9 MPH 32.0 KPH	26.5 MPH 42.6 KPH	33.1 MPH 53.3 KPH	39.7 MPH 63.9 KPH
3rd	20.8 MPH 33.5 KPH	31.1 MPH 50.0 KPH	41.5 MPH 66.8 KPH	51.9 MPH 83.5 KPH	62.3 MPH 100.3 KPH
3rd Overdrive (LH)	25.3 MPH 40.7 KPH	38.0 MPH 61.1 KPH	50.6 MPH 81.4 KPH	63.3 MPH 101.9 KPH	76.0 MPH 122.3 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
4th Overdrive (LH)	32.1MPH 51.7 KPH	48.2 MPH 77.6 KPH	70.0 MPH 112.6 KPH	80.3 MPH 129.2 KPH	96.4 MPH 155.1 KPH
Reverse	2.4 MPH 3.9 KPH	3.6 MPH 5.8 KPH	4.7 MPH 7.6 KPH	5.9 MPH 9.5 KPH	7.1 MPH 11.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.1 MPH 16.2 KPH	15.2 MPH 24.5 KPH	20.3 MPH 32.7 KPH	25.3 MPH 40.7 KPH	30.3 MPH 48.8 KPH
2nd	14.2 MPH 22.8 KPH	21.3 MPH 34.3 KPH	28.4 MPH 45.7 KPH	35.5 MPH 57.1 KPH	42.6MPH 68.6 KPH
3rd	22.2 MPH 35.7 KPH	33.4 MPH 53.7 KPH	44.5 MPH 71.6 KPH	55.6 MPH 89.5 KPH	66.7 MPH 107.3 KPH
3rd Overdrive (LH)	27.1 MPH 43.6 KPH	40.7 MPH 65.5 KPH	54.3 MPH 87.4 KPH	67.8 MPH 109.0 KPH	81.4 MPH 131.0 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
4th Overdrive (LH)	37.5 MPH 60.3 KPH	56.2 MPH 90.4 KPH	75.0 MPH 120.7 KPH	93.7 MPH 150.8 KPH	112.5 MPH 181.0 KPH
Reverse	2.5 MPH 4.0 KPH	3.8 MPH 6.1 KPH	5.1 MPH 8.2 KPH	6.3 MPH 10.1 KPH	7.6 MPH 12.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.7 MPH 17.2 KPH	16.1 MPH 25.9 KPH	21.4 MPH 34.4 KPH	26.8 MPH 43.1 KPH	32.1 MPH 51.1 KPH
2nd	15.0 MPH 24.1 KPH	22.5 MPH 36.2 KPH	30.0 MPH 48.3 KPH	37.5 MPH 60.3 KPH	45.0 MPH 72.4 KPH
3rd	23.5 MPH 37.8 KPH	35.3 MPH 56.8 KPH	47.1 MPH 75.8 KPH	58.8 MPH 94.6 KPH	70.6 MPH 113.6 KPH
3rd Overdrive (LH)	28.7 MPH 46.2 KPH	43.1 MPH 69.4 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
4th Overdrive (LH)	39.7 MPH 63.8 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	2.7 MPH 4.3 KPH	4.0 MPH 6.4 KPH	5.4 MPH 8.7 KPH	6.7 MPH 10.8 KPH	8.1 MPH 13.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.9 MPH 17.5 KPH	16.4 MPH 26.4 KPH	21.8 MPH 35.1 KPH	27.3 MPH 43.9 KPH	32.7 MPH 52.6 KPH
2nd	15.3 MPH 24.6 KPH	33.0 MPH 53.1 KPH	30.6 MPH 49.2 KPH	38.2 MPH 61.5 KPH	45.9 MPH 73.9 KPH
3rd	24.0 MPH 38.6 KPH	36.0 MPH 57.9 KPH	48.0MPH 77.2 KPH	60.0 MPH 96.6 KPH	72.0 MPH 115.9 KPH
3rd Overdrive (LH)	29.3 MPH 47.1 KPH	43.9 MPH 70.6 KPH	58.5 MPH 89.8 KPH	73.1 MPH 117.6 KPH	87.8 MPH 141.3 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
4th Overdrive (LH)	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	2.7 MPH 4.3 KPH	4.1 MPH 6.6 KPH	5.5 MPH 8.8 KPH	6.8 MPH 10.9 KPH	8.2 MPH 13.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion gearset (Special Tuning Part # ATB 7240)[1], For The Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	11.2 MPH 18.0 KPH	16.9 MPH 27.2 KPH	22.5 MPH 36.2 KPH	28.1 MPH 45.2 KPH	33.7 MPH 54.2 KPH
2nd	15.7 MPH 25.3 KPH	23.6 MPH 38.0 KPH	31.5 MPH 50.7 KPH	39.4 MPH 63.4 KPH	47.2 MPH 76.0 KPH
3rd	24.7 MPH 39.7 KPH	37.0 MPH 59.5 KPH	49.4 MPH 79.5 KPH	61.7 MPH 99.3 KPH	74.1 MPH 119.2 KPH
3rd Overdrive (LH)	30.1 MPH 48.4 KPH	45.2 MPH 72.7 KPH	60.2 MPH 96.9 KPH	75.2 MPH 121.0 KPH	90.3 MPH 145.3 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
4th Overdrive (LH)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	2.8 MPH 4.5 KPH	4.2 MPH 6.8 KPH	5.6 MPH 9.0 KPH	7.0 MPH 11.3 KPH	8.5 MPH 13.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.8 MPH 19.0 KPH	17.7 MPH 28.5 KPH	23.6 MPH 38.0 KPH	29.5 MPH 47.5 KPH	35.4 MPH 57.0 KPH
2nd	16.5 MPH 26.5 KPH	24.8 MPH 39.9 KPH	33.0 MPH 53.1 KPH	41.3 MPH 66.5 KPH	49.5 MPH 79.7 KPH
3rd	25.9 MPH 41.7 KPH	38.8 MPH 54.4 KPH	51.8 MPH 83.4 KPH	64.7 MPH 104.1 KPH	77.7 MPH 125.0 KPH
3rd Overdrive (LH)	31.6 MPH 50.8 KPH	47.3 MPH 76.1 KPH	63.2 MPH 101.7 KPH	79.0 MPH 127.1 KPH	94.8 MPH 152.6 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
4th Overdrive (LH)	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	3.0 MPH 4.8 KPH	4.4 MPH 7.1 KPH	5.9 MPH 9.5 KPH	7.4 MPH 11.9 KPH	8.9 MPH 14.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset (Autogear Part # CWP033)[1] Hardy-Spicer Banjo-type Rear Axle, (BMC Part # BTB1244)[2] Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.4 MPH 19.9 KPH	18.7 MPH 30.1 KPH	24.9 MPH 40.1 KPH	31.1 MPH 50.0 KPH	37.3 MPH 60.0 KPH
2nd	17.4 MPH 28.0 KPH	26.2 MPH 42.2 KPH	34.9 MPH 56.2 KPH	43.6 MPH 70.2 KPH	52.3 MPH 84.2 KPH
3rd	27.4 MPH 44.1 KPH	41.0 MPH 66.0 KPH	54.7 MPH 88.0 KPH	68.4 MPH 110.1 KPH	82.1 MPH 132.1 KPH
3rd Overdrive (LH)	33.4 MPH 53.7 KPH	50.0 MPH 80.5 KPH	66.7 MPH 107.3 KPH	83.4 MPH 134.2 KPH	100.1 MPH 161.1 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
4th Overdrive (LH)	46.1 MPH 74.2 KPH	69.2 MPH 11.42 KPH	92.2 MPH 148.42 KPH	115.2 MPH 185.42 KPH	138.3 MPH 222.62 KPH
Reverse	3.1 MPH 5.0 KPH	4.7 MPH 7.6 KPH	6.2 MPH 10.0 KPH	7.8 MPH 12.5 KPH	9.4 MPH 15.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.9 MPH 20.8 KPH	19.3 MPH 31.0 KPH	25.7 MPH 41.4 KPH	32.2 MPH 51.8 KPH	38.6 MPH 62.1 KPH
2nd	18.0 MPH 29.0 KPH	27.0 MPH 43.4 KPH	36.1 MPH 58.1 KPH	45.1 MPH 72.6 KPH	54.1 MPH 87.1 KPH
3rd	28.3 MPH 45.5 KPH	42.4 MPH 68.2 KPH	56.5 MPH 90.9 KPH	70.7 MPH 113.8 KPH	84.8 MPH 136.5 KPH
3rd Overdrive (LH)	34.5 MPH 55.5 KPH	51.7 MPH 83.2 KPH	69.0 MPH 111.0 KPH	86.2 MPH 138.7 KPH	103.5 MPH 166.6 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
4th Overdrive (LH)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	3.2 MPH 5.1 KPH	4.8 MPH 7.7 KPH	6.5 MPH 10.5 KPH	8.1 MPH 13.0 KPH	9.7 MPH 15.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	13.0 MPH 20.9 KPH	19.5 MPH 31.4 KPH	26.0 MPH 41.8 KPH	32.5 MPH 52.3 KPH	39.0 MPH 62.8 KPH
2nd	18.2 MPH 29.3 KPH	27.3 MPH 43.9 KPH	36.5 MPH 58.7 KPH	45.6 MPH 73.4 KPH	54.7 MPH 88.0 KPH
3rd	28.6 MPH 46.0 KPH	42.9 MPH 69.0 KPH	57.2 MPH 92.0 KPH	71.5 MPH 115.1 KPH	85.8 MPH 138.1 KPH
3rd Overdrive (LH)	34.9 MPH 56.2 KPH	52.3 MPH 84.2 KPH	69.7 MPH 112.2 KPH	87.2 MPH 140.3 KPH	104.6 MPH 168.3 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
4th Overdrive (LH)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	3.3 MPH 5.3 KPH	4.9 MPH 7.9 KPH	6.5 MPH 10.5 KPH	8.2 MPH 13.2 KPH	9.8 MPH 15.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900) [1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.9 MPH 22.4 KPH	20.9 MPH 33.6 KPH	27.9 MPH 44.9 KPH	34.8 MPH 56.0 KPH	41.8 MPH 67.3 KPH
2nd	19.5 MPH 31.4 KPH	29.3 MPH 47.1 KPH	39.0 MPH 62.8 KPH	48.8 MPH 78.5 KPH	58.6 MPH 94.3 KPH
3rd	30.1 MPH 48.4 KPH	45.9 MPH 73.9 KPH	61.2 MPH 98.5 KPH	76.5 MPH 123.6 KPH	91.8 MPH 147.7 KPH
3rd Overdrive (LH)	37.3 MPH 60.0 KPH	56.0 MPH 90.1 KPH	74.7 MPH 120.2 KPH	93.3MPH 150.1 KPH	112.0 MPH 180.2 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
4th Overdrive (LH)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	3.5 MPH 5.6 KPH	5.2 MPH 8.4 KPH	7.0 MPH 11.3 KPH	8.7 MPH 14.0 KPH	10.5 MPH 16.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	15.0 MPH 24.1 KPH	22.5 MPH 36.2 KPH	30.0 MPH 48.3 KPH	37.5 MPH 60.3 KPH	45.0 MPH 72.4 KPH
2nd	21.0 MPH 33.8 KPH	31.5 MPH 50.7 KPH	42.0 MPH 67.6 KPH	52.6 MPH 84.6 KPH	63.1 MPH 101.5 KPH
3rd	33.0 MPH 53.1 KPH	49.4 MPH 79.5 KPH	65.9 MPH 106.1 KPH	82.4 MPH 132.6 KPH	98.9 MPH 159.2 KPH
3rd Overdrive (LH)	40.2 MPH 64.7 KPH	60.3 MPH 97.0 KPH	80.4 MPH 129.4 KPH	100.5 MPH 161.7 KPH	120.6 MPH 194.1 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
4th Overdrive (LH)	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	1.5 MPH 2.4 KPH	2.3 MPH 3.7 KPH	3.1 MPH 5.0 KPH	3.9 MPH 6.3 KPH	4.6 MPH 7.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

1977-1980 (Late MKII) (Side Fill Version) Transmission, w/ LH-type Overdrive:

1st	3.333 : 1
2nd	2.167 : 1
3rd	1.382 : 1
3rd Overdrive	1.133 :1
4th	1.000 : 1
4th Overdrive	0.820 : 1
Reverse	12.098 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.166	1,050 RPM	2,200 RPM	1,924 RPM	3,576 RPM
2nd-3rd	.785	1,087 RPM	2,163 RPM	1,992 RPM	3,508 RPM
3rd-3rd O.D.	.882	586 RPM	2,664 RPM	991 RPM	4,509 RPM
3rd-4th	.382	829 RPM	2,421 RPM	1,520 RPM	3,980 RPM
3rd O.D.-4th	1.133	382 RPM	2,868 RPM	646 RPM	4,854 RPM
4th-4th O.D.	.180	585 RPM	2,665 RPM	990 RPM	4,510 RPM

3rd O.D.-4th O.D.	.313	586 RPM	2,664 RPM	991 RPM	4,509 RPM

These gearsets can be found as Original Equipment in transmission assemblies with engines whose engine numbers start with:

18V846, 18V847, 18V883, 18V884, 18V890, 18V891, 18V892, 18V893

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], for the Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.2 MPH 13.2 KPH	12.3 MPH 19.8 KPH	16.4 MPH 26.4 KPH	20.5 MPH 33.0 KPH	24.6 MPH 39.6 KPH
2nd	12.5 MPH 20.1 KPH	18.8 MPH 30.3 KPH	25.1 MPH 40.4 KPH	31.3 MPH 50.4 KPH	37.6 MPH 60.5 KPH
3rd	19.7 MPH 31.7 KPH	29.6 MPH 47.6 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	56.2 MPH 90.4 KPH
3rd Overdrive (LH)	24.1 MPH 38.8 KPH	36.1 MPH 58.1 KPH	48.2 MPH 77.6 KPH	60.2 MPH 96.9 KPH	72.3 MPH 116.4 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
4th Overdrive (LH)	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	2.3 MPH 3.7 KPH	3.4 MPH 5.5 KPH	4.5 MPH 7.2 KPH	5.6 MPH 9.0 KPH	6.8 MPH 10.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.6 MPH 13.8 KPH	12.9 MPH 20.8 KPH	17.2 MPH 27.7 KPH	21.5 MPH 34.6 KPH	25.8 MPH 41.5 KPH
2nd	13.2 MPH 21.2 KPH	19.9 MPH 32.0 KPH	26.5 MPH 42.6 KPH	33.1 MPH 53.3 KPH	39.7 MPH 63.9 KPH
3rd	20.8 MPH 33.5 KPH	31.1 MPH 50.0 KPH	41.5 MPH 66.8 KPH	51.9 MPH 83.5 KPH	62.3 MPH 100.3 KPH
3rd Overdrive (LH)	25.3 MPH 40.7 KPH	38.0 MPH 61.1 KPH	50.6 MPH 81.4 KPH	63.3 MPH 101.9 KPH	76.0 MPH 122.3 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
4th Overdrive (LH)	32.1MPH 51.7 KPH	48.2 MPH 77.6 KPH	70.0 MPH 112.6 KPH	80.3 MPH 129.2 KPH	96.4 MPH 155.1 KPH
Reverse	2.4 MPH 3.9 KPH	3.6 MPH 5.8 KPH	4.7 MPH 7.6 KPH	5.9 MPH 9.5 KPH	7.1 MPH 11.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.2 MPH 14.6 KPH	13.8 MPH 22.2 KPH	18.4 MPH 29.6 KPH	23.1 MPH 37.2 KPH	27.7 MPH 44.6 KPH
2nd	14.2 MPH 22.8 KPH	21.3 MPH 34.3 KPH	28.4 MPH 45.7 KPH	35.5 MPH 57.1 KPH	42.6MPH 68.6 KPH
3rd	22.2 MPH 35.7 KPH	33.4 MPH 53.7 KPH	44.5 MPH 71.6 KPH	55.6 MPH 89.5 KPH	66.7 MPH 107.3 KPH
3rd Overdrive (LH)	27.1 MPH 43.6 KPH	40.7 MPH 65.5 KPH	54.3 MPH 87.4 KPH	67.8 MPH 109.0 KPH	81.4 MPH 131.0 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
4th Overdrive (LH)	37.5 MPH 60.3 KPH	56.2 MPH 90.4 KPH	75.0 MPH 120.7 KPH	93.7 MPH 150.8 KPH	112.5 MPH 181.0 KPH
Reverse	2.5 MPH 4.0 KPH	3.8 MPH 6.1 KPH	5.1 MPH 8.2 KPH	6.3 MPH 10.1 KPH	7.6 MPH 12.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.8 MPH 15.8 KPH	14.6 MPH 23.5 KPH	19.5 MPH 31.4 KPH	24.4 MPH 39.3 KPH	29.3 MPH 47.1 KPH
2nd	15.0 MPH 24.1 KPH	22.5 MPH 36.2 KPH	30.0 MPH 48.3 KPH	37.5 MPH 60.3 KPH	45.0 MPH 72.4 KPH
3rd	23.5 MPH 37.8 KPH	35.3 MPH 56.8 KPH	47.1 MPH 75.8 KPH	58.8 MPH 94.6 KPH	70.6 MPH 113.6 KPH
3rd Overdrive (LH)	28.7 MPH 46.2 KPH	43.1 MPH 69.4 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
4th Overdrive (LH)	39.7 MPH 63.8 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	2.7 MPH 4.3 KPH	4.0 MPH 6.4 KPH	5.4 MPH 8.7 KPH	6.7 MPH 10.8 KPH	8.1 MPH 13.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.9 MPH 15.9 KPH	14.9 MPH 24.0 KPH	19.9 MPH 32.0 KPH	24.9 MPH 40.0 KPH	29.8 MPH 48.0 KPH
2nd	15.3 MPH 24.6 KPH	33.0 MPH 53.1 KPH	30.6 MPH 49.2 KPH	38.2 MPH 61.5 KPH	45.9 MPH 73.9 KPH
3rd	24.0 MPH 38.6 KPH	36.0 MPH 57.9 KPH	48.0MPH 77.2 KPH	60.0 MPH 96.6 KPH	72.0 MPH 115.9 KPH
3rd Overdrive (LH)	29.3 MPH 47.1 KPH	43.9 MPH 70.6 KPH	58.5 MPH 89.8 KPH	73.1 MPH 117.6 KPH	87.8 MPH 141.3 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
4th Overdrive (LH)	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	2.7 MPH 4.3 KPH	4.1 MPH 6.6 KPH	5.5 MPH 8.8 KPH	6.8 MPH 10.9 KPH	8.2 MPH 13.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.2 MPH 16.4 KPH	15.3 MPH 24.6 KPH	20.5 MPH 33.0 KPH	25.6 MPH 41.2 KPH	30.7 MPH 49.4 KPH
2nd	15.7 MPH 25.3 KPH	23.6 MPH 38.0 KPH	31.5 MPH 50.7 KPH	39.4 MPH 63.4 KPH	47.2 MPH 76.0 KPH
3rd	24.7 MPH 39.7 KPH	37.0 MPH 59.5 KPH	49.4 MPH 79.5 KPH	61.7 MPH 99.3 KPH	74.1 MPH 119.2 KPH
3rd Overdrive (LH)	30.1 MPH 48.4 KPH	45.2 MPH 72.7 KPH	60.2 MPH 96.9 KPH	75.2 MPH 121.0 KPH	90.3 MPH 145.3 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
4th Overdrive (LH)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	2.8 MPH 4.5 KPH	4.2 MPH 6.8 KPH	5.6 MPH 9.0 KPH	7.0 MPH 11.3 KPH	8.5 MPH 13.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.7 MPH 17.2 KPH	16.1 MPH 25.9 KPH	21.5 MPH 34.6 KPH	26.8 MPH 43.1 KPH	32.2 MPH 51.8 KPH
2nd	16.5 MPH 26.5 KPH	24.8 MPH 39.9 KPH	33.0 MPH 53.1 KPH	41.3 MPH 66.5 KPH	49.5 MPH 79.7 KPH
3rd	25.9 MPH 41.7 KPH	38.8 MPH 54.4 KPH	51.8 MPH 83.4 KPH	64.7 MPH 104.1 KPH	77.7 MPH 125.0 KPH
3rd Overdrive (LH)	31.6 MPH 50.8 KPH	47.3 MPH 76.1 KPH	63.2 MPH 101.7 KPH	79.0 MPH 127.1 KPH	94.8 MPH 152.6 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
4th Overdrive (LH)	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	3.0 MPH 4.8 KPH	4.4 MPH 7.1 KPH	5.9 MPH 9.5 KPH	7.4 MPH 11.9 KPH	8.9 MPH 14.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset (Autogear Part # CWP033)[1] Hardy-Spicer Banjo-type Rear Axle, (BMC Part # BTB1244)[2] Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.3 MPH 18.2 KPH	17.0 MPH 27.4 KPH	22.7 MPH 36.5 KPH	28.4 MPH 45.7 KPH	34.0 MPH 54.7 KPH
2nd	17.4 MPH 28.0 KPH	26.2 MPH 42.2 KPH	34.9 MPH 56.2 KPH	43.6 MPH 70.2 KPH	52.3 MPH 84.2 KPH
3rd	27.4 MPH 44.1 KPH	41.0 MPH 66.0 KPH	54.7 MPH 88.0 KPH	68.4 MPH 110.1 KPH	82.1 MPH 132.1 KPH
3rd Overdrive (LH)	33.4 MPH 53.7 KPH	50.0 MPH 80.5 KPH	66.7 MPH 107.3 KPH	83.4 MPH 134.2 KPH	100.1 MPH 161.1 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
4th Overdrive (LH)	46.1 MPH 74.2 KPH	69.2 MPH 11.42 KPH	92.2 MPH 148.42 KPH	115.2 MPH 185.42 KPH	138.3 MPH 222.62 KPH
Reverse	3.1 MPH 5.0 KPH	4.7 MPH 7.6 KPH	6.2 MPH 10.0 KPH	7.8 MPH 12.5 KPH	9.4 MPH 15.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.7 MPH 18.8 KPH	17.6 MPH 28.3 KPH	23.5 MPH 37.8 KPH	29.3 MPH 47.1 KPH	35.2 MPH 56.6 KPH
2nd	18.0 MPH 29.0 KPH	27.0 MPH 43.4 KPH	36.1 MPH 58.1 KPH	45.1 MPH 72.6 KPH	54.1 MPH 87.1 KPH
3rd	28.3 MPH 45.5 KPH	42.4 MPH 68.2 KPH	56.5 MPH 90.9 KPH	70.7 MPH 113.8 KPH	84.8 MPH 136.5 KPH
3rd Overdrive (LH)	34.5 MPH 55.5 KPH	51.7 MPH 83.2 KPH	69.0 MPH 111.0 KPH	86.2 MPH 138.7 KPH	103.5 MPH 166.6 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
4th Overdrive (LH)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	3.2 MPH 5.1 KPH	4.8 MPH 7.7 KPH	6.5 MPH 10.5 KPH	8.1 MPH 13.0 KPH	9.7 MPH 15.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.8 MPH 19.0 KPH	17.8 MPH 28.6 KPH	23.7 MPH 38.1 KPH	29.6 MPH 47.6 KPH	35.6 MPH 57.3 KPH
2nd	18.2 MPH 29.3 KPH	27.3 MPH 43.9 KPH	36.5 MPH 58.7 KPH	45.6 MPH 73.4 KPH	54.7 MPH 88.0 KPH
3rd	28.6 MPH 46.0 KPH	42.9 MPH 69.0 KPH	57.2 MPH 92.0 KPH	71.5 MPH 115.1 KPH	85.8 MPH 138.1 KPH
3rd Overdrive (LH)	34.9 MPH 56.2 KPH	52.3 MPH 84.2 KPH	69.7 MPH 112.2 KPH	87.2 MPH 140.3 KPH	104.6 MPH 168.3 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
4th Overdrive (LH)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	3.3 MPH 5.3 KPH	4.9 MPH 7.9 KPH	6.5 MPH 10.5 KPH	8.2 MPH 13.2 KPH	9.8 MPH 15.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.7 MPH 20.4 KPH	19.0 MPH 30.6 KPH	25.4 MPH 40.9 KPH	31.7 MPH 51.0 KPH	38.1 MPH 61.3 KPH
2nd	19.5 MPH 31.4 KPH	29.3 MPH 47.1 KPH	39.0 MPH 62.8 KPH	48.8 MPH 78.5 KPH	58.6 MPH 94.3 KPH
3rd	30.1 MPH 48.4 KPH	45.9 MPH 73.9 KPH	61.2 MPH 98.5 KPH	76.5 MPH 123.6 KPH	91.8 MPH 147.7 KPH
3rd Overdrive (LH)	37.3 MPH 60.0 KPH	56.0 MPH 90.1 KPH	74.7 MPH 120.2 KPH	93.3MPH 150.1 KPH	112.0 MPH 180.2 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
4th Overdrive (LH)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	3.5 MPH 5.6 KPH	5.2 MPH 8.4 KPH	7.0 MPH 11.3 KPH	8.7 MPH 14.0 KPH	10.5 MPH 16.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	13.7 MPH 22.0 KPH	20.5 MPH 33.0 KPH	27.3 MPH 43.9 KPH	34.2 MPH 55.0 KPH	41.0 MPH 66.0 KPH
2nd	21.0 MPH 33.8 KPH	31.5 MPH 50.7 KPH	42.0 MPH 67.6 KPH	52.6 MPH 84.6 KPH	63.1 MPH 101.5 KPH
3rd	33.0 MPH 53.1 KPH	49.4 MPH 79.5 KPH	65.9 MPH 106.1 KPH	82.4 MPH 132.6 KPH	98.9 MPH 159.2 KPH
3rd Overdrive (LH)	40.2 MPH 64.7 KPH	60.3 MPH 97.0 KPH	80.4 MPH 129.4 KPH	100.5 MPH 161.7 KPH	120.6 MPH 194.1 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
4th Overdrive (LH)	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	1.5 MPH 2.4 KPH	2.3 MPH 3.7 KPH	3.1 MPH 5.0 KPH	3.9 MPH 6.3 KPH	4.6 MPH 7.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

The ratios that were chosen to be used in the mass-market MGBs were obviously chosen less for performance-oriented driving and more for Daily Driver applications: A very low first gear ratio (Excepting the 1975-1976 transmission) for starting off on a steep hill with a heavy load on board, a relatively low second gear ratio for puttering in downtown traffic, a relatively high jump into a third gear ratio suitable for urban roads, and a fourth gear ratio that was essentially an Overdrive for use on rural County Roads and State Highways back in the days before Interstate Highways (Motorways) with their higher posted speed limits. An optional Laycock de Normanville Overdrive unit was available for those who desired their cars to be appropriate for high-speed use. Because of its taller first gear ratio, the ratios used in the 1975-1976 transmission produce the smallest ratio jump between its first and second gears and as such is the best available choice of the mass production gear ratios that were used in the MGB for keeping an engine "on the boil" when it has been equipped with a camshaft lobe profile that will result in a narrower powerband. Not surprisingly, these gearsets are much sought-after by those who are performance-oriented drivers.

There are still other options for those seeking alternative gear ratios. The MGC used what was essentially the same four-synchro transmission design, of which there were two basic versions. These differ from the transmission of the MGB only in their bellhousings, the clutch, the clutch actuating fork, the clutch actuating fork boot, the clutch slave cylinder, the output flange, and the ratios of their gearsets. Everything else is the same as that found as Original Equipment in the contemporary MGB transmissions, and as such their complete gearsets are fully interchangeable with those of the MGB transmissions. The gear ratios of the gearsets used on the 1968 model without an Overdrive unit are the same as those for the 1968-1974 MGB. However, for the Overdrive unit-equipped 1968 model, as well as for all of the 1969 models, the gear ratios for first and second gear are unique to the MGC. Offering a greater amount of overlap, they are always of interest to those who are planning to use a camshaft lobe profile that will result in a narrower powerband:

MGC (Top Fill Version) Transmission, w/ LH-type Overdrive:

1st	2.980 : 1
2nd	2.058 : 1
3rd	1.382 : 1
3rd Overdrive	1.133 : 1
4th	1.000 : 1
4th Overdrive	0.820 : 1
Reverse	12.098 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	.932	928 RPM	2,322 RPM	1,702 RPM	3,798 RPM
2nd-3rd	.860	985 RPM	2,265 RPM	2,007 RPM	3,493 RPM
3rd-3rd O.D.	.882	586 RPM	2,664 RPM	991 RPM	4,509 RPM
3rd-4th	.382	829 RPM	2,421 RPM	1,520 RPM	3,980 RPM
3rd O.D.-4th	1.133	382 RPM	2,868 RPM	646 RPM	4,854 RPM
4th-4th O.D.	.180	585 RPM	2,665 RPM	990 RPM	4,510 RPM
3rd O.D.-4th O.D.	.313	586 RPM	2,664 RPM	991 RPM	4,509 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.2 MPH 14.8 KPH	13.7 MPH 22.1 KPH	18.3 MPH 29.4 KPH	22.9 MPH 36.8 KPH	27.5 MPH 44.3 KPH
2nd	12.2 MPH 19.6 KPH	18.3 MPH 29.4 KPH	24.4 MPH 39.3 KPH	30.4 MPH 48.9 KPH	36.5 MPH 58.7 KPH
3rd	19.7 MPH 31.7 KPH	29.6 MPH 47.6 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	59.2 MPH 95.3 KPH
3rd Overdrive (LH)	24.1 MPH 38.8 KPH	36.1 MPH 58.1 KPH	48.2 MPH 77.6 KPH	60.2 MPH 96.9 KPH	72.2 MPH 116.2 KPH
4th	27.3 MPH 43.9 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
4th Overdrive (LH)	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	2.3 MPH 3.7 KPH	3.4 MPH 5.5 KPH	4.5 MPH 7.2 KPH	5.6 MPH 9.0 KPH	6.8 MPH 10.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L) [1], Hardy-Spicer Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.6 MPH 15.5 KPH	14.4 MPH 23.2 KPH	19.3 MPH 31.1 KPH	24.1 MPH 38.8 KPH	28.9 MPH 46.5 KPH
2nd	13.9 MPH 22.4 KPH	20.9 MPH 33.6 KPH	27.9 MPH 44.9 KPH	34.9 MPH 56.2 KPH	41.8 MPH 67.3 KPH
3rd	20.8 MPH 33.5 KPH	31.1 MPH 50.0 KPH	41.5 MPH 66.8 KPH	51.9 MPH 83.5 KPH	62.3 MPH 100.3 KPH
3rd Overdrive (LH)	25.3 MPH 40.7 KPH	38.0 MPH 61.2 KPH	50.6 MPH 81.4 KPH	63.3 MPH 101.9 KPH	75.0 MPH 120.7 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.7 MPH 115.4 KPH	86.1 MPH 138.6 KPH
4th Overdrive (LH)	35.0 MPH 56.3 KPH	52.5 MPH 84.5 KPH	70.0 MPH 112.6 KPH	87.5 MPH 140.8 KPH	105.0 MPH 169.0 KPH
Reverse	2.4 MPH 3.9 KPH	3.6 MPH 5.8 KPH	4.7 MPH 7.6 KPH	5.9 MPH 9.5 KPH	7.1 MPH 11.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.3 MPH 16.6 KPH	15.5 MPH 25.3 KPH	20.6 MPH 33.3 KPH	25.8 MPH 41.5 KPH	30.9 MPH 49.8 KPH
2nd	14.9 MPH 24.0 KPH	22.4 MPH 35.8 KPH	29.9 MPH 48.1 KPH	37.3 MPH 60.1 KPH	44.8 MPH 72.1 KPH
3rd	22.2 MPH 35.7 KPH	33.4 MPH 53.7 KPH	44.5 MPH 71.6 KPH	55.6 MPH 89.5 KPH	66.7 MPH 107.3 KPH
3rd Overdrive (LH)	27.1 MPH 43.6 KPH	40.7 MPH 65.5 KPH	54.3 MPH 87.4 KPH	67.8 MPH 109.1 KPH	81.4 MPH 131.0 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.8 MPH 123.6 KPH	92.2 MPH 148.4 KPH
4th Overdrive (LH)	37.5 MPH 60.3 KPH	56.2 MPH 90.4 KPH	74.9 MPH 120.5 KPH	93.7 MPH 150.8 KPH	112.5 MPH 181.0 KPH
Reverse	2.5 MPH 4.0 KPH	3.8 MPH 6.1 KPH	5.1 MPH 8.2 KPH	6.3 MPH 10.1 KPH	7.6 MPH 12.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production

MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion gearset (BMC Part # 88G 283)[1], Hardy-Spicer Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.9 MPH 17.5 KPH	16.7 MPH 26.9 KPH	21.8 MPH 35.1 KPH	27.3 MPH 43.9 KPH	32.7 MPH 52.6 KPH
2nd	15.8 MPH 25.4 KPH	23.7 MPH 38.1 KPH	31.6 MPH 50.8 KPH	39.5 MPH 63.6 KPH	47.4 MPH 76.3 KPH
3rd	23.5 MPH 37.8 KPH	35.3 MPH 56.9 KPH	47.1 MPH 75.8 KPH	58.8 MPH 94.6 KPH	70.7 MPH 113.8 KPH
3rd Overdrive (LH)	28.7 MPH 46.2 KPH	43.1 MPH 69.4 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.1 KPH
4th Overdrive (LH)	39.7 MPH 63.9 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	2.7 MPH 4.3 KPH	4.0 MPH 6.4 KPH	5.4 MPH 8.7 KPH	6.7 MPH 10.8 KPH	8.1 MPH 13.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.1 MPH 17.9 KPH	16.7 MPH 26.9 KPH	22.2 MPH 35.7 KPH	27.8 MPH 44.7 KPH	33.4 MPH 53.7 KPH
2nd	16.1 MPH 25.9 KPH	24.2 MPH 38.9 KPH	32.2 MPH 51.8 KPH	40.3 MPH 64.9 KPH	48.3 MPH 77.7 KPH
3rd	24.0 MPH 38.6 KPH	36.0 MPH 57.9 KPH	48.0 MPH 77.2 KPH	60.0 MPH 96.6 KPH	72.0 MPH 115.9 KPH
3rd Overdrive (LH)	29.3 MPH 47.2 KPH	43.9 MPH 70.1 KPH	58.5 MPH 94.1 KPH	73.1 MPH 117.6 KPH	87.8 MPH 141.3 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 113.4 KPH	99.4 MPH 160.0 KPH
4th Overdrive (LH)	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	2.7 MPH 4.3 KPH	4.1 MPH 6.6 KPH	5.5 MPH 8.8 KPH	6.8 MPH 10.9 KPH	8.2 MPH 13.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.4 MPH 18.3 KPH	17.2 MPH 27.7 KPH	22.9 MPH 36.8 KPH	28.6 MPH 46.0 KPH	34.4 MPH 55.4 KPH
2nd	16.6 MPH 26.7 KPH	24.9 MPH 40.1 KPH	33.2 MPH 53.4 KPH	41.4 MPH 66.6 KPH	49.7 MPH 79.0 KPH
3rd	24.7 MPH 39.7 KPH	37.0 MPH 59.5 KPH	49.4 MPH 79.5 KPH	61.7 MPH 99.3 KPH	74.1 MPH 119.5 KPH
3rd Overdrive (LH)	30.2 MPH 48.6 KPH	45.2 MPH 72.7 KPH	60.2 MPH 96.9 KPH	75.3 MPH 121.2 KPH	90.3 MPH 145.3 KPH
4th	34.2 MPH 55.0 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
4th Overdrive (LH)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	2.8 MPH 4.5 KPH	4.2 MPH 6.8 KPH	5.6 MPH 9.0 KPH	7.0 MPH 11.3 KPH	8.5 MPH 13.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.0 MPH 19.3 KPH	18.0 MPH 29.0 KPH	24.0 MPH 38.6 KPH	30.0 MPH 48.3 KPH	36.0 MPH 57.9 KPH
2nd	17.4 MPH 28.0 KPH	26.0 MPH 41.8 KPH	34.8 MPH 56.0 KPH	43.5 MPH 70.0 KPH	52.2 MPH 84.0 KPH
3rd	25.8 MPH 41.5 KPH	38.8 MPH 62.4 KPH	51.8 MPH 83.4 KPH	64.7 MPH 104.1 KPH	77.7 MPH 125.0 KPH
3rd Overdrive (LH)	31.6 MPH 50.8 KPH	47.4 MPH 76.3 KPH	63.2 MPH 101.7 KPH	79.0 MPH 127.1 KPH	94.7 MPH 152.4 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.5 MPH 144.0 KPH	107.4 MPH 172.8 KPH
4th Overdrive (LH)	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	3.0 MPH 4.8 KPH	4.4 MPH 7.1 KPH	5.9 MPH 9.5 KPH	7.4 MPH 11.9 KPH	8.9 MPH 14.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset, (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.7 MPH 20.4 KPH	19.0 MPH 30.6 KPH	25.4 MPH 40.9 KPH	31.7 MPH 51.0 KPH	38.1 MPH 61.3 KPH
2nd	18.4 MPH 29.6 KPH	27.5 MPH 44.3 KPH	36.7 MPH 59.1 KPH	45.9 MPH 73.9 KPH	55.1 MPH 88.7 KPH
3rd	27.4 MPH 44.1 KPH	41.0 MPH 66.0 KPH	54.7 MPH 88.3 KPH	68.4 MPH 110.1 KPH	82.1 MPH 132.1 KPH
3rd Overdrive (LH)	33.4 MPH 53.7 KPH	50.1 MPH 80.6 KPH	66.7 MPH 107.3 KPH	83.4 MPH 134.2 KPH	100.1 MPH 161.1 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.7 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
4th Overdrive (LH)	46.1 MPH 74.2 KPH	69.2 MPH 11.4 KPH	92.2 MPH 148.4 KPH	115.2 MPH 185.4 KPH	138.3 MPH 222.6 KPH
Reverse	3.1 MPH 5.0 KPH	4.7 MPH 7.6 KPH	6.2 MPH 10.0 KPH	7.8 MPH 12.5 KPH	9.4 MPH 15.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.1 MPH 28.1 KPH	19.7 MPH 31.7 KPH	26.2 MPH 42.2 KPH	32.8 MPH 52.8 KPH	39.3 MPH 63.4 KPH
2nd	19.0 MPH 30.6 KPH	28.5 MPH 45.9 KPH	38.0 MPH 61.1 KPH	47.5 MPH 76.4 KPH	57.0 MPH 91.7 KPH
3rd	28.3 MPH 45.5 KPH	42.4 MPH 68.2 KPH	56.5 MPH 90.9 KPH	70.7 MPH 113.8 KPH	84.8 MPH 136.5 KPH
3rd Overdrive (LH)	34.5 MPH 55.5 KPH	51.7 MPH 83.2 KPH	69.0 MPH 111.0 KPH	86.2 MPH 138.7 KPH	103.5 MPH 166.5 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
4th Overdrive (LH)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	3.2 MPH 5.1 KPH	4.8 MPH 7.7 KPH	6.5 MPH 10.5 KPH	8.1 MPH 13.0 KPH	9.7 MPH 15.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.3 MPH 21.4 KPH	19.9 MPH 32.0 KPH	26.5 MPH 42.6 KPH	33.1 MPH 53.3 KPH	39.8 MPH 64.1 KPH
2nd	19.2 MPH 30.9 KPH	28.8 MPH 46.3 KPH	38.4 MPH 61.8 KPH	48.0 MPH 77.2 KPH	57.6 MPH 92.7 KPH
3rd	28.6 MPH 46.0 KPH	42.9 MPH 69.0KPH	57.2 MPH 92.0 KPH	71.5 MPH 115.1 KPH	85.8 MPH 138.1 KPH
3rd Overdrive (LH)	34.9 MPH 56.1 KPH	52.3 MPH 84.2 KPH	69.8 MPH 112.3 KPH	87.2 MPH 140.3 KPH	104.6 MPH 168.4 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
4th Overdrive (LH)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	3.3 MPH 5.3 KPH	4.9 MPH 7.9 KPH	6.5 MPH 10.5 KPH	8.2 MPH 13.2 KPH	9.8 MPH 15.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	14.2 MPH 22.8 KPH	21.3 MPH 34.3 KPH	28.4 MPH 45.7 KPH	35.5 MPH 57.1 KPH	43.6 MPH 70.2 KPH
2nd	20.5 MPH 32.0 KPH	30.8 MPH 49.6 KPH	41.1 MPH 66.1 KPH	51.4 MPH 82.7 KPH	61.7 MPH 99.3 KPH
3rd	30.6 MPH 49.2 KPH	45.9 MPH 73.9 KPH	61.2 MPH 98.5 KPH	76.5 MPH 123.1 KPH	91.8 MPH 147.7 KPH
3rd Overdrive (LH)	37.3 MPH 60.0 KPH	56.0 MPH 90.1 KPH	74.7 MPH 120.2 KPH	93.3 MPH 150.1 KPH	112.0 MPH 180.2 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
4th Overdrive (LH)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	3.5 MPH 5.6 KPH	5.2 MPH 8.4 KPH	7.0 MPH 11.3 KPH	8.7 MPH 14.0 KPH	10.5 MPH 16.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion rear axle (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	15.3 MPH 24.6 KPH	22.9 MPH 36.8 KPH	30.6 MPH 49.2 KPH	38.2 MPH 61.5 KPH	45.9 MPH 73.9 KPH
2nd	22.1 MPH 35.6 KPH	33.2 MPH 53.4 KPH	44.3 MPH 71.3 KPH	55.3 MPH 89.0 KPH	66.4 MPH 106.9 KPH
3rd	33.0 MPH 53.1 KPH	49.4 MPH 79.5 KPH	65.9 MPH 106.1 KPH	82.4 MPH 132.6 KPH	99.0 MPH 159.3 KPH
3rd Overdrive (LH)	40.2 MPH 64.7 KPH	60.3 MPH 97.0 KPH	80.4 MPH 129.4 KPH	100.5 MPH 161.7 KPH	120.6 MPH 194.1 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
4th Overdrive (LH)	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	1.5 MPH 2.4 KPH	2.3 MPH 3.7 KPH	3.1 MPH 5.0 KPH	3.9 MPH 6.3 KPH	4.6 MPH 7.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Yet another option for those seeking alternative gear ratios could be those used in the transmission of the MGB GT V8 model. The MGB GT V8 used what was essentially the same BMC four-synchro transmission design. This differs from the transmission of the MGB only in its bellhousing, the clutch, the clutch actuating fork, the clutch actuating fork boot, the clutch slave cylinder, the input shaft (first motion shaft), the laygear, the output flange, and the ratios of their gearsets. Everything else is the same as that found as Original Equipment in the contemporary MGB transmissions, and as such their parts are fully interchangeable with those of the MGB transmissions. For those who are looking for something different from any of the usual Original Equipment gear ratios, or are contemplating a Rover V8 conversion, they are well worth considering. However, be aware that due to the stress limitations of the transmission design, in the Rover V8 application the remote control tower always incorporated an Overdrive inhibitor switch (BMC Part # 13H 2154) in order to preclude the use of the Overdrive in third gear.

MGB GT V8 (Top Fill Version) Transmission, w/ LH-type Overdrive:

1st	3.138 : 1
2nd	1.974 : 1
3rd	1.259 : 1
3rd Overdrive	1.032 : 1
4th	1.0 : 1
4th Overdrive	0.82 : 1
Reverse	2.819 : 1

These gear ratios create the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.589	1,205 RPM	2,045 RPM	2,039 RPM	3,461 RPM
2nd-3rd	1.568	1,177 RPM	2,072 RPM	1,992 RPM	3,508 RPM
3rd-3rd O.D.	1.220	111 RPM	3,149 RPM	171 RPM	5,329 RPM
3rd-4th	1.259	669 RPM	2,581 RPM	1,132 RPM	4,368 RPM
3rd O.D.-4th	1.535	1,133 RPM	2,117 RPM	1,917 RPM	3,583 RPM
4th-4th O.D.	0.820	584 RPM	2,666 RPM	988 RPM	4,512 RPM
3rd O.D.-4th O.D.	1.258	1,514 RPM	1,736 RPM	2,562 RPM	2,938 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.7 MPH 14.0 KPH	13.0 MPH 20.9 KPH	17.4 MPH 28.0 KPH	21.7 MPH 34.9 KPH	26.1 MPH 42.0 KPH
2nd	13.9 MPH 22.4 KPH	20.7 MPH 33.3 KPH	27.6 MPH 44.4 KPH	34.6 MPH 55.7 KPH	41.5 MPH 66.8 KPH
3rd	21.7 MPH 34.9 KPH	32.5 MPH 52.3 KPH	43.4 MPH 69.8 KPH	54.2 MPH 87.2 KPH	65.0 MPH 104.6 KPH
3rd Overdrive (LH)	26.4 MPH 42.5 KPH	39.7 MPH 63.9 KPH	52.9 MPH 85.1 KPH	66.1 MPH 106.4 KPH	79.3 MPH 127.6 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
4th Overdrive (LH)	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	9.7 MPH 15.6 KPH	13.0 MPH 20.9 KPH	19.4 MPH 31.2 KPH	24.2 MPH 38.9 KPH	29.0 MPH 46.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	9.1 MPH 14.6 KPH	13.7 MPH 22.0 KPH	18.3 MPH 29.4 KPH	22.9 MPH 36.8 KPH	27.4 MPH 44.1 KPH
2nd	14.5 MPH 23.3 KPH	21.8 MPH 35.1 KPH	29.1 MPH 46.8 KPH	36.3 MPH 58.4 KPH	43.6 MPH 70.2 KPH
3rd	22.8 MPH 36.7 KPH	34.2 MPH 55.0 KPH	45.6 MPH 73.4 KPH	57.0 MPH 91.7 KPH	68.4 MPH 110.1 KPH
3rd Overdrive (LH)	27.8 MPH 44.7 KPH	41.7 MPH 67.1 KPH	55.6 MPH 89.5 KPH	69.5 MPH 111.8 KPH	83.4 MPH 134.2 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
4th Overdrive (LH)	32.1MPH 51.7 KPH	48.2 MPH 77.6 KPH	70.0 MPH 112.6 KPH	80.3 MPH 129.2 KPH	96.4 MPH 155.1 KPH
Reverse	10.2 MPH 16.4 KPH	15.3 MPH 24.6 KPH	20.3 MPH 32.7 KPH	25.4 MPH 40.9 KPH	30.5 MPH 49.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.8 MPH 15.8 KPH	14.7 MPH 23.7 KPH	19.6 MPH 31.5 KPH	24.5 MPH 39.4 KPH	29.4 MPH 47.3 KPH
2nd	15.6 MPH 25.1 KPH	23.4 MPH 37.7 KPH	31.1 MPH 50.0 KPH	38.9 MPH 62.6 KPH	46.7 MPH 75.2 KPH
3rd	24.4 MPH 39.3 KPH	36.6 MPH 58.9 KPH	48.8 MPH 78.5 KPH	61.0 MPH 98.2 KPH	73.2 MPH 117.8 KPH
3rd Overdrive (LH)	29.8 MPH 48.0 KPH	44.7 MPH 71.9 KPH	59.6 MPH 95.9 KPH	74.5 MPH 119.9 KPH	89.4 MPH 143.9 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
4th Overdrive (LH)	37.5 MPH 60.3 KPH	56.2 MPH 90.4 KPH	75.0 MPH 120.7 KPH	93.7 MPH 150.8 KPH	112.5 MPH 181.0 KPH
Reverse	2.5 MPH 4.0 KPH	3.8 MPH 6.1 KPH	5.1 MPH 8.2 KPH	6.3 MPH 10.1 KPH	7.6 MPH 12.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.4 MPH 16.7 KPH	15.5 MPH 29.4 KPH	20.7 MPH 33.3 KPH	25.9 MPH 41.7 KPH	31.1 MPH 50.0 KPH
2nd	16.5 MPH 26.5 KPH	24.7 MPH 39.7 KPH	33.0 MPH 53.1 KPH	41.2 MPH 66.1 KPH	49.4 MPH 79.5 KPH
3rd	25.8 MPH 41.5 KPH	38.8 MPH 62.4 KPH	51.7 MPH 83.2 KPH	64.6 MPH 104.0 KPH	77.5 MPH 124.7 KPH
3rd Overdrive (LH)	31.5 MPH 50.7 KPH	47.3 MPH 76.1 KPH	63.0 MPH 101.4 KPH	78.8 MPH 126.8 KPH	94.6 MPH 152.2 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
4th Overdrive (LH)	39.7 MPH 63.8 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	2.7 MPH 4.3 KPH	4.0 MPH 6.4 KPH	5.4 MPH 8.7 KPH	6.7 MPH 10.8 KPH	8.1 MPH 13.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.6 MPH 17.1 KPH	15.8 MPH 25.4 KPH	21.1 MPH 34.0 KPH	26.4 MPH 42.5 KPH	31.7 MPH 51.0 KPH
2nd	16.8 MPH 27.0 KPH	25.2 MPH 40.6 KPH	33.6 MPH 54.1 KPH	42.0 MPH 67.6 KPH	50.4 MPH 81.1 KPH
3rd	26.3 MPH 42.3 KPH	39.5 MPH 63.6 KPH	52.7 MPH 84.8 KPH	65.8 MPH 105.9 KPH	79.0 MPH 127.1 KPH
3rd Overdrive (LH)	32.1 MPH 51.7 KPH	48.1 MPH 77.4 KPH	64.2 MPH 103.3 KPH	80.3 MPH 129.2 KPH	96.4 MPH 155.1 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
4th Overdrive (LH)	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	11.8 MPH 19.0 KPH	17.6 MPH 28.3 KPH	23.5 MPH 37.8 KPH	29.4 MPH 47.3 KPH	35.3 MPH 56.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.9 MPH 17.5 KPH	16.3 MPH 26.2 KPH	21.7 MPH 34.9 KPH	27.2 MPH 43.8 KPH	32.6 MPH 52.5 KPH
2nd	17.3 MPH 27.8 KPH	25.9 MPH 41.7 KPH	34.6 MPH 55.7 KPH	43.2 MPH 69.5 KPH	51.8 MPH 83.4 KPH
3rd	27.1 MPH 43.6 KPH	40.6 MPH 65.3 KPH	54.2 MPH 87.2 KPH	67.7 MPH 109.0 KPH	81.3 MPH 130.8 KPH
3rd Overdrive (LH)	33.1 MPH 53.3 KPH	49.6 MPH 79.8 KPH	66.1 MPH 106.4 KPH	82.7 MPH 133.1 KPH	99.2 MPH 159.6 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
4th Overdrive (LH)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	12.1 MPH 19.5 KPH	18.1 MPH 29.1 KPH	24.2 MPH 38.8 KPH	30.3 MPH 48.8 KPH	36.3 MPH 58.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.4 MPH 18.3 KPH	17.1 MPH 27.5 KPH	22.8 MPH 36.7 KPH	28.5 MPH 45.9 KPH	34.2 MPH 55.0 KPH
2nd	18.1 MPH 29.1 KPH	27.2 MPH 43.8 KPH	36.3 MPH 58.4 KPH	45.3 MPH 72.9 KPH	54.4 MPH 87.5 KPH
3rd	28.4 MPH 45.7 KPH	42.6 MPH 68.9 KPH	56.8 MPH 91.4 KPH	71.1 MPH 114.4 KPH	85.3 MPH 137.3 KPH
3rd Overdrive (LH)	34.7 MPH 55.8 KPH	52.0 MPH 83.7 KPH	69.3 MPH 111.5 KPH	86.7 MPH 139.5 KPH	104.0 MPH 167.4 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
4th Overdrive (LH)	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	12.7 MPH 20.4 KPH	19.0 MPH 30.6 KPH	25.4 MPH 40.9 KPH	31.7 MPH 51.0 KPH	38.1 MPH 61.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset (Autogear Part # CWP033)[1] Hardy-Spicer Banjo-type Rear Axle, (BMC Part # BTB1244)[2] Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.0 MPH 19.3 KPH	18.1 MPH 29.1 KPH	24.1 MPH 38.8 KPH	30.1 MPH 48.4 KPH	36.1 MPH 58.1 KPH
2nd	19.1 MPH 30.7 KPH	28.7 MPH 46.2 KPH	38.3 MPH 61.6 KPH	47.9 MPH 77.1 KPH	57.4 MPH 92.4 KPH
3rd	30.0 MPH 48.3 KPH	45.0 MPH 72.4 KPH	60.0 MPH 96.6 KPH	75.0 MPH 120.7 KPH	90.0 MPH 144.8 KPH
3rd Overdrive (LH)	36.6 MPH 58.9 KPH	54.9 MPH 88.5 KPH	73.2 MPH 117.8 KPH	91.6 MPH 147.4 KPH	109.9 MPH 176.9 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
4th Overdrive (LH)	46.1 MPH 74.2 KPH	69.2 MPH 11.42 KPH	92.2 MPH 148.42 KPH	115.2 MPH 185.42 KPH	138.3 MPH 222.62 KPH
Reverse	13.4 MPH 21.6 KPH	20.1 MPH 32.3 KPH	26.8 MPH 43.1 KPH	33.5 MPH 53.9 KPH	40.2 MPH 64.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.4MPH 20.0 KPH	18.7 MPH 30.0 KPH	24.9 MPH 40.0 KPH	31.1 MPH 50.0 KPH	37.3 MPH 60.0 KPH
2nd	19.8 MPH 31.9 KPH	29.7 MPH 47.8 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	59.3 MPH 95.4 KPH
3rd	31.0 MPH 49.9 KPH	46.5 MPH 74.8 KPH	62.0 MPH 99.8 KPH	77.5 MPH 124.7 KPH	93.0 MPH 149.7 KPH
3rd Overdrive (LH)	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.7 MPH 121.8 KPH	94.6 MPH 152.2 KPH	113.5 MPH 182.7 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
4th Overdrive (LH)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	3.2 MPH 5.1 KPH	4.8 MPH 7.7 KPH	6.5 MPH 10.5 KPH	8.1 MPH 13.0 KPH	9.7 MPH 15.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.6 MPH 20.3 KPH	18.9 MPH 30.4 KPH	25.1 MPH 40.4 KPH	31.4 MPH 50.5 KPH	37.9 MPH 70.0 KPH
2nd	20.0 MPH 32.2 KPH	30.0 MPH 48.3 KPH	40.0 MPH 64.4 KPH	50.0 MPH 80.5 KPH	60.0 MPH 96.6 KPH
3rd	31.3 MPH 50.4 KPH	47.0 MPH 75.6 KPH	62.7 MPH 100.9 KPH	78.4 MPH 126.2 KPH	94.0 MPH 151.3 KPH
3rd Overdrive (LH)	38.2 MPH 61.5 KPH	57.3 MPH 92.2 KPH	76.5 MPH 123.1 KPH	95.6 MPH 153.8 KPH	114.7 MPH 184.6 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
4th Overdrive (LH)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	3.3 MPH 5.3 KPH	4.9 MPH 7.9 KPH	6.5 MPH 10.5 KPH	8.2 MPH 13.2 KPH	9.8 MPH 15.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	13.5 MPH 21.7 KPH	20.2 MPH 32.5 KPH	27.0 MPH 43.4 KPH	33.7 MPH 54.2 KPH	40.3 MPH 64.9 KPH
2nd	21.4 MPH 34.4 KPH	32.1 MPH 51.7 KPH	42.9 MPH 69.0 KPH	53.6 MPH 86.3 KPH	64.0 MPH 103.0 KPH
3rd	33.6 MPH 54.1 KPH	50.4 MPH 81.1 KPH	67.2 MPH 108.0 KPH	84.0 MPH 135.2 KPH	100.8 MPH 162.2 KPH
3rd Overdrive (LH)	41.0 MPH 66.0 KPH	61.5 MPH 99.0 KPH	82.0 MPH 132.0 KPH	102.5 MPH 165.0 KPH	123.0 MPH 198.0 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
4th Overdrive (LH)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	15.0MPH 24.1 KPH	22.5 MPH 36.2 KPH	30.0 MPH 48.2 KPH	37.5 MPH 60.3 KPH	45.0 MPH 72.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	14.5 MPH 23.3 KPH	21.8 MPH 35.1 KPH	29.0 MPH 46.7 KPH	36.3 MPH 58.4 KPH	43.6 MPH 70.2 KPH
2nd	23.1 MPH 37.2 KPH	34.6 MPH 55.7 KPH	46.2 MPH 74.3 KPH	57.7 MPH 92.8 KPH	69.2 MPH 111.4 KPH
3rd	36.2 MPH 58.3 KPH	54.3 MPH 87.4 KPH	72.4 MPH 116.5 KPH	90.5 MPH 145.6 KPH	108.6 MPH 174.8 KPH
3rd Overdrive (LH)	44.1 MPH 71.0 KPH	66.2 MPH 106.5 KPH	88.3 MPH 142.1 KPH	110.4 MPH 177.7 KPH	132.4 MPH 213.1 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
4th Overdrive (LH)	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	16.2 MPH 26.1 KPH	24.2 MPH 38.9 KPH	32.4 MPH 52.1 KPH	40.4 MPH 65.0 KPH	48.5 MPH 78.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

The non-synchro first gear being largely irrelevant on a racetrack, many racers prefer the earlier three-synchro transmission over the later four-synchro transmission because of its marginally lower weight (75 lbs w/ clutch), lower internal friction, and the wider range of crownwheel / pinion gear ratios that are available. In addition, its low 18.588:1 reverse gear ratio gives essentially the same maximum reverse speed with the highest final drive ratio available (3.071:1) as does the 12.098:1 reverse gear ratio of the four-synchro transmission with the lowest final drive ratio (5.125:1) available, thus making it much more manageable in the pits at a racetrack.

The most desirable of these three-synchro transmissions is the version that was introduced in the month of March in 1967 starting with engine number GB74720 (Standard transmission) and GB74529 (Overdrive transmission). It makes use of a larger-diameter (.668" / 18.967mm) layshaft (second motion shaft) (BMC Part # 22H 571) and an accompanying laygear (BMC Part # 22H 1301), a modified distance piece for the layshaft (second motion shaft) (BMC Part #22H 672), and four sets of caged needle-roller bearings (BMC Part # 22H 471) in order to support the laygear instead of the three uncaged sets of needle-roller bearings (BMC Part # 3H 2865) of the earlier design, all of which enable the transmission to absorb heavy abuse better than the earlier smaller-diameter (.643" / 16.3322mm) layshaft (second motion shaft) (BMC Part # 1H 3305) with its lesser three-support bearings. This heavier duty layshaft (second motion shaft) assembly was originally developed and introduced as a competition part by the racing department and subsequently was standardized on the mass production cars in order to meet the homologation rules of racing associations. The need for a stronger, more durable layshaft (second motion shaft) as well as an increase in support from its needle-roller bearings in order to better cope with the greater power of the 1800cc version of the B Series engines compelled this redesign. In addition, the circlip that retained the uncaged needle-roller bearings of the laygear did not allow the fine metal particles created by wear to be flushed out of the laygear assembly, thereby accelerating the rate of wear. In some cases of hard abuse, these components could wear out in as little as 50,000 miles, although with normal use they could be expected to endure for a good 70,000 miles of reliable service. However, when wear became excessive, the loose needle-roller bearings would destroy both the layshaft as well as their bore inside of the laygear, resulting in expensive repair work. When the transmission is in fourth gear, the power goes through the transmission in a straight line. That is, the power goes through the transmission from the input shaft (first motion shaft) directly to the mainshaft (third motion shaft) which is locked directly to it, without transferring through the gear on the layshaft (second motion shaft). Because the gear on the layshaft (second motion shaft) is idling, its bearings are not under a load. If the layshaft is badly worn, the bearings will then make a metallic "fizzing" sound. A set of double-paired caged needle-roller bearings (BMC Part # 22H 471), along with a larger-diameter layshaft (BMC Part # 88G 400) and spacer tube (BMC Part # 22H 672), were seen to be the solution to this problem. This configuration is sometimes referred to as the "four-hole" layshaft (second motion shaft) because of its four holes for feeding lubricating oil to its four sets of caged needle-roller bearings that support its larger-diameter .668" (16.9672mm) layshaft and spacer tube, while its earlier, smaller-diameter .643" (16.332mm) counterpart (BMC Part# 1H 3305) and spacer tube (BMC Part # 11G 3026) is sometimes referred to as the "three-hole" layshaft (second motion shaft) because of its three holes for feeding lubricating oil to its uncaged three sets of support bearings (BMC Part # AAU 3252).

It is possible to modify the earlier three-synchro transmissions (found with the 18GB 74529 and earlier engines) in order to install this stronger, larger-diameter layshaft (second motion shaft) assembly by boring out and line reaming the layshaft (second motion shaft) mounting holes to .6693" +/- .0005". (17.0002mm +/- .0127mm). The best, and easiest, way to do this is to use an "end bell reamer" that is used for reaming the holes in the end bells of electric motors. This is a piloted, adjustable reamer that is adjustable (over a small range) to give the proper diameter and clearance. To keep things in line, it pilots in one hole while it cuts the other. They are commonly found at shops that rebuild electric motors, and the technicians there should do the job for a nominal fee. You will, of course, need to use the corresponding later laygear in order to fit onto the larger-diameter layshaft (second motion shaft), along with its front and rear thrust washers, caged needle-roller bearings, and its larger-diameter distance piece.

There is another, seldom-used way to upgrade the needle-roller bearing assembly of the three-synchro transmission without making the conversion to the later four bearing assembly, although it is not as good an upgrade as the one that the factory engineers chose. By a happy coincidence, the longer needle-roller bearings between the input gear of the Input shaft (first motion shaft) and the nose of the mainshaft (third motion shaft) have the same diameter needle-roller bearing and journal dimensions as that of the earlier smaller-diameter (.643" / 16.332mm) layshaft (second motion shaft). The needle-roller bearings are a nominal 3mm diameter (0.1181"). The clearance should be -.0001" to -.0003" (.00254mm to .00762mm), making the size range .1178" to .1180" (2.992mm to 2.9972mm). These needle-roller bearings present the advantage of being longer than the needle-roller bearings of the layshaft (second motion shaft), thus spreading the loading over a greater area. Their use will require that the step inside of the laygear be delicately relocated to a point deeper inside of the bore of the laygear, and that the spacer tube of the layshaft (second motion shaft) will need to be shortened by the same amount in order to accommodate use of these longer needle-roller bearings in the layshaft assembly.

The original layshaft (second motion shaft) needle bearing assemblies of the three-synchro transmission consisted of two end plates and twenty loose needle-roller bearings for each of the three bearing assemblies. These must have been a real joy to assemble on the factory's mass production line. The needle-roller bearings have conical ends. The retaining side plates have an L-shaped cross section, thus providing a small flange around the Internal Diameter on one side in order to guide the conical ends of the needle-roller bearings and retain them in position. Drop one side plate into the bore of the laygear. Insert the greased layshaft (second motion shaft) so that the side plate will hold it centered inside of the laygear and the end of the layshaft (second motion shaft) is near the end of the bore. Next, carefully insert the 20 individual needle-roller bearings, followed by the second side plate, after which you can remove the layshaft (second motion shaft) and install either a snap ring or the tube spacer, and the rollers will then be held securely in place without the shaft. Including the tube spacer and the three snap rings, you will have to load no fewer than 70 loose pieces inside of the laygear in order to assemble all of the needle-roller bearings for the layshaft (second motion shaft).

Do not be tempted to substitute caged roller bearings for the loose 20 individual needle-roller bearings that support the smaller-diameter (.643" / 16.332mm) layshaft (second motion shaft). While much easier to install, caged roller bearings require some space between the rollers in order to provide for the webs of the cage. This being the case, there may be space enough left for only sixteen roller bearings, or perhaps less, depending upon the design of the cage. As a consequence, the load will be distributed across fewer rollers on one side of the layshaft. The attendant consequence of having fewer roller bearings carrying the bulk of the load is increased loadings on the surface of the layshaft (second motion shaft), thus increasing the rate of wear. This can make a significant difference in the service life of the layshaft. Using loose needle-roller bearings without a cage allowed the installation of twenty load-bearing needle-roller bearings around the layshaft with just enough space between them for a film of oil. This permitted about eight needle-roller bearings per set to carry the load in one direction (40% of the circle), with six needle-roller bearings carrying the bulk of the load, while the two adjacent needle-roller bearings carried a reduced proportion of the load.

For those who want to keep their B as original as possible and retain the quaint usefulness of the Laycock de Normanville Overdrive unit, yet still yearn for a close ratio transmission, Cambridge Motorsport offers straight-cut close-ratio gearsets for both the three-synchro and four-synchro transmissions. However, for the three-synchro transmission, they also offer the same gear ratios in a helically-cut gearset, that uses the more desirable larger-diameter (.668" / 16.9672mm) layshaft (second motion shaft) with its four bearing support of the layshaft, while the straight-cut gearset uses the smaller-diameter (.643" / 16.3322mm) layshaft (second motion shaft) with its three bearing support of the layshaft. Straight-cut gears are the more efficient transmitters of power of the two choices, thus absorbing less power, but are extremely noisy. In order to understand just how much noisier straight-cut gears are than helically-cut gears, simply remember that your reverse gearset is straight-cut. In all cases, these are the same close-ratio gear ratios that were offered by the MG factory's Special Tuning Department, making them quite acceptable to both Vintage Racing associations and many Purists alike.

Cambridge Motorsport's ratios for the Three-Synchro transmission, w/ D-type Overdrive:

1st	2.450 : 1
2nd	1.620 : 1
3rd	1.268 : 1
3rd Overdrive	1.102 : 1
4th	1.000 : 1
4th Overdrive	0.802 : 1
Reverse	18.588: 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.442	1,016 RPM	2,234 RPM	1,863 RPM	3,637 RPM
2nd-3rd	.352	652 RPM	2,598 RPM	1,195 RPM	4,305 RPM
3rd-3rd O.D.	.90826	845 RPM	2,605 RPM	1,092 RPM	4,408 RPM
3rd-4th	.286	652 RPM	2,563 RPM	1,195 RPM	4,813 RPM
3rd O.D.-4th	.102	584 RPM	2,676 RPM	550 RPM	4,990 RPM
4th-4th O.D.	.198	643 RPM	2,607 RPM	1,089 RPM	4,510 RPM

3rd OD- 4th O.D.	.300	883 RPM	2,367 RPM	1,494 RPM	4,006 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.1 MPH 17.9 KPH	16.7 MPH 27.0 KPH	22.3 MPH 35.9 KPH	27.9 MPH 44.9 KPH	33.4 MPH 53.7 KPH
2nd	16.8 MPH 27.0 KPH	25.3 MPH 40.7 KPH	33.7 MPH 54.2 KPH	42.1 MPH 67.7 KPH	50.6 MPH 81.4 KPH
3rd	21.5 MPH 34.6 KPH	32.3 MPH 52.0 KPH	43.1 MPH 69.4 KPH	53.8 MPH 86.6 KPH	64.6 MPH 104.0 KPH
3rd Overdrive (D)	24.8 MPH 39.9 KPH	37.2 MPH 59.9 KPH	49.5 MPH 79.7 KPH	61.9 MPH 99.6 KPH	74.3 MPH 119.6 KPH
4th	27.3 MPH 43.9 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
4th Overdrive (D)	34.0 MPH 54.7 KPH	51.0MPH 82.1 KPH	68.1 MPH 109.6 KPH	85.1 MPH 137.0 KPH	102.1 MPH 164.3 KPH
Reverse	1.5 MPH 2.4 KPH	2.2 MPH 3.5 KPH	2.9 MPH 4.7 KPH	3.7 MPH 5.9 KPH	4.4 MPH 7.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production

MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 crownwheel (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	11.7 MPH 18.8 KPH	17.6 MPH 28.3 KPH	23.4 MPH 37.7 KPH	29.3 MPH 47.1 KPH	35.1 MPH 56.5 KPH
2nd	17.7 MPH 28.5 KPH	26.6 MPH 42.8 KPH	35.4 MPH 57.0 KPH	44.3 MPH 71.3 KPH	53.1 MPH 85.5 KPH
3rd	22.6 MPH 36.4 KPH	34.0 MPH 54.7 KPH	45.3 MPH 72.9 KPH	56.6 MPH 91.1 KPH	67.9 MPH 109.3 KPH
3rd Overdrive (D)	26.0 MPH 41.8 KPH	39.1 MPH 62.9 KPH	52.1 MPH 83.8 KPH	65.1 MPH 104.8 KPH	78.1 MPH 125.7 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.7 MPH 115.4 KPH	86.1 MPH 138.6 KPH
4th Overdrive (D)	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
Reverse	1.6 MPH 2.6 KPH	2.3 MPH 3.7 KPH	3.1 MPH 5.0 KPH	3.9 MPH 6.3 KPH	4.6 MPH 7.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.5 MPH 20.1 KPH	18.8 MPH 30.3 KPH	25.1 MPH 40.4 KPH	31.4 MPH 50.5 KPH	37.6 MPH 60.5 KPH
2nd	19.0 MPH 30.6 KPH	28.5 MPH 45.9 KPH	37.9 MPH 61.0 KPH	47.4 MPH 76.3 KPH	56.9 MPH 91.6 KPH
3rd	24.2 MPH 38.9 KPH	36.4 MPH 58.6 KPH	48.5 MPH 78.1 KPH	60.6 MPH 97.5 KPH	72.7 MPH 117.0 KPH
3rd Overdrive (D)	27.9 MPH 44.9 KPH	41.8 MPH 67.3 KPH	55.8 MPH 89.8 KPH	69.7 MPH 112.2 KPH	83.7 MPH 134.7 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.8 MPH 123.6 KPH	92.2 MPH 148.4 KPH
4th Overdrive (D)	38.3 MPH 61.6 KPH	57.5 MPH 92.5 KPH	76.7 MPH 123.4 KPH	95.8 MPH 154.2 KPH	115.0 MPH 185.1 KPH
Reverse	1.6 MPH 2.6 KPH	2.5 MPH 4.0 KPH	3.3 MPH 5.3 KPH	4.1 MPH 6.6 KPH	5.0 MPH 8.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion rear axle gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.3 MPH 21.4 KPH	29.9 MPH 48.1 KPH	26.6 MPH 42.8 KPH	33.2 MPH 53.4 KPH	39.8 MPH 64.0 KPH
2nd	20.1 MPH 32.3 KPH	30.1 MPH 48.4 KPH	40.2 MPH 64.7 KPH	50.2 MPH 80.8 KPH	60.2 MPH 96.9 KPH
3rd	25.6 MPH 41.2 KPH	38.5 MPH 62.0 KPH	51.3 MPH 82.6 KPH	64.1 MPH 103.2 KPH	77.0 MPH 123.9 KPH
3rd Overdrive (D)	29.5 MPH 47.5 KPH	44.3 MPH 71.3 KPH	59.0 MPH 94.9 KPH	73.8 MPH 118.8 KPH	88.6 MPH 142.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.1 KPH
4th Overdrive (D)	40.6 MPH 65.3 KPH	60.8 MPH 97.8 KPH	81.1 MPH 130.5 KPH	101.4 MPH 163.2 KPH	121.7 MPH 195.9 KPH
Reverse	1.7 MPH 2.7 KPH	2.6 MPH 4.2 KPH	3.5 MPH 5.6 KPH	4.4 MPH 7.1 KPH	5.2 MPH 8.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.5 MPH 21.7 KPH	20.3 MPH 32.7 KPH	27.1 MPH 43.6 KPH	33.8 MPH 54.4 KPH	40.6 MPH 65.3 KPH
2nd	20.5 MPH 33.0 KPH	30.7 MPH 49.4 KPH	40.9 MPH 65.8 KPH	51.1 MPH 82.2 KPH	61.4 MPH 98.8 KPH
3rd	26.1 MPH 42.0 KPH	39.2 MPH 63.1 KPH	52.3 MPH 84.2 KPH	65.3 MPH 105.1 KPH	78.4 MPH 126.2 KPH
3rd Overdrive (D)	30.1 MPH 48.4 KPH	45.1 MPH 72.6 KPH	60.2 MPH 96.9 KPH	75.2 MPH 121.0 KPH	88.6 MPH 142.6 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 113.4 KPH	99.4 MPH 160.0 KPH
4th Overdrive (D)	41.3 MPH 66.5 KPH	62.0 MPH 99.8 KPH	82.7 MPH 133.1 KPH	103.3 MPH 166.2 KPH	124.0 MPH 199.6 KPH
Reverse	1.8 MPH 2.9 KPH	2.7 MPH 4.3 KPH	3.6 MPH 5.8 KPH	4.5 MPH 7.2 KPH	5.3 MPH 8.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion rear axle gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	13.9 MPH 22.4 KPH	20.9 MPH 33.6 KPH	27.8 MPH 44.7 KPH	34.8 MPH 56.0 KPH	41.8 MPH 67.3 KPH
2nd	21.1 MPH 34.0 KPH	31.6 MPH 50.8 KPH	42.1 MPH 67.7 KPH	52.6 MPH 84.7 KPH	63.2 MPH 101.7 KPH
3rd	26.9 MPH 43.3 KPH	40.4 MPH 65.0 KPH	53.8 MPH 86.6 KPH	67.3 MPH 108.3 KPH	80.7 MPH 129.9 KPH
3rd Overdrive (D)	31.0 MPH 49.0 KPH	46.4 MPH 74.7 KPH	61.9 MPH 99.6 KPH	77.4 MPH 124.6 KPH	92.9 MPH 149.5 KPH
4th	34.2 MPH 55.0 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
4th Overdrive (D)	42.5 MPH 68.4 KPH	63.8 MPH 102.7 KPH	85.1 MPH 137.0 KPH	106.3 MPH 171.1 KPH	127.6 MPH 205.3 KPH
Reverse	1.8 MPH 2.9 KPH	2.7 MPH 4.3 KPH	3.7 MPH 5.9 KPH	4.6 MPH 7.4 KPH	5.5 MPH 12.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	14.6 MPH 23.5 KPH	21.9 MPH 35.2 KPH	29.2 MPH 47.0 KPH	36.5 MPH 58.7 KPH	43.8 MPH 70.5 KPH
2nd	22.1 MPH 35.6 KPH	33.1 MPH 53.3 KPH	44.2 MPH 71.1 KPH	55.2 MPH 88.8 KPH	66.3 MPH 106.7 KPH
3rd	28.2 MPH 45.4 KPH	42.3 MPH 68.1 KPH	56.4 MPH 90.8 KPH	70.5 MPH 113.5 KPH	84.6 MPH 136.1 KPH
3rd Overdrive (D)	32.5 MPH 52.3 KPH	48.7 MPH 78.4 KPH	64.9 MPH 104.4 KPH	81.2 MPH 130.7 KPH	97.4 MPH 156.7 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.5 MPH 144.0 KPH	107.4 MPH 172.8 KPH
4th Overdrive (D)	44.6 MPH 71.8 KPH	66.9 MPH 107.7 KPH	89.2 MPH 143.5 KPH	111.6 MPH 179.6 KPH	133.9 MPH 215.5 KPH
Reverse	3.0 MPH 4.8 KPH	4.4 MPH 7.1 KPH	5.9 MPH 9.5 KPH	7.4 MPH 11.9 KPH	8.9 MPH 14.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset, (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	15.4 MPH 24.8 KPH	23.1 MPH 37.2 KPH	30.9 MPH 49.7 KPH	36.8 MPH 59.2 KPH	46.3 MPH 74.5 KPH
2nd	23.3 MPH 37.5 KPH	35.0 MPH 56.3 KPH	46.7 MPH 75.2 KPH	58.3 MPH 93.8 KPH	89.4 MPH 143.9 KPH
3rd	29.8 MPH 48.0 KPH	44.7 MPH 71.9 KPH	59.6 MPH 95.9 KPH	74.5 MPH 119.9 KPH	89.4 MPH 143.4 KPH
3rd Overdrive (D)	34.3 MPH 55.2 KPH	51.5 MPH 82.9 KPH	68.6 MPH 110.4 KPH	85.8 MPH 138.1 KPH	102.9 MPH 165.6 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.7 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
4th Overdrive (D)	47.1 MPH 75.8 KPH	70.7 MPH 113.8 KPH	94.3 MPH 151.7 KPH	117.8 MPH 189.6 KPH	141.4 MPH 227.6 KPH
Reverse	2.0 MPH 3.2 KPH	3.0 MPH 4.8 KPH	4.1 MPH 6.6 KPH	5.1 MPH 8.2 KPH	6.1 MPH 9.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	16.0 MPH 25.7 KPH	23.9 MPH 38.5 KPH	31.9 MPH 51.3 KPH	39.9 MPH 64.2 KPH	47.8 MPH 76.9 KPH
2nd	24.1 MPH 38.8 KPH	36.2 MPH 58.3 KPH	48.2 MPH 77.6 KPH	60.3 MPH 97.0 KPH	72.4 MPH 116.5 KPH
3rd	30.8 MPH 49.6 KPH	46.2 MPH 74.4 KPH	61.6 MPH 99.1 KPH	77.0 MPH 123.9 KPH	92.4 MPH 148.7 KPH
3rd Overdrive (LH)	35.5 MPH 57.1 KPH	53.2 MPH 85.6 KPH	70.9 MPH 114.1 KPH	88.6 MPH 142.6 KPH	106.4 MPH 171.2 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
4th Overdrive (LH)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	2.1 MPH 3.4 KPH	3.1 MPH 5.0 KPH	4.2 MPH 6.8 KPH	5.2 MPH 8.4 KPH	6.3 MPH 10.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	16.1 MPH 26.0 KPH	24.2 MPH 38.9 KPH	32.3 MPH 52.0 KPH	40.3 MPH 64.9 KPH	48.4 MPH 77.9 KPH
2nd	24.4 MPH 39.3 KPH	36.6 MPH 58.9 KPH	48.8 MPH 78.5 KPH	61.0 MPH 98.2 KPH	73.2 MPH 117.8 KPH
3rd	31.2 MPH 50.2 KPH	46.7 MPH 75.2 KPH	62.3 MPH 100.3 KPH	77.9 MPH 125.4 KPH	93.5 MPH 150.5 KPH
3rd Overdrive (LH)	35.9 MPH 57.8 KPH	53.8 MPH 86.6 KPH	71.7 MPH 115.4 KPH	89.6 MPH 144.9 KPH	107.6 MPH 173.2 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
4th Overdrive (LH)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	2.1 MPH 3.4 KPH	3.2 MPH 5.1 KPH	4.3 MPH 6.9 KPH	5.3 MPH 8.5 KPH	6.4 MPH 10.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production 1962-1968 MGB roadsters, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	17.3 MPH 27.8 KPH	25.9 MPH 41.7 KPH	34.5 MPH 55.5 KPH	43.2 MPH 69.5 KPH	51.8 MPH 83.4 KPH
2nd	26.1 MPH 42.0 KPH	39.2 MPH 63.1 KPH	52.2 MPH 84.0 KPH	65.3 MPH 105.1 KPH	78.3 MPH 126.0 KPH
3rd	33.4 MPH 53.7 KPH	50.0 MPH 80.5 KPH	66.7 MPH 107.3 KPH	83.4 MPH 134.2 KPH	100.1 MPH 161.1 KPH
3rd Overdrive (D)	38.4 MPH 61.8 KPH	57.6 MPH 92.7 KPH	76.8 MPH 123.6 KPH	96.0 MPH 154.5 KPH	115.1 MPH 185.2 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
4th Overdrive (D)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	2.3 MPH 3.7 KPH	3.4 MPH 5.5 KPH	4.5 MPH 7.2 KPH	5.7 MPH 9.2 KPH	6.8 MPH 10.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	18.6 MPH 29.9 KPH	27.9 MPH 44.9 KPH	37.2 MPH 59.9 KPH	46.5 MPH 74.8 KPH	55.8 MPH 89.8 KPH
2nd	28.1 MPH 45.2 KPH	42.2 MPH 67.9 KPH	56.2 MPH 90.4 KPH	70.3 MPH 113.1 KPH	84.4 MPH 135.8 KPH
3rd	35.9 MPH 57.8 KPH	53.9 MPH 86.7 KPH	71.9 MPH 115.7 KPH	89.8 MPH 144.5 KPH	107.8 MPH 173.5 KPH
3rd Overdrive (D)	41.3 MPH 66.5 KPH	62.2 MPH 100.1 KPH	82.7 MPH 133.1 KPH	103.4 MPH 166.4 KPH	124.0 MPH 199.6 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
4th Overdrive (D)	56.8 MPH 91.4 KPH	85.2 MPH 137.1 KPH	113.6 MPH 182.8 KPH	142.0 MPH 228.5 KPH	170.4 MPH 274.2 KPH
Reverse	2.4 MPH 3.8 KPH	3.7 MPH 5.9 KPH	4.9 MPH 7.9 KPH	6.1 MPH 9.8 KPH	7.3 MPH 11.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Cambridge Motorsport's ratios for the Four-Synchro Transmission, w/ LH-type Overdrive:

1st	2.45 : 1
2nd	1.670 : 1
3rd	1.306 : 1
3rd Overdrive	1.025 : 1
4th	1.000 : 1
4th Overdrive	.820 : 1
Reverse	12.098 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	.670	859 RPM	2,391 RPM	1,575 RPM	3,925 RPM
2nd-3rd	.370	755 RPM	2,495 RPM	1,383 RPM	4,117 RPM
3rd-3rd O.D.	.882	586 RPM	2,664 RPM	991 RPM	4,509 RPM
3rd-4th	.250	600 RPM	2,650 RPM	1,100 RPM	4,400 RPM
3rd O.D.-4th	.025	625 RPM	2,600 RPM	1,180 RPM	4,320 RPM
4th-4th O.D.	.180	585 RPM	2,665 RPM	990 RPM	4,510 RPM

3rd O.D.-4th O.D.	.205	586 RPM	2,664 RPM	991 RPM	4,509 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.7MPH 18.8 KPH	17.5 MPH 28.2 KPH	23.3 MPH 37.5 KPH	29.2 MPH 47.0 KPH	35.0 MPH 56.3 KPH
2nd	16.3 MPH 26.2 KPH	24.5 MPH 39.4 KPH	32.6 MPH 52.5 KPH	40.9MPH 65.8 KPH	49.0MPH 78.9 KPH
3rd	21.8 MPH 35.1 KPH	32.7 MPH 52.6 KPH	43.7 MPH 70.3 KPH	54.6 MPH 87.9 KPH	65.5 MPH 105.4 KPH
3rd Overdrive (LH)	26.6 MPH 42.8 KPH	39.9 MPH 64.2 KPH	53.3 MPH 85.8 KPH	66.6 MPH 107.2 KPH	79.9 MPH 128.6 KPH
4th	27.3 MPH 43.9 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
4th Overdrive (LH)	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	2.3 MPH 3.7 KPH	3.4 MPH 5.5 KPH	4.5 MPH 7.2 KPH	5.6 MPH 9.0 KPH	6.8 MPH 10.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production

MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.3 MPH 19.8 KPH	18.4 MPH 29.6 KPH	24.5 MPH 39.4 KPH	30.7 MPH 49.4 KPH	36.8 MPH 59.2 KPH
2nd	17.2 MPH 27.7 KPH	25.8 MPH 41.5 KPH	34.4 MPH 55.4 KPH	43.0 MPH 69.2 KPH	51.5 MPH 82.9 KPH
3rd	23.0 MPH 37.1 KPH	34.4 MPH 55.4 KPH	45.9 MPH 73.9 KPH	57.4 MPH 92.4 KPH	68.9 MPH 110.9 KPH
3rd Overdrive (LH)	28.0 MPH 45.1 KPH	42.0 MPH 67.6 KPH	56.0 MPH 90.1 KPH	70.0 MPH 112.6 KPH	84.0 MPH 135.2 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.7 MPH 115.4 KPH	86.1 MPH 138.6 KPH
4th Overdrive (LH)	35.0 MPH 56.3 KPH	52.5 MPH 84.5 KPH	70.0 MPH 112.6 KPH	87.5 MPH 140.8 KPH	105.0 MPH 169.0 KPH
Reverse	2.4 MPH 3.9 KPH	3.6 MPH 5.8 KPH	4.7 MPH 7.6 KPH	5.9 MPH 9.5 KPH	7.1 MPH 11.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.1 MPH 21.1 KPH	19.7 MPH 31.7 KPH	26.3 MPH 42.3 KPH	32.8 MPH 52.8 KPH	39.4 MPH 63.4 KPH
2nd	18.4 MPH 29.6 KPH	27.6 MPH 44.4 KPH	36.8 MPH 59.2 KPH	46.0 MPH 74.0 KPH	55.2 MPH 88.8 KPH
3rd	24.6 MPH 39.6 KPH	36.9 MPH 59.4 KPH	49.2 MPH 79.2 KPH	61.5 MPH 99.0 KPH	73.8 MPH 118.8 KPH
3rd Overdrive (LH)	30.0 MPH 48.3 KPH	45.0 MPH 72.4 KPH	60.0 MPH 96.6 KPH	75.0 MPH 12.8 KPH	90.0 MPH 144.8 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.8 MPH 123.6 KPH	92.2 MPH 148.4 KPH
4th Overdrive (LH)	37.5 MPH 60.3 KPH	56.2 MPH 90.4 KPH	74.9 MPH 120.5 KPH	93.7 MPH 150.8 KPH	112.5 MPH 181.0 KPH
Reverse	1.6 MPH	2.5 MPH	3.3 MPH	4.1 MPH	5.0 MPH

	2.6 KPH	4.0 KPH	5.3 KPH	6.6 KPH	8.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: Note: This gearset requires the differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.9 MPH 22.4 KPH	20.8 MPH 33.5 KPH	27.8 MPH 44.7 KPH	34.7 MPH 55.8 KPH	41.7 MPH 67.1 KPH
2nd	19.5 MPH 31.4 KPH	29.2 MPH 47.0 KPH	39.0 MPH 62.8 KPH	48.7 MPH 78.4 KPH	58.4 MPH 94.0 KPH
3rd	26.0 MPH 41.8 KPH	39.0 MPH 62.8 KPH	52.0 MPH 83.7 KPH	65.1 MPH 104.8 KPH	78.1 MPH 125.7 KPH
3rd Overdrive (LH)	31.7 MPH 51.0 KPH	47.6 MPH 76.6 KPH	63.5 MPH 102.2 KPH	79.3 MPH 127.6 KPH	95.2 MPH 153.2 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.1 KPH

4th Overdrive (LH)	39.7 MPH 63.9 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	2.7 MPH 4.3 KPH	4.0 MPH 6.4 KPH	5.4 MPH 8.7 KPH	6.7 MPH 10.8 KPH	8.1 MPH 13.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	14.2 MPH 22.8 KPH	21.2 MPH 34.1 KPH	28.3 MPH 45.5 KPH	35.4 MPH 57.0 KPH	42.5 MPH 68.4 KPH
2nd	19.8 MPH 31.9 KPH	29.8 MPH 48.0 KPH	39.7 MPH 63.9 KPH	49.6 MPH 79.8 KPH	59.6 MPH 95.9 KPH
3rd	26.5 MPH 42.6 KPH	39.0 MPH 62.8 KPH	53.0 MPH 85.3 KPH	66.3 MPH 106.7 KPH	79.6 MPH 128.1 KPH
3rd Overdrive (LH)	32.3 MPH 52.0 KPH	48.5 MPH 78.0 KPH	64.7 MPH 104.1 KPH	80.8 MPH 130.0 KPH	97.0 MPH 156.1 KPH
4th	33. MPH	49.7 MPH	66.3 MPH	82.9 MPH	99.4 MPH

	53.3 KPH	80.0 KPH	106.7 KPH	113.4 KPH	160.0 KPH
4th Overdrive (LH)	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	2.7 MPH 4.3 KPH	4.1 MPH 6.6 KPH	5.5 MPH 8.8 KPH	6.8 MPH 10.9 KPH	8.2 MPH 13.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	14.6 MPH 23.5 KPH	20.9 MPH 33.6 KPH	29.2 MPH 47.0 KPH	36.4 MPH 58.6 KPH	43.7 MPH 70.3 KPH
2nd	20.4 MPH 32.8 KPH	30.6 MPH 49.2 KPH	40.9 MPH 65.8 KPH	51.1 MPH 82.2 KPH	61.3 MPH 98.6 KPH
3rd	27.3 MPH 43.9 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
3rd Overdrive (LH)	33.3 MPH 53.6 KPH	49.3 MPH 79.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.8 MPH 160.6 KPH
4th	34.2 MPH 55.0 KPH	51.2 MPH 82.4 KPH	68.3 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
4th Overdrive (LH)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	1.8 MPH 2.9 KPH	2.7 MPH 4.3 KPH	3.7 MPH 5.9 KPH	4.6 MPH 7.4 KPH	5.5 MPH 12.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	15.3 MPH 24.6 KPH	22.9 MPH 36.8 KPH	30.6 MPH 49.2 KPH	38. MPH 61.5 KPH	45.9 MPH 73.9 KPH
2nd	21.4 MPH 34.4 KPH	32.1 MPH 51.7 KPH	42.8 MPH 68.9 KPH	53.6 MPH 86.3 KPH	64.3 MPH 103.5 KPH
3rd	28.6 MPH 46.0 KPH	42.9 MPH 69.0 KPH	57.2 MPH 92.0 KPH	71.6 MPH 115.2 KPH	85.9 MPH 138.2 KPH
3rd Overdrive (LH)	34.9 MPH 56.2 KPH	52.4 MPH 84.3 KPH	69.8 MPH 112.3 KPH	87.3 MPH 140.5 KPH	104.7 MPH 168.5 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.5 MPH 144.0 KPH	107.4 MPH 172.8 KPH
4th Overdrive (LH)	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	3.0 MPH 4.8 KPH	4.4 MPH 7.1 KPH	5.9 MPH 9.5 KPH	7.4 MPH 11.9 KPH	8.9 MPH 14.3 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	16.2 MPH 26.1 KPH	24.2 MPH 38.9 KPH	32.3 MPH 52.0 KPH	40.4 MPH 65.0 KPH	48.5 MPH 78.1 KPH
2nd	22.6 MPH 36.4 KPH	34.0 MPH 54.7 KPH	45.3 MPH 72.9 KPH	56.6 MPH 91.1 KPH	67.9 MPH 109.3 KPH
3rd	30.2 MPH 48.6 KPH	45.4 MPH 73.0 KPH	60.5 MPH 97.4 KPH	75.6 MPH 121.6 KPH	90.7 MPH 145.6 KPH
3rd Overdrive (LH)	36.9 MPH 59.4 KPH	55.3 MPH 89.0 KPH	73.8 MPH 118.8 KPH	92.2 MPH 148.4 KPH	110.6 MPH 178.0 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.7 KPH	94.5 MPH 152. KPH 1	113.4 MPH 182.5 KPH
4th Overdrive (LH)	46.1 MPH 74.2 KPH	69.2 MPH 11.4 KPH	92.2 MPH 148.4 KPH	115.2 MPH 185.4 KPH	138.3 MPH 222.6 KPH
Reverse	3.1 MPH 5.0 KPH	4.7 MPH 7.6 KPH	6.2 MPH 10.0 KPH	7.8 MPH 12.5 KPH	9.4 MPH 15.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980

MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	16.0 MPH 25.7 KPH	23.9 MPH 38.5 KPH	31.9 MPH 51.3 KPH	39.9 MPH 64.2 KPH	47.8 MPH 76.9 KPH
2nd	23.4 MPH 37.7 KPH	35.1MPH 56.5 KPH	46.8 MPH 75.3 KPH	58.5 MPH 94.1 KPH	70.2 MPH 113.0 KPH
3rd	29.9 MPH 48.1 KPH	44.9 MPH 72.3 KPH	59.8 MPH 96.2 KPH	74.8 MPH 120.4 KPH	89.7 MPH 144.4 KPH
3rd Overdrive (LH)	38.1MPH 61.3 KPH	57.2MPH 92.0 KPH	76.2 MPH 122.6 KPH	95.3 MPH 153.4 KPH	114.4 MPH 184.1 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
4th Overdrive (LH)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	3.2 MPH 5.1 KPH	4.8 MPH 7.7 KPH	6.5 MPH 10.5 KPH	8.1 MPH 13.0 KPH	9.7 MPH 15.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	16.1 MPH 26.0 KPH	24.2 MPH 38.9 KPH	32.3 MPH 52.0 KPH	40.3 MPH 64.9 KPH	48.4 MPH 77.9 KPH
2nd	23.7 MPH 38.1 KPH	35.5 MPH 57.1 KPH	47.3 MPH 76.1 KPH	59.1 MPH 95.1 KPH	71.0 MPH 114.3 KPH
3rd	30.3 MPH 48.8 KPH	45.4 MPH 73.1 KPH	60.5 MPH 97.4 KPH	75.6 MPH 121.7 KPH	90.8 MPH 146.3 KPH
3rd Overdrive (LH)	38.5 MPH 62.0 KPH	57.8 MPH 93.0 KPH	77.1 MPH 124.1KPH	96.4 MPH 155.1 KPH	115.6 MPH 186.0 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
4th Overdrive (LH)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	3.3 MPH 5.3 KPH	4.9 MPH 7.9 KPH	6.5 MPH 10.5 KPH	8.2 MPH 13.2 KPH	9.8 MPH 15.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH* w/ 3.307:1 (13/43) crownwheel and pinion gearset, (BMC Part # BTB 841[1] and Special Tuning Part # BTB 900[1]), Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	18.1 MPH 29.1 KPH	27.1 MPH 43.6 KPH	36.1 MPH 58.1 KPH	45.2 MPH 72.7 KPH	54.2 MPH 87.2 KPH
2nd	25.3 MPH 40.7 KPH	38.0 MPH 61.1 KPH	50.7 MPH 81.6 KPH	63.3 MPH 101.9 KPH	76.0 MPH 122.3 KPH
3rd	33.8 MPH 54.4 KPH	50.8 MPH 81.7 KPH	67.7 MPH 108.9 KPH	84.6 MPH 136.1 KPH	101.5 MPH 163.5 KPH
3rd Overdrive (LH)	41.3 MPH 66.5 KPH	61.9 MPH 99.6 KPH	82.5 MPH 132.8 KPH	103.2 MPH 166.1 KPH	123.8 MPH 199.2 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
4th Overdrive (LH)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	3.5 MPH 5.6 KPH	5.2 MPH 8.4 KPH	7.0 MPH 11.3 KPH	8.7 MPH 14.0 KPH	10.5 MPH 16.9 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset, (BMC Part # BTB 841[1] and Special Tuning Part # BTB 900), Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	19.5 MPH 31.4 KPH	29.2 MPH 47.0 KPH	38.9 MPH 62.6 KPH	48.7 MPH 78.4 KPH	58.4 MPH 94.0 KPH
2nd	27.3 MPH 43.9 KPH	40.9 MPH 65.8 KPH	54.7 MPH 88.0 KPH	68.2 MPH 109.8 KPH	81.8 MPH 131.6 KPH
3rd	36.4 MPH 58.6 KPH	54.7 MPH 88.0 KPH	72.9 MPH 117.3 KPH	91.1 MPH 146.6 KPH	109.3 MPH 175.9 KPH
3rd Overdrive (LH)	44.4 MPH 71.4 KPH	66.7 MPH 107.3 KPH	88.9 MPH 143.0 KPH	111.1 MPH 178.8 KPH	133.4 MPH 214.7 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
4th Overdrive (LH)	56.8 MPH 91.4 KPH	85.2 MPH 137.1 KPH	113.6 MPH 182.8 KPH	142.0 MPH 228.5 KPH	170.4 MPH 274.2 KPH
Reverse	1.5 MPH 2.4 KPH	2.3 MPH 3.7 KPH	3.1 MPH 5.0 KPH	3.9 MPH 6.3 KPH	4.6 MPH 7.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 840) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

For those who might deem the factory's synchromesh design to be inappropriate for their particular needs, Jack Knight offers alternative straight-cut close-ratio gearsets for both of the three-synchro and the four-synchro transmissions in the same ratios as those of the original factory close-ratio gearsets that the factory racing team used. It should be noted that these immensely strong transmissions are dog-type designs that use custom-made shifter forks and dispense with the use of synchro hubs, and are specifically intended to withstand the rigors of racing use. Jack Knight has a website at:
http://www.jackknight.co.uk/

For those who do not mind using non-MG components, there is the option of converting to a modern five-speed transmission. This is offered as a kit by Hi-Gear engineering in the UK, and is available through vendors outside the UK, including North America. Perhaps one of the best features of the kit is that installation requires no modifications to the car, including into the narrower transmission tunnel of the MGB MKI (or MGA), and as such is a completely reversible modification that will permit the car to be easily refitted with an Original Equipment transmission.

The Ford Type 9 (Type N) transmission, which was Ford's first five-speed, rear wheel drive transmission, is based on the earlier four-speed Ford Type E transmission. The fifth, or overdrive, gear was added to the pre-existing four-speed transmission design by placing it inside of the extension housing. While this made production easier and cheaper, it effectively limits its torque capacity in its standard version to 200 Ft-lbs. The Ford Type 9 is a popular choice for five-speed conversions because its design incorporates the use of a separate bellhousing so that it can be easily adapted to different engines. There are two different kinds of bellhousings offered for use in the MGB, depending on the engine and starter configuration. It uses a one inch diameter 23 spline input shaft (first motion shaft), and its main gear housing is of cast iron, plus a bolt-on cast aluminum extension housing. The conversion kit for the MGB is satisfyingly complete, including a high-strength LM25TF heat-treated aluminum alloy bell housing that will attach to the engine backplate of the BMC B Series engine without any modifications required, the clutch release lever pivot & bolt, a bell housing to gearbox gasket, an extended spigot bush, a new propshaft (driveshaft) assembly that matches the original manufacturer's specification of handling 422 Ft-lbs of torque for a short duration, and a maximum speed of 7,000 RPM. Unlike the Hardy-Spicer driveshaft (propeller shaft) that is Original Equipment on the MGB, this driveshaft (propeller shaft) is not attached to a flange on the transmission and does not telescope, but instead floats on the end of the output shaft. All of the kits come with a rear rubber gearbox mounting, a new transmission support crossmember, a modified short throw gear lever assembly which places the gear lever in exactly the same place as the original so that the conversion is visibly undetectable, a speedometer cable and circlip, all of the required bolts and fasteners, detailed fitting instructions, and even two types of Loctite. The kit conveniently retains use of the standard MGB clutch cover, clutch driven plate, release bearing, release lever, and clutch slave cylinder. Interestingly, its total weight is thirty-five pounds lighter than that of an Original Equipment overdrive-equipped BMC transmission.

These conversion kits come in variants of two basic versions. The first variant is a rebuilt Ford "Type 9" 5 five-speed transmission as was used in the 4 cylinder English Ford Sierra sedans outside of the US Market. Two different first gear ratios are available. This comes with either a non-modified front cover that uses the standard input sleeve and throw out (release) bearing and collar length, and the "short" input shaft (160mm), or a modified (you specify) front cover. The T9 transmissions came with four different front covers. The front cover actually doubles as the oil seal holder and the throw out (release) bearing guide. Since there were different transmissions (both standard and heavy duty) and different throw out (release) bearing sizes (2.0/2.8 and Scorpio), there were four different front covers made. It is rebuilt with all new synchros, layshaft, bearings, seals and gaskets. It is also available with a choice of two different first gear ratios. For those who have extra-powerful engines or intend to use this version of the transmission for racing applications, there are also equivalent heavy-duty versions available.

The second version is a completely rebuilt Ford Type 9 five-speed "Semi Close Ratio" or a "2.8 Sport" designation transmission as used in the 6 cylinder English Ford Sierra sedans outside of the US Market. These were also the standard transmission on the Mercury Merkur in the USA. This comes with a non-modified or modified (you specify) front cover and the "short" (160mm) input shaft (first motion shaft). It is a rebuilt unit with an all new laygear, layshaft, bearings, seals and gaskets, as well as a new input shaft (first motion shaft), plus synchros in which they replace the brass baulk rings (synchronizer rings) and test the spring loads for broken or weak springs. Two different first gear ratios and three different fifth gear ratios are available.. For those who have an extra-powerful engine with an output of up to 250 Ft-lbs of torque, or intend to use this version of the transmission for racing applications, there is also a heavy-duty version with two different available first gear ratios and four different available fifth gear ratios, thus making it readily adaptable to use on twisty race tracks. The Heavy Duty version of the Ford T9 Transmission has a "split" layshaft with a larger front caged bearing and a larger diameter on the layshaft for its 4th gear end at the front of the transmission. The rear bearing is actually case-mounted and rides on the surface of the laygear itself. This makes it possible to deal with higher torque loads for the transmission. This particular design change was done assuming the stresses involved in racing use. The larger front diameter of the layshaft and larger bearing gives it more strength in 3rd, 4th, and 5th gear, which are the gears normally used for racing. The other gears, synchros, and bearings are the same as in the "standard" T9. All of the "standard" Type 9 bearings are comparatively much heavier than most other British transmission bearings from any other manufacturer other than from Ford itself.

Hi-Gear Engineering Ford Type 9 Five-Speed Transmission, Low 1st Gear Ratio:

1st	3.65 : 1
2nd	1.97 : 1
3rd	1.37 : 1
4th	1.000 :1
5th	0.82 : 1
Reverse	3.66 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.85	1,496 RPM	1,754 RPM	2,527 RPM	2,973 RPM
2nd-3rd	1.44	993 RPM	2,257 RPM	1,681 RPM	3,819 RPM
3rd-4th	1.37	878 RPM	2,372 RPM	1,486 RPM	4,014 RPM
4th-5th	1.21	564 RPM	2,686 RPM	955 RPM	4,545 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	7.5 MPH 12.1 KPH	11.2 MPH 18.0 KPH	15.0 MPH 24.1 KPH	18.7 MPH 30.1 KPH	22.4 MPH 36.0 KPH
2nd	13.8 MPH 22.2 KPH	20.8 MPH 33.5 KPH	27.7 MPH 44.6 KPH	34.6 MPH 55.7 KPH	41.6 MPH 66.9 KPH
3rd	19.9 MPH 32.0 KPH	29.9 MPH 48.1 KPH	39.8 MPH 64.0 KPH	49.8 MPH 80.1 KPH	59.8 MPH 96.2 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
5th	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	7.4 MPH 11.9 KPH	11.2 MPH 18.0 KPH	14.9 MPH 24.0 KPH	18.6 MPH 29.9 KPH	22.4 MPH 36.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 crownwheel (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	7.9 MPH 12.7 KPH	11.8 MPH 19.0 KPH	15.7 MPH 25.3 KPH	19.6 MPH 31.5 KPH	23.6 MPH 38.0 KPH
2nd	14.6 MPH 23.5 KPH	21.8 MPH 35.1 KPH	29.1 MPH 46.8 KPH	36.4 MPH 58.6 KPH	43.7 MPH 70.3 KPH
3rd	20.9 MPH 33.6 KPH	31.4 MPH 50.5 KPH	41.9 MPH 67.4 KPH	52.4 MPH 84.3 KPH	62.8 MPH 101.1 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
5th	32.1MPH 51.7 KPH	48.2 MPH 77.6 KPH	70.0 MPH 112.6 KPH	80.3 MPH 129.2 KPH	96.4 MPH 155.1 KPH
Reverse	7.8 MPH 12.6 KPH	11.8 MPH 19.0 KPH	15.7 MPH 25.3 KPH	19.6 MPH 31.5 KPH	23.5 MPH 37.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.1 MPH 34.0 KPH	25.3 MPH 40.7 KPH
2nd	15.6 MPH 25.1 KPH	23.4 MPH 37.7 KPH	31.2 MPH 50.2 KPH	39.0 MPH 62.8 KPH	46.8 MPH 75.3 KPH
3rd	22.4 MPH 36.0 KPH	33.7 MPH 54.2 KPH	44.9 MPH 72.3 KPH	56.1 MPH 90.3 KPH	67.3 MPH 108.3KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
5th	37.5 MPH 60.3 KPH	56.2 MPH 90.4 KPH	75.0 MPH 120.7 KPH	93.7 MPH 150.8 KPH	112.5 MPH 181.0 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980

MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion rear axle gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.9 MPH 14.3 KPH	13.4 MPH 21.6 KPH	17.8 MPH 28.6 KPH	22.3 MPH 35.9 KPH	26.7 MPH 43.0 KPH
2nd	16.5 MPH 26.6 KPH	24.8 MPH 39.9 KPH	33.0 MPH 53.1 KPH	41.3 MPH 66.5 KPH	49.5 MPH 79.7 KPH
3rd	23.7 MPH 36.5 KPH	35.6 MPH 57.3 KPH	47.5 MPH 76.4 KPH	59.4 MPH 95.6 KPH	71.2 MPH 114.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
5th	39.7 MPH 63.8 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	8.9 MPH 14.3 KPH	13.3 MPH 21.4 KPH	17.8 MPH 28.6 KPH	22.2 MPH 35.7 KPH	26.7 MPH 43.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	9.1 MPH 14.6 KPH	13.6 MPH 21.9 KPH	18.2 MPH 29.3 KPH	22.7 MPH 36.5 KPH	27.2 MPH 43.8 KPH
2nd	16.9 MPH 27.2 KPH	25.2 MPH 40.6 KPH	33.7 MPH 54.2 KPH	42.1 MPH 67.7 KPH	50.5 MPH 81.3 KPH
3rd	24.2 MPH 38.9 KPH	36.3 MPH 58.4 KPH	48.4 MPH 77.9 KPH	60.1 MPH 96.7 KPH	72.6 MPH 116.8KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
5th	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	9.1 MPH 14.6 KPH	13.6 MPH 21.9 KPH	18.1 MPH 29.1 KPH	22.6 MPH 36.4 KPH	27.2 MPH 43.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion rear axle gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.3 MPH 15.0 KPH	14.0 MPH 22.5 KPH	18.7 MPH 30.1 KPH	23.4 MPH 37.7 KPH	28.0 MPH 45.1 KPH
2nd	17.3 MPH 27.8 KPH	26.0 MPH 41.8 KPH	34.6 MPH 55.7 KPH	43.3 MPH 69.7 KPH	52.0 MPH 83.7 KPH
3rd	24.9 MPH 40.1 KPH	37.3 MPH 60.0 KPH	49.8 MPH 80.1 KPH	62.3 MPH 100.3 KPH	74.7 MPH 120.2 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
5th	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	9.3 MPH 15.0 KPH	14.0 MPH 22.5 KPH	18.6 MPH 29.9 KPH	23.3 MPH 37.5 KPH	28.0 MPH 45.1KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.8 MPH 15.8 KPH	14.7 MPH 23.7 KPH	19.6 MPH 31.5 KPH	24.5 MPH 39.4 KPH	29.4 MPH 47.3 KPH
2nd	18.2 MPH 29.3 KPH	27.2 MPH 43.8 KPH	36.3 MPH 58.4 KPH	45.4 MPH 73.1 KPH	54.5 MPH 87.7KPH
3rd	26.1 MPH 42.0 KPH	39.2 MPH 63.1 KPH	52.2 MPH 84.0 KPH	65.3 MPH 105.1 KPH	78.4 MPH 126.2 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
5th	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	9.8 MPH 15.8 KPH	14.7 MPH 23.7 KPH	19.5 MPH 31.4 KPH	24.4 MPH 39.3 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset, (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.4 MPH 16.7 KPH	15.5 MPH 24.9 KPH	20.7 MPH 33.3 KPH	25.9 MPH 41.7 KPH	31.1 MPH 50.0 KPH
2nd	19.2 MPH 30.1 KPH	28.8 MPH 46.3 KPH	38.4 MPH 61.8 KPH	48.0 MPH 77.2 KPH	57.6 MPH 92.7 KPH
3rd	27.6 MPH 44.4 KPH	41.4 MPH 66.6 KPH	55.2 MPH 88.8 KPH	70.0 MPH 112.7 KPH	82.8 MPH 133.2 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
5th	46.1 MPH 74.2 KPH	69.2 MPH 11.42 KPH	92.2 MPH 148.42 KPH	115.2 MPH 185.42 KPH	138.3 MPH 222.62 KPH
Reverse	10.3 MPH 16.6 KPH	15.5 MPH 24.9 KPH	20.7 MPH 33.3 KPH	25.8 MPH 41.5 KPH	30.9 MPH 49.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.7 MPH 17.2 KPH	16.0 MPH 25.7 KPH	21.4 MPH 34.4 KPH	26.7 MPH 43.0 KPH	32.1 MPH 51.7 KPH
2nd	19.8 MPH 31.9 KPH	29.7 MPH 47.8 KPH	39.6 MPH 63.7 KPH	49.5 MPH 79.7 KPH	59.4 MPH 95.6 KPH
3rd	28.5 MPH 45.9 KPH	42.7 MPH 68.7 KPH	57.0 MPH 91.7 KPH	71.2 MPH 114.6 KPH	85.5 MPH 137.6 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
5th	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	10.7 MPH 17.2 KPH	16.0 MPH 25.7 KPH	21.3 MPH 34.3 KPH	26.7 MPH 43.0 KPH	32.0 MPH 51.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.8 MPH 17.4 KPH	16,2 MPH 26.1 KPH	21.6 MPH 34.8 KPH	27.0 MPH 43.5 KPH	32.4 MPH 52.1 KPH
2nd	20.0 MPH 32.2 KPH	30.0 MPH 48.3 KPH	40.1 MPH 64.5 KPH	50.1 MPH 80.6 KPH	60.1 MPH 96.7 KPH
3rd	28.8 MPH 46.3 KPH	43.2 MPH 69.5 KPH	57.6 MPH 92.7 KPH	72.0 MPH 115.9 KPH	86.4 MPH 139.0 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
5th	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	10.8 MPH 17.4 KPH	16.2 MPH 26.1 KPH	21.6 MPH 34.8 KPH	26.9 MPH 43.3 KPH	32.3 MPH 52.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production 1962-1968 MGB roadsters, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.6 MPH 18.7 KPH	17.4 MPH 28.0 KPH	23.2 MPH 37.3 KPH	29.0 MPH 46.7 KPH	34.8 MPH 56.0 KPH
2nd	21.5 MPH 34.6 KPH	32.2 MPH 51.8 KPH	42.9 MPH 69.0 KPH	53.7 MPH 86.4 KPH	64.4 MPH 103.6 KPH
3rd	30.9 MPH 49.7 KPH	46.3 MPH 74.5 KPH	61.7 MPH 99.3 KPH	77.2 MPH 124.2 KPH	92.6 MPH 149.0 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
5th	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	11.6 MPH 18.7 KPH	17.3 MPH 27.8 KPH	23.1 MPH 37.2 KPH	28.9 MPH 46.5 KPH	34.7 MPH 55.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.5 MPH 20.1 KPH	18.7 MPH 30.1 KPH	25.0 MPH 40.2 KPH	31.2 MPH 50.2 KPH	37.4 MPH 60.2 KPH
2nd	23.1 MPH 37.2 KPH	34.7 MPH 55.8 KPH	46.2 MPH 74.3 KPH	57.8 MPH 93.0 KPH	69.4 MPH 111.7 KPH
3rd	33.2 MPH 53.4 KPH	49.9 MPH 80.3 KPH	66.5 MPH 107.0 KPH	83.1 MPH 133.7 KPH	99.7 MPH 160.4 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
5th	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	12.4 MPH 20.0 KPH	18.7 MPH 30.1 KPH	24.9 MPH 40.1 KPH	31.1 MPH 50.0 KPH	37.3 MPH 60.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

High Gear Engineering Ford Type 9 Five-Speed Transmission, High 1st Gear Ratio:

1st	2.98 : 1
2nd	1.97 : 1
3rd	1.37 : 1
4th	1.00 : 1
5th	0.82 : 1
Reverse	3.66 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.51	1,048 RPM	2,152 RPM	1,858 RPM	3,642 RPM
2nd-3rd	1.44	993 RPM	2,257 RPM	1,681 RPM	3,819 RPM
3rd-4th	1.37	878 RPM	2,372 RPM	1,486 RPM	4,014 RPM
4th-5th	1.21	564 RPM	2,686 RPM	955 RPM	4,545 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.2 MPH 14.8 KPH	13.7 MPH 36.2 KPH	18.3 MPH 29.4 KPH	22.9 MPH 36.8 KPH	27.5 MPH 44.2 KPH
2nd	13.8 MPH 22.2 KPH	20.8 MPH 33.5 KPH	27.7 MPH 44.6 KPH	34.6 MPH 55.7 KPH	41.6 MPH 66.9 KPH
3rd	19.9 MPH 32.0 KPH	29.9 MPH 48.1 KPH	39.8 MPH 64.0 KPH	49.8 MPH 80.1 KPH	59.8 MPH 96.2 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
5th	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	7.4 MPH 11.9 KPH	11.2 MPH 18.0 KPH	14.9 MPH 24.0 KPH	18.6 MPH 29.9 KPH	22.4 MPH 36.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 crownwheel (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.6 MPH 15.4 KPH	14.4 MPH 23.2 KPH	19.3 MPH 31.1 KPH	24.1 MPH 38.8 KPH	28.9 MPH 46.5 KPH
2nd	14.6 MPH 23.5 KPH	21.8 MPH 35.1 KPH	29.1 MPH 46.8 KPH	36.4 MPH 58.6 KPH	43.7 MPH 70.3 KPH
3rd	20.9 MPH 33.6 KPH	31.4 MPH 50.5 KPH	41.9 MPH 67.4 KPH	52.4 MPH 84.4 KPH	62.8 MPH 101.1 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
5th	32.1MPH 51.7 KPH	48.2 MPH 77.6 KPH	70.0 MPH 112.6 KPH	80.3 MPH 129.2 KPH	96.4 MPH 155.1 KPH
Reverse	7.8 MPH 12.6 KPH	11.8 MPH 19.0 KPH	15.7 MPH 25.3 KPH	19.6 MPH 31.5 KPH	23.5 MPH 37.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.3 MPH 16.6 KPH	15.5 MPH 24.9 KPH	20.6 MPH 33.1 KPH	25.8 MPH 41.5 KPH	30.9 MPH 49.7 KPH
2nd	15.6 MPH 25.1 KPH	23.4 MPH 37.7 KPH	31.2 MPH 50.2 KPH	39.0 MPH 62.8 KPH	46.8 MPH 75.3 KPH
3rd	22.4 MPH 36.0 KPH	33.7 MPH 54.2 KPH	44.9 MPH 72.3 KPH	56.1 MPH 90.3 KPH	67.3 MPH 108.3KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
5th	37.5 MPH 60.3 KPH	56.2 MPH 90.4 KPH	75.0 MPH 120.7 KPH	93.7 MPH 150.8 KPH	112.5 MPH 181.0 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion rear axle gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.9 MPH 17.5 KPH	16.4 MPH 26.4 KPH	21.8 MPH 35.1 KPH	27.3 MPH 43.9 KPH	32.7 MPH 60.2 KPH
2nd	16.5 MPH 26.6 KPH	24.8 MPH 39.9 KPH	33.0 MPH 53.1 KPH	41.3 MPH 66.5 KPH	49.5 MPH 79.7 KPH
3rd	23.7 MPH 36.5 KPH	35.6 MPH 57.3 KPH	47.4 MPH 76.3 KPH	59.4 MPH 95.6 KPH	71.2 MPH 114.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
5th	39.7 MPH 63.8 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	8.9 MPH 14.3 KPH	13.3 MPH 21.4 KPH	17.8 MPH 28.6 KPH	22.2 MPH 35.7 KPH	26.7 MPH 43.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.1 MPH 17.9 KPH	16.7 MPH 26.9 KPH	22.2 MPH 35.7 KPH	27.8 MPH 44.7 KPH	33.4 MPH 53.7 KPH
2nd	16.9 MPH 27.2 KPH	25.2 MPH 40.6 KPH	33.7 MPH 54.2 KPH	42.1 MPH 67.7 KPH	50.5 MPH 81.3 KPH
3rd	24.2 MPH 38.9 KPH	36.3 MPH 58.4 KPH	48.4 MPH 77.9 KPH	60.1 MPH 96.7 KPH	72.6 MPH 116.8KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
5th	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	9.1 MPH 14.6 KPH	13.6 MPH 21.9 KPH	18.1 MPH 29.1 KPH	22.6 MPH 36.4 KPH	27.2 MPH 43.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion rear axle gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.4 MPH 18.3 KPH	17.2 MPH 27.7 KPH	22.9 MPH 36.8 KPH	28.6 MPH 46.0 KPH	34.3 MPH 55.2 KPH
2nd	17.3 MPH 27.8 KPH	26.0 MPH 41.8 KPH	34.6 MPH 55.7 KPH	43.3 MPH 69.7 KPH	52.0 MPH 83.7 KPH
3rd	24.9 MPH 40.1 KPH	37.3 MPH 60.0 KPH	49.8 MPH 80.1 KPH	62.3 MPH 100.3 KPH	74.7 MPH 120.2 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
5th	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	9.3 MPH 15.0 KPH	14.0 MPH 22.5 KPH	18.6 MPH 29.9 KPH	23.3 MPH 37.5 KPH	28.0 MPH 45.1KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.0 MPH 19.3 KPH	18.0 MPH 29.0 KPH	24.0 MPH 38.6 KPH	30.0 MPH 48.3 KPH	36.0 MPH 57.9 KPH
2nd	18.2 MPH 29.3 KPH	27.2 MPH 43.8 KPH	36.3 MPH 58.4 KPH	45.4 MPH 73.1 KPH	54.5 MPH 87.7KPH
3rd	26.1 MPH 42.0 KPH	39.2 MPH 63.1 KPH	52.2 MPH 84.0 KPH	65.3 MPH 105.1 KPH	78.4 MPH 126.2 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
5th	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	9.8 MPH 15.8 KPH	14.7 MPH 23.7 KPH	19.5 MPH 31.4 KPH	24.4 MPH 39.3 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset, (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.7 MPH 20.4 KPH	19.2 MPH 30.9 KPH	25.4 MPH 40.9 KPH	31.7 MPH 51.0 KPH	38.1 MPH 61.3 KPH
2nd	19.2 MPH 30.1 KPH	28.8 MPH 46.3 KPH	38.4 MPH 61.8 KPH	48.0 MPH 77.2 KPH	57.6 MPH 92.7 KPH
3rd	27.6 MPH 44.4 KPH	41.4 MPH 66.6 KPH	55.2 MPH 88.8 KPH	70.0 MPH 112.7 KPH	82.8 MPH 133.2 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
5th	46.1 MPH 74.2 KPH	69.2 MPH 11.42 KPH	92.2 MPH 148.42 KPH	115.2 MPH 185.42 KPH	138.3 MPH 222.62 KPH
Reverse	10.3 MPH 16.6 KPH	15.5 MPH 24.9 KPH	20.7 MPH 33.3 KPH	25.8 MPH 41.5 KPH	30.9 MPH 49.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	13.1 MPH 21.1 KPH	19.7 MPH 31.7 KPH	26.2 MPH 42.2 KPH	32.7 MPH 52.6 KPH	39.3 MPH 63.2 KPH
2nd	19.8 MPH 31.9 KPH	29.7 MPH 47.8 KPH	39.6 MPH 63.7 KPH	49.5 MPH 79.7 KPH	59.4 MPH 95.6 KPH
3rd	28.5 MPH 45.9 KPH	42.7 MPH 68.7 KPH	57.0 MPH 91.7 KPH	71.2 MPH 114.6 KPH	85.5 MPH 137.6 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
5th	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	10.7 MPH 17.2 KPH	16.0 MPH 25.7 KPH	21.3 MPH 34.3 KPH	26.7 MPH 43.0 KPH	32.0 MPH 51.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	13.2 MPH 21.2 KPH	19.9 MPH 32.0 KPH	26.5 MPH 42.6 KPH	33.1 MPH 53.3 KPH	39.7 MPH 63.9KPH
2nd	20.0 MPH 32.2 KPH	30.0 MPH 48.3 KPH	40.1 MPH 64.5 KPH	50.1 MPH 80.6 KPH	60.1 MPH 96.7 KPH
3rd	28.8 MPH 46.3 KPH	43.2 MPH 69.5 KPH	57.6 MPH 92.7 KPH	72.0 MPH 115.9 KPH	86.4 MPH 139.0 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
5th	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	10.8 MPH 17.4 KPH	16.2 MPH 26.1 KPH	21.6 MPH 34.8 KPH	26.9 MPH 43.3 KPH	32.3 MPH 52.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production 1962-1968 MGB roadsters, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	14.2 MPH 22.8 KPH	21.3 MPH 34.3 KPH	28.4 MPH 45.7 KPH	35.5 MPH 57.1 KPH	42.6 MPH 68.6 KPH
2nd	21.5 MPH 34.6 KPH	32.2 MPH 51.8 KPH	42.9 MPH 69.0 KPH	53.7 MPH 86.4 KPH	64.4 MPH 103.6 KPH
3rd	30.9 MPH 49.7 KPH	46.3 MPH 74.5 KPH	61.7 MPH 99.3 KPH	77.2 MPH 124.2 KPH	92.6 MPH 149.0 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
5th	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	11.6 MPH 18.7 KPH	17.3 MPH 27.8 KPH	23.1 MPH 37.2 KPH	28.9 MPH 46.5 KPH	34.7 MPH 55.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	15.3 MPH 24.6 KPH	22.9 MPH 36.8 KPH	30.6 MPH 49.2 KPH	38.2 MPH 61.5 KPH	45.8 MPH 73.7KPH
2nd	23.1 MPH 37.2 KPH	34.7 MPH 55.8 KPH	46.2 MPH 74.3 KPH	57.8 MPH 93.0 KPH	69.4 MPH 111.7 KPH
3rd	33.2 MPH 53.4 KPH	49.9 MPH 80.3 KPH	66.5 MPH 107.0 KPH	83.1 MPH 133.7 KPH	99.7 MPH 160.4 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
5th	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	12.4 MPH 20.0 KPH	18.7 MPH 30.1 KPH	24.9 MPH 40.1 KPH	31.1 MPH 50.0 KPH	37.3 MPH 60.0 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Hi-Gear Engineering Ford Type 9 2.8 Closer Ratio Five-Speed Transmission, Low 1st Gear Ratio:

1st	3.36 : 1
2nd	1.81 : 1
3rd	1.26 : 1
4th	1.000 : 1
5th	0.825 : 1
Reverse	3.87 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.86	1,503 RPM	1,747 RPM	2543 RPM	2,957 RPM
2nd-3rd	1.44	993 RPM	2,257 RPM	1,681 RPM	3,819 RPM
3rd-4th	1.26	671 RPM	2,579 RPM	1,135 RPM	4,365 RPM
4th-5th	1.21	564 RPM	2,686 RPM	955 RPM	4,545 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.1 MPH 13.0 KPH	12.2 MPH 19.6 KPH	16.2 MPH 26.1 KPH	20.3 MPH 32.7 KPH	24.4 MPH 39.3 KPH
2nd	13.6 MPH 21.9 KPH	20.3 MPH 32.7 KPH	27.1 MPH 43.6 KPH	33.9 MPH 54.6 KPH	40.7 MPH 65.5 KPH
3rd	21.7 MPH 34.9 KPH	32.5 MPH 52.3 KPH	43.3 MPH 70.2 KPH	54.1 MPH 87.1 KPH	65.0 MPH 104.6 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
5th	33.1 MPH 53.3 KPH	49.6 MPH 79.8 KPH	66.2 MPH 10.0 KPH	82.7 MPH 133.9 KPH	99.2 MPH 159.6 KPH
Reverse	7.0 MPH 11.3 KPH	10.6 MPH 17.1 KPH	14.1 MPH 22.7 KPH	17.6 MPH 28.3 KPH	21.2 MPH 34.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 crownwheel (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	8.5 MPH 13.7 KPH	12.8 MPH 20.6 KPH	17.1 MPH 27.5 KPH	21.3 MPH 34.3 KPH	25.6 MPH 41.2 KPH
2nd	15.8 MPH 25.4 KPH	23.8 MPH 38.3 KPH	31.7 MPH 51.0 KPH	39.6 MPH 63.7 KPH	47.5 MPH 76.4 KPH
3rd	22.8 MPH 36.7 KPH	34.2 MPH 55.0KPH	45.5 MPH 73.2 KPH	56.9 MPH 91.6 KPH	68.3 MPH 109.9 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
5th	34.8 MPH 56.0 KPH	52.2 MPH 84.0 KPH	69.6 MPH 112.0 KPH	86.9 MPH 139.8 KPH	104.3 MPH 167.8 KPH
Reverse	7.8 MPH 12.5 KPH	11.7 MPH 18.8 KPH	15.7 MPH 25.3 KPH	19.6 MPH 31.5 KPH	23.5 MPH 37.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.1 MPH 14.6 KPH	13.7 MPH 22.0 KPH	18.3 MPH 29.4 KPH	22.9 MPH 36.8 KPH	27.4 MPH 44.1 KPH
2nd	17.0 MPH 27.4 KPH	25.5 MPH 41.0 KPH	34.0 MPH 54.7 KPH	42.5 MPH 68.4 KPH	51.0 MPH 82.1 KPH
3rd	24.4 MPH 39.3 KPH	36.6 MPH 58.9 KPH	48.8 MPH 78.5 KPH	61.0 MPH 98.2 KPH	73.2 MPH 117.8 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
5th	37.3 MPH 60.0 KPH	55.9 MPH 90.0 KPH	74.5 MPH 120.0 KPH	93.2 MPH 150.0 KPH	111.8 MPH 180.0 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion rear axle gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	9.7 MPH 15.6 KPH	14.5 MPH 23.3 KPH	19.4 MPH 31.2 KPH	24.2 MPH 38.9 KPH	29.0 MPH 46.7 KPH
2nd	18.0 MPH 29.0 KPH	27.0 MPH 43.5 KPH	35.9 MPH 57.8 KPH	44.9 MPH 72.3 KPH	53.9 MPH 86.7 KPH
3rd	25.8 MPH 41.5 KPH	38.7 MPH 62.3 KPH	51.6 MPH 83.0 KPH	64.5 MPH 103.8 KPH	77.5 MPH 124.7 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
5th	39.4 MPH 63.4 KPH	59.1 MPH 95.1 KPH	78.9 MPH 127.0 KPH	98.6 MPH 158.7 KPH	118.3 MPH 190.4 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.8 MPH 15.8 KPH	14.8 MPH 23.8 KPH	19.7 MPH 31.7 KPH	24.7 MPH 39.7 KPH	29.6 MPH 47.6 KPH
2nd	18.3 MPH 29.4 KPH	27.5 MPH 44.3 KPH	36.6 MPH 58.9 KPH	45.8 MPH 73.7 KPH	54.9 MPH 88.3 KPH
3rd	26.3 MPH 42.3 KPH	39.5 MPH 63.6 KPH	52.6 MPH 84.6 KPH	65.8 MPH 105.9 KPH	78.9 MPH 127.0 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
5th	40.2 MPH 64.7 KPH	60.3 MPH 97.0 KPH	80.3 MPH 129.2 KPH	100.4 MPH 161.6 KPH	120.5 MPH 193.9 KPH
Reverse	8.6 MPH 13.8 KPH	12.8 MPH 20.6 KPH	17.1 MPH 27.5 KPH	21.4 MPH 34.4 KPH	25.7 MPH 41.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion rear axle gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.1 MPH 16.1 KPH	15.2 MPH 24.5 KPH	20.3 MPH 32.7 KPH	25.4 MPH 40.9 KPH	30.5 MPH 49.1 KPH
2nd	18.8 MPH 30.3 KPH	28.2 MPH 45.4 KPH	37.7 MPH 60.7 KPH	47.1 MPH 75.8 KPH	56.5 MPH 90.9 KPH
3rd	27.1 MPH 43.6 KPH	40.6 MPH 65.3 KPH	54.1 MPH 87.1 KPH	67.7 MPH 109.0 KPH	81.2 MPH 130.7 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
5th	41.3 MPH 66.5 KPH	62.0 MPH 99.8 KPH	82.7 MPH 133.1 KPH	103,4 MPH 166.4 KPH	124.0 MPH 199.6 KPH
Reverse	8.8 MPH 14.2 KPH	13.2 MPH 21.2 KPH	17.6 MPH 28.3 KPH	22.0 MPH 35.4 KPH	26.4 MPH 42.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.6 MPH 17.1 KPH	16.0 MPH 25.7 KPH	21.3 MPH 34.3 KPH	26.6 MPH 42.8 KPH	31.9 MPH 51.3 KPH
2nd	19.8 MPH 31.9 KPH	29.7 MPH 47.8 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	59.3 MPH 95.4 KPH
3rd	28.4 MPH 45.7 KPH	42.6 MPH 68.6 KPH	56.8 MPH 91.4 KPH	71.0 MPH 114.3 KPH	85.2 MPH 137.1 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
5th	43.4 MPH 69.8 KPH	65.1 MPH 104.8 KPH	86.7 MPH 139.5 KPH	108.4 MPH 174.4 KPH	130.1 MPH 209.4 KPH
Reverse	9.2 MPH 14.8 KPH	13.9 MPH 22.4 KPH	18.5 MPH 29.8 KPH	23.1 MPH 37.2 KPH	27.7 MPH 44.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset, (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	11.2 MPH 18.0 KPH	16.9 MPH 27.2 KPH	22.5 MPH 36.2 KPH	28.1 MPH 45.2 KPH	33.7 MPH 54.2 KPH
2nd	19.8 MPH 31.9 KPH	29.7 MPH 47.8 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	59.3 MPH 95.4 KPH
3rd	30.0 MPH 48.3 KPH	45.0 MPH 72.4 KPH	60.0 MPH 96.6 KPH	75.0 MPH 120.7 KPH	90.0 MPH 144.8 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
5th	45.8 MPH 73.7 KPH	68.7 MPH 110.6 KPH	91.6 MPH 147.4 KPH	114.6 MPH 184.4 KPH	137.5 MPH 221.3 KPH
Reverse	9.8 MPH 15.8 KPH	14.6 MPH 23.5 KPH	19.5 MPH 31.4 KPH	24.4 MPH 34.4 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	11.6 MPH 18.7 KPH	17.4 MPH 28.0 KPH	23.2 MPH 37.3 KPH	29.0 MPH 46.7 KPH	34.9 MPH 56.2 KPH
2nd	21.6 MPH 34.8 KPH	32.3 MPH 52.0 KPH	43.1 MPH 69.4 KPH	53.9 MPH 86.7 KPH	64.7 MPH 104.1 KPH
3rd	31.0 MPH 49.9 KPH	46.5 MPH 74.8 KPH	62.0 MPH 99.8 KPH	77.5 MPH 124.7 KPH	92.9 MPH 149.5 KPH
4th	39.1 MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
5th	47.3 MPH 76.1 KPH	71.0 MPH 114.3 KPH	94.6 MPH 152.2 KPH	118.3 MPH 190.4 KPH	142.0 MPH 228.5 KPH
Reverse	11.5 MPH 18.5 KPH	17.3 MPH 27.8 KPH	23.0 MPH 37.0 KPH	28.8 MPH 46.3 KPH	34.6 MPH 55.7 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	11.7 MPH 18.8 KPH	17.6 MPH 28.3 KPH	23.5 MPH 37.8 KPH	29.4 MPH 47.3 KPH	35.2 MPH 56.6 KPH
2nd	21.8 MPH 35.1 KPH	32.7 MPH 52.6 KPH	43.6 MPH 70.2 KPH	54.5 MPH 87.7 KPH	65.4 MPH 105.3 KPH
3rd	31.3 MPH 50.4 KPH	47.0 MPH 75.6 KPH	62.6 MPH 100.7 KPH	78.3 MPH 126.0 KPH	93.9 MPH 151.1 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
5th	47.8 MPH 76.9 KPH	71.7 MPH 115.4 KPH	95.7 MPH 154.0 KPH	119.6 MPH 192.5 KPH	143.5 MPH 231.0 KPH
Reverse	10.2 MPH 16.4 KPH	15.3 MPH 24.6 KPH	20.4 MPH 32.8 KPH	25.5 MPH 41.0 KPH	30.6 MPH 49.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production 1962-1968 MGB roadsters, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.6 MPH 20.3 KPH	18.9 MPH 30.4 KPH	25.2 MPH 40.6 KPH	31.5 MPH 50.7 KPH	37.8 MPH 60.8 KPH
2nd	23.4 MPH 37.7 KPH	35.0 MPH 56.3 KPH	46.7 MPH 75.2 KPH	58.4 MPH 94.0 KPH	70.1 MPH 112.8 KPH
3rd	33.6 MPH 54.1 KPH	50.4 MPH 81.1 KPH	67.1 MPH 108.0 KPH	83.9 MPH 135.0 KPH	100.7 MPH 162.0 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
5th	51.3 MPH 82.6 KPH	76.9 MPH 123.8 KPH	102.5 MPH 165.0 KPH	128.2 MPH 206.3 KPH	153.8 MPH 247.5 KPH
Reverse	10.9 MPH 17.5 KPH	16.4 MPH 26.4 KPH	21.9 MPH 35.2 KPH	27.3 MPH 43.9 KPH	32.8 MPH 51.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.6 MPH 21.9 KPH	20.3 MPH 32.7 KPH	27.1 MPH 43.6 KPH	33.9 MPH 54.6 KPH	40.7 MPH 65.5 KPH
2nd	25.2 MPH 40.6 KPH	37.7 MPH 60.7 KPH	50.3 MPH 81.0 KPH	62.9 MPH 101.2 KPH	75.5 MPH 121.5 KPH
3rd	36.1 MPH 58.1 KPH	54.2 MPH 87.2 KPH	72.3 MPH 116.4 KPH	90.4 MPH 145.5 KPH	108.4MPH 174.4 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
5th	55.2 MPH 88.8 KPH	82.8 MPH 133.2 KPH	110.4 MPH 177.7 KPH	138.0 MPH 222.0 KPH	165.6 MPH 266.5 KPH
Reverse	11.8 MPH 19.0 KPH	17.6 MPH 28.3 KPH	23.5 MPH 37.8 KPH	29.4 MPH 47.3 KPH	35.3 MPH 56.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Hi-Gear Engineering Ford Type 9 2.8 Closer Ratio Five-Speed Transmission, High 1st Gear Ratio:

1st	2.83 :1
2nd	1.81 : 1
3rd	1.26 : 1
4th	1.00 : 1
5th	0.87 : 1
5th	0.85 : 1
5th	0.825 : 1
Reverse	3.87 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.56	1,167 RPM	2,083 RPM	1,974 RPM	3,526 RPM
2nd-3rd	1.44	993 RPM	2,257 RPM	1,681 RPM	3,819 RPM
3rd-4th	1.26	671 RPM	2,579 RPM	1,135 RPM	4,365 RPM
4th-5th (.87)	1.15	424 RPM	2,826 RPM	718 RPM	4,782 RPM
4th-5th (.85)	1.18	496 RPM	2,754 RPM	839 RPM	4,661 RPM
4th-5th (.825)	1.21	564 RPM	2,686 RPM	955 RPM	4,545 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	6,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM
1st	9.6 MPH 15.5 KPH	14.5 MPH 23.3 KPH	19.2 MPH 30.9 KPH	24.1 MPH 38.8 KPH	28.9 MPH 46.5 KPH
2nd	13.6 MPH 21.9 KPH	20.3 MPH 32.7 KPH	27.1 MPH 43.6 KPH	33.9 MPH 54.6 KPH	40.7 MPH 65.5 KPH
3rd	21.7 MPH 34.9 KPH	32.5 MPH 52.3 KPH	43.3 MPH 70.2 KPH	54.1 MPH 87.1 KPH	65.0 MPH 104.6 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
5th (.87)	31.4 MPH 50.5 KPH	47.1 MPH 75.8 KPH	62.7 MPH 100.9 KPH	78.4 MPH 126.2 KPH	94.1 MPH 151.4 KPH
5th (.85)	32.1 MPH 51.7 KPH	48.2 MPH 77.6 KPH	64.2 MPH 103.3 KPH	80.3 MPH 129.2 KPH	96.3 MPH 155.0 KPH
5th (.825)	33.1 MPH 53.3 KPH	49.6 MPH 79.8 KPH	66.2 MPH 10.0 KPH	82.7 MPH 133.9 KPH	99.2 MPH 159.6 KPH
Reverse	7.0 MPH 11.3 KPH	10.6 MPH 17.1 KPH	14.1 MPH 22.7 KPH	17.6 MPH 28.3 KPH	21.2 MPH 34.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 crownwheel (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.1 MPH 16.3 KPH	15.2 MPH 24.5 KPH	20.3 MPH 32.7 KPH	25.3 MPH 40.7 KPH	30.4 MPH 48.9 KPH
2nd	15.8 MPH 25.4 KPH	23.8 MPH 38.3 KPH	31.7 MPH 51.0 KPH	39.6 MPH 63.7 KPH	47.5 MPH 76.4 KPH
3rd	22.8 MPH 36.7 KPH	34.2 MPH 55.0 KPH	45.5 MPH 73.2 KPH	56.9 MPH 91.6 KPH	68.3 MPH 109.9 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
5th (.87)	33.0 MPH 53.1 KPH	49.5 MPH 79.7 KPH	66.0 MPH 106.2 KPH	82.4 MPH 132.6 KPH	98.9MPH 159.2 KPH
5th (.85)	33.8 MPH 54.4 KPH	50.6 MPH 81.4 KPH	67.5 MPH 108.6 KPH	84.4 MPH 135.8 KPH	101.3 MPH 163.0 KPH
5th (.825)	7.8 MPH 12.5 KPH	11.7 MPH 18.8 KPH	15.7 MPH 25.3 KPH	19.6 MPH 31.5 KPH	23.5 MPH 37.8 KPH
Reverse	7.8 MPH 12.5 KPH	11.7 MPH 18.8 KPH	15.7 MPH 25.3 KPH	19.6 MPH 31.5 KPH	23.5 MPH 37.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining

required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.9 MPH 17.5 KPH	16.3 MPH 26.2 KPH	21.7 MPH 34.9 KPH	27.2 MPH 43.8 KPH	32.6 MPH 52.5 KPH
2nd	17.0 MPH 27.4 KPH	25.5 MPH 41.0 KPH	34.0 MPH 54.7 KPH	42.5 MPH 68.4 KPH	51.0 MPH 82.1 KPH
3rd	24.4 MPH 39.3 KPH	36.6 MPH 58.9 KPH	48.8 MPH 78.5 KPH	61.0 MPH 98.2 KPH	73.2 MPH 117.8 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
5th (.87)	35.3 MPH 56.8 KPH	53.0 MPH 85.3 KPH	70.7 MPH 113.8 KPH	88.3 MPH 142.1 KPH	106.0 MPH 170.6 KPH
5th (.85)	36.1 MPH 58.1 KPH	54.2 MPH 87.2 KPH	72.3 MPH 116.4 KPH	90.4 MPH 145.5 KPH	108.5 MPH 174.6 KPH
5th (.825)	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.5 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion rear axle gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.5 MPH 18.5 KPH	16.3 MPH 26.2 KPH	21.7 MPH 34.9 KPH	27.2 MPH 43.8 KPH	32.6 MPH 52.5 KPH
2nd	18.0 MPH 29.0 KPH	27.0 MPH 43.5 KPH	35.9 MPH 57.8 KPH	44.9 MPH 72.3 KPH	53.9 MPH 86.7 KPH
3rd	25.8 MPH 41.5 KPH	38.7 MPH 62.3 KPH	51.6 MPH 83.0 KPH	64.5 MPH 103.8 KPH	77.5 MPH 124.7 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
5th (.87)	37.4 MPH 60.2 KPH	56.1 MPH 90.3 KPH	74.8 MPH 120.4 KPH	93.5 MPH 150.5 KPH	112.2 MPH 180.6 KPH
5th (.85)	38.3 MPH 61.6 KPH	57.4 MPH 92.4 KPH	76.5 MPH 123.1 KPH	95.7 MPH 154.0 KPH	114.8 MPH 184.8 KPH
5th (.825)	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.6 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.7 MPH 18.8 KPH	17.6 MPH 28.3 KPH	23.4 MPH 37.7 KPH	29.3 MPH 47.1 KPH	35.1 MPH 56.5 KPH
2nd	18.3 MPH 29.4 KPH	27.5 MPH 44.3 KPH	36.6 MPH 58.9 KPH	45.8 MPH 73.7 KPH	54.9 MPH 88.3 KPH
3rd	26.3 MPH 42.3 KPH	39.5 MPH 63.6 KPH	52.6 MPH 84.6 KPH	65.8 MPH 105.9 KPH	78.9 MPH 127.0 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
5th (.87)	38.1 MPH 61.3 KPH	57.1 MPH 91.9 KPH	76.2 MPH 122.6 KPH	95.2 MPH 153.2 KPH	114.3 MPH 183.9 KPH
5th (.85)	39.0 MPH 62.8 KPH	58.5 MPH 94.1 KPH	78.0 MPH 125.5 KPH	97.5 MPH 156.9 KPH	117.0 MPH 188.3 KPH
5th (.825)	8.6 MPH 13.8 KPH	12.8 MPH 20.6 KPH	17.1 MPH 27.5 KPH	21.4 MPH 34.4 KPH	25.7 MPH 41.4 KPH
Reverse	8.6 MPH 13.8 KPH	12.8 MPH 20.6 KPH	17.1 MPH 27.5 KPH	21.4 MPH 34.4 KPH	25.7 MPH 41.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion rear axle gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.0 MPH 19.3 KPH	18.1 MPH 29.1 KPH	24.1 MPH 38.8 KPH	30.1 MPH 48.4 KPH	36.2 MPH 48.4 KPH
2nd	18.8 MPH 30.3 KPH	28.2 MPH 45.4 KPH	37.7 MPH 60.7 KPH	47.1 MPH 75.8 KPH	56.5 MPH 90.9 KPH
3rd	27.1 MPH 43.6 KPH	40.6 MPH 65.3 KPH	54.1 MPH 87.1 KPH	67.7 MPH 109.0 KPH	81.2 MPH 130.7 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
5th (.87)	39.2 MPH 63.1 KPH	58.8 MPH 94.6 KPH	78.4 MPH 126.2 KPH	98.0 MPH 157.7 KPH	117.6 MPH 189.2 KPH
5th (.85)	40.1 MPH 64.5 KPH	60.2 MPH 96.9 KPH	80.3 MPH 129.2 KPH	100.3 MPH 161.4 KPH	120.4 MPH 193.8 KPH
5th (.825)	41.3 MPH 66.5 KPH	62.0 MPH 99.8 KPH	82.7 MPH 133.1 KPH	103,4 MPH 166.4 KPH	124.0 MPH 199.6 KPH
Reverse	8.8 MPH 14.2 KPH	13.2 MPH 21.2 KPH	17.6 MPH 28.3 KPH	22.0 MPH 35.4 KPH	26.4 MPH 42.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.6 MPH 20.3 KPH	19.0 MPH 30.6 KPH	25.3 MPH 40.7 KPH	31.6 MPH 50.9 KPH	37.9 MPH 61.0 KPH
2nd	19.8 MPH 31.9 KPH	29.7 MPH 47.8 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	59.3 MPH 95.4 KPH
3rd	28.4 MPH 45.7 KPH	42.6 MPH 68.6 KPH	56.8 MPH 91.4 KPH	71.0 MPH 114.3 KPH	85.2 MPH 137.1 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
5th (.87)	41.1 MPH 66.1 KPH	61.7 MPH 99.3 KPH	82.3 MPH 132.4 KPH	102.8 MPH 165.4 KPH	123.4 MPH 198.6 KPH
5th (.85)	42.1 MPH 67.7 KPH	63.1 MPH 101.5 KPH	84.2 MPH 135.5 KPH	105.2 MPH 169.4 KPH	126.3 MPH 203.3 KPH
5th (.825)	43.4 MPH 69.8 KPH	65.1 MPH 104.8 KPH	86.7 MPH 139.5 KPH	108.4 MPH 174.4 KPH	130.1 MPH 209.4 KPH
Reverse	9.2 MPH 14.8 KPH	13.9 MPH 22.4 KPH	18.5 MPH 29.8 KPH	23.1 MPH 37.2 KPH	27.7 MPH 44.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset, (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.4 MPH 21.6 KPH	20.0 MPH 32.2 KPH	26.7 MPH 43.0 KPH	33.4 MPH 53.7 KPH	40.1 MPH 64.5 KPH
2nd	19.8 MPH 31.9 KPH	29.7 MPH 47.8 KPH	39.5 MPH 63.6 KPH	49.4 MPH 79.5 KPH	59.3 MPH 95.4 KPH
3rd	30.0 MPH 48.3 KPH	45.0 MPH 72.4 KPH	60.0 MPH 96.6 KPH	75.0 MPH 120.7 KPH	90.0 MPH 144.8 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
5th (.87)	43.5 MPH 70.0 KPH	65.2 MPH 104.9 KPH	86.9 MPH 139.9 KPH	108.6 MPH 174.7 KPH	130.4 MPH 209.9 KPH
5th (.85)	44.5 MPH 71.6 KPH	66.7 MPH 107.3 KPH	89.0 MPH 143.2 KPH	111.2 MPH 179.0 KPH	133.4 MPH 214.7 KPH
5th (.825)	45.8 MPH 73.7 KPH	68.7 MPH 110.6 KPH	91.6 MPH 147.4 KPH	114.6 MPH 184.4 KPH	137.5 MPH 221.3 KPH
Reverse	9.8 MPH 15.8 KPH	14.6 MPH 23.5 KPH	19.5 MPH 31.4 KPH	24.4 MPH 34.4 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	14.2 MPH 22.8 KPH	20.7 MPH 33.3 KPH	27.6 MPH 44.4 KPH	34.5 MPH 55.5 KPH	41.4 MPH 66.6 KPH
2nd	21.6 MPH 34.8 KPH	32.3 MPH 52.0 KPH	43.1MPH 69.4 KPH	53.9 MPH 86.7 KPH	64.7 MPH 104.1 KPH
3rd	31.0 MPH 49.9 KPH	46.5 MPH 74.8 KPH	62.0 MPH 99.8 KPH	77.5 MPH 124.7 KPH	93.0 MPH 149.7 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
5th (.87)	44.8 MPH 72.1 KPH	67.3 MPH 108.3 KPH	89.7 MPH 144.3 KPH	112.2 MPH 181.4 KPH	134.6 MPH 216.6 KPH
5th (.85)	45.9 MPH 73.8 KPH	68.9 MPH 110.8 KPH	91.9 MPH 147.9 KPH	114.8 MPH 184.7 KPH	137.8 MPH 221.8 KPH
5th (.825)	47.3 MPH 76.1 KPH	71.0 MPH 114.3 KPH	94.6 MPH 152.2 KPH	118.3 MPH 190.4 KPH	142.0 MPH 228.5 KPH
Reverse	9.8 MPH 15.8 KPH	14.6 MPH 23.5 KPH	19.5 MPH 31.4 KPH	24.4 MPH 34.4 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.9 MPH 22.4 KPH	20.9 MPH 33.6 KPH	27.9 MPH 44.9 KPH	34.9 MPH 56.2 KPH	41.8 MPH 67.3 KPH
2nd	21.8 MPH 35.1 KPH	32.7 MPH 52.6 KPH	43.6 MPH 70.2 KPH	54.5 MPH 87.7 KPH	65.4 MPH 105.3 KPH
3rd	31.3 MPH 50.4 KPH	47.0 MPH 75.6 KPH	62.6 MPH 100.7 KPH	78.3 MPH 126.0 KPH	93.9 MPH 151.1 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
5th (.87)	45.4 MPH 73.1 KPH	68.0 MPH 109.4 KPH	90.7 MPH 146.0 KPH	113.4 MPH 182.5 KPH	136.1 MPH 219.0 KPH
5th (.85)	46.4 MPH 74.7 KPH	69.6 MPH 112.0 KPH	92.8 MPH 149.3 KPH	116.1 MPH 186.8 KPH	139.3 MPH 224.2 KPH
5th (.825)	47.8 MPH 76.9 KPH	71.7 MPH 115.4 KPH	95.7 MPH 154.0 KPH	119.6 MPH 192.5 KPH	143.5 MPH 231.0 KPH
Reverse	10.2 MPH 16.4 KPH	15.3 MPH 24.6 KPH	20.4 MPH 32.8 KPH	25.5 MPH 41.0 KPH	30.6 MPH 49.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production 1962-1968 MGB roadsters, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as

Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	14.9 MPH 24.0 KPH	22.4 MPH 36.0 KPH	29.9 MPH 48.1 KPH	37.4 MPH 60.2 KPH	44.8 MPH 72.1 KPH
2nd	23.4 MPH 37.7 KPH	35.0 MPH 56.3 KPH	46.7 MPH 75.2 KPH	58.4 MPH 94.0 KPH	70.1 MPH 112.8 KPH
3rd	33.6 MPH 54.1 KPH	50.4 MPH 81.1 KPH	67.1 MPH 108.0 KPH	83.9 MPH 135.0 KPH	100.7 MPH 162.0 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
5th (.87)	48.6 MPH 78.2 KPH	72.9 MPH 117.3 KPH	97.2 MPH 156.4 KPH	121.5 MPH 195.5 KPH	145.9 MPH 234.8 KPH
5th (.85)	49.8 MPH 80.1 KPH	74.6 MPH 120.1 KPH	99.5 MPH 160.1 KPH	124.4 MPH 200.2 KPH	149.3 MPH 240.3 KPH
5th (.825)	51.3 MPH 82.6 KPH	76.9 MPH 123.8 KPH	102.5 MPH 165.0 KPH	128.2 MPH 206.3 KPH	153.8 MPH 247.5 KPH
Reverse	10.9 MPH 17.5 KPH	16.4 MPH 26.4 KPH	21.9 MPH 35.2 KPH	27.3 MPH 43.9 KPH	32.8 MPH 51.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	16.1 MPH 25.9 KPH	24.1 MPH 38.8 KPH	32.2 MPH 51.8 KPH	40.2 MPH 64.7 KPH	48.3 MPH 77.7 KPH
2nd	25.2 MPH 40.6 KPH	37.7 MPH 60.7 KPH	50.3 MPH 81.0 KPH	62.9 MPH 101.2 KPH	75.5 MPH 121.5 KPH
3rd	36.1 MPH 58.1 KPH	52.2 MPH 84.0 KPH	72.3 MPH 116.4 KPH	90.4 MPH 145.5 KPH	108.4 MPH 174.4 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
5th (.87)	52.4 MPH 84.3 KPH	78.5 MPH 126.3 KPH	104.7 MPH 168.5 KPH	130.9 MPH 210.7 KPH	157.1 MPH 252.8 KPH
5th (.85)	53.6 MPH 86.2 KPH	80.4 MPH 129.4 KPH	107.2 MPH 172.5 KPH	134.0 MPH 215.6 KPH	160.8 MPH 258.8 KPH
5th (.825)	55.2 MPH 88.8 KPH	82.8 MPH 133.2 KPH	110.4 MPH 177.7 KPH	138.0 MPH 222.0 KPH	165.6 MPH 266.5 KPH
Reverse	11.8 MPH 19.0 KPH	17.6 MPH 28.3 KPH	23.5 MPH 37.8 KPH	29.4 MPH 47.3 KPH	35.3 MPH 56.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Hi-Gear Engineering Ford Type 9 2.8 Sporting Close Ratio Heavy Duty Only Five-Speed Transmission, Low 1st Gear Ratio:

1st	2.75: 1
2nd	1.75: 1
3rd	1.26: 1
4th	1.00 : 1
5th	0.89 : 1
5th	0.86 : 1
5th	0.84 : 1
5th	0.82: 1
Reverse	3.87 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.57	1,180 RPM	2,070 RPM	1,997 RPM	3,503 RPM
2nd-3rd	1.39	912 RPM	2,338 RPM	1,543 RPM	3,957 RPM
3rd-4th	1.26	671 RPM	2,579 RPM	1,135 RPM	4,365 RPM
4th-5th (.89:1)	1.12	348 RPM	2,902 RPM	589 RPM	4,911 RPM
4th-5th (.86:1)	1.16	448 RPM	2,802 RPM	758 RPM	4,741 RPM
4th-5th (.84:1)	1.19	519 RPM	2,731 RPM	878 RPM	4,622 RPM
4th-5th (.82:1)	1.22	586 RPM	2,664 RPM	992 RPM	4,508 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	9.9 MPH 15.9 KPH	14.9 MPH 24.0 KPH	19.8 MPH 31.9 KPH	24.8 MPH 39.9 KPH	29.8 MPH 48.0 KPH
2nd	15.6 MPH 25.1 KPH	23.4 MPH 37.7 KPH	31.2 MPH 50.2 KPH	39.0 MPH 62.8 KPH	46.8 MPH 75.3 KPH
3rd	21.7 MPH 34.9 KPH	32.5 MPH 52.3 KPH	43.3 MPH 70.2 KPH	54.1 MPH 87.1 KPH	65.0 MPH 104.6 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
5th (.89)	30.7 MPH 49.4 KPH	46.0 MPH 74.0 KPH	61.3 MPH 98.6 KPH	76.7 MPH 123.4 KPH	92.0 MPH 148.1 KPH
5th (.86)	31.7 MPH 51.0 KPH	47.6 MPH 76.6 KPH	63.5 MPH 102.2 KPH	79.3 MPH 127.6 KPH	95.2 MPH 153.2 KPH
5th (.84)	32.5 MPH 52.3 KPH	48.7 MPH 78.4 KPH	65.0 MPH 104.6 KPH	81.2 MPH 130.7 KPH	97.5 MPH 156.9 KPH
5th (.82)	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	7.0 MPH 11.3 KPH	10.6 MPH 17.1 KPH	14.1 MPH 22.7 KPH	17.6 MPH 28.3 KPH	21.2 MPH 34.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 crownwheel (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.4 MPH 16.7 KPH	15.6 MPH 25.1 KPH	20.9 MPH 33.6 KPH	26.1 MPH 42.0 KPH	31.3 MPH 50.4 KPH
2nd	16.4 MPH 26.4 KPH	24.6 MPH 39.6 KPH	32.8 MPH 52.8 KPH	41.0 MPH 70.0 KPH	49.2 MPH 79.2 KPH
3rd	22.8 MPH 36.7 KPH	31.2 MPH 50.2 KPH	45.5 MPH 73.2 KPH	57.0 MPH 91.7 KPH	68.3 MPH 109.9 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
5th (.89)	32.2 MPH 51.8 KPH	48.3 MPH 77.7 KPH	64.5 MPH 103.8 KPH	80.6 MPH 129.7 KPH	96.7 MPH 155.6 KPH
5th (.86)	33.4 MPH 53.7 KPH	50.0 MPH 80.5 KPH	66.7 MPH 107.3 KPH	83.4 MPH 134.2 KPH	100.1 MPH 161.1 KPH
5th (.84)	34.2 MPH 55.0 KPH	51.2 MPH 82.4 KPH	68.3 MPH 109.9 KPH	85.4 MPH 137.4 KPH	102.5 MPH 165.0 KPH
5th (.82)	32.1 MPH 51.7 KPH	48.2 MPH 77.6 KPH	70.0 MPH 112.6 KPH	80.3 MPH 129.2 KPH	96.4 MPH 155.1 KPH
Reverse	7.8 MPH 12.5 KPH	11.7 MPH 18.8 KPH	15.7 MPH 25.3 KPH	19.6 MPH 31.5 KPH	23.5 MPH 37.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.2 MPH 18.0 KPH	16.8 MPH 27.0 KPH	22.4 MPH 36.0 KPH	27.9 MPH 44.9 KPH	33.5 MPH 53.9 KPH
2nd	17.6 MPH 28.3 KPH	26.3 MPH 42.3 KPH	35.1 MPH 56.5 KPH	43.9 MPH 70.6 KPH	52.7 MPH 84.8 KPH
3rd	24.4 MPH 39.3 KPH	36.6 MPH 58.9 KPH	48.8 MPH 78.5 KPH	61.0 MPH 98.2 KPH	73.2 MPH 117.8 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
5th (.89)	34.5 MPH 55.5 KPH	51.8 MPH 83.4 KPH	69.1 MPH 111.2 KPH	86.4 MPH 139.0 KPH	103.6 MPH 166.7 KPH
5th (.86)	35.7 MPH 57.4 KPH	53.6 MPH 86.3 KPH	71.5 MPH 115.1 KPH	89.4 MPH 143.9 KPH	107.2 MPH 172.5 KPH
5th (.84)	36.6 MPH 58.9 KPH	54.9 MPH 88.3 KPH	73.2 MPH 117.8 KPH	91.5 MPH 147.2 KPH	109.8 MPH 176.7 KPH
5th (.82)	37.5 MPH	56.2 MPH	75.0 MPH	93.7 MPH	112.5 MPH

	60.3 KPH	90.4 KPH	120.7 KPH	150.8 KPH	181.0 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion rear axle gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.8 MPH 19.0 KPH	17.7 MPH 28.5 KPH	23.7 MPH 38.1 KPH	29.6 MPH 47.8 KPH	35.5 MPH 57.1 KPH
2nd	18.6 MPH 29.9 KPH	27.9 MPH 44.9 KPH	37.2 MPH 59.9 KPH	46.5 MPH 74.8 KPH	55.8 MPH 89.8 KPH
3rd	25.8 MPH 41.5 KPH	38.7 MPH 62.3 KPH	51.6 MPH 83.0 KPH	64.5 MPH 103.8 KPH	77.4 MPH 124.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
5th (.89)	36.6 MPH 58.9 KPH	54.8 MPH 88.2 KPH	73.1 MPH 117.6 KPH	91.4 MPH 147.1 KPH	109.6 MPH 176.4 KPH
5th (.86)	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.7 KPH	94.6 MPH 152.2 KPH	113.5 MPH 182.7 KPH
5th (.84)	38.7 MPH 62.3 KPH	58.1 MPH 93.5 KPH	77.4 MPH 124.6 KPH	96.8 MPH 155.8 KPH	116.2 MPH 187.0 KPH
5th (.82)	39.7 MPH 63.8 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.1 MPH 19.5 KPH	18.1 MPH 29.1 KPH	24.1 MPH 38.8 KPH	30.1 MPH 48.4 KPH	36.1 MPH 58.1 KPH
2nd	18.9 MPH 30.4 KPH	29.4 MPH 47.3 KPH	37.9 MPH 61.0 KPH	47.3 MPH 76.1 KPH	56.8 MPH 91.4 KPH
3rd	26.3 MPH 42.3 KPH	39.5 MPH 63.6 KPH	52.6 MPH 84.6 KPH	65.8 MPH 105.9 KPH	78.9 MPH 127.0 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
5th (.89)	37.2 MPH 59.9 KPH	55.9 MPH 90.0 KPH	74.5 MPH 119.9 KPH	93.1 MPH 149.8 KPH	111.7 MPH 179.8 KPH
5th (.86)	38.5 MPH 62.0 KPH	57.8 MPH 93.0 KPH	77.1 MPH 124.1 KPH	96.4 MPH 155.1 KPH	115.6 MPH 186.0 KPH
5th (.84)	39.4 MPH 63.4 KPH	59.2 MPH 95.3 KPH	78.9 MPH 127.0 KPH	98.7 MPH 158.8 KPH	118.4 MPH 190.5 KPH
5th (.82)	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	8.6 MPH 13.8 KPH	12.8 MPH 20.6 KPH	17.1 MPH 27.5 KPH	21.4 MPH 34.4 KPH	25.7 MPH 41.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion rear axle gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.4 MPH 20.0 KPH	18.6 MPH 29.9 KPH	24.8 MPH 39.9 KPH	31.0 MPH 49.9 KPH	37.2 MPH 59.9 KPH
2nd	19.5 MPH 31.4 KPH	29.2 MPH 47.0 KPH	39.0 MPH 62.8 KPH	48.7 MPH 78.4 KPH	58.5 MPH 94.1 KPH
3rd	27.0 MPH 43.4 KPH	40.6 MPH 65.3 KPH	54.1 MPH 87.1 KPH	67.7 MPH 109.0 KPH	81.2 MPH 130.7 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
5th (.89)	38.3 MPH 61.6 KPH	57.5 MPH 92.5 KPH	76.7 MPH 123.4 KPH	95.8 MPH 154.2 KPH	115.0 MPH 185.1 KPH
5th (.86)	39.7 MPH 63.9 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
5th (.84)	40.6 MPH 65.3 KPH	60.9 MPH 98.0 KPH	81.3 MPH 130.8 KPH	101.5 MPH 163.3 KPH	121.8 MPH 196.0 KPH
5th (.82)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	8.8 MPH 14.2 KPH	13.2 MPH 21.2 KPH	17.6 MPH 28.3 KPH	22.0 MPH 35.4 KPH	26.4 MPH 42.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**6,000 RPM**	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**
1st	13.0 MPH 20.9 KPH	19.5 MPH 31.4 KPH	26.0 MPH 41.8 KPH	32.5 MPH 52.3 KPH	39.0 MPH 62.8 KPH
2nd	20.4 MPH 32.8 KPH	30.7 MPH 49.4 KPH	40.9 MPH 65.8 KPH	51.1 MPH 82.2 KPH	61.3 MPH 98.6 KPH
3rd	28.4 MPH 45.7 KPH	42.6 MPH 68.6 KPH	56.8 MPH 91.4 KPH	71.0 MPH 114.3 KPH	85.2 MPH 137.1 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
5th (.89)	40.2 MPH 64.7 KPH	60.3 MPH 97.0 KPH	80.4 MPH 129.4 KPH	100.5 MPH 161.7 KPH	120.6 MPH 194.1 KPH
5th (.86)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
5th (.84)	42.6 MPH 68.6 KPH	63.9 MPH 102.8 KPH	85.2 MPH 137.1 KPH	106.5 MPH 171.4 KPH	127.8 MPH 205.7 KPH
5th (.82)	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	9.2 MPH	13.9 MPH	18.5 MPH	23.1 MPH	27.7 MPH

	14.8 KPH	22.4 KPH	29.8 KPH	37.2 KPH	44.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset, (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	13.7 MPH 22.0 KPH	20.6 MPH 33.1 KPH	27.5 MPH 44.3 KPH	34.4 MPH 55.4 KPH	41.2 MPH 66.3 KPH
2nd	21.6 MPH 34.8 KPH	32.4 MPH 52.1 KPH	43.2 MPH 69.5 KPH	54.0 MPH 86.9 KPH	64.8 MPH 104.3 KPH
3rd	30.0 MPH 48.3 KPH	45.0 MPH 72.4 KPH	60.0 MPH 96.6 KPH	75.0 MPH 120.7 KPH	90.0 MPH 144.8 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
5th (.89)	42.5 MPH 68.4 KPH	63.7 MPH 102.5 KPH	85.0 MPH 136.8 KPH	106.2 MPH 170.9 KPH	127.4 MPH 205.0 KPH
5th (.86)	44.0 MPH 70.8 KPH	65.9 MPH 106.1 KPH	87.9 MPH 141.5 KPH	109.9 MPH 176.9 KPH	131.9 MPH 212.3 KPH
5th (.84)	45.0 MPH 72.4 KPH	67.5 MPH 108.6 KPH	90.0 MPH 144.8 KPH	112.5 MPH 181.1 KPH	135.0 MPH 217.3 KPH
5th (.82)	46.1 MPH 74.2 KPH	69.2 MPH 11.42 KPH	92.2 MPH 148.42 KPH	115.2 MPH 185.42 KPH	138.3 MPH 222.62 KPH
Reverse	9.8 MPH 15.8 KPH	14.6 MPH 23.5 KPH	19.5 MPH 31.4 KPH	24.4 MPH 34.4 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	14.2 MPH 22.9 KPH	21.3 MPH 34.3 KPH	28.4 MPH 45.7 KPH	35.5 MPH 57.1 KPH	42.6 MPH 68.6 KPH
2nd	22.3 MPH 35.9 KPH	33.5 MPH 53.9 KPH	44.6 MPH 71.8 KPH	55.8 MPH 89.8 KPH	66.9 MPH 107.7 KPH
3rd	31.0 MPH 49.9 KPH	46.5 MPH 74.8 KPH	62.0 MPH 99.8 KPH	77.5 MPH 124.7 KPH	93.0 MPH 149.7 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
5th (.89)	43.9 MPH 70.6 KPH	65.8 MPH 105.9 KPH	87.7 MPH 141.1 KPH	109.7 MPH 176.5 KPH	131.6 MPH 211.8 KPH
5th (.86)	45.4 MPH 73.1 KPH	68.1 MPH 109.6 KPH	90.8 MPH 146.2 KPH	113.5 MPH 182.7 KPH	136.1 MPH 217.4 KPH
5th (.84)	46.5 MPH 73.1 KPH	69.7 MPH 112.2 KPH	93.0 MPH 149.7 KPH	116.2 MPH 187.0 KPH	139.4 MPH 224.3 KPH
5th (.82)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	9.8 MPH 15.8 KPH	14.6 MPH 23.5 KPH	19.5 MPH 31.4 KPH	24.4 MPH 34.4 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	14.3 MPH 23.0 KPH	21.5 MPH 34.6 KPH	28.7 MPH 46.2 KPH	35.9 MPH 57.8 KPH	43.0 MPH 69.2 KPH
2nd	22.5 MPH 36.2 KPH	33.8 MPH 54.4 KPH	45.1 MPH 72.3 KPH	56.4 MPH 90.8 KPH	67.6 MPH 108.8 KPH
3rd	31.3 MPH 50.4 KPH	47.0 MPH 75.6 KPH	62.6 MPH 100.7 KPH	78.3 MPH 126.0 KPH	93.9 MPH 151.1 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
5th (.89)	44.3 MPH 71.3 KPH	66.5 MPH 107.0 KPH	88.7 MPH 142.7 KPH	110.8 MPH 178.3 KPH	133.0 MPH 214.0 KPH
5th (.86)	45.9 MPH 73.9 KPH	68.8 MPH 110.7 KPH	91.8 MPH 147.7 KPH	114.7 MPH 184.6 KPH	137.6 MPH 221.4 KPH
5th (.84)	47.0 MPH 75.6 KPH	70.5 MPH 113.5 KPH	93.9 MPH 151.1 KPH	117.4 MPH 188.9 KPH	140.9 MPH 226.8 KPH
5th (.82)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	10.2 MPH 16.4 KPH	15.3 MPH 24.6 KPH	20.4 MPH 32.8 KPH	25.5 MPH 41.0 KPH	30.6 MPH 49.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production 1962-1968 MGB roadsters, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	15.4 MPH 24.8 KPH	23.1 MPH 37.2 KPH	30.8 MPH 49.6 KPH	38.4 MPH 61.8 KPH	46.1 MPH 74.2 KPH
2nd	24.2 MPH 43.9 KPH	36.2 MPH 65.8 KPH	48.3 MPH 87.9 KPH	60.4 MPH 109.8 KPH	72.5 MPH 131.8 KPH
3rd	33.6 MPH 54.1 KPH	50.4 MPH 81.1 KPH	67.1 MPH 108.0 KPH	83.9 MPH 135.0 KPH	100.7 MPH 162.0 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
5th (.89)	47.5 MPH 76.4 KPH	71.3 MPH 114.7 KPH	95.0 MPH 152.9 KPH	118.8 MPH 191.2 KPH	142.6 MPH 229.5 KPH
5th (.86)	49.2 MPH 79.2 KPH	73.8 MPH 118.8 KPH	98.4 MPH 158.4 KPH	123.0 MPH 197.9 KPH	147.5 MPH 237.4 KPH
5th (.84)	50.3 MPH 81.0 KPH	75.5 MPH 121.5 KPH	100.7 MPH 162.1 KPH	125.9 MPH 202.6 KPH	151.1 MPH 243.2 KPH
5th (.82)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	10.9 MPH 17.5 KPH	16.4 MPH 26.4 KPH	21.9 MPH 35.2 KPH	27.3 MPH 43.9 KPH	32.8 MPH 51.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	16.6 MPH 26.7 KPH	24.8 MPH 39.9 KPH	33.1 MPH 53.3 KPH	41.4 MPH 66.6 KPH	49.7 MPH 80.0 KPH
2nd	26.0 MPH 41.8 KPH	39.0 MPH 62.8 KPH	52.1 MPH 83.8 KPH	65.1 MPH 104.8 KPH	78.1 MPH 125.7 KPH
3rd	36.1 MPH 58.1 KPH	52.2 MPH 84.0 KPH	72.3 MPH 116.4 KPH	90.4 MPH 145.5 KPH	108.4MPH 174.4 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
5th (.89)	51.2 MPH 82.4 KPH	76.8 MPH 123.6 KPH	102.4 MPH 164.8 KPH	127.9 MPH 205.8 KPH	153.5 MPH 247.0 KPH
5th (.86)	53.0 MPH 85.3 KPH	79.4 MPH 127.8 KPH	105.9 MPH 170.4 KPH	132.4 MPH 213.1 KPH	158.9 MPH 255.7 KPH
5th (.84)	54.2 MPH 87.2 KPH	81.3 MPH 130.8 KPH	108.4 MPH 174.5 KPH	135.6 MPH 218.2 KPH	162.7 MPH 261.8 KPH
5th (.82)	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	11.8 MPH 19.0 KPH	17.6 MPH 28.3 KPH	23.5 MPH 37.8 KPH	29.4 MPH 47.3 KPH	35.3 MPH 56.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Hi-Gear Engineering Ford Type 9 2.8 Sporting Close Ratio Heavy Duty Only Five-Speed Transmission, High 1st Gear Ratio:

1st	2.66 : 1
2nd	1.75 : 1
3rd	1.26 : 1
4th	1.00 : 1
5th	0.89 : 1
5th	0.86 : 1
5th	0.84 : 1
5th	0.82 : 1
Reverse	3.87 : 1

These gear ratios make for the following ratio and engine speed changes when up-shifting:

		@ 3,250 RPM		**@ 5,500 RPM**	
	Ratio Change	**Drops:**	**To:**	**Drops:**	**To:**
1st-2nd	1.52	1,112 RPM	2,138RPM	1,882 RPM	3,618 RPM
2nd-3rd	1.39	912 RPM	2,338 RPM	1,543 RPM	3,957 RPM
3rd-4th	1.26	671 RPM	2,579 RPM	1,135 RPM	4,365 RPM
4th-5th (.89)	1.12	348 RPM	2,902 RPM	589 RPM	4,911 RPM
4th-5th (.86)	1.16	448 RPM	2,802 RPM	758 RPM	4,741 RPM
4th-5th (.84)	1.19	519 RPM	2,731 RPM	878 RPM	4,622 RPM
4th-5th (.82)	1.22	586 RPM	2,664 RPM	992 RPM	4,508 RPM

The available rear axle crownwheel and pinion gearsets produce the following results in terms of Engine Speed vs. Road Speed:

Road Speed* in MPH / KPH w/ 5.125:1 (8/41) crownwheel and pinion gearset (BMC Part # 102258)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	10.3 MPH 16.6 KPH	15.4 MPH 24.8 KPH	20.5 MPH 33.0 KPH	25.7 MPH 41.4 KPH	30.8 MPH 49.6 KPH
2nd	15.6 MPH 25.1 KPH	23.4 MPH 37.7 KPH	31.2 MPH 50.2 KPH	39.0 MPH 62.8 KPH	46.8 MPH 75.3 KPH
3rd	21.7 MPH 34.9 KPH	32.5 MPH 52.3 KPH	43.3 MPH 70.2 KPH	54.1 MPH 87.1 KPH	65.0 MPH 104.6 KPH
4th	27.3 MPH 43.8 KPH	40.9 MPH 65.8 KPH	54.6 MPH 87.9 KPH	68.2 MPH 109.8 KPH	81.9 MPH 131.8 KPH
5th (.89)	30.7 MPH 49.4 KPH	46.0 MPH 74.0 KPH	61.3 MPH 98.6 KPH	76.7 MPH 123.4 KPH	92.0 MPH 148.1 KPH
5th (.86)	31.7 MPH 51.0 KPH	47.6 MPH 76.6 KPH	63.5 MPH 102.2 KPH	79.3 MPH 127.6 KPH	95.2 MPH 153.2 KPH
5th (.84)	32.5 MPH 52.3 KPH	48.7 MPH 78.4 KPH	65.0 MPH 104.6 KPH	81.2 MPH 130.7 KPH	97.5 MPH 156.9 KPH
5th (.82)	33.3 MPH 53.6 KPH	49.9 MPH 80.3 KPH	66.6 MPH 107.2 KPH	83.2 MPH 133.9 KPH	99.9 MPH 160.8 KPH
Reverse	7.0 MPH 11.3 KPH	10.6 MPH 17.1 KPH	14.1 MPH 22.7 KPH	17.6 MPH 28.3 KPH	21.2 MPH 34.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.875:1 crownwheel (8/39) crownwheel and pinion gearset (BMC Part # ATB 7056L)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	10.8 MPH 17.4 KPH	16.2 MPH 26.1 KPH	21.6 MPH 34.8 KPH	27.0 MPH 43.5 KPH	32.4 MPH 52.1 KPH
2nd	16.4 MPH 26.4 KPH	24.6 MPH 39.6 KPH	32.8 MPH 52.8 KPH	41.0 MPH 70.0 KPH	49.2 MPH 79.2 KPH
3rd	22.8 MPH 36.7 KPH	31.2 MPH 50.2 KPH	45.5 MPH 73.2 KPH	57.0 MPH 91.7 KPH	68.3 MPH 109.9 KPH
4th	28.7 MPH 46.2 KPH	43.0 MPH 69.2 KPH	57.4 MPH 92.4 KPH	71.8 MPH 115.5 KPH	86.1 MPH 138.6KPH
5th (.89)	32.2 MPH 51.8 KPH	48.3 MPH 77.7 KPH	64.5 MPH 103.8 KPH	80.6 MPH 129.7 KPH	96.7 MPH 155.6 KPH
5th (.86)	33.4 MPH 53.7 KPH	50.0 MPH 80.5 KPH	66.7 MPH 107.3 KPH	83.4 MPH 134.2 KPH	100.1 MPH 161.1 KPH
5th (.84)	34.2 MPH 55.0 KPH	51.2 MPH 82.4 KPH	68.3 MPH 109.9 KPH	85.4 MPH 137.4 KPH	102.5 MPH 165.0 KPH
5th (.82)	35.0 MPH 56.3 KPH	52.5 MPH 84.5 KPH	70.0 MPH 112.6 KPH	87.5 MPH 140.8 KPH	105.0 MPH 169.0 KPH
Reverse	7.8 MPH 12.5 KPH	11.7 MPH 18.8 KPH	15.7 MPH 25.3 KPH	19.6 MPH 31.5 KPH	23.5 MPH 37.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.55:1 (9/41) crownwheel and pinion gearset (BMC Part # 88G 284)[1] for Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # C-BTB 966)[2] for Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	11.6 MPH 18.7 KPH	17.3 MPH 27.8 KPH	23.1 MPH 37.2 KPH	28.9 MPH 46.5 KPH	34.7 MPH 55.8 KPH
2nd	17.6 MPH 28.3 KPH	26.3 MPH 42.3 KPH	35.1 MPH 56.5 KPH	43.9 MPH 70.6 KPH	52.7 MPH 84.8 KPH
3rd	24.4 MPH 39.3 KPH	36.6 MPH 58.9 KPH	48.8 MPH 78.5 KPH	61.0 MPH 98.2 KPH	73.2 MPH 117.8 KPH
4th	30.7 MPH 49.4 KPH	46.1 MPH 74.2 KPH	61.5 MPH 99.0 KPH	76.9 MPH 123.8 KPH	92.2 MPH 148.4 KPH
5th (.89)	34.5 MPH 55.5 KPH	51.8 MPH 83.4 KPH	69.1 MPH 111.2 KPH	86.4 MPH 139.0 KPH	103.6 MPH 166.7 KPH
5th (.86)	35.7 MPH 57.4 KPH	53.6 MPH 86.3 KPH	71.5 MPH 115.1 KPH	89.4 MPH 143.9 KPH	107.2 MPH 172.5 KPH
5th (.84)	36.6 MPH 58.9 KPH	54.9 MPH 88.3 KPH	73.2 MPH 117.8 KPH	91.5 MPH 147.2 KPH	109.8 MPH 176.7 KPH
5th (.82)	37.5 MPH	56.2 MPH	74.9 MPH	93.7 MPH	112.5 MPH

	60.3 KPH	90.4 KPH	120.5 KPH	150.8 KPH	181.0 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.3:1 (10/43) crownwheel and pinion rear axle gearset (BMC Part # 88G 283)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.2 MPH 19.6 KPH	18.3 MPH 29.4 KPH	24.5 MPH 39.4 KPH	30.6 MPH 49.2 KPH	36.7 MPH 59.1 KPH
2nd	18.6 MPH 29.9 KPH	27.9 MPH 44.9 KPH	37.2 MPH 59.9 KPH	46.5 MPH 74.8 KPH	55.8 MPH 89.8 KPH
3rd	25.8 MPH 41.5 KPH	38.7 MPH 62.3 KPH	51.6 MPH 83.0 KPH	64.5 MPH 103.8 KPH	77.4 MPH 124.6 KPH
4th	32.5 MPH 52.3 KPH	48.8 MPH 78.5 KPH	65.1 MPH 104.8 KPH	81.3 MPH 130.8 KPH	97.6 MPH 157.0 KPH
5th (.89)	36.6 MPH 58.9 KPH	54.8 MPH 88.2 KPH	73.1 MPH 117.6 KPH	91.4 MPH 147.1 KPH	109.6 MPH 176.4 KPH
5th (.86)	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.7 KPH	94.6 MPH 152.2 KPH	113.5 MPH 182.7 KPH
5th (.84)	38.7 MPH 62.3 KPH	58.1 MPH 93.5 KPH	77.4 MPH 124.6 KPH	96.8 MPH 155.8 KPH	116.2 MPH 187.0 KPH
5th (.82)	39.7 MPH 63.9 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
Reverse	8.4 MPH 13.5 KPH	12.6 MPH 20.3 KPH	16.8 MPH 27.0 KPH	21.0 MPH 33.8 KPH	25.2 MPH 40.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 4.22:1 (9/38) crownwheel and pinion rear axle gearset (BMC Part # C-BTB 975)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	12.5 MPH 20.1 KPH	18.7 MPH 30.1 KPH	24.9 MPH 40.1 KPH	31.1 MPH 50.1 KPH	37.4 MPH 60.2 KPH
2nd	18.9 MPH 30.4 KPH	29.4 MPH 47.3 KPH	37.9 MPH 61.0 KPH	47.3 MPH 76.1 KPH	56.8 MPH 91.4 KPH
3rd	26.3 MPH 42.3 KPH	39.5 MPH 63.6 KPH	52.6 MPH 84.6 KPH	65.8 MPH 105.9 KPH	78.9 MPH 127.0 KPH
4th	33.1 MPH 53.3 KPH	49.7 MPH 80.0 KPH	66.3 MPH 106.7 KPH	82.9 MPH 133.4 KPH	99.4 MPH 160.0KPH
5th (.89)	37.2 MPH 59.9 KPH	55.9 MPH 90.0 KPH	74.5 MPH 119.9 KPH	93.1 MPH 149.8 KPH	111.7 MPH 179.8 KPH
5th (.86)	38.5 MPH 62.0 KPH	57.8 MPH 93.0 KPH	77.1 MPH 124.1 KPH	96.4 MPH 155.1 KPH	115.6 MPH 186.0 KPH
5th (.84)	39.4 MPH 63.4 KPH	59.2 MPH 95.3 KPH	78.9 MPH 127.0 KPH	98.7 MPH 158.8 KPH	118.4 MPH 190.5 KPH
5th (.82)	40.4 MPH 65.0 KPH	60.6 MPH 97.5 KPH	80.8 MPH 130.0 KPH	101.1 MPH 162.7 KPH	121.3 MPH 195.2 KPH
Reverse	8.6 MPH 13.8 KPH	12.8 MPH 20.6 KPH	17.1 MPH 27.5 KPH	21.4 MPH 34.4 KPH	25.7 MPH 41.4 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 4.1:1 (10/41) crownwheel and pinion rear axle gearset (Special Tuning Part # ATB 7240)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	12.8 MPH 20.6 KPH	19.2 MPH 30.9 KPH	25.6 MPH 41.2 KPH	32.1 MPH 51.7 KPH	38.5 MPH 62.0 KPH
2nd	19.5 MPH 31.4 KPH	29.2 MPH 47.0 KPH	39.0 MPH 62.8 KPH	48.7 MPH 78.4 KPH	58.5 MPH 94.1 KPH
3rd	27.0 MPH 43.4 KPH	40.6 MPH 65.3 KPH	54.1 MPH 87.1 KPH	67.7 MPH 109.0 KPH	81.2 MPH 130.7 KPH
4th	34.1 MPH 54.9 KPH	51.2 MPH 82.4 KPH	68.2 MPH 109.8 KPH	85.3 MPH 137.3 KPH	102.3 MPH 164.6 KPH
5th (.89)	38.3 MPH 61.6 KPH	57.5 MPH 92.5 KPH	76.7 MPH 123.4 KPH	95.8 MPH 154.2 KPH	115.0 MPH 185.1 KPH
5th (.86)	39.7 MPH 63.9 KPH	59.5 MPH 95.8 KPH	79.3 MPH 127.6 KPH	99.2 MPH 159.6 KPH	119.0 MPH 191.5 KPH
5th (.84)	40.6 MPH 65.3 KPH	60.9 MPH 98.0 KPH	81.3 MPH 130.8 KPH	101.5 MPH 163.3 KPH	121.8 MPH 196.0 KPH
5th (.82)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
Reverse	8.8 MPH 14.2 KPH	13.2 MPH 21.2 KPH	17.6 MPH 28.3 KPH	22.0 MPH 35.4 KPH	26.4 MPH 42.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.909:1 (11/43) crownwheel and pinion gearsets (BMC Part # BTB 586)[1] for the Hardy-Spicer B Series Banjo-type rear axle, and (BMC Part # BTB 653)[2] for the Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	13.4 MPH 21.6 KPH	20.2 MPH 32.5 KPH	26.9 MPH 43.3 KPH	33.6 MPH 54.1 KPH	40.4 MPH 65.0 KPH
2nd	20.4 MPH 32.8 KPH	30.7 MPH 49.4 KPH	40.9 MPH 65.8 KPH	51.1 MPH 82.2 KPH	61.3 MPH 98.6 KPH
3rd	28.4 MPH 45.7 KPH	42.6 MPH 68.6 KPH	56.8 MPH 91.4 KPH	71.0 MPH 114.3 KPH	85.2 MPH 137.1 KPH
4th	35.8 MPH 57.6 KPH	53.7 MPH 86.4 KPH	71.6 MPH 115.2 KPH	89.4 MPH 143.9 KPH	107.3 MPH 172.7 KPH
5th (.89)	40.2 MPH 64.7 KPH	60.3 MPH 97.0 KPH	80.4 MPH 129.4 KPH	100.5 MPH 161.7 KPH	120.6 MPH 194.1 KPH
5th (.86)	41.6 MPH 66.9 KPH	62.4 MPH 100.4 KPH	83.2 MPH 133.9 KPH	104.0 MPH 167.4 KPH	124.8 MPH 200.8 KPH
5th (.84)	42.6 MPH 68.6 KPH	63.9 MPH 102.8 KPH	85.2 MPH 137.1 KPH	106.5 MPH 171.4 KPH	127.8 MPH 205.7 KPH
5th (.82)	43.6 MPH 70.2 KPH	65.5 MPH 105.4 KPH	87.3 MPH 140.5 KPH	109.1 MPH 175.6 KPH	130.9 MPH 210.7 KPH
Reverse	9.2 MPH	13.9 MPH	18.5 MPH	23.1 MPH	27.7 MPH

	14.8 KPH	22.4 KPH	29.8 KPH	37.2 KPH	44.6 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5" (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production MGB roadsters 1962-1968, and all use the same carrier, bearings, and shims. There is no need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

[2] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.7:1 (10/37) crownwheel and pinion gearset, (BMC Part # BTB 1244)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	14.2 MPH 22.9 KPH	21.3 MPH 34.3 KPH	28.4 MPH 45.7 KPH	35.5 MPH 57.1 KPH	42.6 MPH 68.6 KPH
2nd	21.6 MPH 34.8 KPH	32.4 MPH 52.1 KPH	43.2 MPH 69.5 KPH	54.0 MPH 86.9 KPH	64.8 MPH 104.3 KPH
3rd	30.0 MPH 48.3 KPH	45.0 MPH 72.4 KPH	60.0 MPH 96.6 KPH	75.0 MPH 120.7 KPH	90.0 MPH 144.8 KPH
4th	37.8 MPH 60.8 KPH	56.7 MPH 91.2 KPH	75.6 MPH 121.6 KPH	94.5 MPH 152.1 KPH	113.4 MPH 182.5 KPH
5th (.89)	42.5 MPH 68.4 KPH	63.7 MPH 102.5 KPH	85.0 MPH 136.8 KPH	106.2 MPH 170.9 KPH	127.4 MPH 205.0 KPH
5th (.86)	44.0 MPH 70.8 KPH	65.9 MPH 106.1 KPH	87.9 MPH 141.5 KPH	109.9 MPH 176.9 KPH	131.9 MPH 212.3 KPH
5th (.84)	45.0 MPH 72.4 KPH	67.5 MPH 108.6 KPH	90.0 MPH 144.8 KPH	112.5 MPH 181.1 KPH	135.0 MPH 217.3 KPH
5th (.82)	46.1 MPH 74.2 KPH	69.2 MPH 11.4 KPH	92.2 MPH 148.4 KPH	115.2 MPH 185.4 KPH	138.3 MPH 222.6 KPH
Reverse	9.8 MPH 15.8 KPH	14.6 MPH 23.5 KPH	19.5 MPH 31.4 KPH	24.4 MPH 34.4 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.583:1 (12/43) crownwheel and pinion gearset (BMC Part # ?)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	14.7 MPH 23.7 KPH	22.0 MPH 35.4 KPH	29.4 MPH 47.3 KPH	36.7 MPH 59.1 KPH	44.0 MPH 70.8 KPH
2nd	22.3 MPH 35.9 KPH	33.5 MPH 53.9 KPH	44.6 MPH 71.8 KPH	55.8 MPH 89.8 KPH	66.9 MPH 107.7 KPH
3rd	31.0 MPH 49.9 KPH	46.5 MPH 74.8 KPH	62.0 MPH 99.8 KPH	77.5 MPH 124.7 KPH	93.0 MPH 149.7 KPH
4th	39.1MPH 62.9 KPH	58.6 MPH 94.3 KPH	78.2 MPH 125.8 KPH	97.7 MPH 157.2 KPH	117.2 MPH 188.6 KPH
5th (.89)	43.9 MPH 70.6 KPH	65.8 MPH 105.9 KPH	87.7 MPH 141.1 KPH	109.7 MPH 176.5 KPH	131.6 MPH 211.8 KPH
5th (.86)	45.4 MPH 73.1 KPH	68.1 MPH 109.6 KPH	90.8 MPH 146.2 KPH	113.5 MPH 182.7 KPH	136.1 MPH 217.4 KPH
5th (.84)	46.5 MPH 73.1 KPH	69.7 MPH 112.2 KPH	93.0 MPH 149.7 KPH	116.2 MPH 187.0 KPH	139.4 MPH 224.3 KPH
5th (.82)	47.7 MPH 76.8 KPH	71.5 MPH 115.1 KPH	95.3 MPH 153.4 KPH	119.1 MPH 191.7 KPH	142.9 MPH 230.0 KPH
Reverse	9.8 MPH 15.8 KPH	14.6 MPH 23.5 KPH	19.5 MPH 31.4 KPH	24.4 MPH 34.4 KPH	29.3 MPH 47.1 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage (BMC Part # BTB 866) of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1969-1980 MGB Roadsters, 1965-1980 MGB GTs, 1968 MGCs with Overdrive, and all 1969 MGCs.

Road Speed* in MPH / KPH w/ 3.545:1 (11/39) crownwheel and pinion gearset (BMC Part # ATC 7564)[1], Hardy-Spicer B Series Banjo-type rear axle, @:

	2,000 RPM	**3,000 RPM**	**4,000 RPM**	**5,000 RPM**	**6,000 RPM**
1st	14.8 MPH 23.8 KPH	22.2 MPH 35.7 KPH	29.7 MPH 47.8 KPH	37.1 MPH 59.7 KPH	44.5 MPH 71.6 KPH
2nd	22.5 MPH 36.2 KPH	33.8 MPH 54.4 KPH	45.1 MPH 72.3 KPH	56.4 MPH 90.8 KPH	67.6 MPH 108.8 KPH
3rd	31.3 MPH 50.4 KPH	47.0 MPH 75.6 KPH	62.6 MPH 100.7 KPH	78.3 MPH 126.0 KPH	93.9 MPH 151.1 KPH
4th	39.5 MPH 63.7 KPH	59.3 MPH 95.4 KPH	79.0 MPH 127.1 KPH	98.8 MPH 159.0 KPH	118.5 MPH 190.7 KPH
5th (.89)	44.3 MPH 71.3 KPH	66.5 MPH 107.0 KPH	88.7 MPH 142.7 KPH	110.8 MPH 178.3 KPH	133.0 MPH 214.0 KPH
5th (.86)	45.9 MPH 73.9 KPH	68.8 MPH 110.7 KPH	91.8 MPH 147.7 KPH	114.7 MPH 184.6 KPH	137.6 MPH 221.4 KPH
5th (.84)	47.0 MPH 75.6 KPH	70.5 MPH 113.5 KPH	93.9 MPH 151.1 KPH	117.4 MPH 188.9 KPH	140.9 MPH 226.8 KPH
5th (.82)	48.2 MPH 77.6 KPH	72.3 MPH 116.4 KPH	96.4 MPH 155.1 KPH	120.5 MPH 194.0 KPH	144.6 MPH 232.7 KPH
Reverse	10.2 MPH 16.4 KPH	15.3 MPH 24.6 KPH	20.4 MPH 32.8 KPH	25.5 MPH 41.0 KPH	30.6 MPH 49.2 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: All of the Hardy-Spicer B Series banjo-type rear axles require the Hardy-Spicer differential cage (BMC Part # BTB 328) found as Original Equipment in mass production 1962-1968 MGB roadsters, and all use the same carrier, bearings, and shims. There is no

need to change a bearing unless it is worn. All of the Hardy-Spicer crownwheels can be simply bolted onto the Original Equipment Hardy-Spicer differential cages that are found as Original Equipment in the mass production MKI MGB Roadsters with no machining required, with minor adjustments of alignment being made by means of shims.

Road Speed* in MPH / KPH w/ 3.307:1 (13/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	15.9 MPH 25.6 KPH	23.8 MPH 38.3 KPH	31.8 MPH 51.2 KPH	39.7 MPH 63.9 KPH	47.7 MPH 76.8 KPH
2nd	24.2 MPH 43.9 KPH	36.2 MPH 65.8 KPH	48.3 MPH 87.9 KPH	60.4 MPH 109.8 KPH	72.5 MPH 131.8 KPH
3rd	33.6 MPH 54.1 KPH	50.4 MPH 81.1 KPH	67.1 MPH 108.0 KPH	83.9 MPH 135.0 KPH	100.7 MPH 162.0 KPH
4th	42.3 MPH 68.1 KPH	63.4 MPH 102.0 KPH	84.6 MPH 136.1 KPH	105.7 MPH 170.1 KPH	126.9 MPH 204.2 KPH
5th (.89)	47.5 MPH 76.4 KPH	71.3 MPH 114.7 KPH	95.0 MPH 152.9 KPH	118.8 MPH 191.2 KPH	142.6 MPH 229.5 KPH
5th (.86)	49.2 MPH 79.2 KPH	73.8 MPH 118.8 KPH	98.4 MPH 158.4 KPH	123.0 MPH 197.9 KPH	147.5 MPH 237.4 KPH
5th (.84)	50.3 MPH 81.0 KPH	75.5 MPH 121.5 KPH	100.7 MPH 162.1 KPH	125.9 MPH 202.6 KPH	151.1 MPH 243.2 KPH
5th (.82)	51.6 MPH 83.0 KPH	77.4 MPH 124.6 KPH	103.2 MPH 166.1 KPH	130.0 MPH 209.2 KPH	154.7 MPH 249.0 KPH
Reverse	10.9 MPH 17.5 KPH	16.4 MPH 26.4 KPH	21.9 MPH 35.2 KPH	27.3 MPH 43.9 KPH	32.8 MPH 51.5 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Road Speed* in MPH / KPH w/ 3.071:1 (14/43) crownwheel and pinion gearset (BMC Part # BTB 841 and Special Tuning Part # BTB 900)[1], Salisbury Tube-type rear axle, @:

	2,000 RPM	3,000 RPM	4,000 RPM	5,000 RPM	6,000 RPM
1st	17.1 MPH 27.5 KPH	25.7 MPH 41.4 KPH	34.2 MPH 55.0 KPH	42.8 MPH 68.9 KPH	51.4 MPH 82.7 KPH
2nd	26.0 MPH 41.8 KPH	39.0 MPH 62.8 KPH	52.1 MPH 83.8 KPH	65.1 MPH 104.8 KPH	78.1 MPH 125.7 KPH
3rd	36.1 MPH 58.1 KPH	52.2 MPH 84.0 KPH	72.3 MPH 116.4 KPH	90.4 MPH 145.5 KPH	108.4 MPH 174.4 KPH
4th	45.6 MPH 73.4 KPH	68.3 MPH 109.9 KPH	91.1 MPH 146.6 KPH	113.9 MPH 183.3 KPH	136.7 MPH 220.0 KPH
5th (.89)	51.2 MPH 82.4 KPH	76.8 MPH 123.6 KPH	102.4 MPH 164.8 KPH	127.9 MPH 205.8 KPH	153.5 MPH 247.0 KPH
5th (.86)	53.0 MPH 85.3 KPH	79.4 MPH 127.8 KPH	105.9 MPH 170.4 KPH	132.4 MPH 213.1 KPH	158.9 MPH 255.7 KPH
5th (.84)	54.2 MPH 87.2 KPH	81.3 MPH 130.8 KPH	108.4 MPH 174.5 KPH	135.6 MPH 218.2 KPH	162.7 MPH 261.8 KPH
5th (.82)	55.6 MPH 89.5 KPH	83.3 MPH 134.1 KPH	111.1 MPH 178.8 KPH	138.9 MPH 223.5 KPH	166.7 MPH 268.3 KPH
Reverse	11.8 MPH 19.0 KPH	17.6 MPH 28.3 KPH	23.5 MPH 37.8 KPH	29.4 MPH 47.3 KPH	35.3 MPH 56.8 KPH

* Presumes road wheels that have an Original Equipment rolling radius of 23.5” (596.9mm).

[1] Note: This gearset requires the Salisbury differential cage BMC Part # BTB 840 of the Salisbury tube-type rear axle, found as Original Equipment in mass production 1968 MGCs with Overdrive and 1969 MGCs without Overdrive.

Understanding The Transmission

If you intend to be putting serious power through your car and vigorously piloting it down winding roads as a sports car was meant to be driven, then logic dictates that it would be prudent to first disassemble the four-synchro transmission and carefully scrutinize its components. Many owners will intrepidly tear down their engines, yet balk at this prospect despite the fact that a transmission is far less challenging to work on than an engine is. Why? Because in their ignorance of the machinations of the transmission, they perceive it as being a mysterious device of awesome complexity. In fact, it is actually much easier to comprehend than the complex interrelationships of an engine and is comparatively easy to work on if you just take your time and follow proper procedures.

It is best to tear away the veil of ignorance pertaining to the presumed complexities of the transmission before starting on this quest. It might help to have an exploded view of a four-synchro transmission at hand, such as is found in the Bentley manuals. You will want to be able to identify the location of the third and fourth synchronizer hub, the input shaft (first motion shaft), and the first and second synchronizer hub.

In the front and rear of the main transmission casing there are large caged ball bearings. Aft of the main transmission casing, inside of the rear extension, are the speedometer drive gearset and the rear support bearing of the mainshaft (third motion shaft), as well as an oil pump that lubricates both the bushings and the bearings of the transmission.

When the side cover has been removed from the transmission, you will see that at the top of the transmission there are three selector fork rods with a selector fork (saddle) and a selector attached to each of them. The selector fork rods and their attached selector forks (saddles) are individually moved fore or aft as units by means of selectors attached to the selector fork rods in order to engage (select) a desired gearset. The top selector fork rod is for selecting either first or second gear. The bottom selector fork rod as for selecting either third or fourth gear. The middle selector fork rod is for selecting reverse gear. Engagement of reverse gear and first gear occurs at the rear of the transmission, engagement of second gear and third gear occurs in the middle of the transmission, and engagement of fourth gear occurs at the front of the transmission.

At the extreme front inside of the transmission, you will see the input gear which rides in a large caged ball bearing. The input gear rotates clockwise whenever the engine is running with the clutch engaged. The input shaft is referred to as the "first motion shaft". The input shaft (first motion shaft) is very short, and has only one gear (the input gear) on its inner end. It drives a cluster gear on the layshaft. The layshaft is referred to as the "second motion shaft". On the layshaft (second motion shaft) is a cluster gear of single-unit design that spins on needle bearings. This all-of-one-piece cluster gear is referred to as the "laygear".

In the three-synchro transmission with its non-synchronized first gear, the laygear comprises the input-driven fourth gear at its front, followed by helically-cut third and second driving gears in the center, and a small straight-cut gear at its rear that drives both the first gear and the reverse gear.

In the four-synchro transmission with its synchronized first gear, the laygear comprises the input-driven fourth gear at its front, followed by helically-cut third and second driving gears in the center, and a helically-cut first gear behind them with a straight-cut reverse gear at the rear.

The mainshaft is frequently referred to as the "output shaft", or, more properly, as the "third motion shaft". The front end of the mainshaft (third motion shaft) is supported by a set of needle-roller bearings that are located inside of the input gear of the first motion shaft, and extends all the way to the back end of the transmission where, in the case of non-Overdrive transmissions, it couples directly to the driveshaft (propeller shaft) via a flange in order to drive the rear axle assembly. It logically follows that the input gear of the first motion shaft (input shaft) drives the laygear on the layshaft (second motion shaft), which in turn drives the individual output gears on the mainshaft (third motion shaft / output shaft).

You should note that the helical gear teeth of the gears on the mainshaft (third motion shaft) are in constant mesh with those of the laygear on the layshaft (second motion shaft). The laygear is driven counterclockwise (anticlockwise) by the clockwise motion of the input gear of the first motion shaft. The sidethrust generated by these helically-cut gears results in a linear motion of the mainshaft along its axis, thus requiring that the motion be limited to within .002" to .003" (.0508mm to .0762mm) by the use of shims.

Whenever neutral is selected, first, second, and third gears are in a disengaged state and freewheel clockwise on the mainshaft (third motion shaft). There are synchronizer hubs in between these gears. On the rear side of the input gear, the front side of the third gear, the rear side of the second gear, and the front side of the first gear are small pointed cogs that can connect to female splines in the adjacent synchronizer hub. Moving the synchronizer hub into engagement with these small pointed cogs will lock the gear to the mainshaft (third motion shaft) in order to drive it. In order to better understand the mechanics of this engagement, it is best to describe the workings of the synchronizer hub assembly.

The synchronizer hubs are splined to the mainshaft (third motion shaft). The side of the gear that is adjacent to the synchronizer hub has small pointed cogs that point toward the synchronizer hub. The baulk rings (synchro rings) have identical cogs that are all oriented in the same direction. The baulk rings (synchro rings) interlock with the cogs of the synchronizer hub, although the baulk rings (synchro rings) are allowed a small amount of rotational movement, as well as a slight amount of fore and aft movement. When not engaged, they float away from the beveled surface on the side of the gear, with just a miniscule amount of contact. The baulk ring (synchro ring) is essentially a conical clutch, and as such, the tapers must make solid contact in order to lock the two rotating parts together at a constant speed. Very light friction from the spinning gear causes the baulk ring (synchro ring) to rotate slightly. The sliding synchronizer sleeve has pointed cogs similar to those of the baulk ring (synchro ring). When actuated by the selector fork (saddle), the synchronizer sleeve slides either back or forth on the splines on the outside of the synchronizer hub. As this rapidly progresses, the angled faces on the pointed cogs of the synchronizer sleeve act upon and mesh with the slightly offset angled faces of the baulk ring's (synchro ring's) pointed cogs, causing the cogs of the baulk ring (synchro ring) to rotate into alignment with the cogs of the synchronizer sleeve. The force transmitted by the angles between the two intermeshing sets of pointed cogs thrust the baulk ring (synchro ring) toward the gear, forcing the conical surfaces of both the baulk ring (synchro ring) and of the gear into contact. Because the synchronizer hub is splined to the mainshaft (third motion shaft), this side thrust forces the gear's rotational speed to become synchronous with that of the mainshaft (third motion shaft). As the speed of the gear and that of the main shaft (first motion shaft) become synchronous, the cogs of the gear come into alignment with those of the synchronizer sleeve, and the cogs of the synchronizer sleeve then slide in and mesh with the cogs on the gear, thus locking the gear to the synchronizer hub.

Also while the speeds are becoming synchronous, the three spring-loaded ball detent assemblies hold the synchronizer hub and the sliding synchronizer sleeve in juxtaposition so that they may function as a single unit once the gear is engaged, and prevents the synchronizer sleeve from rubbing against the selector fork (saddle) which, in turn, is also held in place by a ball and spring detent in the mechanism of the shifting linkage. It should be mentioned that if the ball and spring detents were absent, then the side thrust would tend to force the sliding synchronizer sleeve back off of the cogs of the gear. At this point, the synchronized gearshift is accomplished, and all that remains is for the clutch to be engaged in order for power to be transmitted through the transmission.

A fair amount of mass is being brought up to a synchronized rotational speed, and the differences in rotational speed are progressively greater between the individual gearsets of the lower gear ratios. This is the reason that the baulk rings (synchro rings) of the gearset for the second speed tends to wear out first. From a purely technical standpoint, first gear is theoretically the most vulnerable to wear because of its higher mass differential (the lower the gear ratio, the larger and heavier one of the two gears is in relation to its mating gear), but it is the least used because it is engaged only when downshifting. In practice, the second gear baulk rings (synchro rings) carry the burden of having to synchronize every up-shift from first and every downshift from third. If you speed-shift a lot, do not be surprised if you see more wear on the second gear baulk rings (synchro rings) than on the others.

In the three-synchro transmission with its non-synchronized first gear, first gear is the large straight-cut gear that is splined to the mainshaft (third motion shaft) at the rear on the mainshaft. Being splined to the mainshaft (third motion shaft), it always rotates in unison with the mainshaft, clockwise for forward drive, and counterclockwise (anticlockwise) for reverse drive. First gear is also the outer moving part of the synchronizer hub. The medium-sized straight-cut gear in the lower rear corner of the transmission is the reverse gear. This is actually a one-piece pair of gears that rotates on a separate short shaft with a slightly larger straight-cut gear on its rear. When neutral is selected, the reverse gear is not engaged and remains motionless. When shifting into first gear the large straight-cut gear is slid rearward on the mainshaft (third motion shaft)in order to engage with the small straight-cut gear on the layshaft. It will also simultaneously engage reverse gear. The reverse gearset will freewheel whenever the gearset for first gear is in operation. If you want to be kind to your transmission, shift into these non-synchronized gears only when the car is motionless so that you do not grate or grind the gears and chip little fragments off of the corners from the teeth of these straight-cut gears. The secret that all 3-synchro box owners know is to just lean the gearlever against second gear for about 1 second without actually engaging second gear, then engage first gear. What this does is to slow down the moving parts making for a quiet engagement of first gear.

It should be noted that the baulk ring (synchronizer ring) for engaging the second gear of the three-synchro transmissions underwent a change of materials during the course of production. The initial baulk ring (synchronizer ring) (BMC Part # 11G 3063) was made of brass, while its successor was made of more durable steel (BMC Part # 22H 249). The second gear that uses the brass baulk ring (synchronizer ring) has a shiny, smooth finish to the surface of its engagement cone, while the second gear that uses the steel baulk ring (synchronizer ring) has a dull, matt, sintered finish to the surface of its engagement cone that is needed in order to attain the correct friction characteristics with the steel baulk ring (synchronizer ring). Needless to say, these second gears and their matching baulk rings (synchronizer rings) are interchangeable only as matched assemblies. The brass baulk ring (synchronizer ring) cannot be used with the later second gear, nor can the steel baulk ring (synchronizer ring) be used with the earlier second gear. Note that all four-synchro transmissions use only a different steel baulk ring (synchronizer ring) (BMC Part # 88G 397) for engaging both the first and second gears.

In the four-synchro transmission with a synchronized first gear, first gear is a separate helically cut gear that freewheels on the mainshaft (third motion shaft) until it is locked to the mainshaft (third motion shaft) by the synchronizer hub. It is located immediately forward of the large straight-cut reverse gear that, like its counterpart in the three-synchro transmission, is splined to the mainshaft (third motion shaft) at the rear.

When shifting into second gear, the rear synchronizer hub slides forward to lock second gear to the mainshaft (third motion shaft). When shifting into third gear, the front synchronizer hub slides rearward in order to lock the third gear to the mainshaft (third motion shaft). When shifting into fourth gear, the front synchronizer hub slides forward in order to lock the input gear to the mainshaft (third motion shaft).

You will notice that the synchronizer hub for both third gear and fourth gear is located over the rotating interface of the first motion shaft and the mainshaft (third motion shaft).

Fourth gear does not impart a large amount of sidethrust, but third gear sure does. When you downshift into third, the sidethrust produced by the helically-cur gears forces the input shaft (first motion shaft) and the mainshaft (third motion shaft) to try to form an angle instead of a straight line, which is the reason why the transmission will jump out of gear if there is too much endplay (endfloat).

Now, imagine the path that power takes through the transmission in fourth gear as being a straight line through the transmission with no side loads induced from the laygear on the layshaft (second motion shaft). In fourth gear the input shaft (first motion shaft) and the mainshaft (third motion shaft) are locked together to rotate as one continuous output shaft. In fourth gear, the first motion shaft continues to rotate the gears on the layshaft, but they do not perform any function as none of the mainshaft (third motion shaft) gears that they are meshed to are connected to the mainshaft (third motion shaft) by their synchronizer hubs. Instead, these gears simply spin freely (freewheel) on the mainshaft (third motion shaft) by means of their bushings. W hen traveling down the road the transmission will thus be at its quietest in fourth gear even if there is a bad bearing somewhere in the transmission.

To shift into reverse, the reverse gear cluster is slid forward. This engages its smaller driven reverse gear with the first gear on the mainshaft (third motion shaft), while simultaneously engaging its larger driving reverse gear with the small straight-cut gear of the laygear on the layshaft (second motion shaft). By connecting this reverse idler gear in between the laygear and first gear, the direction of the rotation of the mainshaft (third motion) shaft is reversed (hence the name “reverse gear”).

So you see, the interrelated actions of the transmission are actually far easier to understand than the interrelated complexities of the engine. Note that the transmission is a constant-mesh type, thus when you “shift gears”, you are actually moving the synchronizer hubs to enable the desired gearset to transfer power, rather than causing the gears themselves to move in and out of engagement with each other as in the case of an antiquated non-synchro “crashbox” which should always be double-clutched in order to synchronize the relative speeds of the gearset being selected.

Conversion From A Three-Synchro Transmission

To A Four-Synchro Transmission

There are several reasons for an owner to convert from a three-synchro transmission to a four-synchro transmission. Some critical parts have become very hard to obtain. Also, the convenience of a synchronized first gear has a very strong appeal.

The simplest way is to use the engine backplate (BMC Part # 12H 2189) and gaskets for an 18GD and onwards engine, plus its flywheel (BMC Part # 12H 2184), and flywheel locktab (BMC Part # 12H 1303), ring gear (BMC Part # 12H 2900), rear main oil seal (BMC Part # 13H 2457), and its retainer (BMC Part # 12H 1547). Note that from 18V 883 engines onwards, the flywheels had only one locating dowel hole, so should you use this flywheel, it will then be necessary to machine an additional dowel hole. You will also need a new clutch, a new crankshaft spigot pilot bushing, and a 1968 and onwards Lucas 2M100 pre-engaged-type starter assembly (BMC Part # 13H 6130, Moss Motors Part # 131-220). Be aware that the Moss Motors oil filter conversion kit (Moss Motors Part # 235-940 will not fit when using this starter motor, so if you have one already installed, then you will need to replace it with a top-loading oil filter stand (BMC Part #12H 3273), the 1/2" x 3" USS oil filter stand center bolt, and its copper washer. The hardware required is as follows: five 5/16" x 2 3/4" SAE bolts, two 5/16" x 3" SAE bolts, seven 5/16" lockwashers, seven 5/16" SAE nuts, one 3/8" x 1" SAE bolt, one 3/8" x 1 1/2" SAE socket head cap screw, two 3/8" lockwashers, and two 3/8" SAE nuts.

However, if you cannot obtain the necessary later-model parts, you can attach your existing engine backplate for the three-synchro transmission to the four-synchro transmission, and then drill and tap the upper bolt hole for the upper starter mounting bolt to 3/8" USS. Use a 3/8" x 1 3/4" bolt and lockwasher in order to attach the starter motor. Next, use a 3/8" drill bit to drill a relief indent into the transmission flange to a depth of 1/4" for the lower mounting bolt of the Lucas M418G inertia-type electric starter assembly (BMC Part # 13H 4561, Moss Motors Part # 140-165).

Remove the engine backplate and helicoil the bottom mounting hole for the starter motor to 3/8" USS. Note that the existing hole is already conveniently sized for this purpose. Use a 3/8" X 1" USS bolt and lockwasher in order to attach the starter motor.

You will need six 5/16" X 2 3/4" SAE bolts with nuts and lockwashers in order to attach the transmission to the engine backplate. When retaining the use of the earlier flywheel of the three-synchro-equipped engines, installation of the Original Equipment Lucas M418G inertia-type electric starter assembly (BMC Part # 13H 4561, Moss Motors Part # 140-165) requires that the transmission housing of the four-synchro transmission be bored out (2.5" +) in order to provide clearance for the Bendix drive of the starter motor. You can also use the MGA starter motor, which will require a smaller, 2 1/8" hole, although you should be made aware of the fact that this lower-powered starter motor is a poor choice for cranking over a higher compression or larger displacement engine.

There are two ways to solve the problem of matching the pilot shaft section of the input shaft (first motion shaft) of the four-synchro transmission with the .620" Inside Diameter (I.D.) crankshaft spigot pilot bushing of the three-main-bearing engine. If you are performing a rebuild of the engine, then it would be more convenient to counter-bore the crankshaft to a depth of 1.5" to 1.625" and an Inside Diameter (I.D.) of 1.123". Due to small variations in Outside Diameter (O.D.) of spigot pilot bushings, be sure to supply the machinist with the actual spigot pilot bushing that you intend to use. Use the 1" long spigot pilot bushing as used on the MGB 18V engine (BMC Part # 22H 1416, Moss Motors Part # 330-415). Before installing the new spigot pilot bushing, soak it in 10W oil for several days or weeks. The longer it soaks, the better.

An alternative approach is to retain the original size crankshaft spigot pilot bushing and have the pilot section of the input shaft (first motion shaft) of the four-synchro transmission with its Outside Diameter (O.D.) of .850" reduced to an Outside Diameter (O.D.) dimension of .6215" to .6220", and a total length of 1.500" to 1.625". Be sure to install a new spigot pilot bushing into the crankshaft, the dimensions of which should be a Length of .6875", an Outside Diameter (O.D.) of .814", and an Inside Diameter (I.D.) of .625" to.626", for a working clearance of approximately .003".

Be aware that if your transmission is to be equipped with an Overdrive unit, you will then need to install an Overdrive lockout switch, complete with its adjustment washers. One critical item is the number and thickness of the washers underneath the Overdrive lockout switch. Normally, there must be two washers; otherwise it will not function correctly. Adjust the Overdrive lockout switch so that it operates in 3rd & 4th gear only. Note that the

1977 and later overdrive operated only while in 4th gear. This was achieved by the installation of a micro-switch attached to the selector mechanism.

The existing transmission support crossmember assembly used with the three-synchro transmission may be employed in conjunction with new rubber mounts. However, your original restraint assembly cannot be used. You may wish to install the left hand and right hand engine restraint brackets as found on the 1968 to 1974 MK II model MGB. Yet another option would be to fit the 1975 and later crossmember assembly (BMC Part # BHH 1543), complete with all of its brackets, mounts, bushes, buffer pads, engine restraint rod, spacers and front exhaust strap.

There is a convenient way to solve any transmission interference to adjacent body problems, i.e., in the area of the starter motor. Before assembling the engine backplate to the engine, temporarily fit it to the all-synchro transmission, and then remove the transmission material that would have accommodated the top securing bolt of the starter motor until the 3/8" x 1 1/2" SAE socket head cap screw can be inserted from the rear directly through and butt up against the back plate. This will mean one less bolt securing the transmission to the engine back plate assembly, but should not present a problem. NOTE: if you prefer to not use this weaker method of starter motor attachment, then a four-pound hammer will accomplish the task of providing additional space inside of the transmission tunnel.

Remove the original starter solenoid to starter motor cable, enlarge the cable end to fit the new starter solenoid terminal, then refit and re-route the cable.

Depending upon which transmission/rear axle combination you use, you could either lengthen or shorten your existing (propeller shaft) as required, or use an appropriate-length Original Equipment Hardy-Spicer driveshaft (propeller shaft).

Working On The Four-Synchro Transmission

In order to remove the transmission from the car, it is necessary to remove the engine and transmission as a single assembled unit. Never attempt to remove the transmission from the car by cutting the structural crossmember beneath it. This will seriously damage the rigidity of the chassis.

After having removed the electric starter motor and its solenoid, remove all of the machine bolts that secure the bellhousing to the engine, and then, while ensuring that no load is placed upon the input shaft (first motion shaft), smoothly pull the transmission straight off of the engine. When you remove the transmission from the engine you may notice that at the very bottom of the bellhousing there is a small drain hole, and that in that drain hole is a cotter pin (split pin). Do not discard it. Normally there is no oil inside of the bellhousing, but if an oil seal starts to leak (the crankshaft rear oil seal is usually the culprit), or if the owner overfills the oil sump, then oil gets onto the engine side of flywheel. Centrifugal force flings it outward and all around the inside of the bellhousing, whereupon gravity takes over and it drains down to the bottom of the bellhousing. The drain hole is located at the bottom of the bellhousing in order to allow any oil to drain away without any build up that may effect the operation of the clutch. The cotter pin (split pin), often called a “jiggle pin”, is fitted into this drain hole and moves enough to clear away any muck that would clog the drain hole.

Turn the circlips that retain the carbon clutch release bearing, and then withdraw the carbon clutch release bearing from the clutch withdrawal lever. Remove the nut that retains the pivot pin of the clutch withdrawal lever, and then remove the pin, the clutch release arm, the clutch withdrawal lever, and its pushrod. Inspect both the bushing (BMC Part # 11G 3195) inside of the clutch withdrawal lever and the pivot pin (BMC Part # 11G 3196) for wear. If you find any, rebush the clutch withdrawal lever and replace the pivot pin. This will greatly reduce slop in the action of the clutch pedal and make for a much more consistent clutch action.

Prior to draining the transmission, make sure that you can remove the filler/level plug or dipstick before draining the oil out. You can live with not changing the oil for a bit while you ponder how to remove it, but not if you have already drained the oil out and cannot refill it. Slacken each of the machine bolts of the transmission sump pan in sequence bit by bit, much as you would with cylinder head nuts in order to preclude the chance of warping. In order to prevent the transmission sump pan from suddenly falling and become damaged, free the transmission sump pan with at least one of the machine bolts on opposite sides still loosely fitted. You will need to exert a bit of gentle leverage in order to get the transmission sump pan parted from the body of the Overdrive unit, but fortunately there is a handy tab on the transmission sump pan adjacent to the large hex plug of the relief valve that is there expressly for this purpose. Be ready for more oil to drain out once you have broken the seal of the transmission sump gasket. Leave it to drain at that point rather than trying to get the transmission sump pan completely off and fill your sleeves with oil. After having drained the oil from the transmission, stand it on end (bellhousing downwards) in order to allow any oil remaining inside of the rear extension of the main transmission casing or inside of the Overdrive unit to drain forward into the main housing, and then drain out the remaining oil. This extra draining step will make the rest of the disassembly a much less messy affair.

Take out all of the machine bolts that secure the side cover, remove the side cover, and then carefully remove all of the old gasket material from the mating surfaces of both the side cover and the transmission. Reinstall the screws back into their threaded holes in the side of the main transmission casing so that they will not get lost.

After removing the side cover, and before disassembling anything, you will need to establish certain key clearances within the components. While you are measuring the clearances, you want everything dry and clean enough to eat off of, so spray everything clean with carburetor cleaner first. Inspect the gear teeth. If any of them are worn, chipped, missing, or blackened, then the mainshaft (third motion shaft) assembly will have to be disassembled and the damaged parts replaced. Next, gauge the endplay (endfloat) of the laygear. It should be between .002" and .003" (.05mm to .08mm). Finally, gauge the endplay (endfloat) clearances of the first speed, second speed, and the third speed gears and their thrust washers. Each should be between .005" and .008" (.13mm and .20mm). If any of them are outside of specifications, then the mainshaft (third motion shaft) will have to be disassembled and the worn thrust washer(s) replaced.

When assembling transmissions, there are a number of tolerances that must always be respected. However, it is not always indicated why these tolerances are critical. One such critical dimension is the endplay (endfloat) is between third and second gear. The critical reason for a precise setting of the endplay (endfloat) is that the torque that is transmitted by the gears during a downshift at high speed does change substantially in both magnitude and direction, and any possible movement of the third speed gear results in it being accelerated into a high-speed movement with high inertia. When the third speed gear moves, it takes the synchronizer hub with it. If the gear is being stopped dead by impacting against the thrust washer, then the synchronizer hub is loaded with kinetic energy and wants to continue moving. If the endplay (endfloat) is in excess of the specified limit of .005”-.008” (.13mm-.20mm), this energy can to be enough to overcome the resistance of worn locating balls and springs. As a consequence, the synchronizer hub becomes disengaged from the third speed gear and then pushes the gear change lever into neutral. Wear of the bushing of the third speed gear and of the thrust washers caused by the tilting of the gear will aggravate this situation, as will wear to the gear change fork that is caused by the irritated driver trying to hold the gear lever in position.

Now, check the condition of the baulk rings (synchro rings) of the synchronizer hubs. Move the synchronizer hub away from baulk ring (synchro ring) in order to expose the doghouse-shaped cogs and the ring. Press the baulk ring (synchro ring) as tightly as possible against the surface of the mating cone of its adjacent gear, and then try to slide an .020” (.508mm) thickness gauge in between the flat surface of the baulk ring (synchro ring) and the adjacent side surface of the gear. If the feeler gauge fits without binding, the synchronizer hub will probably function in a satisfactory manner. Having performed that operation, use the feeler gauge to measure the baulk ring (synchro ring)-to-hub gap. If the baulk ring (synchro ring)-to-hub gap is less than .0625” (1.5875mm), then the mainshaft (third motion shaft) will have to be disassembled and the baulk ring (synchro ring) replaced. This could be a result of wear on the gear, or an incorrectly machined baulk ring (synchro ring). Weak or missing detent springs/balls will not actuate the synchronizer hubs correctly. It is not rare for people to lose one or more of these upon previous disassembly, and thus to have them missing. The synchro mechanism has to accelerate the entire gear train from crankshaft to the gear on the third motion shaft in order to function properly. As a result, any drag in the system will cause trouble. Clutch drag, gearbox/engine misalignment, bad spigot pilot bushing in the crankshaft, bad bearings, etc can result in this problem. If the synchromesh is weak on any of the gears, then replacing the related baulk ring (synchro ring) is always worthwhile.

Examine the angled surfaces on the triangular projections of the cogs (doghouse-shaped when new) of the baulk rings (synchro rings), and make a comparison between the most worn ones (probably on the second gear baulk ring / synchro ring) and the least worn ones (usually on the fourth gear baulk ring / synchro ring). A close look often reveals more wear on the downshift side of the cog (top angled surface) than on the upshift side (lower angled surface). The third and fourth gear baulk rings (synchro rings) usually do not exhibit much wear, certainly considerably less than the second gear baulk ring (synchro ring) does. In most cases only a small amount of wear is found on the on third gear baulk ring (synchro ring), and almost negligible wear on fourth gear baulk ring (synchro ring).

Remove the retaining bolts from the gearlever extension housing, and then noting the two dowels that locate it so that it will have to be rocked up and down in order to loosen it, lift it off of the main transmission casing. Carefully remove all gasket material from the mating surfaces, and then screw the retaining bolts back into their threaded holes in the top of the main transmission casing so that they will not get lost. Note that there is a breather on the top of the gear change lever (gear shift lever) extension housing that tends to become plugged up with road crud. The transmission dipstick has a small oil seal on its neck, so air is trapped inside of the transmission whenever the breather becomes clogged. When the transmission reaches its normal operating temperature, causing the air that is trapped inside to expand, the oil seals start to leak as a consequence of the buildup of internal pressure. Cleaning the breather is a simple affair, but most owners do not know that it is there at the front of the top of the gear change lever (gear shift lever) extension housing. Just clean around the top of the gear change lever (gear shift lever) extension housing so that crud will not get into the threads, unscrew it, and spray it out with carburetor cleaner. Carefully clean the threads with an old toothbrush, and then reinstall it after it dries.

Inspect both the bushing on the end of the gearlever and the bushing in the top of the selector lever. If either of these bushings is worn and you fail to replace it, then you will not attain the correct amount of engagement travel of the synchronizer hub, which can in turn result in the car popping out of gear.

Now, remove the selector-interlocking arm. That is the stamped metal bar with a “crab claw” metal bracket attached to it that prevents the simultaneous engagement of two gears. The selector-interlocking arm is in fact a bit fiddly to get in and out, but if it is not removed, you will not be able to remove the extension housing from the main transmission casing. Note that at times you have to rotate the extension housing while removing it from the main transmission casing in order to enable it to clear the ends of the shifter rods.

If you are working on a four-speed-only transmission, unscrew the machine bolts that secure the rear extension to the main transmission casing, remove the nuts and the machine washers, and then remove the rear extension, taking care that any shims that have been fitted to the third motion shaft do not become lost. After you separate the rear extension from the main transmission casing, you will find a housing at the back of the main transmission casing that holds the large rear bearing. That housing will have an alignment dowel that is used to rematch to the rear extension. Before removing the bearing housing, scribe marks on the main transmission casing and the rear extension so that you can align the dowel with the rear extension. Remove the bearing, and then carefully examine it. Should it need to be replaced, remove its retaining circlip (BMC Part #) and press out the bearing from the extension. Remove the old oil seal and all remnants of the old gasket from the mating surfaces of both the rear extension and the rear of the main transmission casing.

If you are working on a four-speed-with-Overdrive transmission, remove the eight nuts that secure the Overdrive unit. The Overdrive unit will have to be eased away from the adapter housing at the rear of the main transmission casing in order to release some of the nuts. Now, remove the Overdrive unit, and then carefully remove all gasket material from the mating surfaces of both the Overdrive unit and its adaptor housing. Replace the machine bolts and their nuts into their mounting holes on the adapter housing so that they will not get lost. Next, remove the machine bolts and their nuts that secure the adapter housing to the main transmission casing, and then remove the adapter housing. Replace the machine bolts and their nuts into their mounting holes on the adapter housing so that they will not get lost.

Next, unfasten the front cover plate nuts and remove the front cover plate, retaining any thrust washers that are between it and the bearing that supports the first motion shaft that is located behind the plate. The condition of these thrust washers will tell you if the outer race of the input bearing has been rotating in its housing and, if so, may require the attention of a machinist. Next, carefully remove the old oil seal and completely remove all of the old gasket from the mating surfaces of both the inside of the front cover and the transmission. Now, remember that old oil seals are always a problem waiting to happen. Have the forethought of installing a new oil seal into the front cover. This will save you from the future headache of having to take everything apart in order to pull the engine / transmission package out just to fix an old, leaking oil seal. Now, replace the front plate nuts back into the front of the main transmission casing so that they will not get lost.

Remove the three retaining bolts and their machine washers on the top of the front end of the main transmission casing, and then extract the springs and the detent plungers. Set the springs (BMC Part # 22G 327) and detent plungers (BMC Part # 22B 408) aside so that they will not become lost, and then screw their retaining bolts and their machine washers back into the top of the main transmission casing so that they will not become lost. Now, remove the retaining bolt and its washer of the reverse gear selector detent, and then withdraw its ball spring (BMC Part # 22B 614), ball (BMC Part # BLS 110), detent plunger (BMC Part # 22B 396), and plunger spring (BMC Part # 1G 3863). Clean them thoroughly

Check the bore holes where the selector fork rods slide in the aluminum housing. They should be a very close fit with little or no perceptible clearance. If their fit is sloppy, then the bore holes should be machined oversize and bronze sleeves installed. While noting their positions and orientations of the three selector forks (saddles) and also of the three selectors, release the locknut and machine bolt that secure each of the three selector forks (saddles). Next, unfasten the locking bolts on the selector forks (saddles), and then unfasten the locking bolts on the selector fork rods. Remove the plugs, detent plungers, and springs from the selector forks (saddles), taking care to keep them together in matched sets. At this point it is important to do things in the correct order. First, withdraw the Reverse gear selector fork rod from the rear of the transmission, and then remove the selector fork and selector out through the opening in the side of the main transmission casing. Second, withdraw the Third / Fourth gear selector fork rod through the rear of the transmission, and then remove the selector fork (saddle), and then the selector out through the side of the main transmission casing. Third, withdraw the First / Second gear selector fork rod from the rear of the transmission, and then remove the selector fork (saddle), as well as the selector out through the side of the main transmission casing. As you remove the selector fork rods out through the back of the main transmission casing, be sure to screw the retaining bolts of the selector forks (saddles) and the selectors back into each of them so that they will not get lost, and then place the selector forks (saddles) and selectors onto their corresponding selector fork rods in their original positions and orientations in order to make the reassembly process easier. If you should accidentally get the selector forks (saddles) mixed, they can be identified by their shapes. The small selector fork (saddle) and the longest, deepest reach selector are for selecting reverse gear. The large selector fork with the deepest reach and the widest selector are for the synchronizer hub that selects First / Second gear. The large selector fork with the shallowest reach and the narrowest selector are for the synchronizer hub that selects third / fourth gear.

It is important to determine that the selector forks (saddles), selectors, and the selector fork rods that they are screwed onto, are not badly worn, deformed or bent. If the corners of these parts are worn and rounded off, that is acceptable as it just endows the transmission with slicker shifting. However, if you find that there are no longer any flat surfaces left, that is, they are completely rounded off, then the shifting will be quite sloppy. In order to check the selector forks (saddles), drag your fingernail along the edge of the selector fork (saddle) that wears against the groove in the synchronizer hub. If there is much of a lip on the selector fork (saddle), or if it is wearing thin, then you will need to replace the selector fork (saddle) because if there is a lip, then it cannot cause the gear that is being selected to fully engage, in which case the transmission will tend to pop out of gear. The straightness of the selector fork rods can be checked by using the previously mentioned old petroleum-jelly-and-a-mirror trick. (First / Second Gear Selector Fork, BMC Part # 22B 284; First / Second Gear Selector, BMC Part # 22B 379); First / Second Gear Selector Fork Rod, BMC Part # 22B 375; Third / Fourth Gear Selector Fork, BMC Part # 22B 285; Third / Fourth Gear Selector, 22B380,Third / Fourth Gear Selector Fork Rod, BMC Part # 22B 377; Reverse Gear Selector Fork, BMC Part # 22B 283; Reverse Gear Selector, BMC Part # 22B 378; Reverse Gear Selector Fork rod, BMC Part # 22B 376). If you are having problems shifting into first with first/second motion shaft and laygear near complete stop, then you could have a problem with a worn gear lever bushing (nylon), worn selectors, or worn and/or loose selector forks (saddles). These latter two items are made of bronze and will wear.

In order to disassemble the transmission, first bend back the tab washer of the reverse idler gear locking bolt, and then remove the bolt. Withdraw the reverse gear shaft out through the rear of the main transmission casing while sliding it out of its gear, and then remove the gear through the opening in the side of the main transmission casing. Examine the bore of the reverse gear bushing (BMC Part # 22H 310) that is located inside of the reverse gear for wear, and then examine the teeth of the reverse gear (BMC Part # 22H 308). If any of them are worn, chipped, missing, or blackened, then the reverse gear will have to be replaced. Any scoring of the reverse gear shaft (BMC Part # 88G 467) or the reverse gear bushing (BMC Part # 22H 310) will likewise indicate the need for replacement. Lay this entire assembly aside as a group.

After having gauged and noted the endplay (endfloat) of the layshaft (second motion shaft) (it should be .002" to .003" / .0508mm to .0762mm), use a soft hammer and a wooden drift to carefully drift the layshaft (second motion shaft) forward out of its mounting inside of the main transmission casing. Withdraw the layshaft (second motion shaft) through the front of the main transmission casing, and then recover the thrust plates of the layshaft (second motion shaft). Inspect the layshaft (second motion shaft) carefully for signs of wear. If the wear appears to be serious, do not be tempted to replace the needle-roller bearings with a brass bushing. Brass bushings do not work properly without forced lubrication! Place the laygear into the bottom of the main transmission casing so that it will not interfere with the removal of the input shaft (first motion shaft).

Loosen and then rotate the retaining clips of the clutch release bearing and then remove the clutch release bearing. Next remove the rubber boot from the clutch lever. Unscrew the nuts that retain the front cover. Now, withdraw the front cover assembly with the input shaft (first motion shaft) along with its bearing through the front of the main transmission casing. Next, measure how far the bearing of the input shaft (first motion shaft) protrudes from the casing. Take care not to misplace the spring ring and shim(s) that are installed behind the bearing. The purpose of the shims is to clamp the outer bearing race so that it will not move back or forth. The bearing needs to be secured in place in order to prevent forward and backward movement generated by the sidethrust from the helical teeth of the gears. Should any of the shims become lost or damaged, use a depth micrometer or the tail of a vernier caliper in order to measure the depth of the recess from the face of the front cover. Add .012" (.3048mm) for the thickness of the joint washer to the amount of the depth dimension from the face of the front cover. Fit shims between the front cover and the bearing of the input shaft (first motion shaft) in order to bring the distance that the bearing protrudes to .000" to .001" (.000mm to .0508mm) greater than the depth of the front cover. Shims are available in thicknesses of .002" (.0508 mm) (BMC Part # 6K 778, Moss Motors Part # 462-020) and .005" (.127 mm) (BMC Part # 6K 779, Moss Motors Part # 462-025).

Next, inspect the input shaft (first motion shaft) for wear. Pay particular attention while inspecting the machined recess in the main transmission casing for wear and / or cracks. In order to replace the bearing of the input shaft (first motion shaft), bend back the lockwasher unfasten the retaining nut (Beware! This is a left-hand thread!), and then press off the bearing. Carefully inspect the input shaft (first motion shaft) for signs of wear and damage to its gear teeth or to its splines (BMC Part # 22B 556). Press on a new bearing (BMC Part # 6K 777) with the circlip facing toward the front, and then reinstall the retaining nut and secure it with the lockwasher (BMC Part # 22H 798).

Remove the retaining circlip (BMC Part # 37H 1911) that secures the Overdrive pump drive cam from the mainshaft (third motion shaft), and then remove both the Overdrive pump drive cam (BMC Part # 22B 611) and its locating ball (BMC Part # BLS 106) from the mainshaft. Carefully clean them and store them away. Remove the rear flange nut (It is torqued to 140 Ft-lbs on non-Overdrive transmissions). If you are converting a four-speed-only transmission to a four-speed-with-Overdrive, the flange itself can be held with a large pipe wrench or in a vice as it will be discarded. If you are not performing such a conversion, protect the distance piece by placing two pieces of soft wood in the jaws of the vice, and then clamp the mainshaft (third motion shaft) between them while you remove the rear flange nut.

Now, place a pair of milk crates under the bellhousing so that the input shaft (first motion shaft) does not fall and hit the ground when the bearing housing slides free. Ensuring that the laygear is safely away from the gears on the mainshaft (third motion shaft), withdraw the mainshaft (third motion shaft) and its bearings out through the rear of the main transmission casing by holding it up by the rear end of mainshaft (third motion shaft)and tapping the main transmission casing gently downwards with a soft-faced mallet.

Recover the needle-roller bearings (BMC Part # 22H 523) of the mainshaft (third motion shaft) from inside of the input shaft (first motion shaft) assembly as well as the synchronizer hub for both third and fourth gears, along with its baulk rings (synchro rings). Inspect the needle-roller bearings of the mainshaft (third motion shaft) and the nose of the mainshaft (third motion shaft) for pitting and / or wear. If any is found, then the mainshaft (third motion shaft) assembly will have to be dismantled and subsequently be re-tipped. It should be noted that because the needle roller bearings of the first motion shaft (input shaft) run on the nose of the mainshaft (third motion shaft), any wear of the needle bearings or of the input shaft bearing will cause high frequency radial flexion of the induction hardened mainshaft (third motion shaft), eventually resulting in it snapping off at the ring groove. Consequently, these should be carefully inspected before making any decision concerning reusing them.

Now, remove the laygear and its needle-roller bearings from the main transmission casing. Pay careful attention as you inspect both the layshaft (second motion shaft) (BMC Part # 22B 280) and its roller bearings (BMC Part # 22H 523). If either have become pitted or worn, then you will need to replace them.

Using a pair of bocks of soft wood in order to protect the mainshaft from damage, grip the mainshaft (third motion shaft) rear distance tube (BMC Part # 22B 425) in a vice and bend back the locking tab in the slots of the front retaining nut. In order to disassemble the mainshaft (third motion shaft), release the lockwasher behind the front retaining nut, and then unscrew the front retaining nut. Carefully noting their order and orientation, slide off the gears, mainshaft (third motion shaft), sleeve, synchronizer hubs, baulk rings (synchro rings), distance piece, and the thrust washers from the mainshaft (third motion shaft), keeping them all in their proper original order. The first gear, the reverse gear, and the main bearing are all press-fitted onto the rear of the mainshaft (third motion shaft), and should not be removed unless one of them is in need of replacement. Clean and inspect all of the components of the mainshaft (third motion shaft) assembly for wear, especially if any of the first, second, or third gear endplay (endfloat) clearances (.005" to .008" / .127mm to .2032mm) were outside of the specified tolerances. Check to be sure that the oil restrictor located at the end of the mainshaft (third motion shaft) is clean and unobstructed. Replace any damaged or worn gears and / or thrust washers. Examine the bores of the bushings inside of the gears for wear, as well as the areas of the mainshaft (third motion shaft) upon which they rotate. Any sign of scoring and / or wear indicates the need for replacement. (First Gear Bushing, BMC Part # 22H 281; Second Gear Bushing, BMC Part # 22H 274; Third Gear Bushing, BMC Part # 22H 281).

Because the gear bushings are the same size on both the first and the third gears, these gears can be accidentally interchanged on the mainshaft (third motion shaft). If this is done, their teeth will not properly mesh with those of the laygear. You will have to pull the mainshaft (third motion shaft) out, heat up the bearing, pull everything apart, put it back in again. Discovering that you have reversed the order of these gears is a great lesson in humility.

Slide the First / Second synchronizer hub back and forth. If resistance to this movement is weak, then both the mainshaft (third motion shaft) and the hub will have to be disassembled and its springs replaced. The synchromesh hub springs should have a free length (uncompressed length) of .72" (18.288mm) and a compressed length of .385" (9.779mm). The same holds true for a loose Third / Fourth synchronizer hub. Be careful whenever you are disassembling a synchronizer hub, otherwise its springs and balls will fly across the garage with sufficient velocity to disappear through a space / time warp into the parallel dimension of lost parts (Ball, BMC Part # BLS 109; Spring, BMC Part # 22H 827). A good bit of preventive wisdom is to place the synchronizer hub inside of a large clear plastic bag in order to disassemble it so that the inevitable projectiles are trapped. Next, closely inspect the cogs of the synchronizer hubs and their baulk rings (synchro rings). If they are badly rounded or worn, then these parts will have to be replaced (First / Second gear baulk ring (synchro ring), BMC Part # 88G 397; First / Second Synchronizer hub, BMC Part # 22H 1167; Third / Fourth gear baulk ring (synchro ring), BMC Part # 22H 1028; Third / Fourth Synchronizer hub, BMC Part # 22H 1168). Note that these baulk rings (synchro rings) are not interchangeable. A First / Second speed baulk ring (synchro ring) is identifiable by means of a drill point on the reverse side of one of the lugs and / or by means of the fillets at the inside base of the lugs. Inspect the main bearing for roughness. This is a double-row bearing (BMC Part # 13H 7268) that is over-engineered and rarely fails. Examine the distance tube. Should this prove to be loose, the lock tab can be released and the nut retightened before resecuring the lock tab (BMC Part # 22H 773). Inspect the teeth of the laygear for signs of excessive wear and / or chipping. Also, closely examine the caged needle-roller bearings and the bore of the laygear in which they run. If you find any defects, replace the laygear.

In order to reassemble the transmission, you must first clean all of the parts thoroughly. The main transmission casing should be cleaned only with petrochemical solvents. Do not use any caustic cleaner (such as oven cleaner) as such caustic chemicals will corrode the aluminum alloy. Lightly oil all of the parts.

Including the thrust washers, install the front cover using a new gasket. Next, measure the gap between the cover and the main transmission casing, subtracting .0012" (03048mm) from this figure, and noting down the result. Refit the thrust washers to within +/- .001" (+/- .0254mm) of the thickness of the result, and then refit the front cover along with the new gasket. Unfortunately, there are a lot of poor-quality Asian-made bearings and thrust washers on the market right now. If you tear down the transmission and find that yours are faulty, contact The Roadster Factory as they carry thrust washers made to the original specifications by the British Motor Industry Heritage Trust, as well as Timken bearings (the best). They have a website at http://www.the-roadster-factory.com .

Place the needle-roller bearings that support the forward end of the mainshaft (third motion shaft) into the recessed bore of the input shaft (first motion shaft), holding them in place with reassembly lubricant or white petroleum jelly. Insert the input shaft (first motion shaft) and slide the roller bearing onto it, and then gently tap the bearing into place until it seats. Next, place the needle-roller bearings inside of the laygear, holding them in place with reassembly lubricant or white petroleum jelly, and then place the resulting laygear assembly into the bottom of the main transmission casing in a position where it will be safely clear when you install the mainshaft (third motion shaft).

Installing the balls and their springs into the synchronizer hub mechanism can be easily accomplished with a simple homemade tool. Get a section of PVC pipe tube that has the same Internal Diameter as that of the Outside Diameter (O.D.) of the slider hub. Drill a hole slightly larger than the size of the ball bearing into the PVC tube about half way down its length. When you have the hole in the tube aligned, insert a ball bearing and its spring, then rotate the tube to the next position, and then insert another pair, and so on until you have all of them installed. Slide on the slider while pushing the PVC tube off at the same time.

Prior to installing the mainshaft (third motion shaft) assembly, you will first need to prepare the output bearing assembly. The large output bearing is pressed into its bearing housing, and then afterwards the entire assembly is inserted into the rear of the main transmission casing. When the output bearing is pressed into place in the bearing housing, it must be pressed in past the face of the housing until it seats against the bottom of its recess in the bearing housing, so do not try to install it on a flat surface. Instead, use a raised support and press the bearing fully into the bearing housing. If you press the bearing into place in its bearing housing on a flat surface, then it will not be seated inside of the recess of its bearing housing. As a result, you will find that things will not align properly because of the extra space that you did not take up.

Now, stand the main transmission casing on end upon some milk crates so that there will be sufficient clearance. Insert the mainshaft (third motion shaft) through the rear of the main transmission casing. Next, you will need to assemble the baulk rings (synchro rings) into the First / Second gear (rear) synchronizer hub. The Third / Fourth gear (front) synchronizer hub can be easily identified by the groove cut into its outer circumference, while the First / Second gear (rear) synchronizer hub can be readily identified by the absence of such a groove on its outer circumference.

Insert the second gear baulk ring (synchro ring) into the synchronizer hub for first / second gear with its cogs pointing inward, and then insert first gear baulk ring (synchro ring) into the synchronizer hub for first / second gear with its cogs pointing inward. Now, place the First / Second (rear) synchronizer hub assembly onto the mainshaft (third motion shaft). Next, install the second gear thrust washer with its grooves facing toward the First / Second gear synchronizer hub assembly. Next, slide the second gear assembly onto the mainshaft (third motion shaft) with its synchro lugs facing toward the rear of the transmission main casing.

Hold the third gear so that its synchro lugs will face toward the rear of the main transmission casing, and then insert the distance piece into the third gear with its lugs facing toward the second gear. Next, fit the interlocking notches of the thrust washer onto the lugs of the distance piece. You will need to get this right the first time around. This is something that you can easily get wrong, which of course will ensure that it will not be the last time that you will get to disassemble the transmission. This is an easy step to miss and, if omitted, will result in misalignment of the gearsets.

Now, slide the third gear assembly onto the mainshaft (third motion shaft) with the thrust washer facing toward the rear of the main transmission casing. Place the third gear baulk ring (synchro ring) into the synchronizer hub for Third / Fourth gears with its cogs pointing inward , and then place the fourth gear baulk ring (synchro ring) into the synchronizer hub for third / fourth gears with its cogs pointing inward. Now, place the

Third / Fourth (front) synchronizer hub assembly onto the mainshaft (third motion shaft). Next, carefully align the dowel in the main transmission casing with the cutout in the bearing housing. While guiding the mainshaft (third motion shaft) into the needle-roller bearing inside of the input shaft (first motion shaft) at the forward end of the transmission, gently press or tap the output bearing housing into its recess in the rear of the main transmission casing. Check to be sure that both the output bearing and the input bearing are both fully seated inside of their machined recesses and that there is no excessive endplay (endfloat) on the Third / Fourth synchronizer hub (.005" to .008" / .127mm to .2032mm). Finally, install a new lockwasher (BMC Part # 22H 773) and the ring nut (BMC Part # 22H 772) onto the forward end of the mainshaft, and then bend and stake the new lockwasher into its recesses in the ring nut.

Be aware that the layshaft thrust washers employed in the four-synchro transmissions that were used on the later 18V engines are of a different design than those used in earlier four-synchro transmissions. The earlier transmission casings (BMC Part # 22H 1044) used a simple, circular thrust washer while the later transmission casings (BMC Part # 22H 1463) used a forked type. These forked thrust washers were for use in the transmissions that were joined with 18V 846F, 18V 847F, 18V 797AE, 18V 798AE, 18V 801AE, and 18V 802AE engines. The earlier part number is presumably for all transmissions prior to those. The 18V 797, 18V 847, and 18V 802 engines were for use in Overdrive-equipped versions of the 18V 796, 18V 846 and 18V 801 engined non-Overdrive equipped models. The forked type layshaft thrust washer was an improvement in that its forks fit over small bosses on the inside of the casing, thus preventing the thrust washers from rotating along with the laygear, which would result in a groove being worn into the layshaft as well as enlarge the hole in the thrust washer. The introduction of this redesign of the casing incidentally coincided with a redesign of the mounting for the electric starter mechanism on the bellhousing. While the earlier version is rounded, the later version is flat and much stronger. In all cases the thrust washer at the front of the laygear was available in one size only, while the thrust washer used at the rear of the laygear was available in four different sizes for the earlier transmissions and five different sizes for the later transmissions in order to adjust endplay (endfloat).

With the largest gear of the laygear facing toward the front of the main transmission casing, align the laygear with the mounting holes for the layshaft (second motion shaft) (either a long piece of wire or a wooden dowel rod is a great tool for this). Insert the layshaft (second motion shaft), plain end first, from the front of the main transmission casing while ensuring that the each of the thrust washers is in place at its respective end of the laygear and that the spring ring is forward of the front thrust washer. Align the cutout in the front end with the recess in the front plate, and then gently drift the layshaft (second motion shaft) into place. Recheck the endplay (endfloat) of the laygear and adjust the clearance as necessary by using a different thickness rear thrust washer.

Refit the clutch withdrawal lever, along with its pivot bolt, bushing, and operating rod. When you do this, make sure that its pivot bushing is not ovaled; otherwise you will probably have problems getting the proper amount of clutch free play. People who forget to do this usually go crazy when they try to rectify the resulting problem by repeatedly bleeding the clutch slave cylinder, all to no avail!

Be aware that the reverse gear can be installed backwards on mainshaft (third motion shaft). It is one of the completely reversible parts that creates a disaster when installed backwards. You need to pay attention to the gear teeth and put their rounded off edges facing in the right direction. Catch this one early and avoid having to pull everything apart again! Insert the reverse gear inside of the main transmission casing, and then, using the slot in the rear of the shaft in order to align the shaft with its locking bolt, insert the shaft through the rear of the transmission. Align the locking hole (the slot in the end can be of assistance in turning the shaft), and then install the locking bolt and its lockwasher (tab washer). Be sure to use a new lockwasher (tab washer) (BMC Part # 1B 3363) under the bolt head, and then secure the shaft.

Place the selectors (saddles) onto their synchronizer hubs and onto the reverse gear, and then insert the selector fork rods through the rear of the main transmission casing. Insert the selector shafts through their selectors, then through their selector forks (saddles), and then into the front of the main case. The position of the lockscrew holes in the selector shafts will help you to identify each selector shaft. Align the threaded holes in the selector forks (saddles) and the selectors with their holes in the selector shafts, and then screw in the lock bolts with a locking nut and star washer on them, but do not tighten them just yet.

The shaft of the gear change lever (gear shift lever) on the non-Overdrive-equipped three-synchro transmissions (BMC Part # 22H 152) was chrome plated and had two bends so that the longer middle section of the gear change lever (gear shift lever) shaft was inclined towards the rear, while the top part with the shifter knob was vertical (BMC Part # 1G 3706). The shaft of the gear change lever (gear shift lever) on the Overdrive-equipped three-synchro transmissions had only one bend, located near the bottom of the shaft so that the top section of the shaft (BMC Part # 22H 251) inclined the shifter knob (BMC Part # 1G 3706) towards the rear. These three-synchro gear change levers (gearshift levers) are not interchangeable with any of the gear change levers (gear shift levers) used on the four-synchro transmissions, nor are their 5/16”-24 UNF threaded shifter knobs.

Should you decide that you would prefer to use the later cranked gear change lever (gear shift lever) of the 1977-1980 models with the Overdrive switch that is integral to its shift knob, be aware that with minor modification to the gear change lever (gear shift lever) extension housing, the gear change lever can become interchangeable with those of the earlier four-synchro transmissions. The need for modification is due to the fact that both the gear change lever (gear shift lever) extension housings and the ball ends of the gear change levers (gear shift levers) are different. The extension housing (BMC Part # 22B 522) of the earlier 1968 through 1976 transmissions uses two pivot bolts in order to align the gear change lever (gear shift lever), while that of the later 1977 through 1980 transmissions (BMC Part # 22H 1457) uses only a single pivot bolt. This being the case, the extension housing of the earlier four-synchro transmission will need to have one of the pivot bolts removed in order to mount the cranked gear change lever (gear shift lever) of the later transmissions. The alternative is to interchange the appropriate gear change lever (gear shift lever) extension housing. If you are considering mounting the later shift knob with its integral Overdrive control switch onto an earlier gear change lever (gear shift lever), be aware that the earlier gear change lever (gear shift lever) has a 3/8”-16 UNC threaded shaft while the later cranked gear change lever (gear shift lever) has a 7/16”-20 UNF threaded shaft; thus the shift knobs are not interchangeable.

In order to install the later cranked gear change lever (gear shift lever) of the 1977-1980 models with the Overdrive switch that is integral to its shift knob, pop off the top of the knob (there is a screwdriver slot on the front side where you can't normally see it. There are two wires with round bullet type connectors that will pull off the switch. The switch is held to the top cap with two little self tapping screws. The switch comes apart fairly easily, with a brass roller making contact inside which is probably covered in old hard grease. Clean this old grease away with CRC QD Electronic Cleaner, and then regrease it with dielectric grease. Since the lever moves every gear change, the wire can chafe at the base of the lever and break or short. Being careful to install the anti-rattle plunger located at the 5 o'clock position on the rear of the transmission, gently lower the shift lever in place with the transmission in neutral. Slowly install the three 7/16" bolts around the shift lever, being careful to include the spring washers that are underneath the bolts. These serve a very important function. Slide the shift lever retaining plate toward the shifter and attach it. Finally, install the four screws that hold the chrome bezel in place. Next, connect the two wires that run down the shift lever from the Overdrive switch. Now, If you ever need to replace the wire, then you will need to remove the knob by undoing the chrome lock nut just below the knob (spanner flats), and then undoing the ring nut inside the knob with a screwdriver. At that point you can see how the cable gets up the side of the lever and into the knob. It is well worth adding an in-line fuse to the positive (+) feed from the wiring loom where it goes down towards the gearbox from the engine bay.

Next, be sure to check the play of the remote control shaft inside of its bores in the body of the extension housing. Wiggle the remote control shaft up and down as well as side-to-side. There should be almost no play. If there is any, the bores should be sleeved. This will restore the feel of a nice, precision gear change.

Now, check to be sure that the anti-rattle damper mechanism inside of the extension housing for the gear change lever (gear shift lever) is clean and well-lubricated. This mechanism consists of a plunger (BMC Part # 22A 84) with a spring (BMC Part # AEG 3123) that sits at about the Four o'clock position in the gear change lever (gear shift lever) extension housing where the large, rounded part of the gear change lever (gear shift lever) goes.

Using a new gasket, replace the interlock plate and the gear change lever (gear shift lever) extension housing, ensuring that the plastic bushing (BMC Part # 22B 295) is on the end of the gear change lever (gear shift lever) and that it fits properly into the remote control shaft.

Note that if the gear selectors are adjusted when the gear change lever (gear shift lever) is not in its neutral position, then you will not attain the correct amount of travel of the synchronizer hub, which will result in the transmission popping out of gear. Insert the springs, the detent plungers, and the plugs with their fiber washers under their plugs into the selectors. Tighten each of the lock bolts of the selectors into their holes in the selector shaft, and then tighten the locking nut, and then safety wire them. If a lock bolt is insufficiently tightened, then the selector will vibrate loose, hence the need for safety wire. With the transmission in neutral, wiggle the synchronizer hubs in order to make sure that the selector forks (saddles) are properly centered on their synchronizer hubs and are not putting the synchronizer hubs under tension. Next, tighten the lock bolts and safety wire them. Next, replace the three selector fork rod plungers, their springs, caps (fiber washer under the cap head) and their retaining bolts.

Note that the mainshaft (third motion shaft) passes through a tight seal on the rear half of the transmission case. When you reinstall the rear case onto a non-Overdrive transmission, the gears will be in a bind until you torque up the driveshaft (propeller shaft) flange on the output shaft. If you start checking for proper shifting before the flange is properly torqued in place, you will swear that you have reassembled the transmission incorrectly.

If you are converting a four-speed-only transmission to a four-speed-with-Overdrive, remove the blanking plug from the gearlever extension housing and insert the Overdrive inhibitor switch operating shaft and its spring. Align the flat nearest the head of the shaft with the locating pinhole and drive the locating pin in from the top. Screw the Overdrive inhibitor switch into the extension in place of the blanking plug. Its position is critical as it must only be in far enough to make contact with the selector rod. This spacing is achieved by the use of fiber washers and the correct number must be determined by trial and error. Problems can arise because the Overdrive inhibitor switch has a habit of loosening itself. Coarse threads are easily moved, and with time, the fiber washers can become brittle and lose their grip. The terminals are also prone to corrosion and becoming loose. Finally, install and tighten the six machine bolts that secure the gearlever extension housing to the transmission. The flange nut should be torqued to 55-60 Ft-lb. Inside of the engine bay, there are three harnesses joining together near the fusebox along the passenger side (starboard) inner wing in the mass of connectors. The biggest is the main wiring loom (harness), the next biggest is the rear wiring loom (harness) going down and to the back of the car, and the smallest is the transmission wiring loom (harness) going down under the tunnel closer to the middle of the car. This contains the wires for the reversing light as well as the wire for the Overdrive unit. All cars with the Overdrive manual switch in the knob had the Overdrive manual switch and the yellow wire, regardless of whether or not they were equipped with the Overdrive. It is not uncommon for owners to use that yellow wire for the fuel pump instead of the white wire, thus enabling them to use the manual Overdrive switch as a form of immobilizer circuit for the fuel pump. While the manual Overdrive switch was on the dashboard and column the wire from the main wiring loom (harness) was yellow, it became white when the Overdrive switch moved to the gear change lever (gear shift lever) in June of 1976. You will find a fistful of bullet connectors. Look for an unused bullet connector hiding somewhere in that wiring loom. That is where the Overdrive connection is made. Since the wires are prone to breaking from flexure at the point where they exit the gear change lever (gear shift lever), it would be a good idea to install an inline fuse along the unprotected Overdrive circuit.

If you are working on a four-speed-only transmission, calculate the needed thickness of the shims that will be placed between the front bearing of the third motion shaft and the rear extension casing by measuring the depth (a) from the front bearing and its housing, then add the thickness (c) of the joint washer. Next, measure the depth (b) from the joint face to the bearing register face of the front extension. Install the shims between the face of the front bearing and the extension in order to bring the dimension .000” to .001” (.000mm to .0254mm) less than the dimension (a). Now you will need to calculate the thickness of the shims required between the distance tube and the rear bearing of the third motion shaft. Measure the depth (d) between the rear face of the distance tube. Next, measure the depth (e) of the rear bearing register from the rear face of the extension. Install shim washers in between the distance tube and the bearing in order to bring the dimension measured in (a) from .000” to .001” (.000mm to .0254mm) less than the dimension measured in (b). Finally, press the new bearing into the extension, and then replace the circlip. Taking care to align the marks that you scribed on the rear extension and the main transmission casing, install the rear extension. The flange nut should be torqued to 150 Ft-lb.

Do not be surprised if there appears to be some binding when you engage a gear and attempt to turn the input shaft (first motion shaft) by hand. What actually happens is the synchronizer mechanisms actually hang up until everything is properly centered. However, the whole thing will tend to flop around until either the flange of the Overdrive adapter housing or the flange of the rear extension is torqued in place. Wait until it is torqued in place before you replace the inspection cover. At that point, any problems can be spotted by observing through the opening. Verify that each selector gives the proper throw necessary to engage and release each synchronizer hub. If there is a problem, usually it is a synchronizer hub that is hanging from insufficient throw, or the fact that the whole layshaft (second motion shaft) is not quite centered up yet as a result of loose rear extension flange bolts, or possibly an input shaft (first motion shaft) problem.

After everything measures as being within tolerance, you should take the time to disassemble the transmission a second time, and then reassemble it with a good prelude in order to be sure that there is no dry metal-to-metal contact anywhere. Upon final assembly, it is also a good idea to refill it somewhat through the inspection cover, and then smear high quality camshaft break-in lubricant over all of the gears, the selector forks (saddles) and the selectors. Camshaft break-in lubricant is oil-soluble, and thus will present no problems in the future. You can just use gear oil too, although the success of this method of lubrication is dependent upon how long it will be before you run the car. Just strictly adhere to the principle of “no dry metal-to-metal contact on any moving parts”, and you should experience no problems.

Refit the side cover with a new gasket, but do not apply any sealant as the mechanisms of the transmission may need to be inspected at some future date. Be sure to install new spring washers under the 5/16” UNC machine bolts so that the torque readings will be consistent, thus discouraging warpage of the cover and resultant leakage. Refit the oil drain plug, and then install a new gasket onto the rear of the transmission. Slide the Overdrive adapter housing over the mainshaft (third motion shaft), and then secure it with its nuts and machine washers. Finally, install the selector interlock plate.

Place a new gasket on the rear of the Overdrive adapter housing. Refit the Overdrive pump cam which is located by the ball bearing held in the hole in the mainshaft (third motion shaft), then fit the circlip onto the mainshaft (third motion shaft). Rotate the mainshaft (third motion shaft) so that the cam lobe points at the top of the transmission, then carefully slide the Overdrive unit onto the mainshaft (third motion shaft). If it fails to reach the studs, then the internal splines are not properly aligned. In this case, remove the Overdrive unit and, while using two wrenches to lift the operating bars, use a long screwdriver to rotate the sliding annular clutch (conical clutch) (conical clutch) until the splines are aligned. If they fail to mate within ¼” to 3/8” (6.35mm to 9.525mm) of movement, then the pump-operating roller is catching on the cam. Should this prove to be the case, use a long, thin screwdriver to depress the pump plunger. Refit the retaining nuts and their machine washers, noting that some must be fitted with the Overdrive unit slightly raised from the adapter housing.

Install the drain plug and, making sure that the transmission is level, refill the transmission. It is actually easier to do this prior to installing the transmission back into the car. If your transmission is equipped with an Overdrive unit, do not use a detergent oil as it will foam and cause chatter of the sliding annular clutch (conical clutch) , thus reducing the life of the lining of the sliding annular clutch (conical clutch) and interfering with the function of the Overdrive unit. Note that both the transmission and the Overdrive unit share engine oil of the same grade as for the engine, e.g., 20W/50. The three-synchro transmission takes 5.6 US pints (4.5 Imperial pints / 2.56 liters), and the three-synchro transmission plus Overdrive unit takes 6 US pints (5.33 Imperial pints / 3.36 liters. The 4-synchro transmission takes 7 US pints (5.25 Imperial pints / 3 liters), and the 4-synchro transmission with Overdrive unit takes 7 US pints (6 Imperial pints / 3.4 liters). Be aware that these quantities are for clean and dry transmissions. A drain and a refill can be expected to take a little less than that, as some of the old oil is bound to be left behind.

The Overdrive Units

Be advised that two different, non-interchangeable basic types of Laycock de Normanville Overdrive units were used on the MGB. Both were a basic two-speed planetary transmission that gave a choice of direct drive from the transmission when disengaged (1.00:1), or a reduced ratio when engaged. The first was the D-type Overdrive unit that produced a reduction ratio of .802:1. This Overdrive unit was used on the three-synchro transmissions and had an external linkage for the solenoid. It can be readily identified by its identification numbers 25/3308 (sometimes 63308). The more rugged and durable LH-type Overdrive unit was used on the four-synchro transmissions and came in two versions. The LH-type Black Label Overdrive unit used from the 1969 through the 1974 model years with an identification number of 22/61972 which had a 21 tooth white speedometer pinion drive gear (BMC Part # 37H 3464) and pinion (BMC Part # 37H 3463) appropriate for the 1,280 tpm (turns per minute) speedometer. The later LH-type Blue Label Overdrive unit used from the 1975 through the 1980 model years with an identification number of 22/62005 with a 20 tooth red speedometer pinion drive gear (BMC Part # 37 H 8844) and pinion (BMC Part # 37 H 8845) appropriate for the 1,000 tpm speedometer. Fortunately for those adapting an Overdrive to a different series transmission, these speedometer pinion drive and pinion gearsets are interchangeable. When engaged, both versions of the LH-type Overdrive unit produce a reduction ratio of 0.82:1, but on 1977 and later models, due to the use of a switch in the shift mechanism inside of the gearlever extension housing, the Overdrive unit can be engaged only with the transmission in fourth gear. Aside from their two different speedometer drive ratios, the LH-type Overdrive units are interchangeable. However, their white and red speedometer pinion drive gears are not easily interchangeable. You have to perform a complete disassembly of the Overdrive unit in order to replace the driving gear on the output shaft, as well as the pinion gear. It should be noted that there should be a copper washer fitted to ensure that the drive shaft of the speedometer angle drive pinion does not bottom out in the speedometer drive pinion, causing premature wear and failure.

Be aware that the later LH-type Blue Label Overdrive unit has a weaker thrust washer for the sun gear. Instead of combining the input shaft (first motion shaft) bushing and the thrust washer into one piece as in the Black Label Overdrive units, these later Overdrive units use a two-piece assembly consisting of a spacer and a thin phosphor-bronze washer with oil grooves in it. These washers tend to fracture along their oil grooves. This thrust washer cannot be replaced. The only method of repair is to have the casing modified in order to accept the earlier and sturdier one-piece thrust bushing of the LH-type Black Label version.

Many people are under the impression that installing an Overdrive unit is simply a matter of bolting it onto the rear of a non-Overdrive transmission. Sadly, this is not so. The transmission will require a longer mainshaft (third motion shaft), but you can still use all of your old gears, synchronizer hubs, baulk rings (synchro rings), thrust washers, and the distance piece. The engineers deliberately designed it this way in order to create the strongest possible couple at the junction of the transmission and the Overdrive unit, and to keep the number of different parts involved in the assembly of the two types of the transmission as few as possible.

For those whose cars lack the much-coveted Overdrive unit, Chris Betson of Octarine Services in Essex, England supplies a complete Overdrive conversion kit for the four-synchro transmission. The all-important longer replacement mainshaft (third motion shaft) is supplied, along with the needed mainshaft (third motion shaft) bearing and its carrier, plus both the required rear extension housing as well as a fully reconditioned Overdrive unit. The Overdrive unit itself is fully reconditioned by Overdrive Repair Services in Sheffield, England. This is the company that is the successor to the Original Equipment supplier Laycock de Normanville, having purchased all of the original tooling and continuing production with some of the original staff. For all practical purposes, the Overdrive units are basically as good as new. The casing is first bead blasted, and then remachined, the thrust washer on the later Blue Label Overdrive units being replaced by the lubrication bushing of the earlier Black Label Overdrive units. The annulus is also remachined, and the sliding clutch is relined with new friction material that is fully riveted in place. All of the bearings are replaced with new bearings, and all other parts are carefully inspected, tested, and replaced as necessary. All new filters, seals and gaskets are fitted, and the Overdrive unit is tested after careful reassembly. If it passes inspection, it is then given a circular stamp on the bottom with "ORS" and the date that it passed inspection. Just for the sake of completeness, it is even supplied with a new wiring loom so that you will not have any electrical problems. Both the early LH-type Black Label Overdrive unit with the 1,280 tpm speedometer drive unit and the late LH-type Blue Label Overdrive unit with the 1,000 tpm speedometer drive unit are available.

For those whose intention it is to develop an extremely high performance version of the B Series engine, yet are concerned about the ability of the Overdrive unit to handle the increased power output, it should be noted that the LH-type Overdrive unit was also available in a specially-modified version for competition use. This version used the internal clutch springs from the LH-type Overdrive unit that was used in the more powerful MGC in order to enable a boost of the internal pressure to 510 PSI to 530 PSI when engaged, as opposed to the 400 PSI to 420 PSI of the common LH-type Overdrive unit used in the mass production MGBs, as well as a different freewheel cage and uprated friction material on the annulus. It also used the output flange of the MGC in order to take advantage of its larger machine bolts for the front U-Joint (Universal Joint) of the driveshaft.

How An Overdrive Works

The Overdrive is operated by an electric solenoid that is controlled by a switch, usually mounted on the steering column or fascia panel. An Overdrive inhibitor switch is invariably fitted in the electrical circuit in order to prevent engagement of Overdrive in reverse and some or all of the lower speed gears.

The overdrive gears are epicyclic (planetary) and consist of a sunwheel, three planet gears, and an internally toothed ring gear or annulus. All of the gears are constantly in mesh. The planet carrier is attached to the input shaft. Carried on a splined extension of the sunwheel, a double sided sliding annular clutch (conical clutch) engages with either a stationary brake ring or with the outside surface of the annulus. A caged unidirectional clutch assembly (freewheel) connects the input and output shafts.

Oil is drawn from the sump by the reciprocating pump through the suction filter and is thus delivered to the pressure filter. It then passes through to the relief valve and acts upon the closed solenoid valve. Oil passes through the relief valve spill port and is conducted into the main drive shaft in order to lubricate the running gear. Excessive pressure build up in the lubrication passages is prevented by the lube relief valve which operates at a predetermined pressure.

Since oil supply to the dashpot is withheld by the solenoid valve, the dashpot piston is held down by its spring(s). Movement of the relief valve piston is resisted by the spring, the strength of which determines the residual pressure within the system when the unit is in direct drive. It is necessary to maintain this residual pressure in order to initiate action of the system when engagement is signaled, as well as for lubrication.

When the solenoid is energized, the valve opens and oil at residual pressure is allowed to enter the passage which leads to the dashpot. An increase in system pressure results from the "out-of-balance" between the dashpot piston and the relief valve piston due to their differing areas. This causes the dashpot assembly to compress the residual spring. The dashpot then continues to move, compressing its spring until it contacts the dashpot cup.

The reaction of the increasing load of the relief valve spring being applied to the relief valve piston progressively increases the system pressure. The dashpot piston continues to its stop by which time the relief valve spring has been compressed to its working length, thus giving full system pressure.

At this point the sliding annular clutch (conical clutch) is moved forward by hydraulic pressure in the operating cylinders so that the outer friction lining of the clutch comes into contact with the stationary brake ring. Since the sliding annular clutch (conical clutch) is attached to the sunwheel, both come to a halt and the sunwheel then becomes the reaction member for the planetary train. Since the planet carrier is splined to the input shaft and driven by it, the planet wheels orbit the stationary sunwheel. In so doing, they rotate the annulus and the output shaft at a speed greater than that of the carrier and input shaft. The caged unidirectional clutch enables the output member to rotate faster than the input member. At this point, the car is said to be in Overdrive

The forward drive is transmitted directly through the unidirectional sliding annular clutch (conical clutch) . The sliding annular clutch (conical clutch) is held rearwards by spring pressure exerted through a thrust ring and ball race, loading its inner friction lining in contact with the Outside Diameter (O.D.) of the annulus on the output shaft. This loading is further increased by the reverse thrust of the helical sunwheel. Thus the gear train is locked and overrun or reverse torque is taken by the sliding annular clutch (conical clutch) , without which, the caged unidirectional clutch would give a freewheeling condition.

When the solenoid is de-energized the valve is returned by a spring, thus allowing oil to exhaust from the dashpot. The relief valve spring is able to gradually relax towards its direct drive position. The system pressure will progressively decrease, eventually allowing the sliding annular clutch (conical clutch) to separate from the stationary brake ring and move gently into contact with the annulus. The supplementary dashpot spring provides additional load upon the piston and allows relative rotation between the sliding annular clutch (conical clutch) and the stationary brake ring as the engine speed increases and the caged unidirectional clutch takes up direct drive.

Smooth overrun disengagements are obtained by a progressive increase of the load on the inner clutch linings, compared with that provided by the free action of the clutch springs, by varying the hydraulic pressure which offsets the clutch spring load and thus determines the net loading at the clutch face. This is accomplished by controlling the reduction of pressure controlled by the action of the dashpot piston, exhausting oil through the discharge orifice. When the contents of the dashpot have been fully discharged, the system pressure has then fallen to its original residual value.

The D-Type Overdrive Vacuum Switch

The Overdrive vacuum switch on the MKI MGB is not actually required for the use of its D-type Overdrive unit. The vacuum switch was an additional, afterthought item (it was not on all the cars with a D-type Overdrive unit, just some years of the MKI models). According to the Leyland Workshop Manual, the overdrive vacuum switch was to prevent Overdrive being disengaged under conditions of "high manifold vacuum", i.e., to prevent high reverse torque from damaging the Overdrive. Subsequent discussion with the designer of the transmission system for the MGB confirmed that the vacuum switch was indeed designed to prevent disengagement unless the car was accelerating, but to give a smoother disengagement rather than to prevent damage. However, they found that the vacuum switches were unreliable and so deleted them, opting for 'driver education' instead.

The later LH-type Overdrive does not have this vacuum switch and relay, presumably the designers feel it is strong enough to take disengagement under conditions of high manifold vacuum without damage. The LH-type Overdrive accomplishes the same thing by making the operating oil mix with the output of the pump enroute from the piston chambers to the sump, thus damping the sliding annular clutch (conical clutch) movement and providing a smoother change. It was wired in line with the Overdrive circuit and was used in order to keep the Overdrive unit engaged under conditions of high intake manifold vacuum even when the driver had turned the manually switched the Overdrive unit off, thus protecting the Overdrive from sudden torque reversals when switching out on the over run. Without it, the transmission would downshift out of Overdrive when the throttle was opened hard. In essence, this made the Overdrive unit shift not unlike the passing gear downshift of an automatic transmission when used for more power on the highway. Unfortunately, this creates high reverse torque that could damage the D-type Overdrive unit. Selecting a non-Overdrive gear would turn it off instantly, but as this usually meant the clutch had also been operated, there was no reverse torque to be concerned about. The later LH-type Overdrive unit is stronger and does not need this feature; hence the vacuum switch was discontinued with the advent of the MKII model. Most of these switches are no longer functional, and they are no longer manufactured due to the fact that there is too little demand for them and they are too complicated to make at a reasonable price in small-scale production. However, it is obviously a worthwhile item and should be made functional.

The vacuum switch for the D-type Overdrive unit is mounted on the driver's side of the firewall (bulkhead) of the Left Hand Drive MGBs. There are two holes in the firewall (bulkhead) next to the heater box near the throttle cable aperture on the vertical wall. There are also two holes above this for mounting the relay. The vacuum switch has a yellow wire one side and a yellow / red wire on the other. Both of these go to the relay, the yellow wire goes to the relay W1 terminal (together with one from the manual switch) and the yellow / red wire goes to the C1 terminal, along with one from the transmission switch. The relay C2 terminal has a white wire from the fusebox and the W2 terminal has a ground (earth) wire.

In the arrangement of connectors as shown in the Leyland Workshop Manual there is a spare socket in the 6-way connector for the white wires, so strictly speaking the 4-way bullet connector is not required. Note that the wire between the two may go into the harness and straight back out again.

So, what happens from an electrical standpoint? When the manual switch is closed the relay is operated, and the relay contact energizes the solenoid via the transmission switch if closed, and Overdrive is then engaged. At this point the condition of the vacuum switch - open or closed - is immaterial.

Assume now that the engine is doing high engine speeds with the Overdrive engaged, but the throttle is closed. This will create a high vacuum in the intake manifold which will close the vacuum switch. If the driver now opens the manual switch, the vacuum switch being closed will continue to maintain a 12 Volt supply to the relay winding from the relay contact, independently of the manual switch, so the relay remains operated, and thus the Overdrive remains engaged regardless of the fact that the manual switch has been turned "Off".

Imagine now that either the speed of the car has slowed so that the engine revs are closer to idle, or that the clutch is released so allowing the engine speed to fall to idle, or the throttle is opened again. In all cases, the vacuum in the intake manifold will reduce, allowing the vacuum switch to open. This causes the relay to release (the manual switch being already open) and its contact disconnects 12 Volts from the Overdrive solenoid, thus disengaging the Overdrive.

Of course, if the transmission should be taken out of an Overdrive gear, then the transmission switch will ensure that the Overdrive is disengaged instantly, regardless of the position of the manual or vacuum switches. 'Normal' gear changes, say from 3rd to 2nd, will usually allow the Overdrive to disengage safely and not encounter the mechanical stresses that the vacuum switch and relay are designed to avoid.

The vacuum switch on its own cannot operate the relay and so cause Overdrive to be engaged, the manual switch must be closed first.

Be aware that the manual switch will operate the relay, and the vacuum switch will keep the relay operated under conditions of high manifold vacuum even if the manual switch is turned “Off”, when the transmission is *not* in an Overdrive gear. All this means is that when an Overdrive gear is selected, the solenoid will be energized and Overdrive engaged as normal.

It should be noted that if you intend to fuse the Overdrive circuit to protect the wiring, both the white feeds from the fusebox to the manual switch and the relay must be fused.

The hose (flexible pipe) from the vacuum switch attaches to a nipple fitting on the intake manifold. If a fitting is not already present, you will need to add one. Do not attempt to simply install a T fitting into the vacuum line from the carburetor to the distributor as the vacuum *has to be from the intake manifold*, otherwise it would not be able do what it is supposed to do, i.e., keep Overdrive engaged on the overrun, as there is no vacuum in the carburetor / distributor line when the throttle is closed.

With the cap removed, you will discover an extremely simple switching mechanism. There is a black plastic disk that is about 9mm thick to which the two electrical connectors are attached 180^{o} apart. Between these two connectors runs an inset copper bridge that is attached to one terminal (rivet) and having potential to make contact with the other terminal through bent tension in the bridge. This switching disk is held in place with four screws that are easily removed. With the switching disk removed, you will find a rubber diaphragm that is apparently glued in place. It looks much like what you see when you remove the bottom part of a SU fuel pump. In the middle of this rubber diaphragm protrudes a collared metal nipple, 7mm in length. If you push on this nipple with your finger, you will see that its height above the metal base (about 10mm) is maintained by a spring underneath the diaphragm. This metal nipple passes through a hole in the center of the black switching disk. Around this hole is a recess that accommodated the nipple's collar. With the switching disk unmounted, the copper bridge provides continuity between the two terminals. On the bottom side of the switch housing, centered under the diaphragm, is a plugged tube. The plugging material is much like hardened plumber's putty and can be easily removed to expose a brass adjustment screw. Backing off on this screw counterclockwise (anticlockwise) lowers the vacuum threshold required to close the switch. Additionally, removing the sealant had no apparent effect on the integrity of the unit (i.e., no air leaks under vacuum). Except for perhaps a smashed housing, there is nothing in this switching unit that cannot be easily refurbished or repaired. Even the rubber diaphragm should be easy to fabricate and replace.

So, how does this switch work? The switching disk is 9mm thick, but the copper bridge is recessed and lays slightly less than 7mm from the bottom of the disk. When mounted, the metal diaphragm nipple passes up through the disk, pushing up on the copper bridge, thus breaking the contact between the two terminals. However, when under vacuum the metal nipple is withdrawn from inside the switching disk, allowing the copper bridge to fall and make contact between the two terminals to complete the electrical circuit. Thus, if there is no vacuum, the switch is off. When vacuum is applied, the switch is on. The vacuum switch de-energizes the Overdrive solenoid, which then disengages the Overdrive unit. By now, however, you have realized just how simple this switch really is. For the cleaning of the black contact disk, just place the whole thing in some diluted acetic acid (about 25%). Undiluted vinegar (5% to 18% acetic acid) will work as well, albeit a bit more slowly. The nice thing about acetic acid is it seems to attack the oxides while leaving the base metals relatively untouched. You can leave things in for days without causing any noticeable damage.

Electrical Troubleshooting Of The D-type Overdrive:

The earlier D-type unit has both a vacuum switch to prevent the driver from disengaging OD under certain conditions, such as high engine speed and no throttle, which could overstress the unit., and a relay, as well as the manual and transmission switches. The manual switch and the vacuum switch operate the relay, which in turn operates the solenoid. Obviously, you will have to check both circuits if the Overdrive isn't working. On the D-type Overdrive you *should* be able to hear both the relay and the solenoid clicking as you connect and disconnect power while the car is stationary, so you will have to differentiate between them. The D-type solenoid has a two-part winding and contact - a low resistance/high current (0.7 Ohms/17 Amps) 'pull in' winding, which when the solenoid has moved a certain distance then opens a contact which leaves just a high resistance/low current (6 Ohms/2Aamps) 'hold in' winding energized. There is an adjustment provided to make sure this contact opens when it should. If it does not, then the solenoid will almost certainly overheat and could possibly burn out. In order to check this, remove the cover plate by the solenoid (three bolts). With the ignition on, 4th gear selected, and the Overdrive switched on, the solenoid should move a lever. This lever has a hole near the solenoid spindle, and another hole in the casting behind it. With the solenoid correctly adjusted, a 3/16" bar should pass through the hole in the lever into the hole in the casting. If it does not, while holding the spindle by the two flats machined into its shank, adjust the self-locking nut on the end of the solenoid spindle so that it does.

In order to measure the current through the solenoid, as opposed to the relay, you will have to insert the Ammeter into either the single white wire where it adjoins the one relay contact or into the two yellow / red wires where they leave the other relay contact. Be aware that there is no fusing in this circuit, so anything shorting to ground (earth) may well damage the wiring looms.

Electrical Troubleshooting Of The LH-type Overdrive:

On the later LH-type Overdrive you probably will *not* be able to hear both the relay and the solenoid clicking as you connect and disconnect power while the car is stationary. When applying power to the solenoid, the moving part tries to moves towards the middle of the solenoid coil, pressing a ball-bearing against the end of a nozzle in order to block the flow of oil. This causes the pressure in the system to rise which engages the Overdrive. When power is disconnected with the car running, oil pressure causes the moving part to move very slightly away from the ball-bearing and nozzle. With the car stationary, there will be no oil pressure, and while you may hear a faint click the first time the solenoid is energized, upon being de-energized it will not move back because there is no oil pressure, and so when it is again energized it will not move at all and thus will not make a noise. However, you can use an Ammeter, test-lamp, or voltmeter to test the circuit and prove its continuity. A test-lamp or a Voltmeter will only indicate a disconnection, thus they cannot differentiate between good and bad (unless very bad) connections. An Ammeter is the preferred instrument to use as it will show the actual current flowing within the circuit and thus will indicate any bad connections (causing a lower than expected current and possible non-operation), as well as a complete disconnection.

If the Overdrive unit is not operating, the best electrical test to perform is one of the few occasions when you should check for current rather than voltage. Even if 12 Volts is measured at the solenoid, you do not know if the solenoid windings are good. You could measure the resistance (about 15 Ohms), but that is at a very low current, and the standard current may be making it go open-circuit. The same problem is likewise encountered with the lockout switch. These both can appear to be of high resistance when measured with a test meter, or appear to have continuity when measuring open-circuit voltage. The only accurate test is performed with both its design voltage and design current, i.e., with 12 Volts and the solenoid as a load. Having voltage at the solenoid isn't enough if the solenoid itself is faulty. This is one of the very few occasions when an ammeter is useful for diagnostics. It is important to perform all testing at the appropriate connection. Just which connection is appropriate depends upon with which version of the wiring system your car is equipped. On a system that employs a dashboard switch, the connection on the back of the switch is best. On a system that employs a switch in the knob of the gear change lever (gear shift lever), find where the connector in the engine bay where the yellow wire from the main wiring loom (harness) joins the yellow or yellow / red wire in the transmission wiring loom (harness). Disconnect this and insert an Ammeter, set to 1 Amp. Turn on both the ignition and the manual switch, and then select an Overdrive gear. You should see 800mA being drawn, indicating that there is full circuit continuity, and so the problem is either mechanical or hydraulic. If you do not have an ammeter, then insert a test-lamp instead, and hopefully that will glow, again confirming continuity although not the correct current.

If no current is flowing only then should you start looking at voltages. The first place that you should examine is at the connectors that you have just parted. There should be 12 Volts coming from the main harness. If not, then you need to check at the manual switch if on the dashboard or on the steering column. If you have voltage at those connectors, then you need to check at the gearbox lockout switch, and the manual switch if on the gearlever.

However, if you find that no current being drawn, then check to be sure that you have 12 Volts on the yellow wire that comes from the main wiring loom (harness). If it is not being drawn, then check both sides of the manual switch on the dashboard. If you find 12 Volts there, then check the resistance to ground (earth) of the yellow / red wire or the yellow wire in the transmission wiring loom (harness). You should read 15 Ohms of resistance, if the circuit is open, then you will need to check both sides of the lock-out switch on the gear change lever (gear shift lever) extension housing of the transmission. This is tricky to get access to since you will need to remove both the center console and the tunnel carpeting, and then remove the access panel on the top of the tunnel. You may even have to undo the machine bolts that fasten the transmission support crossmember to the chassis rails, and then lower the tail of the transmission with a bottle jack. If you see the correct current, push the connectors back together again. Test for 12 Volts on both sides, and then try driving it, as the problem could have been a just bad connection in that particular connector which may have been cleared by disconnecting and then reconnecting it. If you do not see the correct current, even with the correct current applied, the plunger could be sticking, so the first step is to remove the solenoid, taking care not to lose the little ball bearing on top of the plunger. Push the plunger to the bottom of the solenoid, apply power to it, and then check to see if the plunger is pulled to the middle of the solenoid. If that is the result, then the problem is most likely hydraulic in nature.

In order to prevent the Overdrive unit from being activated while in the first / second gear plane or while in the reverse gear plane, Overdrive transmissions are equipped with an Overdrive inhibitor switch operated by a pin driven by a pad on the side of the gear selector mechanism. The only way to get a fourth-only transmission to operate in third is to change this mechanism, which will require removal of the transmission. If you engage reverse gear with the Overdrive unit engaged, then you will likely do damage to the Overdrive unit. If after doing so your Overdrive unit still works normally when going forward with the

Overdrive unit both engaged and disengaged, and is OK in reverse and on the overrun, then you have been very lucky. There is a one-way clutch consisting of rollers in a tapered housing. Run this the wrong way, i.e., when in reverse gear, and the rollers will be wedged into the taper, either jamming them both together permanently or, at the least, distorting them. Disconnect the Overdrive unit by unplugging the yellow / red wire from the transmission loom from where it joins the main loom by the fuse box until you get the Overdrive inhibitor switch fixed.

This, of course, will bring up the basic issue of how can you test to find out if the Overdrive inhibitor switch is functioning? The switch is located on the top of the gear change lever (gear shift lever) extension housing and impossible to get at with the transmission in place in the car. There are two basic approaches: the first approach is mechanical, and the second approach is electrical.

Change from Overdrive third gear into second gear, note the engine speed, and then switch the manual switch to the off position. If the engine speed increases, then the Overdrive unit was engaged in second gear (which should not have happened), and hence would probably also be engaged if you were to shift into reverse gear. This should be prevented by a functioning Overdrive inhibitor switch.

If you want to do a more rigorous test on the Overdrive inhibitor switch, disconnect the yellow / red wire in the transmission loom from where it joins the yellow wire in the main wiring loom by the fuse box and connect a test-lamp or Voltmeter that is switched to its twelve Volt scale from the yellow / red wire to the purple wire on the fuse box. By moving the gear lever back and forth across the gate from the third / fourth gear plane to the first / second gear plane and to the reverse gear planes you should see the test lamp glow or the Voltmeter register twelve Volts in the third / fourth gear plane but not in the first / second gear or reverse gear planes. Wiggle it back and forth, and every time that you take it out of the ¾ plane the test lamp must stop glowing or the Voltmeter must register 0 Volts. If it tends to keep glowing even a bit or register a few Volts, even if only one time in 20, then the Overdrive inhibitor switch is sticking and should be replaced in order to protect the Overdrive unit.

You can pull the solenoid out without draining the oil from the transmission. However, it will drip, so place a pan underneath. Sometimes it will come out as an entire assembly, or it may come out in parts. If this happens, be very careful not to lose the tiny ball bearing that operates the valve. In order to bench test the solenoid, simply apply twelve Volts to it. The plunger should center in the coil. Make sure that the little ball is reinstalled where it belongs.

Testing the LH-Type Overdrive Circuits

If you have the manual switch on the dashboard or the steering column, then find the bullet connector where the yellow wire in the main loom joins the yellow/red wire in the transmission loom at the back of the right-hand inner wing. For a car with the manual switch on the gear lever look for white/browns in that position. Part the connector and insert the Ammeter in the circuit. With the ignition on, the manual switches on, and the transmission in an Overdrive gear you should see a current of about 800 Milliamps, equating to a solenoid resistance of about 15 ohms. If using a test-lamp or meter then with the ignition off and the manual switch off poke the probe of the test-lamp or meter in the connector and the other connection to a 12 Volt supply e.g. the purple (which is fused and hence safer than the brown) at the fusebox. By putting the gear lever into an overdrive gear the test-lamp should glow/meter register 12 Volts which indicates electrical continuity through the transmission loom, transmission switch (and manual switch when on the gear-lever) and solenoid. Now connect the flying lead of the test-lamp or meter to ground instead of to 12 Volts. With the manual switch and the ignition both on, it should glow/register with the transmission both in or out of an Overdrive gear. If it only glows/registers when the transmission is *out* of an overdrive gear, then a bad connection back towards the ignition is indicated, including the manual switch where this is on the dash or column.

If the problem is intermittent, i.e. the Overdrive disengages only after driving for a while, and the static tests indicate a draw about 1 Amp at 14 Volts (running), or about 800mA at 12 Volts (engine off with the ignition on), then you can either do the tests indicated above hoping that it doesn't 'repair' itself whilst doing them, or you can do something a bit more reliable. That is to wire the Ammeter into the circuit semi-permanently, have the instrument in the cabin, then go for a test drive.

The simplest place to insert the Ammeter is at the manual switch on the dash on those cars that have them, but with the manual switch on the column it will have to be at the bullet connector in the yellow wire where the main wiring loom joins the transmission wiring loom by the fusebox. Part the connector, and connect the meter to the two halves of the connection, observing polarity (meter positive (+) to the wire from the main wiring loom and negative (-) to the transmission wiring loom). You will need to have the meter in the cockpit, so you may have to extend the leads. This circuit is unfused, so make sure none of the connections can come into contact with anything else, including the body of the car.

At a normal running voltage of about 14 Volts, the current should be closer to 1 amp. If you see about that with the Overdrive operating, but the Ammeter current suddenly drops when the Overdrive disengages, then you have a intermittent electrical connection somewhere in the circuit, which is good news as it should be the easiest and cheapest to cure. If you have significantly less than those currents to begin with, and it doesn't really change when the Overdrive disengages, then there is *permanent* bad connection somewhere giving a marginal current to the solenoid. This could well cause the Overdrive to disengage as it warms up, but ordinarily I would expect that to cause Overdrive to drift in and out of engagement on its own before disengaging altogether.

Note that if the static tests show significantly *less* than 800mA, then even though the Overdrive may function when it is cold, there may be insufficient current flowing when things get warmer for the solenoid to function. However, in this case, I would expect the Overdrive to go from being engaged, to drifting in and out of engagement, before it disengages altogether.

At this point, in either of the above cases, you will need to start working your way through the circuit with a Voltmeter. If you have either a break or a high-resistance connection somewhere in the circuit, then on the 12 Volt supply side of that fault you will see 12 Volts, but on the solenoid side you will see significantly lower than that, or possibly no Voltage at all. Start at the connection of the main wiring loom to transmission wiring loom as that is in the middle of the chain on most cars. Ideally, you need the Ammeter still in the circuit so you can confirm the fault is still present while doing your voltage tests, but that will require a second instrument, although you should be able get away with a test-lamp.

Test between the connector and a known good ground (earth), i.e., the engine with your Voltmeter or test-lamp. If you see a full 12 Volts, and the Ammeter is still indicating low or no current, then the fault lies closer to the transmission. If the voltmeter or test-lamp shows no or low voltage, then the fault lies back towards the 12 Volt supply.

In the former case the next point in the chain is the transmission switch, which is going to be awkward to get to, especially as the fault may 'repair' itself while you are getting at the switch. In that case I would run a wire up from the transmission switch to a voltmeter sitting in the cabin, and again go for a test drive. If the Ammeter shows a drop in Voltage when the Overdrive disengages, then the problem lies back towards the connector by the fusebox. If it does not, then it lies closer to the solenoid. Depending on what year and market your car is, there could be the transmission and the manual switch connections down there, several bullet connections, as well as the final connection to the wire going into the solenoid. If this last test shows no drop in voltage when the Overdrive disengages, but the current test still shows a drop, then the problem is either inside the solenoid, or in its connection to ground (earth). You may find a dry joint where the wire connects to the winding, but I think the ground (earth) connection is simply a pressure connection between two bits of metal, probably from the end cover that retains it.

If the fault lies back towards the supply then, it could be the manual switch itself, its spade (dash switch) or multi-plug (column switch) connections. Test each of these in turn. Remember to test the back of each bullet as well as the metal sleeve of bullet connectors, both sides of the multi-plug connector, and the wiring connectors, as well the spades on both sides of the switches. If you find you have voltage on one spade of the switch, but not the other, then the switch itself is faulty. If you find that you have voltage on a spade but not the wiring connector that is on that spade, then the connection between the two is faulty. However, this is likely to be disturbed by prodding it and may seem to 'repair' itself there and then.

Remember, all connections must be made (via an Ammeter if appropriate), ignition on, manual switch operated, transmission in an Overdrive gear in order for voltage tests to be valid. If you have an Ammeter in circuit that must also be showing reduced or zero current for the voltage tests to be valid. If the Ammeter shows a normal current, i.e., 800mA, then there is no point doing voltage tests, as *electrically* the circuit is working as it should.

Finally, if the Ammeter is still showing about one Amp when the Overdrive disengages, then the problem is not electrical, but is either mechanical or hydraulic, which is a whole different ball game.

Resistance measurements through switched circuits can be unreliable. Most Ohm-meters only pass a microscopic current through the circuit, and where switches are involved this can sometimes not be enough to 'burn' through the oxide film that can develop on the switch contacts if they have not been used for a while, and instead of showing zero resistance as you would expect you will see a resistance possibly in the tens of Ohms indicated. This may lead you to think that the switch is faulty and go through the cost and effort of changing it whereas passing the real-world current through the circuit will show no or negligible volt-drop across the switch, which is the only valid test. Resistance measurements of *components* though, like of the solenoid itself, are valid.

If all of that tests out OK, then any problems are likely to be hydraulic or mechanical. The first step in diagnosing *that* should be a pressure gauge on the Overdrive, which should show 400 to 420 PSI with Overdrive engaged, and zero when disengaged. This will obviously need the engine and transmission to be run with the rear wheels off the ground, handbrake off, *and the car very safely supported*. Remember to switch Overdrive in and out when doing a pressure test as you will need to note from the spinning of the wheels whether Overdrive is actually engaged when the pressure shows it should be. This will be easier with just one wheel raised and the other on the ground. If the problem is that Overdrive disengages after driving for a while then it will need to be run like this for an equivalent time to allow the fault to develop. Insufficient pressure indicates pump or relief valve problems, or dirt in the passageways. Correct pressure but Overdrive not engaging indicates problems with the actuator pistons, dirt in hydraulic passageways feeding them, or with the sliding annular clutch (conical clutch).

Hydraulic Troubleshooting Of The LH-Type Overdrive

Too many people play Overdrive roulette by installing an untested Overdrive unit, only to find that their Overdrive unit either does not function properly, or does not function at all. You should always test it before you put it back into the car. In order to do this you will need the following equipment: a large drill with a with a large size chuck (5/8" or more), a 5/8" drill bit, a short piece of ¾" rubber hose along with two hose clamps to function as a flexible drive, and a 12 Volt source with a pair of alligator leads. Make sure that the transmission is filled with the correct amount and type of oil (20W/50). Connect the drill bit to the first motion shaft of the transmission via the ¾" rubber hose that is secured to the drill by means of hose clamps. Make sure that the drill is properly secured in the drill chuck in order to assure that the drill bit itself will not spin. The first motion shaft will need to be rotated clockwise, so check to be sure of the rotational setting of the drill so as to not allow the drill motor to rotate the output flange of the Overdrive unit in counterclockwise (anticlockwise) direction, otherwise you may ruin the clutch mechanism that is inside of the Overdrive unit. With the transmission engaged in fourth gear, spin the drill. Allow a few minutes of rotation so that the pump that is inside of the Overdrive unit can build up adequate pressure. If you are testing an LH-type Overdrive unit that is used with the four-synchro transmission of the MGB MKII, then apply 12 Volts to the solenoid. On the other hand, if you are testing a D-type Overdrive unit that is used with the three-synchro transmission of the MGB MKI, then you will not need any electrical power, as all that you have to do is to lift the arm of the solenoid. If the Overdrive unit is functioning correctly, then the output flange should increase in speed (you should hear the change when the solenoid engages). When the current is switched off, or the arm of the solenoid is lowered, the output flange will decrease in speed.

A nonfunctioning LH-type Overdrive unit can be caused by either mechanical, hydraulic, or, most commonly, electrical problems. Sometimes it is a matter of nothing more than insufficient oil in the transmission. Pre-1975 model four-synchro transmissions have a dipstick that can be accessed by lifting the carpet on the transmission tunnel and removing a rubber access plug. The best way to add oil to these transmissions is to use some inexpensive plastic tubing with a funnel inserted into one end. Feed the tubing into the tiny little hole in the transmission, and then slowly pour oil in through the funnel. An alternate method is to use an old-fashioned oil squirt can. Post-1974 models have a side filler plug on the transmission, which is located at the proper level for a full transmission. Remember, the transmission does not hold all that much oil (6 U.S. pints without an Overdrive unit, 7 U.S. pints with an Overdrive unit), so if you have a leak from your transmission, take care to check the oil level often. An inadequate oil level will interfere with the function of the synchronizer hubs, causing the transmission to shift slowly.

Unfortunately, a broken sliding annular clutch (conical clutch) makes no noise at all. On the late-model LH-type Blue Label Overdrive units, its friction material was only bonded on, instead of being riveted. Although the clutch friction material is designed to operate in an oil-soaked environment, 30+ years of saturation has most likely taken its toll. In the case of late-model LH-type Blue Label Overdrive units wherein the clutch friction material is merely bonded to the sliding member, it is not unusual to find the entire clutch material separated from the sliding member and floating intact inside of the Overdrive unit. Symptoms of this type of deterioration are slipping or "free wheeling" in Overdrive. These symptoms can also occur in direct drive. Normally, the friction material of the sliding member is usually more deteriorated that that of the annulus. This is caused by insufficient oil pressure in the Overdrive circuit, causing the clutch to slip, thus overstressing the friction material. Of course, this results in a more rapid deterioration than in the case of the more positive engagement of the spring-driven direct drive. It may also be due to the type of oil that is used. Transmissions lubricated by 90W hypoid gear oil always show a more advanced state of deterioration of the friction material than those that have been lubricated by 20W/50 engine oil. Once the clutch friction material disintegrates, it is rapidly turned into muck by the moving parts within the Overdrive unit. Because both the transmission and the Overdrive unit share the same oil, this muck eventually finds its way throughout both the transmission and the Overdrive unit, jamming synchronizer hubs, blocking oil passages, fouling the oil pump of the Overdrive unit, preventing the seating of the low-pressure and non-return valves, and generally wreaking all sorts of havoc. One cringes when thinking of what this pulverized friction material can do to bearings. This being the case, an Overdrive unit should always be disassembled and carefully inspected prior to being installed onto the transmission.

Be aware that there are certain problems inherent in using 90W hypoid gear oil in an Overdrive unit. The engagement of the Overdrive unit will tend to be more abrupt. The oil passages in the accumulator sleeve being quite small, 90W hypoid gear oil is so thick that it will not escape from the accumulator chamber as quickly as the oil pump can force oil into it. As a result, although the accumulator piston passes the position of the oil relief passage, the pressure continues to increase, forcing the seating pistons against the sliding member and often leaving the mechanism jammed in Overdrive. In addition, the accumulator spring often being compressed to the point that it becomes no longer serviceable, the accumulator piston will bottom out in its sleeve. Other problems common to Overdrive units filled with

90W hypoid gear oil are excessive wear on the roller of the oil pump plunger and the eccentric cam lobe upon which it rides, a result of the higher pressures developed in pumping the heavier oil through the lubrication system of the Overdrive unit. As though this is not bad enough, EP90 GL5 oils are unsuitable as their high sulphur content will attack the bronze metals within the gearbox and the Overdrive unit. These bronze components include all three (3) gear selector forks of both the three-synchro and four-synchro transmissions, possibly the second speed baulk (synchro) ring of the three-synchro transmission, second & third speed gear bushings, and the second to third speed interlocking ring, as well as the bushings of the D-type overdrive unit.

However, the problem is much more likely to be either hydraulic or electrical. Let's tackle the hydraulics first. If you wish, prior to any disassembly you can add a pint of Dexron 3 Automatic Transmission Fluid to the transmission oil in order to act as a cleaning agent. It is highly detergent and will clean out any sticky deposits. I should point out that I have heard of people using nothing but Automatic Transmission Fluid for a while in order to cure a problem. I have also heard of people claiming that after the Automatic Transmission Fluid had cured their problem, they went back to using engine oil and the problem returned, so they use nothing but to Automatic Transmission Fluid in their transmission. In such cases there is still a problem within the Overdrive unit that the use of Automatic Transmission Fluid is merely masking, and they are running the risk of eventually developing an even more serious problem.

Drain both the Overdrive unit and the transmission. Be warned that Overdrive units are very sensitive to dirt. Clean the Overdrive sump cover and the area around it so that dirt and grit do not get inside when you remove it. Unscrew the Overdrive sump cover securing bolts, taking great care with these bolts as their threads can be easily damaged by rough handling; always use the correct sized socket and apply even pressure. Remove the Overdrive sump cover on the sump filter screen, clean all of the metallic particles from the two magnets that are fitted inside of the Overdrive sump cover, and then clean both the Overdrive sump cover and filter screen with carburetor cleaner. If needed, a new combination gasket and filter screen can be obtained from Moss Motors (Moss Motors Part # 466-360).

Now you are ready to tackle the relief valve. It is located in the top left hand corner. Remove both the ½" BSP relief valve plug and its sealing washer, and then withdraw the relief valve assembly. In the following order, remove the filter screen, the spacer tube, the low-pressure valve assembly, and the relief valve spring. Keep them together in that order and in their original orientation. Now, remove the relief valve plunger. Examine both the relief valve plunger and its seat for pitting, scoring, and excessive wear, and then replace any worn or damaged parts. Examine the O-rings of the relief valve body for signs of deterioration and replace if necessary (Moss Motors Part # 290-930 and 290-925). Check the relief valve spring for signs of collapse or weakening. Its free length (uncompressed length) should be 3 cm. Clean the relief valve filter. If the spring is fouled by dirt, or has become weak, the pump will not generate the 400 PSI of pressure that is necessary in order to operate the Overdrive unit. Reassembly is the reverse of the above order. Make sure that the relief valve is installed in its correct order and that its parts are in their correct orientations. I once put a piece in upside down and it took me hours to figure out why the Overdrive unit would refuse to engage.

Now you are ready to check the Solenoid valve. Unscrew the four screws securing the solenoid cover (the name plate), and then remove both the solenoid cover and its gasket. Withdraw both the solenoid and the operating valve assembly by carefully pulling on the solenoid lead. Withdraw the solenoid rod and the operating valve assembly from the solenoid housing. Keep them together in that order and in their original orientation. Press the solenoid coil and the base cap from the housing. Remove both the operating valve plunger and ball by shaking them off of the solenoid rod. Examine the valve ball and its seat for pitting and scoring. Replace all damaged or suspicious parts. The ball may be reseated by lightly tapping it onto its seat by using a wooden dowel rod as a drift punch. Inspect the O-ring seals (Moss Motors Part #'s 290-935, 290-940, and 290-9450) for signs of deterioration and replace if necessary. Reassembly is the reverse of the above order. Be aware that the wire from the solenoid needs to be aligned correctly in the slot of the unit in order to permit the refitting of the cover plate.

Finally, you are ready to tackle the pump and the non-return valve. Remove the O-Ring (Moss Motors Part # 462-620), and then unscrew the pump retaining plug. It is the one with two holes in its face. Remove both the spring and the ball of the non-return valve. Remove the pump body, the pump plunger spring, and the plunger. Keep them together in that order and in their original orientation. Taking care not to damage the bore of the pump body, use a suitable drift to separate the seat of the non-return valve from the pump body. Examine the O-ring seals (Moss Motors Part # 290-915) for signs of deterioration, and replace them if necessary. Examine the ball of the non-return valve and its seat for pitting and scoring. Replace any and all damaged or suspicious parts. The ball may be reseated by lightly tapping it onto the seat with a wooden dowel rod. Carefully reinstall the non-return valve seat into the pump body. Insert the pump into the casing, ensuring that the flat side of the plunger is towards the rear of the Overdrive unit. Reassembly is the reverse of the above order.

One vexing symptom of an Overdrive problem that is not covered by the factory manual is a “pumping” effect during engagement when the car is driven in direct drive, while in Overdrive all appears normal. This is caused by a problem that is very simple to fix. When the system is operating correctly, a buildup of hydraulic pressure from the pump is directed to the two operating pistons and moves the sliding annular clutch (conical clutch) unit, its outer brake surface then comes into contact with the stationary brake ring, and then the complete sliding member and the sunwheel will cease to rotate. With the solenoid deactivated (i.e., Overdrive switches off), the solenoid plunger does not retract fully, and the springiness in the small O-ring at its tip is enough to push the piston and ball back into its seat once the pressure has bled off. This in turn causes the pressure in the actuating system to increase, which in turn causes the Overdrive to engage again. Past a certain level, there is enough pressure to force the plunger back a bit until the pressure bleeds off and the cycle is repeated again (at a frequency of about every 2 seconds). Normally, as the sliding annular clutch (conical clutch) unit of the Overdrive unit starts to move during the engagement process, the Overdrive unit temporarily loses engagement during the moment between when the inner lining of the sliding annular clutch (conical clutch) leaves its seat on the annulus and the outer lining contacts the stationary brake ring. This is event so brief that it is not noticeable, but in this case, the sliding of the annular clutch (conical clutch) never travels far enough for its outer lining to contact the stationary brake ring before it is pushed back again, so for about a second neither lining is in contact. As soon as the inner lining of the of the sliding annular clutch (conical clutch) engages the annulus, engagement returns with a jerk and stays for a second or so until the of the sliding annular clutch (conical clutch) is again pumped away, and the cycle then repeats itself. The solution to this problem is very simple: fit a thicker gasket under the solenoid cover plate. This will allow the piston to move a bit further back before hitting the solenoid cover plate, the extra movement being enough to make the O-ring slide down into its bore instead of simply compressing a bit and functioning like a spring.

When reinstalling the sump cover of the Overdrive unit, tighten the sump bolts bit by bit in a logical alternating-side sequence in order to avoid warpage. I have not seen a specific torque figure for these machine bolts, but the standard figure for 1/4" UNC/UNF threads is 8 Ft-lb to 9 Ft-lb (11 Nm to 12 Nm). Do not exceed that amount of torque or you could either strip the threads or warp the sump pan.

It is not commonly known that the D-type Overdrive units of both the Triumph Spitfire and the Triumph GT6 can be used in the three-synchro transmission of the MGB by doing some parts swapping. While both versions use the same .802:1 Overdrive ratio, the internal mechanisms are slightly different in that the Overdrive units of both the Triumph Spitfire and its GT6 sibling use a lower operating pressure and have a set of rebound springs on their operating pistons. However, the operating pressure can be boosted by simply shimming the balance tube assembly, and the rebound springs can be either left in or removed, as it does not seem to make any difference in the performance of the unit. The major external differences are the rear flange bolt pattern and the gear change lever (gear shift lever) extension housing mount on the MGB D-type rear casting. The flange for the MGB can be easily interchanged with that of either of the Triumph models, or simply redrilled in order to change from the MGB pattern to that of the Triumph. The rear housings can also be interchanged so that the MGB gear change lever (gear shift lever) extension housing can be bolted onto the rear housing.

Whilst the side-fill transmission is easy to clean around the filler hole and get the end of the tubing into it without picking up any dirt, the top-fill transmissions need a bit more care. Loosely insert the dip-stick in order to preclude any dirt falling into the transmission, then from underneath wipe around the dipstick and the top of the casing. Use a long length of plastic tubing up from the filler hole (top or side) into the engine compartment so that you can refill the transmission in relative comfort. Note that both the transmission and the Overdrive unit use engine oil of the same grade as for the engine, e.g., 20W/50. If your transmission is equipped with an Overdrive unit, do not use a detergent oil as it will foam and interfere with the function of the Overdrive unit. The three-synchro transmission takes

5.6 US pints (4.5 Imperial pints / 2.56 liters), and the three-synchro transmission plus Overdrive unit takes 6 US pints (5.33 Imperial pints / 3.36 liters. The 4-synchro transmission takes 7 US pints (5.25 Imperial pints / 3 liters), and the 4-synchro transmission with Overdrive unit takes 7 US pints (6 Imperial pints / 3.4 liters). Be aware that these quantities are for clean and dry transmissions. A drain and a refill can be expected to take a little less than that as some of the old oil is bound to be left behind.

It will take time for the oil to flow throughout the transmission, so do not try to pour in the whole of the recommended amount all in one go or it will probably overflow. Even though a dry-fill takes 3.4 liters, you may find that even after 2 liters you will have to pull the tube up a little bit in order to get any air back up the tube. Testing with the dipstick at that point in the refill process may show that it is just above the “MIN” (minimum) mark even through there is not much more than half the required quantity in yet. Allow the oil to settle into the transmission, then recheck the level and top-off as required, then replace the filler plug/dipstick and check the drain plug is tight before taking it for a run of a few miles. Once you have the transmission refilled with the proper quantity of oil, drive the car until it reaches normal operating temperature. Upon your return, check the level again on a flat and level surface, and check it again when cold after leaving it overnight, rechecking the drain and level/filler plugs to ensure that they are tight. After the next decent run check the level and the plugs again in order to give you the confidence there are no leaks, then you should be fine to leave it alone until the next normal service interval.

If these procedures do not get the Overdrive unit up and functioning, then you will need to tackle the electrical possibilities.

Of course, it is always possible that you will have to completely disassemble the Overdrive unit in order to troubleshoot it. To disassemble the Overdrive Unit, remove the filter screen cover and the filter screen. Remove the relief valve plug, and then remove the relief valve. Remove the pump plug and remove the pump, taking care not to lose the ball bearing. Unfasten the machine bolts securing the ID plate and then remove the solenoid, taking care not to lose the ball bearing. Unfasten the nuts that hold the casing together, then separate the halves. Remove the four nuts that hold the piston operating bars, withdraw the sliding member, and then refit the bars and the nuts in order to retain the springs and the spacers. Using needle nosed pliers, pull out the operating pistons. Remove both the speedometer drive bearing and the speedometer pinion gear. Clamp the drive flange and remove its retaining nut. Withdraw the flange and the rear oil seal.

Examine the teeth inside of the annulus for excessive wear. Inspect the one-way Sprague clutch for correct operation. Examine the pinion gear teeth for excessive wear and check its bearings for smooth operation. Likewise, examine the sun gear teeth for excessive wear. Check the sliding member bearing for smooth operation. Inspect the sliding annular clutch (conical clutch) for burning, loose rivets and wear. Inspect the bronze bushing in the front casing (or thrust washer) for wear or missing parts. Inspect both the relief valve and the pump for wear or heavy scoring. Check the ball valve seats for correct seating. Examine the front casing for loose circlips. Place the pump one-way valve spring in its recess in the pump retaining screw and ensure that it projects 1/16" to 1/8" (1.5875mm to 3.175mm) above the surface. If not, gently stretch the spring by running a thin screwdriver up the coils sideways. Finally, ensure there are no sharp edges on the pump retaining screw that might catch on the spring. If necessary, place it in a drill and use a fine file to lightly chamfer it.

In order to reassemble the Overdrive unit, clean all of the parts thoroughly, then lightly oil the bearings and the gear assemblies. Replace the O-rings on the operating pistons by lightly oiling them, and then refitting them into the front casing. Refit the sliding member with the cutouts on its bars facing forward. Fit a new rear oil seal, and then lightly oil the surface of the seal. Install the drive flange and then torque its retaining nut to 55 Ft-lbs to 60 Ft-lbs. Align the marks on the pinion gear and insert it into the annulus gear. Apply a thin coating of red Loctite to the mating surfaces, and then loosely assemble the front and rear casings. In order to ensure correct alignment, align the internal splines and slide the Overdrive unit temporarily onto the mainshaft (third motion shaft) of the transmission. Tighten the nuts that clamp the two halves together, and then remove the Overdrive unit from the mainshaft (third motion shaft), taking care not to rotate drive flange from this point onward. Lightly oil the O-rings of the relief valve assembly, then replace them onto the relief valve assembly and refit it into the casing. Lightly oil the O-rings on the pump assembly and then replace them onto the pump assembly and refit it into the casing, noting that the flattened side of the pump-operating shaft faces rearwards against the casing. Install the new filter screen and the filter screen cover. Be aware that the positions of both the Overdrive filter screen and its sump can be reversed. Paying attention to the internal layout should give you the proper guidance as to their proper orientation. Lightly oil the O-rings of the solenoid assembly and then replace them onto the solenoid assembly, and then refit it into the casing. Lightly oil the O-ring of the speedometer drive bearing, replace it onto the speedometer drive bearing, and then refit the speedometer pinion and it bearing into the casing.

Place a new gasket onto transmission adapter housing. Refit the circlip, locating ball, and pump drive cam to the transmission mainshaft (third motion shaft). Rotate the mainshaft (third motion shaft) so that the cam lobe is uppermost, and then carefully slide the Overdrive unit onto the mainshaft (third motion shaft). If it fails to reach the studs, then the internal splines are not aligned. Be aware that there are 2 sets of inner splines one of the annulus and another of the sun gear. In this case, remove the Overdrive unit and, while using two wrenches to lift the operating bars, use a long screwdriver to rotate the sliding annular clutch (conical clutch) until the splines are aligned. If they fail to mate within 1/4" to 3/8" (6.35mm to 9.525mm) of movement, then the pump-operating roller is caught on the cam. Should this prove to be the case, use a long, thin screwdriver to depress the pump plunger. Refit the retaining nuts, noting that some must be fitted with the Overdrive unit slightly raised from the adapter housing. Place a new gasket onto the gearlever extension housing, and then refit the housing while locating the plastic bushing into its hole in the selector arm.

Fusing The Overdrive

As with the Fuel Pump, it is a good idea to fuse the Overdrive circuit to avoid loom damage or total loss of the car to a fire if the wire or the Overdrive should short out. Rubber Bumper models with the manual Overdrive switch on the gearlever are particularly vulnerable because of the continual flexing of the wires at the lever due to changing the gears.

For cars with the manual switch on the dashboard, it is best to make up an in-line fuse with a male spade on one end and a female on the other, then simply pull one spade off of the switch and put the male end of the new fuse into that, and the female end of the new fuse back on the switch. Either wire will do, but logic dictates the earlier in the circuit the better, i.e., on the white wire.

For cars with the manual switch on the column, if you want the fuse to be as early in the circuit as possible, you will have to cut one of the wires and crimp/solder male and female spades or bullets onto the cut ends. Note that some years have <u>two</u> white wires coming out of the main loom and going to the plug that feeds the Overdrive switch, and unless you put the fuse in the yellow wire you would have to cut and connect *both* white wires. If you put the fuse on the switch-side of the multi-way plug there is only one white wire (and yellow wire) in all cases.

For UK Market cars with the gearlever manual switch you have to insert the fuse into the brown / white wire by the bullet connectors between the main loom and gearbox loom.

For North American Market cars with the gearlever manual switch there is a double-connector with one white wire from the inertia switch, a white wire to the fuel pump, and a white / brown wire to the Overdrive switch. I suspect this double-connector is with the others where the firewall and right-hand inner wing join, but I am not certain. To fuse just the Overdrive, insert it into the white / brown wire. But while you are at it, you could fuse both the fuel pump **and** the Overdrive if you insert it in the white wire from the inertia switch. If the inertia switch has spade connectors, then you could insert it there by using male and female spades on the in-line fuse instead of bullets

I would use a standard 17amp rated/35 amp blow fuse in the circuit simply because there are (or should be!) a couple of spares of that rating in the main fusebox. That rating may seem higher than required for the Overdrive (and the fuel pump where applicable), but the purpose of the fuse in nearly all cases in the MGB is to protect the wiring and that is the correct rating of fuse for the grade of wire used.

The above relates to cars with the later LH-type Overdrive. Cars with the earlier D-type Overdrive have a vacuum switch and relay as well as the manual switch. On these cars the manual switch operates the relay and it is the relay that operates the Overdrive. The relay has its own white (unfused ignition) wire feed which should be fused as well as that to the manual switch.

Installing An LH-Type Overdrive Sequencer Relay

From time to time you may find that you move from either 3rd gear or 4th gear with the Overdrive engaged to 1st gear or to 2nd gear and forget to switch Overdrive off. All is well until you change up from 2nd to 3rd, then all of a sudden the Overdrive engages again, usually under conditions where it is inappropriate. This means not only do you get the mild and unexpected jolt as it engages, but another one when you then manually switch the Overdrive off until you need it again. The answer to this all-too-human failing is the installation of an Overdrive Sequencer Relay.

The purpose of the circuit is to allow the Overdrive to be engaged if the manual Overdrive switch is operated while in either 3rd gear or in 4th gear, but to lock it out when changing to any other gear, even when changing back to 3rd gear or 4th gear until the manual Overdrive switch is turned “Off” and then “On” again. While this can be achieved with just one relay that has a single normally-closed contact, the disadvantage of such a relay lies in the fact that it requires a connection to the solenoid side of the transmission lockout switch, and that there is the faint possibility that a fault could leave the Overdrive engaged while in reverse gear, which would either damage or destroy the Overdrive if you try to move the car. However, there is another version of an Overdrive Sequencer Relay system that uses some simple electronics in addition to the normally-closed relay, and completely eliminates that possibility. This is sometimes called a “changeover” or “single-pole double-throw” relay (sometimes written as 'SPDT' relay), not a 'switching' relay which is used for spot lights and the like.

What happens is that the normally-closed contact of the relay is wired *in series* with the transmission switch and the manual Overdrive switch. This means that if *any* of the manual switches, the transmission switch, or the relay contact are open, then the Overdrive will remain disengaged, and they all have to be closed before the Overdrive will engage. Thus it is fully fail-safe : any fault in the additional circuitry can only cause the Overdrive to either function exactly as the factory intended, or not at all. It can never cause it to operate when it should not, i.e., if reverse gear, 2nd gear, or 1st gear is selected**.** So as long as the relay remains released, the Overdrive will operate normally, but when the relay operates, it prevents the Overdrive from engaging regardless of the state of the transmission switch and the manual Overdrive switch. The circuit is such that if the manual Overdrive switch is closed and the transmission switch is open, then the relay will operate, and will remain so until either the manual Overdrive switch is opened or until the ignition switch is switched off. Remember that whenever the relay is operated, its normally closed contact is open, thus current flow is interrupted and this prevents the Overdrive solenoid from engaging the Overdrive.

If the transmission is taken out of an Overdrive gear, the transmission switch then opens, the current ceases to flow, the solenoid is switched off, and the Overdrive disengages. Even if an Overdrive gear is selected again and the transmission switch closes, the relay contact ensures that the solenoid will not be energized, and the Overdrive remains disengaged.

It is not until the manual Overdrive switch is turned off, or the ignition is turned off, that the sequencer relay will release, because there is no longer the 12 Volt supply to keep it operating. But this also means that even when the relay releases, there is no voltage supplied to the solenoid, thus the Overdrive remains disengaged.

It is not until the manual Overdrive switch is turned on again while the transmission is in an Overdrive gear that current flows through the sequencer relay (released), the transmission switch (closed) and the solenoid, to engage the Overdrive on again.

Be aware that the electronics are obviously polarity sensitive and so cannot be used as-is on a positive ground (earth) car. However, it should not be beyond anyone capable of building such a circuit to come up with a suitable variant, i.e. reversing the diodes and using NPN transistors. MGB MkI models use the positive ground (earth) D-type Overdrive and I have not tested the circuit with this unit, only with the later negative ground (earth) LH-type. But because the forward-bias voltage of the diodes in the relay is constant, even if the positive ground (earth) D-type solenoid takes a lower current than the LH-type, it may be that it will remain operated in series with the relay, which could wreck your overdrive if you reverse with the manual Overdrive switch still on.

From the 1977 model year all markets had the manual Overdrive switch on the gear lever, and from February 1977 North American Market cars had Overdrive on 4th gear only. Cars for the North American Market other than 'February 1977 -on' cars are wired as before, but the yellow wire and yellow / red wires have to be picked up by the transmission owing to the physical position of the manual Overdrive switch. However, North American Market '4th gear only' cars were wired differently because the transmission switch also controlled the vacuum advance of the ignition through the TCSA switch. In these cars, the current flows from the ignition, on through the transmission switch, and then on through the manual Overdrive switch to the solenoid. With this switch configuration the circuit will not work properly - Overdrive will be locked out by the manual Overdrive switch and not the transmission switch as it should be. It would be possible to come up with an alternative circuit like for the positive ground (earth) MkI version, but you must be aware that it will involve connections to the solenoid side of the transmission switch, so if a fault develops it might be possible for the Overdrive to remain engaged while in reverse gear, and so either damaging or destroying the Overdrive should the car be driven.

Of course, you should test the new system before setting out on the road.

With the ignition switched on but engine stopped and the gearlever in 1st gear, 2nd gear, or reverse gear , turn the manual Overdrive switch on and off a few times. You should hear the relay click as it operates and releases with the manual Overdrive switch.

With the manual Overdrive switch off, shift into 4th gear, and then operate the manual Overdrive switch. You should not hear the relay click at any time.

Shift the gear lever into 1st gear and you should then hear the relay click once as it operates.

Move the gear lever into and out of 4th gear a couple of times and you should not hear the relay click.

Switch the manual Overdrive switch off and you should then hear the relay click once as it releases.

On the road, get into top gear and the Overdrive should engage and disengage as normal as the manual Overdrive switch is turned on and off.

With the manual Overdrive switch in the “On” position and Overdrive engaged, shift into 2nd gear and note the engine speed (which will be higher than in 4th gear, of course). Move the manual Overdrive switch to “Off” and there should be no change in engine speed.

Shift back into 4th gear and move the manual Overdrive switch to the "Off" position, and then back to the "On" position again. If the sequencer relay is functioning, then there will be no increase in engine speed as you switch it off, but there will be a decrease in engine speed as you turn it on again and the Overdrive engages.

If any of these tests fail, do not attempt to drive the car, especially in reverse, with the manual Overdrive switch in the 'On' position. Remove the relay and restore your wiring to normal. Either your relay is too low a resistance, you have wired it incorrectly, or your solenoid remains in operation with a lower current than mine does.

Reinstalling The Transmission

Reuniting the engine and the transmission with each other is an easy task as long as you have the transmission properly supported. When installing the transmission onto the engine, you should first put a thin smear of Lubriplate white lithium grease into the splines of the input shaft (first motion shaft) in order to prevent rusting. Always smear a very thin film of assembly lubricant onto the input shaft (first motion shaft) next. Never use plain grease for this purpose as it will seal the pores of the spigot pilot bushing and prevent the oil contained within it from lubricating the shaft. Do not use too much assembly lubricant as this can cause air to be trapped within the spigot of the crankshaft, creating a pneumatic lock that will cause the air trapped inside of the spigot to compress, keeping the input shaft (first motion shaft) from going all the way into the spigot pilot bushing, and make pulling it out a difficult task at best. Warping the clutch driven plate is a risk, so care in alignment is needed both when installing the clutch assembly and when mating the engine and transmission. The symptom of a warped clutch driven plate is a dragging clutch. This being the case, you will need a clutch alignment tool. The best tool for this purpose is an old input shaft (first motion shaft), although a plastic tool is available. If you do not align the clutch driven plate properly prior to attempting to mate the engine and the transmission, you will likely never make the engine mate to the transmission. If you do not have an alignment tool, then leave the bolts slightly loose, with just enough tension to hold the clutch driven plate in position (about two turns) so that the clutch driven plate can move into proper alignment. Be aware that if you use this method, you will have to remove the transmission in order to tighten the clutch securing bolts. Get the end of the first-motion shaft properly located inside of the carbon clutch release bearing, and, as the two casings are brought together, making sure that the flywheel is square with the transmission casing in both its vertical and horizontal planes, and that the two casings are aligned both vertically and horizontally with the same gap all round. Be prepared to rotate the crankshaft until the splines are properly aligned. Do not lock the transmission in gear. Without that, the shaft turns very easily, and making it easier for the splines to self-align. Remember that the longitudinal centerline of the transmission must be exactly aligned with the longitudinal centerline of the crankshaft. If it is off, even by a small amount, they will not go together. If you can only get the transmission to within 3 inches of the engine, then the problem cannot be the pilot bushing in the spigot of the crankshaft as the first motion shaft will not be anywhere near close to the limit of its travel within the spigot pilot bushing. It is more likely that the splines on the input shaft (first motion shaft) are not lining up with those of the clutch driven plate. If rotating the crankshaft does not produce the desired results, try engaging a gear, as this will allow you to align the splines by rotating the selected gear set by means of rotating the output flange at the rear of the transmission. The static inertia of the gears will prevent the input shaft (first motion shaft) from rotating too easily. If you find that this method is too cumbersome to succeed while working alone, then get an assistant to rotate the output flange at the rear of the transmission while you move the transmission forward onto the engine. Take care to ensure that the upper right bolt hole near the engine block oil outlet fitting and the lower left bolt hole where the brace for the exhaust system attaches are not misaligned. These holes are intentionally a smaller, closer fit on their bolts than the other bolt holes are. These two bolt holes and their bolts serve as locators, similar in manner to that of dowel pins, in order to maintain the concentric alignment of the crankshaft to the transmission input shaft (first motion shaft). If you put those two bolts in first, then the other bolts will slip right in with lots of clearance, and the splines of the input shaft (first motion shaft) and clutch, as well as the starter gear, will all be properly aligned.

Prior to reinstalling the engine along with its attached transmission, you can make things much easier for yourself later on if you have all of the attachment brackets for the wiring loom already bolted onto the transmission. Also, you ought to check both of the switches with an Ohmmeter before you reinstall them. This will serve two functions: If you are in doubt as to which switch performs which function, the right-hand switch for the reverse lights should give continuity on the Ohmmeter only when the transmission is in reverse; whereas the switch on the left-hand Overdrive lockout switch will give continuity only when the transmission is in third or fourth gears. If there is any intermittence in continuity at all, especially by wiggling the shift lever, you can fix the problem from inside the car without having to pull the engine / transmission package out all over again. On the other hand, if the switches are in good condition, it is usually a simple matter of adjusting the number and/or thickness of the washers. Normally, there must be two washers, otherwise it will not function correctly.

If you have a problem with the gear change lever (gear shift lever) rattling after the transmission is reinstalled into the car, check the position of the chrome screws that secure the chrome gaiter ring to the tunnel. One screw is longer than the others are. If that screw goes into the wrong hole, it will bottom out onto the main transmission casing and transmit noise and vibration into the car. The correct position for the longer screw is in the forward screw hole.

When driving, do not rest your hand on the gear change lever (gear shift lever). Doing this will preload the selectors (saddles) against the synchronizer hubs inside of the transmission, causing premature wear of both the baulk rings (synchronizer rings) and the selectors (saddles). Wear of the selectors (saddles) can become so extreme that the shift linkage inside of the gearlever extension housing can become disengaged, leaving the transmission stuck in whatever gear ratio that it happens to be in at the time that this occurs.

When reinstalling the engine/transmission, take care to not omit the installation of the stay rod ob the transmission crossmember. If the bushes (BMC Part # AHH 7854, Moss Motors Part # 280-055; BMC Part # 1B 8347, Moss Motors Part # 280-050) or the buffer pads (BMC Part # 1G 8781, Moss Motors Part # 282-380) have become either hard or cracked, replace them with new parts as their purpose is to reduce the amount of vibration that is transmitted to the cockpit. The purpose of the stay rod is to prevent the engine/transmission from moving forwards during hard braking. If your MGB has its radiator located close to the cooling fan, the results of this forward movement will be a damaged fan and a ruined radiator core. The nuts should be tightened so that you are not putting any tension or loading on either of the engine or the transmission mounts.

The Driveshaft (A.K.A., The Propeller Shaft)

Another concern will be that of the driveshaft (propeller shaft). All those used in the MGB are Hardy-Spicer 1100 Series driveshafts (propeller shafts). While the Original Equipment 2" (50.8mm) diameter Hardy-Spicer driveshaft (propeller shaft) of the MGB has a wall thickness of .064" (1.6256mm) and is of more than adequate strength for reliably transferring the power output of an Original Equipment specification engine, it is wise to consider that the driveshafts (propeller shafts) of the more powerful MGC and the MGB GT V8 are of a more stout .095" (2.413mm) wall thickness (BMC Part # ACH 113, Victoria British Part # 5-5916), plus they have a beefier flange, yoke, and U-Joint (Universal Joint) in order to handle the additional stresses of their more powerful engines (Victoria British Part #'s 5-5950, 5-5951, 5-552, respectively). Long-term reliability counts, especially in a street machine!

It should be noted that 18G and 18GA engines with a three-synchro transmission and a Hardy-Spicer Banjo-type rear axle use a 30" (76.2cm) driveshaft (propeller shaft) without an Overdrive unit (BMC Part # AHH 7488, Hardy-Spicer Part # 114H02771, Victoria British Part # 5-5921, MossMotors Part # 268-080). 18G and 18GA engines with a three-synchro transmission with an Overdrive unit and a Hardy-Spicer Banjo-type rear axle use a 31.125" (78.9cm) driveshaft (propeller shaft) (BMC Part # AHH 7487, Hardy-Spicer Part # 114H02776, Victoria British Part # 5-5922, MossMotors Part # 268-090) with an Overdrive unit. It should be noted that all driveshaft (propeller shaft) length measurements should always be taken Flange to Flange with the driveshaft (propeller shaft) fully compressed in length. Be aware that the protective rubber boot will attempt to extend the driveshaft (propeller shaft).

18GB engines with a three-synchro transmission without an Overdrive unit and a Salisbury tube-type rear axle use a 31.125" (78.9 cm) driveshaft (propeller shaft) (BMC Part # AHH 7487, Hardy-Spicer Part # 114H02776, Victoria British Part # 5-5922, MossMotors Part # 268-090). 18GB engines with a three-synchro transmission that are equipped with a D-type Overdrive unit and a Salisbury tube-type rear axle use a 32" (81.3cm) driveshaft (propeller shaft) (BMC Part # AHH 7486, Hardy-Spicer Part # 114H02779, MossMotors Part # 268-100, Victoria British Part # 5-5924).

All 18GD and later engines with the four-synchro transmission use the same 31.125” (78.9 cm) driveshaft (propeller shaft) (BMC Part # AHH 7487, Hardy-Spicer Part # 114H02776, MossMotors Part # 268-090, Victoria British Part # 5-5922) for both Overdrive and Non-Overdrive applications when used with the Salisbury tube-type rear axle (BMC Part # BTB 1106 for steel wheels, BMC Part # BTB 1107 for wire wheels).

However, the four-synchro transmission uses a 31.125” (76.2 cm) driveshaft (propeller shaft) (BMC Part # AHH 7487, Hardy-Spicer Part # 114H02776, Moss Motors Part # 268-080, Victoria British Part # 5-5921) when used with a Hardy-Spicer Banjo-type rear axle.

Before you start pouring power through your drivetrain, it would be prudent to remove the driveshaft (propeller shaft) and subject it to a careful examination. In all probability, this assembly is original to the car and its maintenance is likely to have been neglected. While the driveshaft (propeller shaft) has been balanced to fine limits, it is very important that you mark all the component parts of the driveshaft (propeller shaft) along a common line, i.e., all four yokes, in order to ensure these can be reassembled in their original alignments. Reassembling either pair of yokes 180° out of phase, or getting the gearbox end 180° out of phase with respect to the axle end, i.e., at the sliding joint, will upset the balance of the driveshaft (propeller shaft).

Be aware that that because there is not enough clearance for a true hex head, the machine bolts on the driven flange at the transmission end of the driveshaft (propeller shaft) of the four-synchro transmissions have very peculiar semi-rounded (D-type) heads. They are a tight fit in the drive flange, and they go in from front to back. They were designed so that you do not need a wrench on the bolt head when tightening their nuts. You will have to remove the drive flange from the rear of the transmission in order to replace the machine bolts, but this is not a big job. This is only the case for the transmission flange, not for the driven flange of the differential. Moss Motors and others sell the correct machine bolts to do the job.

Unscrew the dust cap from its sleeve, and then slide the sleeve off of the driveshaft (propeller shaft). Carefully remove both the steel washer and the cork washer. Next, remove the snap rings that secure the bearing races in the U-joints (Universal Joints). Should any of them seem to be stuck in their grooves, lightly tap the end of its bearing race inward with a wooden dowel rod in order to relieve the pressure on the snap ring. Remove the grease zerks (grease nipples) from the U-Joints (Universal Joints). s as well as from the driveshaft (propeller shaft). Now, tap the radius of the yoke arm with a light hammer in order to loosen the bearings. They should slide out, but if they are stuck in place, use a light hammer and a flat-nosed punch bearing against the shoulders of the races in order to gently tap them out. Take care not to distort the race or damage the needle-roller bearings within it. Once they start to move, turn the yoke over and, in order to avoid losing any of the needle-roller bearings, hold the bearing in a vertical position and pull the bearing out from below with your fingers. Place the trunnion onto wooden blocks and tap the top lug of the flange in order to remove its bearing races. Finally, remove both the gaskets and their retainers from the journal spiders.

Clean all of the components so that they can be carefully inspected. The cleaner, the better. The threads of both the dust cap and the sleeve should be made free of all contaminants by using solvent and a soft nylon toothbrush. Check the splines of both the sleeve and of the shaft extension for indentation or signs of excessive wear. The grooves should have a smooth, almost polished appearance. Secure the yoke of the sleeve in a vise and fit the shaft back onto the dry, ungreased sleeve. Twist the shaft in order to check for sideplay in the splines. There should be very little. Next, inspect the bearing races and their journals for wear. Carefully examine the faces of the flanges for any signs of cracking, as well as the holes in the yokes and the flanges for any signs of cracking or ovality. If you see any of these problems, then these components must be replaced. Ensure that the bearing races are a light, yet tight driving fit in their yokes. If they are not, they must be replaced with new ones.

Once all of the components are in satisfactory condition, you can reassemble the driveshaft (propeller shaft). In order to keep moisture away from the needle-roller bearings, apply a coating of gasket sealer to the gasket retaining shoulders on the journal spiders, then use a hollow drift in order to refit the retainers, and then fit the gaskets. Smear the walls of the races with chilled grease in order to retain the needle-roller bearings in place, insert the needle-roller bearings, and then fill the races with grease.

Be aware that the amount of torque necessary for the flange nut that secures the driveshaft flange at the rear of the transmission varies with the type of transmission:

Manual transmission without Overdrive unit: 150 Ft lbs
Manual transmission with D-type Overdrive unit: 100 to 130 Ft lbs
Manual transmission with LH-type Overdrive unit: 55 to 60 Ft lbs

At this point the U-joints (Universal joints) can be replaced. First, in order to assist in future reinstallation, and because the driveshaft (propeller shaft) is balanced as a unit, mark both of the yoke flanges as well as the transmission and differential flanges so that you will be able to realign them in their original positions when reassembling. This will help to preserve the balance of the driveshaft (propeller shaft). The U-Joints (Universal Joints) selected by the engineers to grace the drivetrain of the MGB (BMC Part # GUJ 115) were made by Hardy-Spicer, complete with their characteristic bent-wire snap rings (retaining clips). With the transmission in gear, remove the nuts, machine washers, and their machine bolts from both the transmission and the differential flanges. Undo the flange bolts on the rear flange first. You may have to put the transmission in neutral and turn the driveshaft in order to gain access to some of the flange bolts. Use a large flat tip screwdriver to pry the rear half of the driveshaft (propeller shaft) forward towards the transmission. It should slide forward enough on the splines of the driveshaft (propeller shaft) to drop down clear and free of the pinion flange, and then lower it. Do not try to pull the driveshaft (propeller shaft) out by its rear end, because it may separate at the sliding joint in its middle. Instead, put one hand on the forward part of the shaft and one on the rear and guide it out. Remove the driveshaft from the car, clean it thoroughly, and put it on your workbench.

Once you have the driveshaft (propeller shaft) clear of the car, spray the outer needle-roller bearing caps (cups) of all the U-Joints (Universal Joints) with a good quality penetrating oil such as PB Blaster in order to loosen any rust that will interfere with removing them. Remove the snap rings (retaining clips) on all four caps. The best method for removing these is to use snap-ring pliers, although a pair of long-nosed pliers may prove to be adequate for the task. Most U-Joints (Universal Joints) come already supplied with new snap rings (retaining clips), so there is no need to be gentle in removing the old ones. While a hydraulic press is ideal for driving the needle-roller bearing caps (cups) out of the ears of the yoke, you should be able to manage to force them out using a husky bench vise or a large C-clamp. Or, you can press the bearing caps (cups) out. Place a socket that is larger than the Outside Diameter (O.D.) of the bearing cup against one ear to accept the bearing cup. Then, with a vise or a large C-clamp, put a smaller socket on the other bearing cup to push it through the hole in the ear. Now, tighten the vise or C-clamp in order to force the bearing cups out through the yoke. Once they are removed, take a little extra time to clean out the retainer grooves in the yokes so that reassembly will be a simpler, more straightforward affair. Note that the grease tends to hold the needle-roller bearings inside of the bearing cap (cup). Make sure the new bearing caps have grease to hold the needle-roller bearings in place during reassembly. When it comes to installing the new joint, first make sure that the bearing cap (cup) is packed with grease. The bearing caps (cups) of the needle-roller bearing should be filled to about one-half of the needle-roller bearing length with the proper grease (usually SAE 140).

Note that for easiest access to the grease fittings of the spider journals, the U-Joints (Universal Joints) should always be installed so that when the axle is hanging at its lowest point, the grease fittings point 180º away from each other to the widest, most open side. This would be pointing upwards towards the front of the car on the front U-Joint (Universal Joint) and pointing downwards towards the rear of the car on the rear U-Joint (Universal Joint). Some U-Joints (Universal Joints) have the grease fitting in the plane of the four spider shafts. These are inappropriate and they should be returned to the vendor. The most common and correct U-Joints (Universal Joints) have the fitting out of this plane, i.e., sticking out toward one side. There are four possible ways to install the in-plane joint, and a total of eight for the out-of-plane one, but only one way is correct. The yoke on the driveshaft is asymmetric, having a forged-in depression at only one point where the "ear" blends into the shaft. This depression is for the nozzle of your grease gun. You must use the out-of-plane type U-Joint (Universal Joint); the fitting must face toward the shaft center, not toward the flange. This eliminates four of the eight possible positions; and the fitting must be located in the depression, which eliminates three more. Now the grease gun will fit. If the U-Joints (Universal Joints) are the in-plane type, you can usually remove the existing fitting (or plug) and install a long type fitting to grease it, but it is no fun. Do not leave the long fitting in place, as it will be broken off in operation! Do not make the mistake of installing Borg Warner's version of the MGB U-Joint (Universal Joint) that comes complete with new clips. Why? Because it has an extended grease zerk (grease nipple) (about ¾" - 1" long) that makes for ease of lubrication, but the grease zerk (grease nipple) is so long that at the car's first encounter with a severe road bump or dip it impacts against the yoke flange and is swiftly snapped off flush at the threads. Instead use either the Hardy-Spicer HS194 U-Joint (Universal Joint), or either the Spicer 5-101X U-Joint (Universal Joint), or the

Spicer 5-102X U-Joint (Universal Joint). It should be noted that the last-mentioned U-Joint (Universal Joint) is a sealed unit that has no provision for injecting new grease into it after it has been installed. This feature is preferred by fleet owners who desire to reduce the expenses involved with required maintenance and who dispose of their vehicles after a specified number of miles have accumulated on them, but is an obvious disadvantage for owners who are concerned with maximizing the lifespan of the components of their cars.

Prior to installing the U-Joint (Universal Joint), clean the yokes and smear a thin coat of chassis grease in the bores. Now, making sure that the grease zerks (grease nipples) are facing away from the yoke flanges, insert the journal spiders into the flange yokes. Using a soft drift such as a dowel rod in order to protect their races from distorting, fit the bearing caps (cups) onto their journals on the journal spiders, and then into the yokes.

While forcing the bearing caps (cups) into place, take care to not dislodge the needle-roller bearings or jam them at an angle. When you install a new snap ring (circlip), it is a good idea to first tap the bearing cap (cup) down just far enough to clear the snap ring (circlip) groove, and then install the snap ring (circlip). Once you have the first snap ring (circlip) installed, turn the driveshaft over and install the opposite ring in the same manner, tapping the bearing cap (cup) down first. That way, you can be sure the snap ring groove will be fully exposed. Once you have the first snap ring installed, turn the driveshaft over and install the opposite ring in the same manner, tapping the bearing cap (cup) down first. When you have the two opposite snap rings installed this way, you may find that the U-Joint (Universal Joint) does not rotate freely. To fix this, turn the driveshaft over again and tap on the center of the U-Joint (Universal Joint) in order to force the first bearing cap (cup) to seat against the snap ring (circlip). When you are sure you have the correct position, carefully remove the upper and lower caps on the U-Joint (Universal Joint). Make sure none of the needle-roller bearings stick to the U-Joint (Universal Joint), and that they are neatly aligned around the inside of the cap (cup) by gently twisting the cap (cup) and feeling for any resistance. Place the U-Joint (Universal Joint) in the yoke from the inside, and then tap the lower cap (cup) into place from the outside. Turn the driveshaft over and tap the other cap (cup) into place. Always be aware that the needle-roller bearings are free to move inside of the cap, and position the U-Joint (Universal Joint) to hold them in place when you are tapping the needle-roller bearing caps (cups) into the yoke.

Install a new cork gasket (BMC Part # 7H 3880), the steel washer, and the dust cap over the ungreased splined section of the sleeve of the driveshaft, then grease the splines on both the shaft and inside of the sleeve. Take care to align the arrows found on both the sleeve and the splined section of the shaft so that the U-joints (Universal joints) will be properly operating aligned parallel to each other in a single plane, and then slide them together. These arrows can sometimes be difficult to locate. One arrow is stamped onto the female sliding sleeve, which is a machined forging. The other arrow is stamped onto the tube of the main shaft just beyond the weld, not onto the male spline shaft. Finally, fit the washer into the dust cap, and then screw the dust cap tightly onto the sleeve.

All that remains now is to clean away any excess grease, and then you can reinstall the driveshaft (propeller shaft) onto the flanges of the transmission and the differential. Do not forget to install the lock washers! It should be noted that whenever a driveshaft (propeller shaft) is installed, the axis of the flange yokes on both the transmission and the differential must be aligned parallel to each other in order to avoid producing unequalized thrust forces that will damage the bearings of the U-Joint (Universal Joint) as well as result in driveline vibration.

The Rear Axle and Differential

The final ingredient in the recipe for putting more power on the ground is the rear axle and differential assembly. During its lifetime the MGB was equipped with two different rear axle / differential assemblies: The Hardy-Spicer banjo-type three-quarter floating axle and the Salisbury tube-type fully floating axle. A three-quarter floating axle has its outer bearing positioned between the wheel hub and the axle, thus eliminating the bending loads of the car's weight, while the fully floating type axle has an additional bearing between the hub and axle to handle the sidethrust of heavy cornering loads. In the case of an MGB powered by a B Series engine, either axle is quite adequate for street use, although the Salisbury tube-type rear axle assembly is both notably quieter and capable of handling heavier loadings, although it is of considerably heavier weight.

The Salisbury tube-type rear axle did not become standardized on all models until the advent of the more powerful MGC in 1968. Until that point, all Roadster models were equipped with the Hardy-Spicer Banjo-type rear axle, and the GT models were all equipped with the Salisbury tube-type rear axle. The principle reason for the higher noise level of the Hardy-Spicer banjo-type rear axle was the profile of the teeth of the gears. They transmit power more efficiently, but are noisier (much as straight-cut gears are more efficient, but noisier, than helically-cut gears are). Another design feature that also makes a difference in noise level is the manner of construction of the rear axle housings. The Salisbury tube-type axle housing has a cast iron differential housing while the Hardy-Spicer banjo-type axle housing is fabricated by welding steel stampings together. Although much heavier, cast iron is a much better sound deadener. The design of the Salisbury tube-type rear axle is not only quieter, but can be inexpensively made to accommodate different width needs by simply lengthening or shortening both the axle tubes and their corresponding enclosed halfshafts (quartershafts). This is a much lower cost solution than the labor-intensive method of fabricating different sheet metal axle housings for each needed width rear axle, plus the additional cost of inventorying and storing multiple sheet metal stampings until they are needed for assembly and welding in jigs. It is only necessary to make sure that the differential mechanism is strong enough to handle the power and weight of the biggest intended vehicle, and the payoff is as-needed production of an axle that lasts practically forever in lightweight applications (as in an MGB). The downside is that while such a generic rear axle design will withstand more power, it will be heavier, making for more unsprung mass for the rear suspension to deal with. In the case of the Salisbury tube-type rear axle which weighs a rather ponderous 175 Lbs against the 115 Lbs of the Hardy-Spicer banjo-type rear axle, this additional penalty in terms of unsprung weight is 60 Lbs.

It should be noted that both the Hardy-Spicer banjo-type rear axles and the Salisbury tube-type rear axles each came in two different widths. This variation was the result of the need to accommodate the fitting of wire wheels and steel wheels. Hardy-Spicer banjo-type rear axles for wire wheels measure 44.5” across, while the rear axles for steel disc wheels measure 46.25“ across. Salisbury tube-type rear axles for wire wheels measure 47” across, while the rear axles for steel disc wheels measure 48.5“ across. This measurement should be taken between the outer rings that have the four holes in the faces on the bearing caps that the back plate bolts onto, i.e., without the backplates installed.

As standard Original Equipment, the differential mechanisms of both the Hardy-Spicer banjo-type and Salisbury tube-type rear axles found on the MGB equipped with a manual transmission both used crownwheel and pinion gearsets that produced the same 3.909:1 final drive ratio, although they were also available from the factory with optional crownwheel and pinion gearsets that produced different final drive ratios that were meant to be appropriate for special applications. Over the years, MG owners with special requirements have also used other ratio crownwheel and pinion gearsets from other model cars found in the BMC stable. The number of respective teeth of both the crownwheel gear and the pinion gear of the Hardy-Spicer banjo-type rear axle are often stamped onto the rear face of the differential casing to the left of the differential cover, i.e., 43 11. In the case of the Salisbury tube-type rear axles, the number of teeth of both the crownwheel gear and the pinion gear are often stamped onto the right hand side of the top of the differential casing. At present, many of these special-purpose Original Equipment specification crownwheel and pinion gearsets are still available from several sources.

Newly-manufactured crownwheel and pinion gearsets for the Hardy-Spicer banjo-type rear axle of the MGB MKI Roadster are currently available in:

3.7:1 (available from Autogear, Part # CWP033)

3.909:1 (available from Moss Motors, Part # 267-165; and Autogear, Part # CWP034)

4.1:1 (available from SC Parts Group and Cambridge Motor Sports)

4.3:1 (available from SC Parts Group and Cambridge Motor Sports)

4.55:1 (available from Moss Motors, Part #267-185) and Cambridge Motor Sports

4.875:1 (available from Victoria British, Part # 5-1049)

Newly-manufactured crownwheel and pinion gearsets for the Salisbury tube-type rear axle of the MGB MKII Roadster and of all MGB GTs are available in:

3.071:1 (available from Autogear, Part # CWP032).

3.7:1 (available from Autogear, Part # CWP030)

3.909:1 (available from Autogear, Part # CWP031)

Be aware that due to the fact that they must be precisely located in exactly the positions that they were in on the gear machines that made them, you cannot interchange Crownwheel and Pinion gearsets (or mix gears of different sets) from one housing to another of any particular design without going through the whole shimming set-up process, otherwise they will gnarl and gnash themselves into scrap metal. The gears are marked to tell you what that position was as compared to the design point. The gears are sets too: the sets are paired in manufacture, and cannot be satisfactorily interchanged, except by dumb luck and usually with a lot of noise and early failure.

It should be noted that in order to maintain speedometer accuracy when using a nonstandard final drive ratio it is necessary to have the speedometer recalibrated. This service is available from Nisonger Automotive. They have a website that can be found at http://www.nisonger.com/ .

The Hardy-Spicer banjo-type rear axle has its differential mechanism assembled into a carrier that is separate from the axle housing and is bolted onto its front, its hubs being press fitted onto the halfshafts (quartershafts). Aside from it being 60 pounds lighter than the Salisbury tube-type rear axle, it is this design feature of easy accessibility that is one of the reasons that this rear axle assembly is so popular with racers: They can quickly remove the carrier assembly without removing the entire rear axle from the car in order to change the crownwheel and pinion gearset. The Salisbury tube-type rear axle has its differential mechanism built into a carrier that fits directly into the axle casing that is sealed by a cover plate. Its hubs are bolted onto the halfshafts (quartershafts).

If you are considering replacing your Salisbury tube-type rear axle with a Hardy-Spicer banjo-type rear axle in order to shed 60 pounds of unsprung weight, be aware that their brake mechanisms differ somewhat in terms of their parts. The only difference between the brake shoes is that the same ones used with the Hardy-Spicer banjo-type rear axle were also used initially on the Salisbury tube-type rear axle, but a redesigned brake shoe was subsequently introduced. However, it appears that only the friction lining material was changed.

Slips in manufacturing quality control have resulted in some Hardy-Spicer banjo-type rear axles being found to have a condition of Toe-out. Toe-in is desirable and can be mechanically adjusted by a skilled craftsman with the right tools, such as Dave Headley of

Fab-Tec. Dave Headley has a website at http://www.fast-mg.com/index.htm. About ½º of negative camber can be added at the same time, which will measurably improve grip. Be aware that more that ½º of Toe-out can cause axle spline distress.

A Quaife Engineering torque-biasing limited-slip differential will assure that the extra power safely gets to the pavement, allowing you to go to full power earlier while traversing a curve. With a normal open differential, fitted as standard on most cars, much precious power is wasted through wheel spin under acceleration. This happens because the open differential shifts power to the wheel with less grip. That is, along the path of least resistance. The Quaife differential, however, does just the opposite. It senses which wheel has the better grip, and automatically biases the power to that wheel. It does this smoothly and constantly, and without ever completely removing power from the other wheel. In addition, it automatically sends a proportionally greater amount of power to whichever wheel is turning at a faster rate, such as on the outside of a curve, thus eliminating torque steer. This allows the driver to begin accelerating earlier, exiting the corner at a higher speed. Because it behaves like an open differential during ordinary driving, the driver will have trouble telling that it is there until pushing the car towards its limits. Furthermore, during heavy braking into a bend when the car is pushed into a potential oversteer attitude, the Quaife differential's ability to bias the torque split in favor of the wheel with the most grip enables easy control and stabilization of the car with minimal steering input. This helps to reduce oversteer during hard braking.

The Quaife design has no plates or clutches to wear out and need costly and regular replacement. Instead, it makes use of sets of floating helical gear pinions meshing in order to provide normal speed differential action, thus providing constant and infinitely variable drive. Disc spring washers pre-load the gear sets so that when a difference in traction occurs, torque bias is generated by the axial and radial thrusts of the pinions in their pockets. The resultant friction force transmits power to its secondary sun gear, and thus to the appropriate wheel. The percentage of torque bias is increased or decreased as needed by varying the helix and pressure angles of the gear teeth. While it is progressive in action, it never locks, and needs both wheels in contact with the ground in order to function properly. There is no maintenance required to the unit, so once installed the unit can be forgotten, apart from the better road responses. Because it behaves like an open differential during ordinary driving, many drivers have trouble telling it is there until pushing the cars towards its limits. It is reputedly hard wearing and gentle in operation. However, its racing applications are limited by its limited performance adjustment. This limited slip differential is available for the Hardy-Spicer Banjo-type rear axle (Quaife Part # QDF15K), and also for the Salisbury tube-type rear axle (Quaife Part # QDF6KB) with a choice of two differential case flanges that permit the fitting of either the 3.909:1 and the 3.7:1 crownwheel and pinion gearsets or the 3.307:1 and the 3.071:1 crownwheel and pinion gearsets. Quaife has websites that can be found at both http://www.quaifeamerica.com/ (USA) and http://www.quaife.co.uk/ (UK).

However, there are drivers who wish to install a tunable clutch-plate-type limited-slip differential in their cars for track days. Clutch-plate-type limited-slip differentials work by clutching the differential gears to the differential cage. The clutch unit will have a relatively small amount of grip when no power is being transmitted, allowing the gears to move within the cage and so the whole unit operates more-or less as a standard open type differential. When full power is applied, these clutches become heavily loaded and the amount of clutch grip increases substantially, effectively locking up the entire internal differential cage assembly and making the car behave as though it has a solid axle from wheel to wheel. At part-throttle openings, the effect will be somewhere between these two extremes. If you think about all this, you will see that the term “limited slip” is an accurate description of its operation. The limited-slip differential will accentuate a slight understeer under power, and oversteer when the throttle is closed. This provides the opportunity to “fine-tune” your line through a corner with the use of the accelerator, which is why they are popular with racers. Unfortunately, the engagement characteristics of a clutch-type limited-slip differential can be a bit hazardous when traction conditions become poor, such as on wet or icy roads. For those drivers who do not mind the extra maintenance involved with such a specialized type of differential, Denis Welch Motorsport offers one for use in the Salisbury tube-type rear axle. Originally designed for use in race cars with heavily uprated engines, it offers more than enough strength for use in any vehicle equipped with a BMC B Series engine. It offers the option of either a 30º / 60º ramp angle or a 45º / 45º ramp angle and can be had in versions for use with either 3.909:1 and 3.7:1 (Part# MCRAX 100A) crownwheel and pinion gearsets, or for 3.307:1 and 3.071:1 (Part# MCRAX 100B) crownwheel and pinion gearsets. Denis Welch Motorsport has a website at http://www.bighealey.co.uk/ .

It is always possible that you may choose to retain your present rear axle. If this is the case, it should be inspected and reconditioned to be sure of its reliability. 1968-1980 MGB MKII Roadsters and virtually all MGB GTs were fitted with the Salisbury tube-type rear axle. Because rear axles are sometimes replaced, an easy way of telling what is fitted to your car is to look at the rear of the axle housing. If there is a stamped sheet metal cover bolted onto the rear face of the differential housing, then your car has a Salisbury tube-type rear axle. If there are no machine bolts on the rear face but instead a series of studs and nuts at the front face of the differential, then your car has the earlier Hardy-Spicer banjo-type rear axle.

Differential Backlash In The Hardy-Spicer Banjo-Type Rear Axle

Setting the correct amount of backlash in the mechanism of the Hardy-Spicer banjo-type rear axle is a rather straightforward affair if proper procedures are followed. First, remove the differential carrier from the axle housing and carefully remove all of the old gasket material.

The input shaft of the pinion gear of the differential is mounted in two bearings that are separated by a spacer. A thrust washer behind the rear bearing inner race is used in order to determine the location of the pinion gear. Shims behind the inner race of the front bearing are used in order to set the preload of the front bearing. Assemble it initially using a thrust washer of a mid-range thickness. Because preload is not critical during initial setup, use enough shims in front to make near zero clearance for the bearings. You can even omit the front shims during the initial setup if you tighten the front nut just enough to remove bearing clearance but not apply too much preload.

There are also shims between the inner races of the carrier bearings and the differential cage. Preload not being critical for initial setup, so set this up initially with equal thickness shims on each side in order to produce a clearance of near zero. There should be some working clearance in the gear teeth. If you find either interference fit or too little clearance for the teeth, adjust the shims between the inner races of the carrier bearings and the differential cage in order to move the crownwheel gear sideways in order to increase tooth clearance. Now, measure the backlash at the input flange. If the backlash is excessive, then adjust the shims in order to adjust the backlash to within the specified range of (.008" +/- .003") (.2032mm +/- .0762mm), that is, 0.005" to 0.011" (.127mm to .2794mm). That is, .008" / .2032mm nominal.

Clean the gears with solvent, blow them dry, and then apply machinist's bluing to the teeth of the crownwheel gear. At 90^o positions on the crownwheel gear, mark two or three teeth only. While you rotate the pinion gear (clockwise for the normal input direction), apply resistance to the crownwheel gear with the palm of your hand, making marks in the bluing on the gear teeth in order to indicate the contact point. The contact points should be of about equal position all of the way around, and the backlash should also be about equal all the way around, thus verifying that the crownwheel gear will run in a flat plane. The maximum permissible run-out is .002" (.508mm). If it is not running in a flat plane, you will need to correct that before continuing to the next step. The crownwheel gear bolts must be torqued to 60 Ft-lbs (8.3 Kg-m) prior to bending their locktabs over.

There are shims between the inner races of the side bearings and the differential cage (often referred to as "side shims"). Perform an initial setup with equal shims on each side, enough to make near zero clearance. Preload is not critical for this initial setup. There should be some working clearance in the gear teeth. If it should turn out to have an interference fit or too little clearance for the teeth, adjust the side shims to move the ring gear over slightly in order to get some tooth clearance. Next, measure the backlash of the gears at the input flange. If there is excessive backlash, then adjust side shims in order to reduce the backlash of the gears (or vice versa) to within the specified range of 0.008" +/- 0.003"

The contact points indicated on the bluing should also be about midway along the teeth of the crownwheel gear. If the contact points are incorrect, compensate for this by adjusting the thicknesses of the thrust washers. Be aware that the diameters of these shims are critical. Their Internal Diameter (ID) must match the smaller diameter of the shaft, and their Outside Diameter (OD) must match the front bearing inner race and spacer. When the contact points fall midway along the gear teeth, re-check the backlash of the gears and re-adjust the side shimmings as necessary in order to obtain the correct amount of backlash. The maximum permissible run-out is .002" (.0508mm) which will produce a corresponding backlash of .002" (.0508mm).

With the front nut tightened to specified torque and without the oil seal in place, adjust the front shims for the specified bearing preload of 10 to 12 In-lbs. Remove the front yoke, install the oil seal, reinstall the front yoke, and then retighten the front nut to the specified torque. The preload torque of the pinion bearing should increase by approximately 3 In.-lbs. for the seal contact for a total of 13 to 15 In.-lbs.

At this point, perform a final check on backlash of the gears, and then check the side clearance of the carrier bearings. Adjust the shims between the inner races of the carrier bearings and the differential cage in order to provide .002" preload pinch on each side on the bearings, and then you are finished. If you like, you can leave the bluing on the gear teeth for possible future reference.

Fit the differential onto the gear carrier, then replace the bearing caps and torque them to 65 Ft-lbs (8.99 kg-m). Finally, place a new gasket onto the sealing face of the differential carrier, and then reinstall it onto the axle housing.

After many tens of thousands of miles, wear inside of the differential is inevitable. In the case of the Salisbury tube-type rear axle this wear normally manifests itself as a “clunk” that is both felt and heard initially upon acceleration and upon deceleration. If your MGB has wire wheels, first check that the hubs and splines are not worn, as this can also be the origin of a “clunk.” A diagram showing profiles for various degrees of wear can be seen at http://www.britishwirewheel.com/faq.htm . If you choose to replace the wire wheels, then you should replace the splined hubs as well, as the mounting splines of the originals will have “mated”. Putting new wire wheels onto the worn mounting splines of the original hubs will cause their new mounting splines to wear out prematurely. If your splines and hubs are sound, or if the car is fitted with steel wheels but the clunk persists, then the condition of both the universal joints of the driveshaft and its splines are worth investigating. If these and the driveshaft are sound, then the noise may be due to wear in the thrust washers of the pinion gears of the differential. If care is taken, replacement of these inexpensive thrust washers is quite straightforward; the two different types used each being of one size only. Doing so will help to prolong the service life of the rear axle.

Rear Axle Clonk in Salisbury Tube-Type Rear Axles

As with all work under the car, first remove the battery ground (earth) lead in order to prevent accidental starting. Next, chock the front wheels, and then lift and secure the rear of the car upon axle stands.

While you have the car up on the axle stands, now is the perfect opportunity to pull off the brake drums and check to see if the halfshaft (quartershaft) seals are leaking. You can also take a fast look at the brake linings to see if they are worn down close to the rivets that secure them to their steel shoes. Take the time to grease the U-joints (Universal joints), the driveshaft splines, and the hand brake cable, too. May as well get it all done at the same time while it is up on the axle stands, right? A Halfshaft (sometimes also called a Quartershaft or an Axle Shaft) is the shaft that transmits power from the differential mechanism to the drive wheels. The halfshaft (quartershaft) oil seal is the seal that keeps the oil inside of the axle from leaking out into your brake drums. Look on page 213 of your Bentley manual. The halfshaft (quartershaft) oil seal is #43. Just look and see if you spot any oil leaking or oozing out. The oil will ruin your brake shoes. Spray the brake system with CRC Brakleen and inspect everything carefully. While you have the drums off, you can clean off the rust and paint them with VHT engine paint. Remember, rust is a heat insulator, and it is heat that triggers the out-gassing that is the cause of brake fade.

Once the rear brakes are reassembled, set the hand brake so that the halfshafts (quartershafts) cannot move. That way your measurements will be as accurate as possible. Grip the differential flange that connects to the rear U-Joint (Universal Joint) of the driveshaft and rotate it to take up any freeplay, and then scribe a mark on its edge and a corresponding mark on the axle housing. Next, rotate the flange in the opposite direction and scribe another corresponding mark on the axle housing. If the marks are 4.5mm apart (6° of rotation), you have a like-new differential. If the marks are 8mm apart (10° of rotation), you have a usable differential. If the marks are 10mm apart (13° of rotation), you have a worn differential.

Of course, before you can do anything with the differential you will need to drain the oil out. This potentially simple operation is made more difficult because the British decided to use a ½" BSP square drive drain plug. This is a rather quaint plumbing item. Of course, the British like to do things their own way, so these are not compatible with any American threading system, plus their plug has a square hole in it for the wrench. The tool necessary to remove this charmingly quaint drain plug from the bottom of the differential housing is a 7/16" extension for a square pipe plug. Do not bother trying to use a hexagonal Allen wrench. You will just end up with a ruined Allen wrench trying to get the plug out. It is well worth purchasing the correct tool for this job as it makes life so much easier. MAC Tools makes them. I know because that is where I got mine. Sometimes you can get a cheap one from a plumbing supply house. Once you have gotten the drain plug out you have the option of swapping it for an American-made stainless steel ½" BSP plug from a hardware store / plumbers shop. They never rust in place, and to remove them all you need is a simple 3/8" Allen wrench.

Do not be surprised at what you see when you drain the old oil out. It is not unusual for this maintenance task to have been totally neglected by the DPO. The Owner's Manual always said to "Check oil level, and top up if necessary." Not a single word about how often to change the oil. Naturally, this led to neglect. It is entirely possible that the oil in it is the original oil. When you drain it out, it may look and smell like something that oozed up out of the ground, prompting you to expect to see old dinosaur bones floating in it, but do not worry too much about it. The Salisbury tube-type axle is a grossly over-engineered piece of design work, originally intended for use in light trucks and delivery vans. Usually, the only thing that damages it is letting the oil level drop too far. This often happens when the breather on the top of the tube on the passenger side (right above the horizontal bracket) is plugged up with road crud. Air then is trapped inside of the axle, the differential gets hot and causes the air trapped inside to expand, and then the gasket starts to leak as a result of the internal pressure. When the axle cools, air is drawn in through the leaking gasket. The process is repeated every time the car is run until the oil is gone, which usually takes a very, very long time. Once in a blue moon a dedicated garage mechanic will check the level and top it off, so outright failures are unusual. Allow the oil to drain into the container and replace the drain plug securely.

Cleaning the breather is a simple affair, but most DPOs do not even know that it is there on the top of the right side of the axle tube. Just clean around the top of the axle tube with cheap carburetor cleaner so that crud will not get into the threads, unscrew it, and spray it out with carburetor cleaner, carefully clean the threads with an old toothbrush, then put it back in after it dries. Simple. Once that is done you can proceed with the replacement of the old cover gasket.

Getting the handbrake assembly off of the differential is actually very easy. Loosen and remove the self-locking nut that secures the compensating lever to the bracket on the differential. Disassemble, clean, and repaint the hand brake compensating lever mechanism. If it does not work properly, then the rear brakes will not apply equal force. See those two cables that traverse the axle and go out to the brakes on each side? They attach to the brake mechanisms. Look and you will see a clevis pin attaching the cable to each of the levers of the brake adjuster mechanism. Remove the cotter (split) pin that secures the clevis pin, then pull the clevis pin out and set it aside, along with its washer. Clean them both and carefully inspect the shank of the clevis pin for signs of wear. If you see any, it is best to replace this inexpensive part (BMC Part #ACB 8715). When you put it back in, be sure that it is pointing downward with the cotter (split) pin on the bottom. Use a stainless steel cotter (split) pin only.

In order to proceed with the replacement of the thrust washers it is necessary to move one of the halfshafts (quartershafts) by about six inches. Using a pair of pliers remove the cotter (split) pin of the main hub nut and use a 1 5/16" thin-wall socket to remove the nut. If your socket has a cupped mouth, you may need to grind it down in order to prevent it from slipping on the shallow nut. You will need to apply the hand brake in order to prevent the halfshaft (quartershaft) from rotating. Next, loosen the brake shoe adjuster by about three quarters of a turn and remove (wire wheels) the four nuts or (steel wheels) the two countersunk screws that help secure the drum in place. Release the hand brake, then gently tap the hub with a rubber hammer or a block of wood and pull it off of the halfshaft (quartershaft). The coned spacer can also be slid off from the shaft and set aside with it.

While it is possible to carry out the next procedure without disconnecting the braking circuit, it is inadvisable to do so. This is due to the high chance of damaging the brake line, which obviously has very severe safety implications. Therefore, first disconnect the handbrake cable from the lever. Next, remove the brake master cylinder top, place some cling film over the opening, and then replace the cap. It is now possible to remove the rear brake line from the rear wheel cylinder with minimum loss of brake fluid. Once the two are separated, plug the brake line in order to prevent brake fluid loss as well as minimizing the amount of air getting into the hydraulic system. The four machine bolts that secure both the brake backing plate and the end cap of the axle in place can now be loosened and removed. Next, lift the backing plate away from the end of the axle. Finally, remove the oil seal collar, bearing hubcap, and the oil seal from the halfshaft (quartershaft). Inspect the oil seal for damage and replace if necessary. Be sure that its lip is facing inward when you do so. Look for a black wear ring near one end of the collar. If you should find such a wear spot, always use a new collar. In order to prevent leakage, be sure to smear some sealant under the square collar when it is reinstalled.

Ideally, a slide hammer can be used to release the bearing and half shaft out of the axle housing. If this tool is not available and the rear axle assembly has been removed from the car, replace the hub and retaining nut back on the half shaft and, using a block of wood in order to protect the hub, hit it with a club hammer on the opposite side until the shaft releases itself. If the rear axle assembly is still on the car, another, better approach, is available. The books show fitting an old brake drum and beating the axle out of the housing, there is a much easier way: once you have removed the brake assembly, dust shield, and the axle bearing cover, replace the flange just far enough to thread on the large flange nut. Place a couple of large sockets between the flange and the differential housing, and then tighten the flange nut. A few turns and the bearing slides right out, no beating, no cussing. Once the bearing is out, set both it and its inner spacer aside. The shaft can now be pulled out about six inches by hand. Be sure to have the foresight to repack the bearing with fresh bearing grease before reinstallation.

Use a wire brush and cheap carburetor cleaner to thoroughly degrease and clean up the area around the rear differential cover plate prior to removing it. Make sure you clean both the bolt face and surrounding area of the axle casing in order to ensure that no dirt falls into the differential. Once the area is clean, release all of the securing bolts making a mental note as to where the handbrake pivot point is attached and the location of the top clips for securing the brake lines (pipes). rear differential cover plate can now be gently pulled away.

You are now able to see the components of the mysterious differential mechanism. Clean everything with cheap carburetor cleaner so that you can inspect the gear teeth. Inspect the crownwheel (the large gear on the left of the differential cage) for any wear lines, cracks or chipping. If there is any visible damage you will need to seriously consider replacing the entire rear axle unit with a used one as that would be less expensive than replacing the crownwheel and its matching pinion gear.

First, rotate the differential cage around until it reveals the 3/16" X 1 ½" roll pin that holds in place the mainshaft (third motion shaft) of the top and bottom pinion gears, and then drift the roll pin out. After the roll pin is removed, turn the differential cage again until the other end of the main pin is facing you. You can now start to drift the main pin out of the carrier. Take care not to push the pin too far through as it is very easy to jam the pin against the casing of the axle with no way of pulling it back, which would render your axle useless! Observe when the pin has started to move and as soon as it does, rotate the differential cage around again so that the pin can be pulled out from the top. Place a thin rod through the roll pin hole in the main pin and use this to pull the main pin completely out of the differential cage.

Once again, slowly and carefully rotate the differential cage and watch the top and bottom gears move away from each other. One will come out at the front while the other tries to fall out at the back. Put your hand in to remove one along with its worn thrust washer and place them on a clean cloth in the same orientation as they were in when they came out of the differential housing. The other gear and its thrust washer should be removed in the same way. Remember that the gears have worn into matched pairs, so take care to keep the pairs separate from each other.

Now that the top and bottom pinion gears have been removed, the other two gears can be removed one at a time, their worn fiber washers removed and replaced with new fiber washers, and then they can be reinstalled.

The top and bottom pinion gears now need to be reinstalled. Should they prove to be badly worn, the easiest way to do this is to turn the differential cage until you can get a hand on either side of the carrier. Then place the two pinion gears opposite each other, hold them in place and have an assistant slowly rotate the differential cage again. You are aiming to be in a position to look down the hole where the main locating pin secures the gears in place and see all the way through. If you are a tooth out with the alignment one of the pinion gears will not line up. Once the pinion gears are in the correct position, slide the new metal thrust washers into place between the carrier and pinion gears. Once all of the pinion gears and washers are positioned correctly, drift the main pin back into position. Hold the new roll pin in a vise and open the Inside Diameter (I.D.) to 7/64" with a drill. This is necessary as the ends are slightly crimped as a result of the pin being cut to length. If a bench vise is not available, hold the pin vertically in vise grips (mole grips) and drill downwards onto a block of wood. Secure the assembly in place with the new roll pin. Insert a 7/64” X 2” cotter (split) pin through the roll pin in order to ensure that it will not come out. In addition, this extra thickness dramatically increases the shear strength of the roll pin and will prevent the pinion pin from rotating and breaking loose.

The halfshaft (quartershaft) can now be felt back into position and, making sure that the mating surfaces are clean, install the axle end cap and the back plate. Use the four machine bolts to pull the whole assembly together slowly by tightening opposite machine bolts a little at a time. Replace the coned spacer, hub and the castellated nut, followed by the brake drum, which needs to be secured with the two Philips screws. When reinstalling the splined hubs of a wire wheeled car, note that the hub with a stamped “RH” goes on the right halfshaft (quartershaft) and that the hub with a stamped “LH” goes on the left halfshaft (quartershaft). The mounting spinners for the wheels are also so marked. This is so that the mounting threads of the hubs will tighten the spinners when the car is moving forward.

The handbrake lever and its cable can now be attached and the brake pipe screwed back into the wheel cylinder. Release the cling film from the brake master cylinder and bleed the brakes. You can get any residual air bubbles in the brake lines (pipes) loose by tapping on the lines with the handle of a screwdriver. With luck you may only have to bleed the side you have removed the brake pipe from. However, if the brake pedal feels spongy, then bleed the whole system. Reset the rear shoes by using the adjuster on the back plate.

The Rear Axle Pinion Oil Seal

While you still have the car up on the axle stands, you might also want to inspect the pinion oil seal for signs of leakage and then decide if you want to replace it (National Part # 224470, Timken Part # 224470). If you choose to do this, be aware that although this can be a mechanically risky undertaking, there should be no problems if proper procedures are adhered to.

Do not be surprised if, upon inspection, you discover a leaking oil seal on the rear axle pinion. However, you will find that a browse through the workshop manual immediately creates apprehension, what with all of the special tools that are required for this and for that, plus the preload for the other (measured in pounds per inch, no less). Small wonder that the task appears to be daunting. Then, of course, there is also the fear that you will crush that almost unobtainable thing referred to as the collapsible spacer (Moss Motors Part# 125-615). Actually, there is a way to replace the pinion oil seal with minimal risk.

The first thing that you will need to do is to purchase the new pinion oil seal, and then mark the flanges of both the driveshaft and the pinion to ensure correct reassembly. Disconnect the driveshaft, and then remove the four nuts/bolts that attach the rear flange of the propeller shaft to the pinion flange. You will then see the nose of the flange. Recessed into it is a 1 1/8" AF Nyloc nut. This is screwed onto the input shaft of the pinion gear. Mark the nut in relation to the input pinion pin with an awl or a sharp nail.

At this point, it is necessary to prevent rotation of the flange as we loosen the nut. This can be done either by locking the handbrake very hard, or by lowering the wheels onto the ground. You can, of course, use the very expensive flange tool. Once you have the pinion marked in relation to the nut, then you have to loosen the nut. You can use a bar placed in between two old bolts positioned in the flange bolt holes and turn the nut against the force of the bar.

Now, having started to loosen the nut (it will initially be very tight), count the exact number of turns required to remove it from the pinion shaft. It will be about 11 or so turns but the precise number of revolutions is required. Saying “10 and a bit”, or “nearly 12 ½”, is not going to be sufficiently accurate. If it is 10 ¼, then that is precisely what is required. Once it has been removed, write down the exact number of turns that was required to remove it from the pinion shaft. Mark the socket with a blob of your wife’s nail polish (she won’t mind you using it for such a noble purpose) on the outside in line with the mark on the nut so that you can see the revolutions of the nut by means of watching the socket.

Having removed the nut, with a sharp tap from behind with a mallet or hammer, the flange will come off, along with the dish-shaped cover. The leaking pinion oil seal will now be exposed, and with the aid of a stout screwdriver, you will be able to pry out the old pinion oil seal. Place a thin smear of red rubber grease onto both the sealing lip of the new pinion oil seal and its seating area on the shaft, and then wipe the pinion sealing area clean, along with the axle case where the new seal will fit. Press in the new pinion oil seal, and then tap it into place with a mallet over a block of wood until the rim of the new pinion oil seal is flush with the edge of the axle case. Clean the seal rubbing area, and then replace the flange.

Put the nut on and screw it up the exact number of turns that were required to remove it. It will tighten up somewhat towards the last half to three quarters of a turn as the preload increases. Just persevere until the scribed marks align. What you have done is to tighten the nut to a degree of rotation as opposed to a torque figure, and, if everything was in proper order before it was removed, then it will be the same now that it has been tightened. Theoretically, the preload on the bearings and crushable spacer should be identical. All that will have been altered will be the oil seal, and that has no bearing on the collapsible spacer at all.

The Halfshaft Oil Seals

If, in the process of attending to your rear brakes, you make the discovery that you need to replace the oil seal of a halfshaft (quartershaft), be advised that replacing one is not a major undertaking. The rear hub should be clamped between two tapered collars that will prevent it from moving relative to the halfshaft (quartershaft) when the hub castle nut is tightened to 150 Ft-lb. Both tapered washers have machined splits in them, two in the oil seal collar and one in the castle nut collar, so that they can be compacted into the tapered bore and lock to the halfshaft (quartershaft). The inner tapered collar is what the oil seal runs on, so if this is grooved you will need to replace it as well as the axle seal. The outer tapered collar is a split washer that is located under the nut. It is necessary for there to be some clearance, i.e., “play” in the splines. Otherwise, you would have to press-fit the hub onto the halfshaft (quartershaft). In practice, it is an easy sliding fit, much the same as centre-lock wheels sliding onto their splined hubs.

Breaking the tapered collar between the hub and the halfshaft (quartershaft) is usually the most difficult part of the job. A screwdriver in the collar split should force it to pop out. Afterwards, use a gear puller in order to remove the hub. It may be very tight on the halfshaft (quartershaft), but it will pull off. With that removed, you may be able to pry the old oil seal out of the end cap as is, but make sure you do not damage the inner taper that it runs on. Failing that, you will have to remove the backplate (four machine bolts), and then lightly replace the hub and tap on the back of it with a mallet to pull the halfshaft (quartershaft), bearing, and end-cap part of the way out. Then you can tap the end-cap off of the bearing so that you can replace the oil seal. Before installing the new oil seal, grease the oil seal and its lip, even though it runs in gear oil.

The hub tapered collars can wear from running loose, and often many are. That in turn results in wear of the splines. However, it should still lock up when tightened. If not, usually the nut is bottomed out on the thread end runout / shoulder. It is sometimes necessary to fit a washer between the nut and the tapered collar in order to cure this. Apply a thin coat of Permatex #2 onto the inner collar (not onto the tapered bit) that is between the collar and the bearing in order to prevent oil from leaking under the collar. Next, apply anti-seize compound onto the tapered washers, splines, and the threads and the face of the castle nut, and then torque the 1 5/16" AF hub nut to 150 Ft-lbs. Correctly done, the tapers themselves would drive the car without splines.

Be aware that a spacer is required between the bearing and the shoulder of the halfshaft (quartershaft). If this is omitted, then there will be a greater splined length sticking through the hub, which would result in the nut tightening down at the bottom of the threads of the halfshaft (quartershaft) without succeeding in clamping the hub. The spacer is "handed" – meaning that it has one face that is flat while its opposite face is concave. The concave face must face against the shoulder on the halfshaft (quartershaft), the flat face must face against the bearing. If you install it the wrong way around, you will find that you have the halfshaft (quartershaft) sticking too far into the differential and not far enough out through the hub.

Rebuilding the Differential

On the other hand, if you are rebuilding the differential, more elaborate procedures are involved. Mark the flanges of both the driveshaft and the pinion in order to ensure correct reassembly, and then disconnect the driveshaft. Measure and record the torque that is required to rotate the pinion with the wheels removed from the rear of the car. While preventing the pinion from rotating, remove the flange retaining nut and its washer, and then remove the pinion flange. You may need to tap the flange forward with a soft-faced hammer, turning it slightly between each tap so nothing becomes bent or stressed. When the flange comes off, you will want to polish the contact surface with some fine-grit emery paper (600 or so will do the job nicely). The old pinion oil seal can be removed either with a narrow pry bar or with a long, heavy screwdriver. Closely examine the oil seal track area on the shank of the of the pinion flange for damage. Grease the periphery and the sealing lip of the new pinion oil seal with red rubber grease, and then proceed to fit the pinion oil seal. Insert the new pinion oil seal and gently tap it into place using a socket until it is flush into the axle casing. Next, refit the pinion flange and washer. At this point it is necessary to proceed strictly according to procedure. Screw on the retaining nut, tightening gradually until resistance is felt. Rotate the pinion in order to settle the bearings, and then measure the amount of torque necessary to rotate the pinion. Begin to tighten the retaining nut while constantly feeling the restriction to the movement of the pinion flange. You are searching for a pre-load, or resistance to turn, of about one foot-pound. The "feel" is that of a very slight tightening. At this point, the difficulty that arises is that the pinion flange rotates independently of the crown wheel by a distance of as little as 1/8" when measured on its circumference, and it is critical that you must measure the preload within that distance. Do not use an air impact tool to turn the retaining nut, as this method gives insufficient control. Instead, use a long-length 1/2" breaker bar. However, you have to keep the pinion flange from turning, so you will also have to make up a tool that you can bolt to the front flange. Use a 3/4"-wide piece of steel, about two feet long, with two 5/16" holes on one end. Insert two 5/16" bolts through the holes, and then secure them tightly with nuts. Do not bolt this tool to the pinion flange, as you will not be able to feel the preload. Eventually you will begin to draw the two taper bearings close together and achieve this very slight resistance to rotation. If the amount of torque necessary to rotate the pinion is less than that which was previously recorded prior to the removal of the pinion oil seal, then tighten the retaining nut a very small amount, then resettle the bearings and recheck the torque reading. Repeat this procedure until a torque reading equal to the recorded amount but not less than 4 to 6 In-lbs is attained. E.g., if the Original Recorded Figure = 9 In-lbs, then adjust torque to this figure (9 In-lbs). If the Original Recorded Figure = 0 In-lbs, then adjust torque to 4 to 6 In-lbs. Caution: Be aware that preload buildup is rapid, so tighten the retaining nut with extreme care. If an Original Recorded Figure that is in excess of 6 In-lbs is exceeded, then the axle will have to be disassembled and a new collapsible spacer installed (Moss Motors Part# 125-615).

It seems that some people are confused by the fact that you only use the torque figure of 140 Ft-lbs on the pinion nut when installing a brand new collapsible spacer. When you tighten the pinion nut, you will also crush a new collapsible spacer to the point that it allows enough space between the inner and outer pinion bearings for them to have the proper preload between them. If you should happen to tighten the pinion nut to 140 Ft-lbs after the collapsible spacer has been previously crushed by means of previous assembly, it will then be collapsed to a point that there is not enough space between the pinion bearings. Excessive preload will be the consequence, because the bearings that support the pinion shaft will have been crushed too close together. The best method is to use a precision tool (I use a Snap-On torque-meter) in order to measure the preload of the pinion. The term "preload" is defined as how much effort it takes to turn the pinion. If it requires too little, then the pinion gear will impact into the crownwheel gear. On the other hand, if it requires too much, then you will overload and burn up the bearings. You need to tighten the pinion nut to the proper preload. That way you can nearly always reuse the collapsible spacer, and very often have more room to tighten. If you start with a preload of 12 In-lbs before you take anything apart, it is perfectly conceivable that you should be able to reinstall everything and end up with a preload of 12 or 13 In-lbs, which is quite within the specification limits, then rebuild again to 13 to 14 In-lbs, etc, etc. As you can see, with careful reassembly and precision measuring tools, there is a potential for these collapsible spacers to have a very long life span.

Ensure that the mating surfaces of the axle and its cover are clean by removing all of the old gasket, dirt and grease. Hopefully, the old gasket will come off easily. Do not be surprised if it comes off in sections and pieces. In order to have a leak-free rear axle, first you have to get all of the old gasket material off in order to have a clean, oil-free sealing surface. A razor scraper that uses single-edge razor blades does nicely at this task, but use it patiently or you will snap off the blade. Whatever you do, do not make the classic mistake of spraying any of it with solvent. You will remove the oil and it will be as hard as a rock, forcing you to shave it off a few hundredths of an inch at a time. If you make this mistake, you will end up going through a large box of single edge razor blades by the time you are done (Bet'cha cannot guess how I know this?). Once you have the entire old gasket off, clean the metal face of both the differential housing and the sealing flange of the sheet metal cover with good old-fashioned rubbing alcohol. Next, check the sealing flange of the cover for distortion. Whenever a leak develops, the common tendency of DPOs is to put a wrench on the cover bolts and tighten them down to three grunt-pounds, crushing the gasket, distorting the cover, and thus worsening the leak. Drip, drip! Use the old Petroleum-Jelly-and-Mirror technique to check the sealing flange of the cover for flatness. If the sealing flange is distorted (and it often is), you can usually flatten it out by placing it on a flat surface and putting a socket open-end-up over the bulge, placing a piece of wood on the socket, and gently striking it with a hammer. Make sure that the socket is the same size as the bulge. Do not hit it too hard or you will thin out and displace the metal, creating a warp that you will not be able to get out. You will have to purchase another cover from Victoria British (Part # 5-1030, $109.95) if you do. As always, work slowly and carefully and you will be fine. Hopefully, your cover will not be distorted, but do not count on it.

By all means, replace those nasty old nuts and machine bolts. A "torque reading" is really just a measurement of friction between threads. You really cannot get a worthwhile reading if the threads are dirty, rusty, or deformed. Personally, I like to use stainless steel machine bolts and nuts on the underside of the car. I get them at a hardware store because the quality is higher and the price is lower than at an auto parts store. Be sure that all of the bolt holes on the mounting face of the differential housing and their threads are clean. Get one of your children's old tiny toothbrushes and some carburetor cleaner and clean them all out really well. Do it right the first time and the seal will never leak again. I put everything together with antisieze compound on the threads so that if I ever have to take it apart again, it will spin right off. Do not worry, as antisieze compound is not a lubricant. Once properly torqued down, the bolts will not come loose.

The differential cover gaskets from Moss Motors and Victoria British are about as thick as a piece of typing paper. They are typical junk gaskets. I make my own from the best gasket material that I can find at the local auto parts suppliers. They are not hard to make. All you need is a sharpened pencil to trace around the outside of the cover and to scribe the circles for the bolt holes and a cheap extendable razor knife cutting tool. You can pick up one of these at any Home Depot type store for very little money. If you consider that to be too much hassle, you might try getting a new gasket from Brit Tek as theirs are of decent quality, but not near as good as you can make yourself from premium materials. When you total up shipping and parts cost, you are no better off financially than doing it yourself. Just tell the person behind the counter what you are going to use the gasket material for and tell him that it needs to be thick (a little compressibility is always good for getting a better seal).

Using the new gasket and sealer, smear a thin layer of Permatex Ultra Black RTV Silicone Gasket Maker onto the outer edge of the gasket to glue it into place on the differential housing. Apply the sealant only to the outer side of the gasket. Why? When you torque the axle cover down you do not want excess sealant to ooze into the inside of the differential housing where it can break free and damage the internals. Next, apply the gasket and again smear the sealant onto the exposed outer half of the gasket, replace the axle cover and torque the bolts in an alternating pattern a few Ft-lbs at a time to no more than 14 Ft-lbs. Any more than 14 Ft-lbs will deform the sheet metal axle cover and consequently cause it to leak. Be sure that you have cleaned all of the threads, otherwise you will get a false torque reading. When you replace the axle cover, remember that the brake line clips at the top and handbrake pivot on the left.

Refill the rear axle with EP90 hypoid differential oil until it drips out of the filler hole, replace the road wheels, ground (earth) lead, and then lower the car. Make sure that you use EP90 hypoid differential oil, and not gear oil. There is a difference and it does matter. The oil is “EP”, meaning Extreme Pressure, because of the hypoid shape of the gear teeth that “roll” past each rather than on a pin-point thin line. Once the car is on its wheels the hub nut can be fully torqued up to 150 Ft-lbs, and then a locking cotter (split) pin can be inserted and bent into place.

Clunking from the axle should now be much reduced, or barely audible. However, if there is no improvement, then providing you have checked the hubs, wheels, and the driveshaft universal joints, you may need to consider a replacement axle.

You might think that painting the axle while it is on the car is the easy way out, but you will find that in practice it is a pretty miserable experience. Nobody ever tries it again once they have done it before. Do it the easy way. Take the axle off and clean it thoroughly, be sure to plug all of the holes so that the POR-15 will not get into the threads (once it is in, it is there to stay), and paint it in the light so that you can do a proper job. POR-15 has a website at http://www.**por-15**.com . POR-15 likes to adhere to surfaces with “tooth” rather than to smooth, glossy surfaces. A dry, sandblasted finish is excellent for this purpose. It will dry to a rock-hard, non-porous finish that will protect the metal from rusting. First, strip the axle housing to bare metal and then use Marine Clean to clean and degrease the metal surface. When the surface is thoroughly clean, rinse it with water and allow the surface to completely dry. Second, in order to etch the surface so that proper adhesion can occur, spray Metal Ready onto the surface full strength, keeping the surface wet for 15 to 20 minutes, and then rinse it off with water and allow the surface to completely dry. The optimum temperature application range is between 50º to 90º Fahrenheit (10º to 38º Celsius). If you are going to apply POR-15 with a brush, be advised that a 1 to 2-mil coat thickness is best. All successive coatings should be applied when the surface is "tacky". In this case "tacky" is defined as dry to the touch, but with a slight finger-drag remaining. Three coats will provide excellent protection. If you are going to apply POR-15 with a spray gun, be advised that you should use 30 to 35 lbs (206-250 kpa) of pressure for a normal gloss finish. For a lower gloss finish, air pressure should be reduced to 20 to 25 lbs (140 to 170 kpa). POR-15 should be thinned only with its own specially-formulated solvent, and never more than 5%. If a topcoat is to be applied later, apply a very light dust coat of POR-15 Self Etching primer or POR-15 Tie Coat primer over a tacky POR-15 coating and then allow it to dry. Full priming and finishing can then be done later. Use POR-15 solvent or lacquer thinner for cleanup, which must be done before POR-15 dries. Once dry, POR-15 cannot be removed with solvent.

In rebuilding the Salisbury tube-type rear axle, you will find that some parts may seem hard to obtain. The following list should be helpful-

Part	**National Part #**
Pinion Oil seal	224470
Front Pinion Bearing	M88048
Front Pinion Race	M88010
Rear Pinion Bearing	HM801346X
Rear Pinion Race	HM801310
Differential Bearing Set	A36
Front Inner Seal	224820

Rear Wheel Seal (SALISBURY TUBE-TYPE AXLE ONLY)	473234
Rear Transmission Oil seal 22H	473234
Rear Transmission Oil seal (REPAIR SLEEVE) 22H	99168

Axle Tramp

Axle tramp problems are the curse of leaf spring rear suspensions that are coupled to high torque engines. When the torque arrives at the differential, the axle tries to twist along its lateral axis, causing the springs to wrap until the tires lose traction, whereupon the axle is snapped back into its original position by the unwrapping leaf springs. The process is then rapidly repeated, the violent result being what is called “axle tramp”. Actually, while this could be minimized by the installation of a pair of antitramp bars, those currently available for MGBs are all, to the best of my knowledge, junk. They are all solid bars which, being of fixed length, cause the leaf springs to bind when the axle to which they are attached moves rearward as the suspension compresses and the bars swing upward along their arc of travel. To keep the springs from binding, each of the antitramp bars should be of two-piece telescopic design, just like the ones made for Chevrolets and Fords. In addition, they have no provision to allow lateral axle sway without stressing their mounts. Upon full extension they should travel no further than the rearmost position of the axle when the leaf spring is at its limit of upward compression, and upon full compression they should travel no further than the forwardmost position of the axle when the leaf spring is at its limit of downward extension. That way when the torque tries to twist the axle there is some limitation factor, yet the springs can perform without interference. Instead of bushings which can bind, ball joints should be provided in order to allow for lateral movement of the rear axle (axle sway). On a V8 model, that is the only solution short of a rear suspension system that incorporates a four-link trailing arm with coil springs, a Panhard rod, and tubular dampers.

However, the torque effect produced by the four-cylinder BMC B Series engine that is used in the MGB is not as severe as it is in a V8-powered version of the car. Late model

MGB GTs used seven-leaf rear springs and a trailing rear stabilizer bar (BMC Part # BHH 2003) that were directly connected to the rear axle by means of bushed pivots (BMC Part # 21H 6655), the combination of which helped tame axle tramp considerably. The seven-leaf springs are designed to increase their compression resistance more progressively at extreme compression than the original six-leaf design, and thus are naturally less prone to wrapping. The trailing rear stabilizer bar employed in late model MGBs is a torsion spring in its own right and, while willing to twist along its axis, it has considerable resistance to flexure, thus functioning as a semi-antitramp bar. I make use of this refinement in my car with its power-enhanced engine and find that the axle tramp will occur only when I stress the hell out of it in a fast takeoff from a standing start. Even then, it is not terrible; just a hopping feeling instead of the noisy, shuddering, banging that characterizes the no-rear-stabilizer-bar, six-leaf rear suspensions of other Chrome Bumper cars. If you want to go this route, try a 7/8" front stabilizer bar and an Original Equipment 5/8" trailing rear stabilizer bar so that the handling will be neutral. This is presuming, of course, that the roll center of the car has not been altered as a result of the car having been lowered. Of course, you can always have a machine shop make up the two-piece telescopic antitramp bars, fabricate mounting brackets, and weld the brackets in.

Of course, all efforts toward increasing power output and getting it onto the ground will eventually result in further considerations of making improvements in other areas of performance, such as the handling and braking of the car. Fortunately, when the geometries of the steering and the suspension of the MGB were developed, the design work was not hampered by a requirement for the incorporation of components that were originally intended for use in other vehicle designs that were already in the BMC stable. Instead, the engineers at MG were given a free hand in their work to create geometries that were optimum for the MGB. Even the suspension geometries of the Armstrong lever arm dampers were developed specifically for application in the MGB. The consequence of this free-handed approach was superior handling.

Converting a Salisbury Tube-Type Wire Wheel Rear Axle to Disc Wheels

In order to perform this conversion, you will need the following parts: A set of front and rear original MGB 4 bolt hubs (Bolt pattern 4"x 4.5"), 2 Grease caps. Moss part #264-515), four brake drum retaining screws (Moss Motors Part# 323-255), your existing front bearing parts, eight metric wheel studs 12mm x 1.5 (2.5" long), eight metric lug nuts (12mm x 1.5), 8 standard lug nuts (1/2”-20), new cotter pins, and two rear wheel spacers.

The sole difference between the halfshaft (quartershaft) of the wire wheel rear axle and the (quartershaft) disc wheel rear axle is their lengths. The splines on their ends which accept the hubs are identical, so the conversion of the rear axle is straightforward. It is a good idea to spray some PB blaster or another good penetrating lubricant onto the castle nut of the halfshaft (quartershaft) before trying to remove it. Let it soak in for about one hour before removing. There is a drive shaft collar (BMC Part # BTA 243, Moss Motors Part # 125-665) behind the castle nut that should come out with the hub. Do not lose it, as you will need this for the new hub. The torque specification for the castle nut of the halfshaft (quartershaft) is 150 Ft-lbs, so if you don't have an impact gun, then you will most likely need a heavy duty breaker bar in order to loosen it. Be aware that because the castle nut is recessed within the hub, some thick-walled sockets will not fit into the gap between the hub and the halfshaft (quartershaft) nut. Be sure to put the parking brake on good and tight before you attempt to break the halfshaft (quartershaft) nuts loose. Remove the cotter pin and the 1 5/16" castle nut of the halfshaft (quartershaft), and then pull the wire wheel hub off. Next, remove the split oil seal collar (BMC Part # AAA 392, Moss Motors Part # 266-030) and then the oil seal (BMC Part # GHS 179, Moss Motors Part # 120-700).

Since the width of the wire wheel rear axle assembly is 1.5" shorter than that of the disc wheel axle assembly, in order to attain the same track width of the disc wheel rear axle assembly, as well as so that your tires will clear the inner wheel well, spacers will need to be inserted between the hubs and the rear wheels. The size of the spacer depends on both the tire size that you are using and the offset of the wheel. Figure somewhere between .375 & .500 inch. The bolt pattern is 4 x 4.5 (114.3mm). In order to accomplish this modification, the original rear wheel studs will need to be removed and replaced with longer ones as the Original Equipment studs are too short. The metric size is 12mm x1.5 (2.5" long) (ARP Part# 100-7708). You can get them from Summit Racing. The metric size is the only stud that is a direct replacement for the original. The original Studs will need to be either pressed out or hammered out. If you do not have a press, you can pull the studs into the hub by using an open-end lug nut. Tap the stud into place, the place the hub in a vice and turn a reversed lug nut (conical side facing away from the hub) until it makes contact with the hub face. Turning it will pull the stud into place.

The front hubs can be used as-is, but the lug threads are 1/2"-20 which is the Original Equipment size. This is OK but you'll need metric lug nuts for the rear wheels and Original Equipment size for the front wheels. Summit racing sells both sizes of the appropriate lug nuts. If you want to simplify matters by using metric all around, just buy 16 studs and change out the front hubs as well. If you're using closed-end lug nuts, then the fronts will be a bit too long and will need to be cut back about a 1/2".

There is no need to remove any of the brake hydraulic lines or the brake pads. Simply loosen and then remove the two 7/16”-20 shouldered bolts, remove the caliper and, using wire or rope, allow it to hang off to the side out of the way. Do not allow it to hang from the rubber hose (flexible pipe). Wedge a piece of wood between each set of brake pads in order to prevent them from accidentally closing. Next, remove the grease cap, and then remove both the cotter pin and the 1 1/8" hub nut. Remove the keyed washer, followed by the outer bearing. A few shims should come out with the outer bearing. Set them aside as a complete set and do not lose them. Now, slide the hub off. Note that the rotor is bolted to the hub and later will need to be removed and bolted to the new hub. On the backside of the hub is a rubber oil seal that needs to be carefully pried off so as to not mar its seating surfaces inside of the hub. If it is in good condition, it can be reused. If not, order a new pair from (BMC Part # GHS 101, Moss Motors Part #120-610). Now remove the inner bearing followed by the spacer. Please note that there will also be an easily-damaged metal oil seal collar (BMC Part # BTB 183, Moss Motors Part # 264-940) that fits between the rubber oil seal and the radius of the spindle. At this point, you only need to deal with the matter of the wheel bears and then reassemble using the new hubs. Be sure that you fasten the disc brake rotor onto the front hub prior to performing any work on the wheel bearings.

Wheel Bearings

It should be noted that there are a variety of caged bearings that are available to be used as wheel bearings. While these all perform their function as thrust bearings quite well, care must be taken as to their proper installation. If incorrectly installed, the hubs may come loose from their axles, sometimes with catastrophic results, both mechanical and personal. They should always be checked to see if there is any clearance between them and the radius of the stub axle. If any clearance exists, then the radius of the inner cage of the bearing is too small and they should not be used. When the bearing radius is smaller than the radius of the stub axle, the bearing does not fit against the flange. The contact between the bearing and the flange face is designed to help support the axle load. Hence, a gap would provide a flex point which could easily lead to failure of the stub axle.

These two caged wheel bearings, although of different dimensions, are of exactly the same type. Although this type of caged wheel bearing can support lateral side forces, it is intolerant to over-tightening of the inner race against the outer race, as this will result in binding and they will consequently suffer either premature wear or, more likely, a catastrophic failure. On the other hand, the design of this type of caged bearing requires that it be installed in applications that position the inner race against the outer race with just the right amount of clearance in order for the wheel bearings to run freely. If there is too much clearance, then you will have “play”. If there is too little clearance, then the bearing will overheat and fail.

The hub into which the wheel bearings fit has an integral spacer machined into its center. This integral spacer holds the outer races of the two wheel bearings and sets them at a specific distance apart. In addition, another spacer is placed over the stub axle between the two wheel bearings and is used to set the two inner races apart. These two spacers should be measured in order to assure that they are exactly the same length, in order to assure correct alignment of the inner and outer races. Both wheel bearings must then be what is referred to as “face adjusted” with shims when installed. A “face adjusted” bearing is one that has been surface ground so that the surfaces that are in contact with the hub and the spacer are flat and parallel to each other whenever both races are in contact with the bearings as they would be when properly installed. This necessity is due to the fact that bearing manufacturers no longer make the original front wheel bearings in the necessary face-adjusted tolerances, thus it should come as no surprise that they do not fit correctly unless the appropriate thickness shims are used. Some cars were designed to use tapered wheel bearings without spacers and shims, but MGs were designed to run with them. Second-guessing university-trained engineers on safety-related items is a fool’s game.

Wheel bearings can fail for many different reasons. The least common reason for their failure is that they have become just plain worn out. In reality, very few wheel bearings run their entire life and expire from being worn out. The more common reasons for wheel bearing failure include over-loading, dirty or incorrect lubrication that includes too much or too little grease, or, if the bearing happens to fail a very short time after installation, then the most frequent reason is poor installation. If you use proper procedures, then they should have a satisfactory service life. Be sure to pack decent grease into the bearing prior to fitting. Keep the entire operation perfectly clean. It does not take much grit to start a bearing failure.

It is important to understand that the rolling elements in any rolling bearing must have a smooth race to run upon. The wheel bearings should be driven into place by using an outer race to push the outer race of the bearing into place. Be aware that forcing the inner race into the hub can cause it to separate from the outer race. It can be difficult to properly reassemble the bearing without damaging any of its precision components. If available, a press is logically the better alternative to driving them into place because a pushing force is preferable to that of an impact force. However, if you press, or gently tap the outer race into place, then the rolling elements cannot impact against either the inner race or the outer race (an event known as "brinelling"), therefore shortening the service life of the bearings. A future problem occurs when you wish to remove the bearings from the hub, as the easy way is to push the inner race, which then forces the rolling elements against the outer race, and thereby pushing the complete bearing out of the hub. Since this method has the potential for damaging the inner race, the rolling elements, and the outer race, of such bearings should be considered to be questionable for reinstallation unless the removal procedure has caused no brinelling of either the race or of the rolling elements.

It should be noted that it is possible to install wheel bearings in such a manner that it will lead to fracturing of the stub axle. You need to be aware of the fact that stub axles are an ideal candidate for fatigue failure that is caused by the cyclic loadings they typically see in their everyday use. Fatigue cracks will generally start at the base of the stub at the bearing shoulder (i.e. in the radius). They start at the surface at the 0° and 180° positions (i.e., top and bottom), and then simultaneously work their way transversely across the axle inwards towards the center, with the crack growing and forming a lens shape. Eventually there will only be a narrow "eye slit" of uncracked metal remaining at which point they let go. The problem is caused by failure to understand the design of the stub axle and, as a result, installing the assembly with incorrect torque on the bearing retaining nut. The design demands that the retaining nut of the wheel bearing be tightened to a specified torque of 45 Ft-lbs to 70 Ft-lbs. This pre-stresses the stub axle by means of the spacer, and thus enables it to carry the suspension loadings without fracturing. If you fit the Original Equipment wheel bearing assembly, complete with its spacer, tab washer, and wheel bearing retaining nut, the stub axle will be placed under tension and the spacer will be placed under compression as the designer intended. This pre-loaded structure will then easily resist bending stresses because prior to any suspension load bending the stub axle, the load will first have to compress the spacer and / or further stretch the stub axle. Since the materials used in the stub axle and the spacer are stronger when they are respectively in tension and in compression than they are in bending, any loading that would tend to bend the stub axle will be too weak to bend the stronger pre-loaded structure. If you reckon you know more about engineering design than the manufacturer, then leave out the spacer and shims. If you have an accident that kills others and the investigators determine that the cause was that you “failed to maintain the vehicle in a safe and roadworthy state”, not only might your insurance be automatically void, but you will have to live with the rest of the consequences as well.

Wheel bearings that have been marked “Thrust” on their outer races must always be installed with the marking facing their bearing spacer. Wheel bearings that have been marked Thrust on their inner races must always be installed with the marking facing away from their bearing spacer. Wheel bearings that have no such marking must always be installed with the thicker outer part of their outer races facing towards their bearing spacer. The wheel bearings of the MGB are adjusted by installing shims over the shaft between the inner race of the outer bearing and the steel spacer tube in order to adjust the spacing between the inner and outer wheel bearings, thus providing an exact, tight, firm fit. The adjustment with these shims can made down to a minimum clearance of .001”. This will allow the hub nut to be tightened very tight (40 Ft-lbs to 70 Ft-lbs). The torque is applied not through the rollers, but through the shims, bearing inner races, and the spacer tube, which is why it has a negligible effect on any free-play that is felt at the hub. In order to check the amount of play in the wheel bearing, you ideally need a magnetic-mount dial gauge. This should be attached to the hub or the brake rotor and the reading then taken at the hub by pushing in and pulling out the hub on the bearing. The reading should be between .002" and .004". This must be measured with the hub nut at 60lb/ft, using shims to get the clearance correct. If it is greater than this, remove shims in order to correct it or, ideally, replace the wheel bearings and re-shim them. I have found that the easiest way to set them is to fit a surplus of shims initially, measure the endplay (endfloat), and then calculate the thickness of shims that need to be removed in order to arrive at the correct endplay (endfloat).

When replacing the wheel bearings, dry-assemble everything first (i.e. without grease or oil) as this makes it easier to accurately set the endplay (endfloat). The order of parts on the axle is: oil seal collar, inner race of inner bearing, spacer, shims, inner race of outer bearing, bearing retaining washer, hub nut. The first time that you assemble the parts onto the hub, in order to seat the outer races in the hub, leave out the shims and tighten the hub nut until the wheel bearings bind. Next, fit the shims between the spacer and the outer bearing. The objective is to add or subtract shims until you get an endplay (endfloat) of .002” to .004”. Once you have got this correct, dismantle the hub, pack with grease, then refit and torque it up. Tighten the nut until the bearings bind, This will pull the outer races fully against their locating flanges inside the hub.

Note that fitting shims between the outer bearing and the locating washer causes slivers of metal to be shaved off of the shims when the nut is tightened, because in this position they are resting on the threads of the axle. Slivers of metal are not what you want in your new bearing! Using combinations and multiples of shims will give most values in .001” increments with the exception of a few of the smaller values, as follows:

Total Thickness Required	Number of .003” Shims Required	Number of .005” Shims Required	Number of .010” Shims Required
.003”	1	0	0
.005”	0	1	0

.006”	2	0	0
.008”	1	1	0
.009”	3	0	0
.010”	0	0	1
.011”	2	1	0
.012”	4	0	0
.013”	1	0	1
.014”	3	1	0
.015”	0	1	1

Keep combining different groupings of shims until you get two combinations of shims that ideally total only .001” apart, where the thinner combination gives no endplay (endfloat) and the thicker combination gives perceptible endplay (endfloat), i.e., +/- .001”. Use the lower combination and then add another .003” shim. This should give you the required clearance of .002” to .004”. After you have determined the correct combination of shims, remove the races and inject or press grease into only one side of the bearing! Keep going till the grease comes out the other side, and leave a bulge of grease on both sides. Do not be tempted to save time by forcing grease in from both sides, as you will trap air in the middle of the bearing and possibly cause premature failure. Fill the groove in the oil seal that faces the base of the axle shaft with grease, as well as the cavity between the oil seal and the inner bearing. Do not fill the cavity between the wheel bearings or the grease retaining cap with grease. The oil seal should be fitted to the hub with the flat side of the seal facing out from the hub and the lip of the seal facing inwards. This is to keep water off of the spring that provides the tension on the lip of the seal, so preventing it rusting, breaking, and consequently letting water and dirt in and grease out. Reassemble everything and tighten the hub nut to 40 Ft-lbs, and then tighten the hub nut further until one of the holes in the shaft lines up with a slot in the nut. This should occur well before the maximum permissible torque of 70 Ft-lbs is reached. The hub nut should then be locked in with a cotter pin. The system then will be firm and adjusted precisely to the clearance needed between the wheel bearings and their spacers, allowing them to spin very precisely with no play or wobble. This setup yields minimal wear to the wheel bearing over many miles. It is one of the best designs on the entire car.

Once you have the wheel bearings correctly installed, refit the wheel hubs, the disc brake rotors, and the disc brake calipers. Check to see if the rotors are properly centered inside of the calipers. If they are not properly centered, adjust the position of the caliper with shims.

The Brake System

The development and fitting of a more powerful engine is one matter, but ensuring that the car is safe to handle the power is an issue that has to be resolved before the car is used on the road. There are many proven ways of improving the brakes, suspension and other areas of the car, and these must always be considered to be an integral part of any conversion that involves the enhancement of performance.

The braking system being essential for control, it is also highly advisable to have it in excellent condition. The braking system of the MGB was highly advanced when it was introduced in 1962, consisting of solid 10¾" cast iron brake rotors on the front wheels and 10" X 1¾" cast iron drums with single leading shoes on the rear wheels. Properly upgraded, it is fully capable of locking the wheels at high speeds, even against the traction of modern V-rated high-performance tires. Although obviously still adequate for normal driving by today's standards, they can be improved for satisfactory function in high performance driving without resorting to the expense of the adapting of an exotic all-wheel vented disc braking system. Installing vented brake rotors will dissipate heat more quickly and thus forestall brake fade, but will hamper roadholding by increasing undesirable unsprung weight. Remember, it is not the intention of this article to create an exotic braking system suitable for competition on a racetrack.

There are several things that can be done to enhance the performance of the original system for use on the street. First, either rebuild or replace the brake master cylinder, brake slave cylinders, and calipers. Be sure to use stainless steel pistons, as they will not rust or pit, which will cause ruination the seals. If you do not wish to do this yourself, White Post Restorations does this for rear brake slave cylinders using stainless steel pistons, brass sleeves that give a better bite to the seals, and gives a lifetime warranty on their products. They have a website at http://www.whitepost.com . Be aware that because it functions as a seal, the piston of the brake master cylinder has to be in pristine condition with a chrome-like finish, otherwise it will leak.

In rebuilding the brake master cylinder, be aware that initial movement of the master cylinder piston by the brake pedal forces brake fluid up into the reservoir of the master cylinder via the bypass port and does not apply the brakes. As soon as the primary seal covers the bypass port, further movement of the piston pressurizes the fluid in the lines and thus applies the brakes. As the primary seal continues moving forward and clears the bypass port, fluid from the reservoir is free to run into the space behind it, so the secondary seal prevents fluid from leaking out the back of the master cylinder. If the primary seal is faulty, then pressurized fluid can leak back past it into the space between the two seals, and back into the reservoir, which cases the pedal to sink. In extreme cases, a ripped primary seal may develop no pressure at all and the pedal will go straight to the floor (which can also be caused by air in the hydraulic systems. If the secondary seal is faulty, brake fluid, even though it is not under pressure, will leak out back towards the pedal linkage and even run down the brake pedal inside of the cockpit.

When changing master cylinders for either the brake or the clutch on the single circuit brake system of a MKI MGB, you will find that the bolts that secure them in place are difficult to reinstall and tighten. This if due to the necessity of having to perform the work by reaching through a 4" square hole in the firewall (bulkhead) in order to fit the washers and nuts onto their bolts. A 1/4" (.250") drive socket set with a universal joint can sometimes be helpful, but it is still very difficult to reinstall the lockwashers and nuts onto the bolts through that 4" hole. A simple solution is to put the bolts through the master cylinder flanges from the reservoir end so that the threaded ends of the bolts project toward the clevis end of the master cylinder. The next step is to use an adhesive in order to adhere the head of the bolt onto the master cylinder flange. In this manner, you will be able to put either cylinder in place and put the nuts and lockwashers on from the engine compartment side. However, for the MKII models, there is an alternate solution to the problem. Once the master cylinder is removed, do yourself a favor and Helicoil the mounting holes with 5/16 - 24 helicoils in the mounting flange of the master cylinder in order to make reinstalling it much easier (not having to fool with getting nuts started inside of the pedal box where there is severely limited space).

The British have developed what they call "copper brake pipe", which seems to be a seamless tubing that is made of a copper alloy that bends easily, is corrosion-resistant, and is in reliable use for "classic cars". The brake line (pipe) in question is called "Kunifer" or "90-10 Copper-Nickel". It is being used in many new cars of European manufacture; Volvo has been using it since 1976. Its yield strength is good, but lower than that of Bundy tubing. Its expansion rate under pressure is also good, but undesirably higher than that of Bundy tubing. It is almost certainly not as reliable as genuine Bundy tubing, and thus should not be used.

Brake lines (pipes) should always be fabricated from Bundy tubing that has a wall thickness of 0.028" at a minimum. Brake line (pipe) pressures can and often exceed 1,000 pounds per square inch (PSI). On no account should copper, aluminum, or commercial fuel line (pipe) be used, despite the fact that they are easily available and appear to be easy to work with. Pure copper tubing work-hardens and becomes very, very brittle- especially at flare fittings. Under the cyclical loadings that occur in brake applications, and in the presence of moisture (and possibly high concentrations of chlorides, if you live in a road salt area), they will almost certainly tend to crack and fail right at the flare, usually with little or no warning.

Bundy brake lines (pipes) are available in various lengths at most parts stores, although they almost always seem to have the wrong swivel nuts, no matter which one you happen to need at the time. The swivel nuts from the old brake lines (pipes) can sometimes be re-used if they are in good shape. Nevertheless, chances are that you are reading this because you want to upgrade your fittings, rather than just replacing what is already there. Most auto parts stores can supply the standard fittings. Various British parts suppliers can sell you new Girling parts. While you are at it, you might follow the lead of many folks that do racecar preparation and only use steel fittings, even though aluminum is available and a bit less expensive.

Be sure to pull the retaining straps off the bottom of the car, pull the rubber back, and make sure that the brake lines (pipes) are not rusty underneath the rubber. Because it only takes one pin hole in one spot to cause a system failure, make sure that you have inspected the entire system for rust. If you see any rusty, bent, or kinked brake lines (pipes) that you are concerned about, Classic Tube in New York is a fantastic source for replacing your individual or complete brake line system. They have a lot of British patterns already in stock and provide the material in either mild steel or stainless steel with the correct flares and fittings. Plus, their brake lines (pipes) are already pre-bent on computer-controlled machines to the required lengths, angles, and orientations. I am not sure that you could buy the material, fittings, and tools for much less than what they would charge to do it for. If you have ever tried to copy some of the British line, then you know how much fun that can be. Play it safe - if you are in doubt, buy new. Classic Tube has a website at http://www.classictube.com/ .

Note that US market MGBs used different models of brake master cylinders. The first model (1962-early 1968) had provision for only a single-circuit system, while the subsequent models all had provision for safer Federally-mandated dual-circuit systems. The 1975-1976 and 1977-1980 models used two different brake master cylinders that were servo-boosted in order to provide power brakes. Note that none of these servo-boosted models produced more braking power; instead, they simply required less effort at the brake pedal. There are two methods of testing the servo. The first is to pump the brake pedal shortly after having switched off the engine, and initially there will be a wheezing sound and the pedal movement will be "normal", and then after a few pumps, the wheezing sound will fade away and the pedal will become noticeably higher. The second method is to then press down hard on the pedal when starting the engine and you should then feel the pedal go down a bit.

For those individuals whose car has the single circuit brake system of the MKI model, a vacuum-assisted servo unit is available. While never offered on the North American Market, this factory option, which consisted of a Lockheed Type 6 booster servo, was somewhat popular in the UK and European markets. It has recently become available for the North American market through aftermarket vendors. However, be advised that in order to be safely used, the components of the entire braking system must be in excellent condition.

The increased hydraulic pressure created by the vacuum-assisted servo unit could cause failure of a system that has any marginal components. Be aware that the system must be properly bled after installation. If the system is not properly bled, then chatter of the front brakes will occur. Should you encounter this problem, first rebleed the system, as it requires consistent fluid pressure from both sides in order to function properly. This is due to the fact that if there should happen to be an air bubble in either the bore of the control piston or in the bore of the main piston, it will then cause the brake pressure to oscillate uncontrollably, with consequent brake chatter. If there is no change after bleeding, then check for leakage of brake fluid by removing the anti-back flow (non-return) valve (the one that the vacuum line connects to) and check to see if there is brake fluid inside of the bottom of the canister of the vacuum servo. Remove the air valve securing screws that hold the air valve cover onto the brake slave cylinder (this is the small cover with a tube running back to the vacuum canister), and then look for any brake fluid leakage inside of the air valve chamber. Inspect the two air valves inside of chamber of the air valve body. With no pressure on the brake pedal, the top air valve should be seated against the aluminum port. Have an assistant step on the brake pedal. The top air valve should open and the bottom one should seat on the aluminum port. If you find brake fluid in either of these two places, then the piston seal on the air valve piston is leaking and the air valve assembly will need to be taken apart. If you find no brake fluid leakage, then it is possible that one of the two bottom air valve seats may be leaking. The bottom air valve seat controls the vacuum level on the rear of the diaphragm in the vacuum chamber, while the top air valve seat allows air into the rear of the diaphragm in the vacuum chamber. When the brake pedal is not being depressed, the air valve is closed and equal vacuum is applied to both sides of the diaphragm in the chamber. When the brake pedal is depressed, the slave piston moves forward in its bore, displacing brake fluid and consequently moving the air valve piston to close the vacuum port at the top section of the diaphragm of the air valve. In chorus with the movement of the slave piston, the top air valve is opened in order to let air into the rear of the main diaphragm inside of the servo so that the vacuum in the front chamber pulls the main diaphragm forward with its attached pushrod in order to close the bypass port in the slave piston and thus apply increased pressure to the brakes. In order to produce brake chatter in a system that is properly bled, the most likely problem is that the air valve piston seats are leaking, causing brake pressure to oscillate.

In order to test the servo system, you must perform two simple tests. First, pump the pedal, and then press the brake pedal pretty hard in order to feel further downward pressure when you start the engine. It may be easier to feel on the later combined master cylinder and servo as that gives more assistance. The second test is of the vacuum circuit. This must be performed after the engine has been run. Switch the ignition off, and then pump the pedal as if for the test. You should hear hissing and wheezing from the servo as the pedal is operated and the pedal should gradually get higher as you use up all of the vacuum in the reservoir. This should still happen several hours after switching off. If the first test works, but this fails, then there is a leak in the vacuum circuit that will be weakening your fuel / air mixture (which the carburetors may have been tweaked to 'compensate' for).

Brake Hoses

Do not reuse old brake hoses (flexible pipes). As they age, tiny bits of deteriorated rubber on the interior of the brake hydraulic hose (flexible pipe) flake off and float around inside of the brake lines (pipes) until they become stuck inside of a caliper feed port, where they will then perform the function of a one-way non-return check valve. When this occurs, usually one (but in rarer cases, both) of the front calipers will lock up solid and fry the brake pads, as well as heat the brake rotor until cementite forms, the presence of which will cause the brake rotor to take on a bluish hue. The result of cementite formation will be rotors that become increasingly warped as their temperature increases. Instead, install a set of braided stainless steel extruded Teflon-lined brake hoses (flexible pipes). Originally developed to handle the higher line pressures inherent in disc brake systems, these will not expand under pressure and result in a firm brake pedal with greater “feel”, enabling you to more precisely modulate braking forces and more easily tell when the brakes are about to lock up. Due to their resistance to expansion, they also result in swifter brake system reaction time, a definite plus during hard driving on twisty roads or when forced to make a panic stop. In addition, their Teflon lining has the advantage of being impermeable to air, thus largely overcoming the problem presented by the hygroscopic properties of most brake fluids. These can be obtained from Brit Tek at http://www.brittek.com (Part # ABK103). Be aware that it is not the Outside Diameter (O.D.) of the brake hoses (flexible pipes) that is important, it is the Internal Diameter (I.D.). Usually brake hoses (flexible pipes) are made with -3 AN line that has an Internal Diameter (I.D.) of 3/16". Some use -4 AN which has an Internal Diameter (I.D.) of 1/4". The braided stainless steel over Teflon brake hoses (flexible pipes) is of a smaller Outside Diameter (O.D.) because it does not take as much material to give an even greater burst strength than those brake hoses (flexible pipes) that are made of the more common reinforced rubber.

A final comment about Teflon brake hoses (flexible pipes): they may not be seen as having a legal place on your street car, so do not make the mistake of bragging about them to the Motor Vehicle Inspector when you go to get your annual inspection sticker renewed. Many of the Teflon-lined brake hoses (flexible pipes) that you can buy pre-built are probably not D.O.T. approved. The first, as well as the main reason for this lack of approval, is that the manufacturers of such racing-oriented equipment usually do not care to incur the expense that would be required in order to get their hardware approved by the D.O.T. They are building brake hoses (flexible pipes) that are intended for racing applications. The second reason is that the D.O.T.'s expected lifetime for approved brake hoses is five years, which is longer than most racers will continue to use them, and the prudent owner will replace his or her brake hoses at least that often. However, there is an exception to this lack of D.O.T. approval: the Goodridge company has taken the time and effort to obtain D.O.T. certification for their stainless steel braided, Teflon-lined brake hoses, thus making them street-legal in all 50 states.

There was a very good reason for all major automobile manufactures to have switch to dual circuit brake systems during the 1970s. If a brake master cylinder, a brake slave cylinder, or a brake hydraulic hose (flexible pipe) on a dual braking system should happen to fail, then the vehicle will still have half of its braking system remain intact. However, should any of these events occur in a single circuit braking system, then the entire braking system fails, leaving the driver with nothing to stop the car, except for the manual hand brake. Unless you are doing a Concours restoration and trailering your car to show events, you would be wise to convert to the dual circuit braking system that is found on the 1968 and later MK II MGBs.

Converting To A Dual Brake Master Cylinder

Converting a 1974 or earlier car to the later dual circuit servo-boosted brake master cylinder is difficult due to the mounting flange of the brake master cylinder having been turned 90°. The fitting of the redesigned later pedal box (the pattern of the pedal box mounting holes on the sheet metal flange is different), complete with servo booster that matches this brake master cylinder mounting flange pattern will solve this problem, but be advised that you will also need the later version pedals that were designed to be used with it, otherwise the clutch pedal will not depress far enough to disengage the clutch! You will also find installation to be easier if you enlarge the pedal hole in the bulkhead. In addition to the aforementioned items, because the master cylinders on the servo-boosted brake system are located with the master cylinder for the clutch back near the firewall (bulkhead) and the master cylinder for the brake system on the front end of the servo mechanism, the pipes for both the brake system and clutch slave cylinder must be completely replaced. You will need the clutch slave cylinder line (pipe) from a servo-boosted car, plus the engine compartment brake lines (pipes) that run from the brake master cylinder to the front wheels and the brake lines (pipes) from a servo-boosted car that run to the connector under the passenger seat. Unfortunately, these later systems are not without their liabilities. Many owners complain that they interfere with the "feel" of the system, producing a somewhat "numb" feel at the pedal. In addition, on Left Hand Drive cars, they project so far into the engine compartment that they make the mounting of dual carburetors difficult and force the mounting of small, restrictive conical air filters.

Front Disc Brakes

Rebuilding the front brake calipers is a very straightforward task. In order to remove the caliper pistons, oil them with brake fluid, and, using levers and / or a G-clamp, compress one fully in. Next, clamp it in place with an engineer's clamp, or secure it with wire. In place of the brake hoses (flexible rubber pipes), install a grease zerk (grease nipple). Simply pump in the grease in order to force the piston out. The caliper will need to be cleaned very well with solvent afterwards in order to remove all of the grease. Split the two halves of the calipers, and then clean up the grooves. Take care when doing this as the rubber oil seal that joins the brake fluid passage is a special square cross-section O-ring that has an 0.500" Outside Diameter and is 0.110" thick, so be warned that using an ordinary O-ring as a replacement will not work. Remove it carefully and then clean its bore in the caliper with brake fluid. You can get the oil seal from either Pegasus Motorsport (Part # 3596) or Moss Motors (Moss Motors Part # 180-285). Use a pencil, not a screwdriver, to remove the oil seal as its sealing surface on the caliper must be free of defect in order for a proper seal to be attained.

Do not be surprised if you find the bore of the caliper to be rusty. While the bore is routinely cleaned with a hone by professional rebuilders, often naval jelly will suffice to remove the rust. Remember that the critical surface area is on the caliper piston, not in the bore of the caliper itself. Make sure that the bore is perfectly clean and rust-free before you try to fit the seal. It is impossible to fully get the bores rust free without splitting the halves of the caliper. The caliper piston rides back and forth over fixed seals but does not bear upon the caliper bore. Examine the caliper piston for signs of rust and / or pitting. If you find any, do not try to recycle the caliper pistons by derusting them. Once the chrome plating on the piston has been compromised, the pits will rust again. Particles of rust have sharp edges which will cut up your new seals during use, causing the calipers to leak. Be smart and replace the old caliper piston with a new stainless steel caliper piston. Clamp it in place and repeat the process in order to pop the other piston out. New stainless steel caliper pistons, fabricated from 303 Stainless steel alloy, can be obtained from John Farrell Auto Parts (Part # 114713). Unlike other, more commercial suppliers, John Farrell is an enthusiast who goes to the extra effort to machine in an anti-squeal notch like the Original Equipment caliper pistons had. They have a website at http://www.fortunecity.com/silverstone/tread/1046/ .

The outer caliper piston rubber seals are supposed to retract the caliper pistons as you let up on the pedal. Normally, as the pistons extend under the increase of hydraulic pressure, the rubber seals grip onto the piston sides and flex outwards slightly. When the brakes are released, the reduction of hydraulic pressure permits the rubber seals to flex back to their original shape and thus pull the pistons back into their bores about 1/16" (1.6mm), or less. Worn or hardened seals lose their flexibility and are unable to retract the pistons.

Be aware that some aftermarket seals are of a slightly larger Outside Diameter (O.D.) and are difficult to fit into their bores. The AP/Lockheed seals are of the correct dimensions to both fit and function properly. Fit both seals, and then lubricate them with Lockheed Disc Brake Lubricant (Lockheed Part # LPK 102; Moss Motors Part # 220-440) or red rubber grease (Moss Motors Part # 220-442), never with axle or bearing grease. You really do need to use some of this!! If you just wet the seals with brake fluid and install them, then you will be at hazard of splitting the inner lips of the seal. You can buy a special Girling tool for retracting the pistons; I got mine from MGOC spares for about £12. It is basically a flat piece of metal with a nice big handle at right angles to it. You insert it between the piston and the disc pull on the handle. That causes even force to be applied across the face of the piston which then easily slides back. It is definitely a tool worth having! A less expensive alternative is a kind of right angled pliers with jaws that open when the handles are squeezed. It can also be used in the hollow part of the pistons to grip and rotate them to the correct alignment.

Fit the dust seals to the rings, and then lubricate them as well. Fit them over the hollow end of the piston, about ¼" (.635mm) down. Note that the machined relief notches of the piston are present in order to alter the distribution of pressure across the brake pad, preventing them from going tapered. Lubricate the pistons with red rubber grease, and then, making sure that their machined relief notches face in the direction of the hub, gently press them in by hand. Now, use a flat piece of steel and a G clamp to press the pistons fully in, and then press the dust seal steel ring into place. It helps to hold the caliper in a vice at this point. The dust seal retainer ring can be, and usually is, is difficult to install if you lack a proper tool. A standard 'C' clamp and a frieze plug (core plug), of the same outside diameter as the dust seal steel ring, will do quite nicely.

Be warned that the two bolts used to assemble the halves of the calipers are special "Torque-To-Yield" bolts, and as such should not be reused. These bolts are torqued to a specific value, and then turned an additional number of degrees. When loosened they do not return to their original dimension. They are of a super high strength variety in order to sustain the forces of application, but once they have been retorqued, they may be stretched beyond their design limits. They have a black finish, a shouldered head and shank, 7/16-20 UNF x 1.4", and are marked on the head with three hash marks at 120 degree intervals, and the letters "GKN", which refers to the manufacturer, and the letter "S" refers to the tensile strength of the machine bolt (50-55 Tons per square inch). They also have a splash of some hard red thread locking compound that appears on approximately 1/3 of the circumference of the bolt, and commencing on the fourth thread groove and ending on the eighth groove from the end. Consequentially, plan on chasing the threads in the calipers with a rethreading tap (not a cutting tap) in order to remove the pre-existing locking compound, to clean and straighten the threads, and to insure proper torque. New ones can be obtained through a variety of suppliers (Moss Motors Part # 320-135). Using a regular cutting tap and cutting die can cause weakness in the machine bolts by causing stress fractures at the point of cutting. This cutting action will damage the rolled threads and reduce the maximum amount of torque that the machine bolts can sustain. Remember, it is absolutely necessary to prevent an improper torque figure reading due to corrosion or paint in the area of unused threads and pre-existing locking compound in the area of the used threads. An application of High Temperature Loctite from the fourth thread to the 9th or 10th thread from the ends of the machine bolts will do a satisfactory job of holding everything together.

When reassembling the two halves of the caliper, use two pieces of drill stock of the same diameter as that of the brake retainer pinholes inserted through the holes so that the two halves are held in proper alignment, and then clamp the rods in your bench vise (Viola!!! An instant fixture and less chance of ruining the O-ring). In order to minimize the risk of damaging this oil seal, put the oil seal into position in the recess of the caliper half, and then insert the bolts and tighten them simultaneously by hand. Make sure that once the caliper half touches the O-ring you tighten both bolts simultaneously so that the oil seal does not become cocked as you begin compressing the O-ring. After you cannot tighten them any further by hand, get out your torque wrench. Alternately tighten each side in increasing five-pound increments until the proper torque reading of 35.5 to 37 Ft-lbs is reached.

Install the hoses (flexible pipes) along with new copper washers. Smear a very thin coat of compound onto the rear face of the brake pads, and then slide the brake pads into place. Install the retaining clips and secure them with new, preferably stainless steel, cotter (split) pins.

You will find that it is worthwhile to use anti-seize compound on the threads of the SAE 7/16-20 UNF shouldered mounting bolts when you reinstall your calipers onto the car. Refit the calipers to the car using new tab washers, torque their mounting bolts to 43 Ft-lbs, and then bend over the lock tabs with a punch. Finally, refill and bleed the brake system.

Although it is common practice amongst track racers to remove the dust cover plates to promote greater airflow to the disc brakes, this is not recommended for a car that is intended for use on the street as the dust cover plates protect both the brake rotors and the brake pads from road debris, preventing small stones from becoming wedged between the brake rotor and the friction material of the brake pads. If this happens, it will result in scoring of the brake rotor.

If you are reconditioning your original brake rotors, take them to a competent automotive machine shop and have them surface ground to a non-directional 60 microinch finish. Run-out must not exceed .002" (.0508mm) and the thickness must be parallel within .001" (.0254mm. If you are checking the run-out with the brake rotors attached to the car and encounter excessive run-out, then be sure to inspect the hubs in order to ensure that they are not the actual source of the problem. Do not use a brake rotor that has a finished thickness of less than .300" (7.62mm). Having them lathe-turned and finished with sandpaper is obviously highly inadvisable, although this is the method that commercial garages all but universally employ. This crude technique will promote glazing and squealing. However, before committing yourself to having them resurfaced, be sure to examine them carefully. Blue areas of the brake rotor are a localized conversion of cast iron into cementite, an extremely hard substance. This transformation takes place at very high temperatures and is non-reversible. As the blue areas of cementite, being harder, will be less subject to wear than the rest of the surface, the phenomenon will spread with each braking action. This transformation of the cast iron effects the brake rotor to such a depth that a refacing of the surface will not resolve the problem. To make matters worse, these areas of cementite have a different coefficient of expansion and contraction than that of the neighboring cast iron material, so the brake rotor will warp as it heats up under friction. Such rotors should always be discarded as scrap.

In recent years, much has been made of the use of cross-drilled brake rotors. These items were developed primarily for racing use. Some people think that they are intended to facilitate cooling, but rust quickly builds up in the holes, acting as insulation. Other people think that the holes are present for water to be displaced into by the brake pads. Originally, they were intended to allow the venting of gases that formed at high temperatures between the brake pads and the brake rotors. However, modern brake pads have a reduced tendency toward out-gassing, so today the primary purpose of the holes is to reduce unsprung weight in the suspension system. This introduces a problem: as the mass of the brake rotor decreases, so does its ability to contain heat. In addition, since drilling creates stress lines that radiate outwards from the holes, they have a tendency to develop a cracking problem, especially if a dull drill bit is used to create them. In addition, the more holes drilled into the brake rotor, the weaker it becomes. To make the reduction in unsprung weight worthwhile, well over a hundred holes have to be drilled ($$), which results in lots of stress lines. All of these holes should be chamfered slightly to equal depths after surface grinding, then the surface lightly reground to a non-directional finish in order to remove any lipping (more $$), then the brake rotor has to be stress-relieved in a furnace in order to reduce the likelihood of cracks forming along stress lines (more $$). These are the reasons why such racing brake rotors are so expensive. The majority of aftermarket brake rotors that have far fewer holes drilled in them remain sufficiently strong after drilling so that the furnace-dependent stress-relieving process is not as necessary and can be eliminated (thus saving $$$), and are surface-ground only once, this being afterward drilling in order to reduce the need for chamfering (thus saving more $), but in practical terms the total reduction of unsprung weight is insignificant. The brutal truth is that the fewer holes in such brake rotors are in effect nothing but a cosmetic sales gimmick aimed at the “monkey see, monkey do” market niche. The cross-drilled brake rotors used by track racers have so many holes in them that few would buy them for street use because their higher cost would make them too expensive. While their reduced unsprung weight makes them a useful modification for the racetrack, the tendency of the holes to become coated with rust and clogged with brake material make it questionable as to whether or not they are a worthwhile investment for a car that is intended for use on the street. Track racers can take the time after a race to use a radial brush in order to remove these materials from all of those holes, but how often would you perform such maintenance on your street machine?

Directionally grooved slotted brake rotors offer the advantage of being less prone to becoming clogged with brake pad material and are far more efficient in that they use centrifugal force in order to duct water off of their surfaces, making for faster and better brake response under wet driving conditions, thus making them the superior choice for street use. They are also far less prone to glazing the brake material. The downside is faster brake pad wear. The brake rotors manufactured by Tarox and Red Dot are of exceptionally good quality, so much so that warpage even at the highest temperatures is a quite rare experience. These are available from the MG Owners Club in the UK. They have a website at http://www.mgownersclub.co.uk/ . They also sell slotted brake drums as well.

Brake Pads And Shoes

Today's brake pads and shoes are available in a wide variety of materials. Materials intended for racing applications are unsuitable for street use as they perform well only when hot. At the temperatures incurred outside of a racetrack their performance is actually inferior to that of friction materials that are intended for street use. Rather than use racing brake material, install a set of MGB GT V8 brake pads into the calipers. They will fit without modification and, due to their larger surface area that dissipates heat more easily, are more fade-resistant. Be aware that these brake pads are "handed" not in terms of right and left, but of inner and outer.

Avoid the use of brake pads made of the Original Equipment organic compounds as they are the least heat resistant, have the poorest coefficient of friction of .32mu, and produce more brake dust than any other type of material. The scientific definition for coefficient of friction is the ratio of perpendicular force required to move or stop one surface. This is expressed as Mu = F/W Where: Mu = coefficient of friction, F = force required to move one surface over the other, and W = perpendicular force. The industry give brake pads a numeric tag that describes their friction coefficient (Mu) wherein 0 is classified as the worst and 1.0 is the best.

Be aware that the transfer layer of material deposited on the brake rotor is what determines the stability and effectiveness of the brake pad's coefficient of friction. Brake pad material is constantly moving between these two surfaces during brake engagement. Every brake pad leaves a transfer layer. This is one reason why a new brake pad will not work optimally against a brake rotor that had previously run a different type of brake pad. Consequently, if you change to a different type of brake friction material, you should have the brake rotor either resurfaced or, ideally, replace it with a new one.

There are essentially three options for high performance brake friction material. The first and perhaps the most commonly available material marketed for a high performance street application are the Carbon Metallic compounds such as those marketed by Hawk. These seem to come in two categories: those suitable only for racing and those suitable only for street use. Those suitable for street use have a coefficient of friction of .36mu, which is too small an increase in performance (11%) over that of Original Equipment materials to make them worth the additional expense. In addition, they produce a black brake dust that is difficult to remove. The second choice is the Semi-Organic / Semi-Metallic type. These produce less brake dust than Carbon Metallic materials. While being more heat resistant than organic compounds, they also have a superior coefficient of friction of .48 mu, a fifty percent improvement over the .32 mu of the Original Equipment material, and a thirty-three percent improvement over Carbon Metallic materials. These are available from Carbotech Engineering. They have a website at http://www.carbotecheng.com . While these may be popular, there is another material that has an equivalent coefficient of friction, but yet an even greater resistance to heat: the Carbon Kevlar type. This material will not begin to outgas until it reaches a temperature of 1,050º Fahrenheit (565º Celsius), thus making brake fade a factor than can be dispensed with. These are available from TSI Automotive (Brake Pads- Part # CKPMGA/B, Shoes- Part # CKSMGA/B). They have a website at http://www.tsimportedautomotive.com . Be advised that whatever material you choose for the front brakes should also be used on the rear brakes as well so that the coefficients of friction will be equal, otherwise one pair will prematurely lock up their wheels under heavy braking.

It is possible that, under the heavy braking loads generated by stronger brakes, the rear brakes may lock up prematurely, creating tail drift. This can be tuned out of the braking system by installing a proportioning valve into the rear braking system of the Roadster model or by changing the brake slave cylinders of the GT model to those of the Roadster model with their smaller-diameter pistons. The latter change will require exchanging the rear brake backplates of the Roadster model for the ones of the GT model in order to fit the different size brake slave cylinders. Another. Though partial, solution is the fitting of tires with more grip, although this can be said to be treating the symptom rather than the cause.

The Rear Brakes

The rear brakes on an MG are straightforward, being of orthodox design. With a proper manual, you should be able to do the work yourself. First, loosen the adjustment of the hand brake cable. The hand brake adjuster is in the transmission tunnel under the lever. Screw the brass nut in or out in order to adjust the hand brake while holding the screw thread still with a pair of pliers or grips. Remove the cotter pin (split pin) from the hand brake lever on each brake drum backplate. In order to remove the drum you need to jack the car up (for safety, always use axle stands). Take off the wheel. If your car is equipped with wire wheels, remove the four nuts that secure the brake drum to the hub. If your car is equipped with disk wheels, remove the two screws that secure the brake drum to the hub. You may need to use an impact driver if they are very tight. You will probably find removal to be a lot easier if you back off the adjusters of the brake shoes before you attempt to remove the drum. With judicious prying / tapping, you should be able to remove the brake drum. At this point it is easiest to clean the entire brake assembly with CRC Brakleen.

Fully slacken the brake shoe adjuster located at the top of the brake assembly backplate. Depress each retaining washer of the brake shoe steady springs, and then rotate them until they release from their steady pins, and then remove the retainer washers and the brake shoe steady springs. Pull the trailing brake shoe against the load of the brake shoe return springs, and then disengage the brake shoe return springs. Upon releasing the tension of the brake shoe return springs, the brake shoes should fall away from the backplate of the brake assembly.

In order to remove the brake slave cylinder, first disconnect the brake hydraulic pipe. Remove the locknut that secures the union of the pipe to the bracket. At that point, you may unscrew the pipe from the brake slave cylinder. Remove both the Dreaded E-Clip and the retaining washer that together anchor the brake slave cylinder at the rear of the backplate.

Be aware that the rear brake slave cylinders on the 4-cylinder GT model are of a larger bore, 7/8" (BMC Part # GWC 1122), as opposed to the 0.8" bore (BMC Part # GWC 1103, applicable to both Hardy-Spicer Banjo-type and Salisbury tube-type axles) of the rear brake slave cylinders of the Roadster. This larger-diameter piston gives more braking effect under the additional rear weight of the GT model without locking up the wheels. They differ externally in the position of the locating peg, and hence the back-plates are also different from GT model to the Roadster model. If you are experiencing rear wheel lockup under heavy braking, more efficient braking up front will result in lower hydraulic line pressures with less chance of the rear brakes locking up. The basic facts that you need to be aware of when dealing with brake system pressures is how they are effected by the components that you use. The amount of hydraulic pressure produced at the pedal is inversely proportional to the master cylinder bore, and the pressure produced at the caliper or wheel cylinder is the opposite of that. Therefore, a master cylinder with a smaller diameter piston will develop a higher line pressure than a slave cylinder with a larger diameter piston at the working end of the brake system. Conversely, a smaller diameter caliper piston or slave cylinder will develop lower line pressure at the working end of the brake system.

To reinstall the Dreaded E-Clip that secures the slave cylinder, the slave cylinder must be held tightly against the brake backplate. This is best achieved by an assistant levering it back against the brake backplate with a large screwdriver. However, if you are working alone, you can clamp a pair of Vise Grip pliers onto the wheel flange of the axle so that the back of the Vise Grip pliers braces the slave cylinder against the brake backing plate and holds it in place. Next, fit one end and the middle of the E-Clip over the retention groove of the brake slave cylinder. Next, be gentle, and while trying to cause as little bending as possible, use a smaller screwdriver to twist or pop the last leg of the E-Clip into place. You will notice that all three tabs are not seated fully home. Now, place a socket over the brake line boss that is on the slave cylinder. Thread on a 3/8"-24 nut fully home onto a 3/8"-24 machine bolt, and then slip the washer on after it. Install the 3/8"-24 machine bolt / nut / washer assembly onto where the brake line would normally go onto the slave cylinder (this will all be placed through an 11/16" socket). Note that there is no need to torque the machine bolt tight with a torque wrench. Once the machine bolt has seated, turn the 3/8" nut and washer against the socket. This will push the E-clip home. Do this until you see all three tabs engage the groove on the wheel cylinder. Give the E-clip a few gentle taps with a lightweight hammer and drift it inward in order to ensure that all three tabs are properly engaged and fully seated. Be aware that early Honda Civics and Honda Accords have a nice rubber boot seal that is found on the adjuster studs of their rear brake backing plates. Purchase a pair of these rubber boot seals from a Honda dealership. They fit perfectly over MGB rear brake adjusters, keeping dirt and corrosion off of them.

Some people have the experience of the rear brakes on one side going out of adjustment very quickly. When this occurs and measurements are done, the friction material on the brake shoes are usually found to be of unequal thickness. The most common cause of this is brake fluid leaking out of the brake slave cylinder and soaking into the shoes, resulting in uneven wear. Check both brake slave cylinders for signs of leakage. In addition, check the axle seals for signs of leakage, as well. Do not fail to replace the dust excluder caps (piston boots), even if the seals and the pistons do appear to be in good shape. Brake dust is highly abrasive and consequently will ruin the seals of your brake slave cylinder pistons very quickly.

Remove the cotter pins from the clevis pins (split pins) from the hand brake mechanism, and then remove the clevis pins in order to release the hand brake cable from the hand brake mechanism. Be aware that the handbrake cable attachment is completely different on the 77 and later model years. It includes a much simpler, lower-maintenance brake compensation arrangement of just a flap of rubber with a frame, rather than the comparatively expensive pivot and reaction rods arrangement used previously. Whereas the pivot mechanism is mounted by using two of the bolts of the differential cover, the rubber flap is simply bolted to a flange bracket that is welded onto the casing of the axle near the left-hand side. A longer brake cable acts directly on the left rear brake, the outer cable sheath being secured into the end of a frame-shaped bracket that is attached to the rear axle by means of a rubber strap. Attached to the opposite end of the frame-shaped bracket is a cable that is also attached to the manual brake mechanism on the right rear brake. Brake balance is provided by the movement of the frame, the rubber strap allowing the required flexibility while still supporting the frame and the cables. Whereas an earlier cable can be fitted to a later axle once you have obtained the pivot mechanism, the later cable will need the axle flange bracket, or something akin to it. It has been claimed that the change in handbrake cable had something to do with the provision of the new rear stabilizer bar which arrived at the same time, but this is not the case. The changes for the cable and the anti-roll bar have no interdependence. Remove the rubber dust cover from the hand brake mechanism, and then remove the hand brake mechanism. Clean it thoroughly with CRC Brakleen. Be aware that there are two different length cables for the handbrake system. The disk wheel axle is longer than the wire wheel axle, and therefore the cable is longer. Usually when the handbrake no longer holds, it is time to adjust the brake shoes a little closer to the drums. Firstly, you must always totally back off the handbrake before you attempt to adjust the rear brakes.

Preferably, remove the clevis pin where the handbrake cable is attached to the compensator mounted to the diff housing. This ensures that the adjustment at the wheels is not interfered with by the handbrake being partly on.

The brake shoe adjusters are among those parts that receive very little attention until seizure occurs. It should be an annual task to remove the drum and shoes, remove the adjuster by screwing it out of the backplate from the backside. They have a square ¼" machine bolt head protruding through the brake back-plate. The adjuster is removed by screwing it in towards the brake drum, not out through the back-plate. It is possible to strip it by trying to remove it through the outside of backplate, i.e., towards the centerline of the car. If you attempt this, it should eventually stop, but if the threads are worn and you force it, then you will strip them. Put a light coat of antisieze compound onto the threads, and then screw it all the way back into the adjuster casting (i.e., towards the centerline of the car). Next, refit the brake shoes and the brake drums.

If the drums have been turned, then, ideally, the brake shoes should be ground to match the increased brake drum diameter. A good brake shop should be able to do this. Otherwise, the shoes will have a reduced contact area, accelerating heat buildup and consequent brake fade. If you examine the friction lining material on the shoes, then you will see that it only covers part of the metal shoe. The leading brake shoe (the one nearest the front of the car) should have the unlined section at the top, while the trailing shoe should have the unlined section at the bottom. When a set of shoes have been correctly assembled you will note that the lever of the hand brake system that connects to the trailing shoe should pass through the hole that is bell-shaped. The leading shoe (the one closest to the front of the car) should be oriented so that the bell-shaped hole is located at the top of the backing plate. The brakes should be assembled in the same manner on both sides of the car. That is, with the "bell" of the leading shoe facing upwards, and with the "bell" of the trailing shoe facing downwards. Thus, when they have been installed, the bare metal end of each brake shoe should be directly opposite the covered end of the opposite shoe. If the orientation of the shoes are flipped, the consequence will be that the hand brake lever can become wedged in the narrow horizontal slot, jamming the hand brake mechanism, and can cause premature shoe wear as well. Be sure to install the return springs of the brake shoes on the inside of the brake shoes, not on the outside of them. Be advised that the pull-off spring that has two coils (one at each end) (BMC Part # 27H 7995, left rear brake; 27H 7994, right rear brake) is installed at the brake slave cylinder end of the shoes, while the pull-off spring that has a single coil in its middle (BMC Part # 27H 6478) is installed at the brake shoe adjuster end of the brake shoes.

After you have adjusted the brakes, you can then re-connect the handbrake and adjust it. You will often find that the clevis pin will no longer fit, and you have to back off the handbrake cable so that it will come through enough to install the pin. If this is the case, you have just demonstrated to yourself the problem previously referred to.

Due to the Hardy-Spicer banjo-type rear axle and the Salisbury tube-type rear axle rear axle having different hub designs, the brake backplates and brake drums are different and are not interchangeable. The handbrake cables are also different for the two types of rear axles. The handbrake levers (which are handed) for both rear axles are the same part numbers (BMC Part # 17H 2005 and 37H 2006), but have to be fitted to the opposite side on a Hardy-Spicer banjo-type rear axle in relation to a Salisbury tube-type rear axle, otherwise the handbrake adjustment and operation will be adversely effected. The levers will foul on the back plates, preventing the adjusters from having enough travel to pull the brake shoes far enough outward to make effective contact with the brake drums. On the brake mechanism of a Hardy-Spicer banjo-type rear axle, the short lever that goes through the front of the brake backplate must be located above the long lever that fits between the shoes. On the brake mechanism of a Salisbury tube-type rear axle, the hand brake lever must be installed into the brake backplate the opposite way around, i.e. the short lever must be below the longer lever that fits between the brake shoes. This can be accomplished by simply switching the parts from one side to the other side. Their pull-off springs for the handbrake levers, however, are quite different parts. The pull-off springs (BMC Part# 17H8737) for the handbrake levers of the Hardy-Spicer banjo-type rear axle have a single coil that runs along its entire length, while the pull-off springs (BMC Part # 27H 6479) for the handbrake levers of the Salisbury tube-type rear axle has dual coils, one at each end, somewhat similar to the pull-off spring used at the brake cylinder end of the brake shoes. Beyond these parts, all of the rest of the parts of the rear brake mechanisms are the same in both cases.

Until your next change of brake linings, you can adjust your brakes at the wheels without having to disconnect the handbrake. But the next time that you fit new linings, you will need to go through this process again. Why is all this necessary? Simple: Owners (and mechanics) who see the handbrake lever pointing to the sky and the rear brakes not coming on, assume that they have a problem with the handbrake. So they adjust the handbrake cable to solve the problem. Many cars will have the adjustment right to the end of the threaded part of the cable, and the handbrake is still not fantastic. But the problem was not the handbrake at all - it was poor adjustment of the brakes themselves at the wheels. So now the handbrake mechanisms inside the brake drums are taking the place of the brake shoe adjustment, with the result that neither of them work correctly. This has probably happened to every MG at some time during its lifetime. Now is the time to zero everything and get it right.

Install the hand brake lever and its return spring, then screw the brake shoe adjuster inwards to the correct position. Apply a little grease to the threads that are exposed on the back of the backplate. This last action forms a ring of grease around the junction of threads, which helps to prevent the ingress of moisture.

Once you have the brake shoes correctly adjusted, look at the angle that the operating lever of the hand brake mechanism forms with the backplate when the brake shoes start to make contact with the friction surface of the brake drum. Ideally, for the greatest leverage (and hence the greatest braking force), they should be parallel to each other. However, over time, the notches in the operating levers wear deeper, necessitating that the lever be pulled past the parallel point before it starts to move the shoes, thus reducing braking effect, as well as using up all of the adjustment at the handbrake lever end.

Brake Fluids

Fluid Fade is caused by the boiling of the brake fluid in the calipers, producing bubbles in the brake system. Since the bubbles are compressible, this makes for a soft spongy pedal. In worse cases, the pedal can plunge to the floor with very little braking! Fluid fade can be avoided by running a high-grade brake fluid and / or frequent changes of brake fluid. Also, if you change the brake pads before they get super thin, the remaining friction material will help insulate the calipers from the heat.

Brake fluids containing Polyglycol ethers are regarded as DOT 3, DOT 4, and DOT 5.1. These brake fluids are hygroscopic, meaning they have an ability to mix with water and still perform adequately. The term “Wet Boiling Point” refers to the minimum temperature at which a brake fluids will begin to boil when the brake system contains 3% water by volume of the system. The term “Dry Boiling Point” refers to the temperature at which brake fluid will boil with no water present in the system. Amongst the brake fluids presently available there are three possible candidates. The first, DOT 3, is a poor choice for high performance driving due to its low dry boiling point of 401° Fahrenheit (205° Celsius) and a wet boiling point of 284° Fahrenheit (140° Celsius) and is now generally considered to be obsolete. Be advised that American DOT 3 brake fluid is formulated differently than British DOT 3 and is incompatible with the aging Original Equipment natural rubber seals that were used throughout the brake system. It will slowly but surely dissolve them! However, after all of these years the original seals have probably long ago been replaced with seals that are made of more modern materials. The second candidate, DOT 4, is much better with a dry boiling point of 446° Fahrenheit (230° Celsius) and a wet boiling point of 311° Fahrenheit (155° Celsius). Of the different brands of DOT 4 brake fluid on the market today, Castrol LMA synthetic with its dry boiling point of 509° Fahrenheit (265° Celsius) and a wet boiling point of 329° Fahrenheit (165° Celsius) appears to be the best available. It is also has the advantage of being markedly less hygroscopic than either DOT 3 or DOT 4 petroleum-based brake fluids.

DOT 5 Silicone-based brake fluid is a poor choice for any automobile as it has problems with air retention, making bleeding of the brake system a real bear, and poor lubrication, sometimes allowing the pistons of calipers and / or brake slave cylinders to bind in their bores and lock up the wheels. With a dry boiling point of 500° Fahrenheit (260° Celsius) and a wet boiling point of 356° Fahrenheit (180° Celsius), its performance is inferior to that of Valvolene SynPower. While it is true that silicone-based brake fluid does not absorb water, water still gets into the system through condensation. Because water is heavier than silicone brake fluid, it will ultimately sink and gather in the lowest point in the system. Should it freeze, line blockage and brake failure becomes possible. It is also highly possible that should the temperature of the brake fluid rise above 212° Fahrenheit (100° Celsius), the water will vaporize, increasing pressure within the system and locking a brake. Brake systems with small orifices or rapid action of the system, such as automatic proportioning valves and antilock braking systems should definitely not use silicon brake fluid since the small orifices and rapid operation of the system will cause airification of the brake fluid as a result of cavitation, which will at best certainly cause a spongy pedal. It also has the disadvantage of being highly compressible, and thus can give the driver a feeling of a spongy pedal. The higher the temperature of the brake system becomes, the more the compressibility of the fluid will increase, and this increases the feeling of a spongy pedal. By comparison, Polyglycol type fluids are 2 times less compressible than silicone type fluids, even when heated. Less compressibility of brake fluid will increase pedal "feel". Should you decide to use silicone-based brake fluid, be sure that all of the seals are in excellent condition as moisture will easily find its way past a leaky oil seal, and air will also get into the system. Diffusion of water throughout the brake fluid occurs over time whenever moisture enters through rubber brake hoses. The use of hoses made from EPDM materials (Ethlene-Propylene-Diene-Materials) will reduce the amount of diffusion, or use steel braided brake hydraulic hose with a non-rubber interior sleeve (usually Teflon) that is impermeable to air in order to greatly reduce the diffusion process. Be sure to flush the system with denatured alcohol prior to refilling it with the silicone brake fluid. Failure to do so will result in the residual glycol-based brake fluid interacting with the silicone brake fluid to form a sludge-like material that can be a real bear to remove from the brake system. This sludge will destroy the seals in the system, resulting in catastrophic brake failure. Perhaps the only argument in favor of silicone brake fluid is that it is slow to damage paint. However, if the brake system is leaking, it should be promptly repaired in the interests of safety, regardless of the type of brake fluid used in it.

Bleeding The Brake System

Brake bleeding has always been an inconvenient and messy job. Actively pulling (vacuum) or forcing (pressure) brake fluid into the brake system are alternative methods to bench bleeding, but they require either an air compressor or a source of vacuum. Commercial shops typically use an expensive pressure bleeder to swiftly do the job. Unfortunately, unless used properly, a pressure bleeder can introduce air bubbles into the system through cavitation. Vacuum bleeders such as a Mity-Vac are much less expensive but can also introduce air into the braking system if the brake master cylinder is not kept full of brake fluid while using it. Personally, I use a Mity-Vac to pull fresh brake fluid into the system as, if proper procedures are adhered to, it eliminates the need for bench bleeding altogether and it minimizes dripping brake fluid onto the car. I then follow this up with a few 3s of conventional pedal bleeding as I find that it helps seat the seals, and also because this action is able to force the brake fluid through the lines more quickly than a Mity-Vac in order to help further dislodge any remaining stubborn bubbles.

The time-honored manual brake bleeding method (still universally used in racing) requires two people: one to pump the pedal, and another to open and close each bleed screw in sequence. As with all brake bleeding methods, proper procedure and coordination of activity are required in order to prevent air inclusion. A Solo-Bleed bleeder screw (Part # 280022ERL) solves both problems. This device makes use of a spring-loaded plunger anti-backflow valve in order to allow one person to safely bleed brakes without a mess and with no chance of air inclusion. It is a direct replacement for the original equipment bleeders. This handy item is marketed by Holley Performance Products and can be found at their website http://www.holley.com/search.asp .

There are always those who do not have access to such sophisticated tools, or do not care to invest in them. Prior to installing a brake master cylinder, you can spare yourself considerable time and effort by bench bleeding it. A normal brake master cylinder bench bleed kit will contain two fittings that screw into the line fittings, and two short lengths of hydraulic hose. Be aware that you will still have to bleed the entire brake system after you install the brake master cylinder, but bench bleeding the brake master cylinder prior to its installation saves you from much of the initial dry pedal stomping and empty (air) bleed screw-turning.

Mount the flange of the brake master cylinder on a sturdy workbench. Next, install the short steel brake lines (pipes) into the outlet port(s). Install clear vinyl hoses onto each end of the short steel brake lines (pipes) and use hose clamps in order to insure a good seal, then route the free ends of the hoses into an empty container. Some people unwisely route the plastic hoses back into the reservoir of the brake master cylinder when bench bleeding, but I do not recommend this as this recycles contaminants (grit, swarf, rubber bits) back into the brake master cylinder. Not a good practice. Fill the brake master cylinder reservoir with fresh brake fluid. Manually push the brake master cylinder piston all the way home, and then hold it in. Pinch either one of the clear vinyl hoses shut with a pair of needle-nosed pliers, or non-locking surgical hemostat. Slowly release the brake master cylinder piston, allowing the brake fluid to be drawn into the cylinder. Release the hose (pliers). Keep the ends of the hoses immersed in the brake fluid in order to prevent air from migrating back into the cylinder. Repeat this process until all of the bubbles are removed from the brake master cylinder. While performing this process, occasionally tap the brake master cylinder lightly in order to help dislodge any trapped bubbles. Be sure to keep the reservoir filled during this process. Go to the other hose and Repeat this process for that hose. You are finished bleeding once all of the air has been expelled from the brake master cylinder. Remove the steel lines and plug the ports with protective plastic caps or rubber stoppers to minimize brake fluid loss. As you do not want to drip brake fluid anywhere on your car, clean the exterior of the brake master cylinder of all brake fluid, then wrap the brake master cylinder in a clean rag prior to carrying it to your car for installation.

Bedding-In The Brakes

Proper bedding-in allows your brakes to reach their full potential. Until they are properly bedded, your brakes simply will not work as well as they could. Proper bedding will improve pedal feel, reduce or eliminate brake squeal, and extend the life of your brake pads and brake rotors. After installing new brake pads and brake rotors, the first few applications of the brake pedal will result in almost no braking power. Brake pads are usually made of different types of heat resistant materials bound together with a phenolic resin binder. These are thermosetting plastic resins with a high heat resistance. On a new brake pad, these resins will out-gas or cure when used hard on their first few heat cycles. The new brake pad can hydroplane on this layer of excreted gas. This type of brake fade is referred to as “green fade”, and is dangerous because many people assume that new brakes are perfect and can be used hard right off the bat. Green fade will typically occur much earlier than normal fade, so it can catch a driver that is used to a certain car’s characteristics unaware. Typically the onset of green fade is rather sudden, further increasing the danger factor. Green fade can be prevented by proper bedding of the brake pads. Fortunately, there is a simple procedure to boil off the resins and break in the brake pads under controlled conditions. The brakes should not be dragged during this procedure.

Gently apply the brakes a few times at low speed in order to build up some grip before blasting down the road at high speed. Otherwise, you may be in for a nasty surprise the first time you hit the brakes at 60 MPH.

Bedding-in is often best done early in the morning, when traffic is light, since other drivers will have no idea what you are up to and will respond in a variety of ways ranging from fear to curiosity to aggression. An officer of the law will probably not understand when you try to explain why you were driving erratically! From a speed of about 60mph, gently apply the brakes a couple of times to bring them up to operating temperature. This prevents you from thermally shocking the brake rotors and brake pads in the next steps. Thermal shock occurs when the brake rotor is heated up and cools down rapidly, causing an uneven heat distribution throughout the brake rotor. The result is the expansion of the brake rotor material in one area, while not expanding in another area. Heat checks in the brake rotor are a symptom of thermal stress. Be careful not to confuse small cracks in the transfer layer on the surface of the brake rotor as evidence of thermal shock.

Make a series of eight near-stops from 60 to about 10 MPH. Do it hard by pressing on the brakes firmly, just shy of locking the wheels. At the end of each slowdown, immediately accelerate back to 60mph. Do not come to a complete stop. If you stop completely and sit for any length of time with your foot on the brake pedal, you will imprint brake pad material onto the hot brake rotors, which can lead to vibration, uneven braking, and could even ruin the brake rotors. The brakes may begin to fade slightly after the seventh or eighth near-stop. This fade will stabilize, but not completely go away until the brakes have fully cooled. A strong smell from the brakes, and even smoke, is normal.

After the eighth near-stop, accelerate back up to speed and cruise for a while, using the brakes as little as possible. The brakes will need 5 to 10 minutes to cool down. Try not to become trapped in traffic or come to a complete stop while the brakes are still hot. There should be a slight blue tint and a light gray film on the brake rotor face after the bed-in cycle. The blue tint will tell you the brake rotor has reached bed-in temperature, and the gray film is brake pad transfer material starting to transfer onto the brake rotor face. It is this transfer that you are trying to accomplish. The best braking occurs when there is an even layer of brake pad material deposited across the face of the brake rotors. This minimizes squealing, increases braking torque, and maximizes both brake pad and brake rotor life. After the first bed-in cycle, the brakes may still not be fully broken in. A second bed-in cycle, after the brakes have cooled down fully from the first cycle, may be necessary before the brakes really start to perform well. This is especially true if you have installed new brake pads on old brake rotors.

Brake squeal is normally the result of vibration between the brake caliper pistons and the steel backing plate of the brake pads. The Original Equipment design brake caliper pistons were fabricated with a cut-away notch that was installed with its cut-away notch facing toward the mounting lugs of the caliper. The advent of anti-squeal pads that are fitted between the piston and the steel backing plate of the brake pads largely eliminated the need for this design feature. However, the combination of these two features usually results in satisfyingly quiet braking.

Handling

Of course, all efforts toward increasing power output and getting it onto the ground will eventually result in further considerations of making improvements in other areas of performance, such as the handling and braking of the car. Fortunately, when the geometries of the steering and the suspension of the MGB were developed, the design work was not hampered by a requirement for the incorporation of components that were originally intended for use in other vehicle designs that were already in the BMC stable. Instead, the engineers at MG were given a free hand in their work to create geometries that were optimum for the MGB. Even the suspension geometries of the Armstrong lever arm dampers were developed specifically for application in the MGB. The consequence of this free-handed approach was superior handling.

The handling qualities of a car are all about control. Superior handling makes for superior control. Without control, high performance driving is suicidal. When seeking the control that better handling provides, it is necessary to first make sure that the steering column and steering rack are in good condition. If any slop or leaks are apparent, then these deficiencies must first be addressed by rebuilding these components before the suspension itself can be raised to a higher performance specification.

The Steering Column

Be aware that the North American Market steering columns from 1968 thru 1980 were of collapsible design and were fabricated by Saginaw (General Motors) in the USA. This same design was used in some non-North American Market MGBs as well. To identify a North American Market specification collapsible column, look under the dashboard in order to examine the outer tube. Collapsible steering columns usually have a plastic wrapping, about 12 inches in length, which covers the section of the column that has been perforated in order to enable it to deform and collapse upon impact. Sometimes the plastic covering is absent, exposing the diamond shaped perforations of the tube. Contrary to appearances, there are no transverse plastic pins in the collapsible column. There are two bands of plastic, injected into the space between the inner and outer steering shafts. This forms a slip joint that allows the steering shaft to collapse telescopically upon impact. The plastic is injected through two opposite pairs of holes in the outer shaft, thus giving the impression that there are two transverse plastic pins. However, one cannot tell what is inside until the steering column is disassembled. Be aware that beating on the end of the shaft in order to loosen the steering wheel so that it can be removed is a very bad practice. Do not do this. Use a proper gear puller in order to do the job. Beating on the shaft can destroy the plastic slip joint. There is a circlip behind the top bearing assembly that will prevent movement. If this circlip is removed or missing you can get as much as 2" (50.8mm) of lengthwise play. If you disconnect the inner shaft from the Universal joint by removing the machine bolt, you can pull it out inside of the car to have a look. Just make sure to have the ignition key in place so that the steering lock plunger is withdrawn.

The Steering Rack

The steering rack of the MGB was made by Cam Gears. Its legend is cast into the aluminum body of the steering rack housing. There were actually four similar, but different steering racks employed in the BMC B Series-engined version of the MGB, with two basic versions and two mirror images of each pattern, the mirror images being for left hand drive and right hand drive. The initial version of the steering rack was used on the Chrome

Bumper models and had a lock-to-lock ratio of 2.93:1. However, as the weight of the car increased as a consequence of the provision of government-mandated safety features and equipment, plus a change to sportier appearing, smaller-diameter steering wheels that burdened the driver with less steering leverage, the need arose to reduce steering effort. The result was the succeeding version of the steering rack installed into the Rubber Bumper models which had a lock-to-lock ratio of 3.57:1. Because of the need to position the universal joint that connects the shaft of the steering column to the pinion shaft further away from the rear of the exhaust manifold on the V8 engine MGB GT V8 models, the steering column of all of the Rubber Bumper models is shorter, but still aligned the same as a Chrome Bumper model, and the pinion shaft is longer. To meet the new, shorter steering column in its new position, the steering box of the Rubber Bumper model has a longer pinion shaft, and the steering rack is mounted onto redesigned mounts that are more askew to the front crossmember. The mount on the passenger's side has its locating holes further forward than those on the driver's side, causing the steering rack to assume an orientation that is non-parallel to the front crossmember. This permits the same steering rack housing to be used on both models. If you wish to fit the steering box of a Chrome Bumper model to a Rubber Bumper model so that its faster steering ratio can be used, then all that needs to be done is to extend its pinion shaft to the same length as that of a Rubber Bumper model.

Before you start any maintenance on any components of the steering assembly, turn the steering wheel so that the front wheels are in the straight-ahead position and check to see if the steering wheel is in a likewise position. Mark the upper section of the steering column and the universal joint at their juncture in order to facilitate easier reassembly should you find it necessary to disassemble these components.

Loosen the two machine bolts that secure the yoke cover plate to the body of the pinion gear housing, and then remove both it and the spacer shims. Take care to not lose the spacer shims. Remove the steering rack support yoke, the damper spring, and the damper pad. Now, check the small brass damper pad for wear. If it has more than about .010" (.254mm) of wear in the form of a groove shaped like the steering rack, it should be replaced. Such wear can cause the steering rack support yoke that it sits inside of to bear against the steering rack and cause binding, not to mention rapid wear of these components. You will find that the oil leaks past any irregularity in the shims or the face of the steering rack, so make sure that everything is spotlessly clean before you proceed with reassembly.

Replace both the rack support yoke and the damper pad back into the pinion housing without the damper spring and the shims. These shims are not present in order to adjust the force of the damper spring. However, if you remove all of the shims, and then tighten down the cover bolts, you will be forcing the plunger solid against the rack in a manner so that the rack will bind and not move. The basic idea is to install enough shims under the cover nut in order to keep the plunger from binding, and to maintain the smallest practical space for motion of the plunger. That way when you hit hard bumps with the tires there is very little room for the rack to move radially, which will minimize rattle and shimmy. The workshop manual calls for .002” to .005” of end clearance for the damper plunger. Gently tighten the two machine bolts that secure the yoke cover plate to the body of the pinion gear housing until slight tightness is felt, and then back it off so that it is just free enough to permit rotation of the pinion shaft by moving the rack. Measure the gap between the yoke cover plate and the pinion housing with a feeler gauge, and then add an additional clearance of .0005” to .003” (.0127mm to .0762mm) to the figure in order to arrive at the correct total thickness of the shims. The damper spring serves to maintain the contact of the rack with that of the pinion gear in order to eliminate any free play in the gear teeth. As the gear teeth have very little friction, this spring force can be substantial without causing much drag. Since the mating angle on the teeth is about 20º, which tends to force the teeth away from each other under conditions of high loading, the spring force must be fairly strong. The damper spring at the other end of the steering rack forces the moving rack against the inside bore of the housing in order to eliminate side play. This side thrust is not as great as that caused by the pinion gear, as the side thrust comes only from the thrust of the tie rod when it is not perfectly aligned with the rack. In this configuration the rack is dragging directly against the bore of the housing, so it is desirable to limit the side loading of the damper to a minimum.

Refit the shims, the rack support yoke, the damper spring, and the damper pad, and then tighten the two machine bolts that secure the yoke cover plate to the body of the pinion gear housing. Check the steering rack movement from end to end. If there is any binding, then install an extra .002" shim. Take care to be sure the shims are all properly aligned and that the steering rack support yoke is riding on the cover plate and not on a shim. This can happen on reassembly after all of your careful measurements. Once you have checked and adjusted the pre-load, any remaining play is wear. If you twist the pinion, there should be no amount of play before it actually moves the steering rack.

A good way to think about the lip of an oil seal is to consider which way any pressure is going to be applied against the oil seal. In other words, you want to install the oil seal such that if any pressure is built up to act against it, it will push the lip harder against the shaft that it is sealing. The flat side of the oil seal faces outward (the side with the dimensions written on it) and the side with the U-shaped channel and the small circular spring around the inner part faces inside the pinion housing. Prior to reassembly, it is a good idea to make sure that the pinion shaft is clean and free of any nicks and / or rough spots that can ruin the new oil seal (BMC Part # 17H6560). Wrap a small piece of smooth paper around the rough pinion shaft, slide the oil seal onto the paper, and then gently slide them down the pinion shaft together. By installing the oil seal in this manner you can safeguard the fragile inner lip of the oil seal from damage.

The steering rack might feel a bit notchy while it is in the air, but should be smooth on the ground. If it is notchy with the wheels on the ground, then the steering damper is probably too tight.

If the gaiters (boots) of the tie rods are not damaged, the inner ball ends of the tie rods are tight with no binding, and the steering rack end bushings have no free play, then it is best not to rebuild the steering rack assembly. Instead, simply remove the steering rack and flush the housing out with WD-30. If the tie rods and steering rack bushings are worn, then you will have parts costs getting close to the cost of a new steering rack. Should you decide to rebuild the steering rack completely, I recommend that the steering column U-Joint (Universal Joint) be replaced at the same time.

However, if the gaiters (boots) are split, then debris has almost certainly entered the assembly and a rebuild may be necessary. A thorough inspection of the components is certainly warranted under such circumstances. Before removing your steering rack from the front crossmember, measure the distance between the two securing bolts of the tie-rod-ends. Then, count the number of turns it takes to remove each end from the tie rods and write it down. You can minimize disturbance of the wheel alignment if you start and finish with the same measurement. Always loosen the lock nuts of the tie rod ends before disconnecting the tie rod ends from the steering arms. Also, mark the pinion shaft and its U-Joint (Universal Joint) so that they can be mated back up the same way. Loosen the pinch bolt on the lower part of the U-Joint (Universal Joint) that attaches the pinion shaft to the steering column.

Do not attempt to move the steering column at this stage of the procedure as to do so will create problems with its alignment to the steering rack. Now remove the four nut and bolts that secure the steering rack assembly to their mounting bracket on the front crossmember. Take care not to lose any of the .015" (.381 mm) shims that you may find under the mounting lugs of the steering rack as they will be need to be reinstalled in their original position during reinstallation of the steering rack in order to proper align the steering rack with the steering column.

Secure the steering rack in a bench vise, and then grab each end of the steering rack (the sliding part) and give it a good wriggle. Any slop will indicate worn steering rack bushings (BMC Part # 17H 8664). Unless they are replaced, you will feel and hear a clonking noise coming from the front end. Note that there is a ball end on the inner section of the tie rod that pivots in a spring-loaded nylon ball socket (BMC Part # 17H 6571). The assembly is held together by a ball housing that is secured from behind with a round locknut. There is also a spring behind the nylon ball socket that fits onto the end the steering rack. The spring (BMC Part # CCA 13) will often break with age and lead to slack in the steering. Do not lose any of these critical ball assembly parts, as they are extremely difficult to obtain separately. If you do lose any of them, then you will most likely find yourself purchasing a complete new steering rack. When reassembling, take care that they are scrupulously clean. When reinstalling the nylon ball socket and its spring, do not apply any lubricant.

When removing the steering rack from its housing, bending back the tab of the tab washer and using a flat screwdriver and a hammer to turn the ball housing will certainly do the trick of removing the ball housings (BMC Part # 17H 6573), but you should consider buying a pin wrench (pin spanner) of the correct size since both, screwdriver and pipe wrench will severely damage the ball housings. This does not matter with the old ball housings, which you should not reuse, but you should avoid damaging the new ones. Furthermore, it is far easier to tighten these ball housings correctly with the right tools. This pin wrench (pin spanner) is made by the Park Tool Co. Go to a good full-line bicycle shop and ask to see a bottom steering bracket wrench (spanner) for a Sugino or Shimano bottom steering racket assembly. The right tool for the job makes things go better. Do not be surprised if you have to apply heat to the ball housing in order to loosen it. During reassembly, the hardest task is getting the ball housings back on without cross-threading the very fine threads. Be sure to apply antisieze compound to the threads in order to make any future disassembly less of an ordeal. Turn the ball housings backwards with gentle pressure, which lines up the two parts of the thread to each other, until you feel a click. This will be the two halves of the thread engaging. At this point, you need only tighten as normal. Once the ball housings are installed, tighten the jam nuts. These are ½-inch fine thread jam nuts found at any good hardware store.

In order to remove the pinion gear, simply remove the two machine bolts on the pinion end cover plate, and then clean all remnants of the old gasket from the mating surfaces of both the pinion end cover plate and the pinion housing. Next, remove the bearing retaining nut along with its joint washer. The pinion shaft and its bearing come out toward the front. You may need to tap them out with a soft hammer. Be sure to use some of your wife's nail polish in order to mark which pinion tooth is meshed with which steering rack tooth. Mismatched wear patterns can cause the steering rack to hang just slightly in one spot, causing the steering to not smoothly return to center after a turn. Once the pinion shaft and its bearing have been removed, carefully clean and inspect them both. Examine the pinion shaft oil seal before you remove it. If the wear is concentric all the way around, you are probably ok. If you see that the oil seal seems to have worn in one area only, you should check rack alignment. Finally, remove the pinion shaft oil seal from the rear of the pinion housing.

Remove the self-tapping screw that secures the old pinion shaft bushing (BMC Part # 17H 6579) and remove it by using an extension to tap it out from below. Install a new one. A 7/64" drill should be inserted into the mounting hole of the self-tapping screw, and then the bushing should be drilled to a depth of .24" (6.1mm). Using a sealing compound in order to prevent leakage of the lubricant inside of the steering rack, smear some jointing compound onto the head of the self-tapping screw and install it.

In order to avoid damaging the steering rack housing bushing, withdraw the rack from the pinion end of the steering rack housing. Loosen the self-tapping screw that retains the rack housing bushing, and then drive out the bushing. A new bushing should be installed so that its outer end is flush with the outer end of the rack housing. A 7/64" drill should be inserted into the mounting hole of the self-tapping screw, and then the bushing should be drilled to a depth of .24" (6.1mm). Using a sealing compound in order to prevent leakage of the lubricant inside of the steering rack, smear some jointing compound onto the head of the self-tapping screw and install it.

Wear of the steering rack occurs much more on the straight-ahead segment than towards either lock, and while removing a shim may compensate for that (although you are likely to have to fiddle about with different sizes before you hit the best balance between play and stiffness), you will may find that the steering rack becomes stiff when moved away from the straight-ahead position - not good. If the teeth of the steering rack are worn in the straight-ahead position, then it needs replacing. Those pinion-housing shims are only present to compensate for manufacturing tolerances, not for compensating for wear. Fractures, hollows, or roughness on any of the teeth of either the pinion or the rack automatically qualify either of the parts for the “Unserviceable” category.

You may find the Moss Motors or other aftermarket gaskets to be too thin, resulting in the oil leaking out. The Original Equipment gasket is about .025” (.635mm) thick, but aftermarket replacements are usually around .011" to .012" thick. Two together in a stack are not thick enough. Instead, fabricate a gasket from a sheet of gasket material over .025” (.635mm) thick. To determine what gasket thickness it is that you need, install the end cover plate without a gasket and use a feeler gauge to measure the gap between the end cover plate and the pinion shaft housing. Add .003" to .005" (.0762mm to .127mm) in order to determine the required gasket thickness if you do not have a new gasket available to measure. In addition, put some thread sealer on the on the threads of the 5/16”-18 UNC bottom cover plate bolts in order to prevent leakage.

Refill the steering rack with 0.4 US pint (½ UK pint, 0.2 liter) of EP90 hypoid gear oil. Do not overfill the steering rack as this thick and viscous oil flows slowly from one side of the steering rack to another. Overfilling can result in a ruptured gaiter (boot).

In order to get the alignment of the front wheels reasonably close to what it should be when you are finished, at this point you should measure the distance from the center of the underside of the steering rack rod end to equal positions on the ends of the tie rods. When fitting new tie rods to the steering rack, or refitting the old tie rods to a new steering rack, you can get reasonably close to your original setting by using this method. I suggest that new tie rods should be used unless the current ones are rather new. With the tie rods adjusted to equal lengths, turn the steering pinion from lock to lock while counting the turns, set pinion shaft at one-half of the distance between the limits. Turn the steering wheel to the desired straight-ahead position, and then assemble the steering column to the steering rack by means of the U-Joint (Universal Joint) of the steering column.

Upon reinstallment of the pinion shaft you might find that it seems to be overly long when you attempt to refit the U-Joint (Universal Joint) of the steering column. If this happens, check to see if the upper steering column has moved forward. There should be evidence of where the column was originally bolted on by the marks on the outer section of the steering column from the column support bracket inside of the cockpit. If this proves to not be the case, be advised that in the case of a collapsible steering column, the inner shaft is composed of two sections. The lower and upper shafts have flat ends with a hard plastic insert that makes the two mating ends slide together with a friction fit, roughly half-way up the steering column. Upon removal of the U-Joint (Universal Joint) with a hammer, it is easily possible for the shaft to work its way out of the column. It should go right back in with a few light taps with a rubber mallet, just be sure to align it with the flat on the upper shaft. This is easy to do by feel as the shaft turns quite easily and slips into the steering column. If you tap on the end of the shaft, it will slide in far enough to give you the proper clearance.

Note that U-joint (Universal joint) of the Chrome Bumper model consists of separate yokes, spider and bearings (needles in a cup), and that the spider and its bearings can be replaced while reusing the existing yokes. On the other hand, for Rubber Bumper models the U-joint (Universal joint) is smaller, which precludes any replacement of separate components, forcing its replacement as a complete assembly.

Proper alignment of the steering column and the pinion shaft is critical if the U-Joint (Universal Joint) of the steering column and the pinion assemblies are not to be damaged. The objective is to get the centerline of the steering column and the centerline of the pinion shaft crossing each other at the centre of the U-Joint (Universal Joint). This is controlled by shims at the mounting points of the steering rack as well as by fore-and-aft and lateral movement of the steering column on slotted holes. First, refit the U-Joint (Universal Joint) to the steering column and to the shaft of the steering rack. Next, loosen the top and bottom mounting bolts of the steering column support bracket inside of the cockpit that secures the steering column to the car in order to permit the steering column to slide lengthwise so that the steering rack rests evenly on the rear mounting holes of its front crossmember bracket. Lightly tighten the rear machine bolts of the steering rack. Shim the front steering rack mounting holes as necessary, and then fit the front machine bolts. At this point, you can torque all four of the steering rack mounting bolts. Tighten the steering column mounts. Check for any binding and readjust the steering column mounts if necessary.

The shank of the tie rod end bolts are actually tapered. For reasons of safety, the angle on the taper was chosen because it will cause the rod to "stick" in its hole. It will not and cannot come undone. Use a plain 3/8"-24 UNF or SAE nut on the tie rod end in order to snug it up so that the taper will be properly seated, then remove the nut and put on a 3/8"-24 UNF or SAE Nyloc nut.

If the steering rack is stiff after performing the above procedures, disconnect both tie rods and see if both swivel hubs swing easily on their kingpins. If they do, and you still have stiffness at the steering wheel, then you know that the cause is not in your recent work. Another test that you can perform is to disconnect the universal joint from the steering column and retest at the steering wheel. Use a good penetrant on the bolts and the shafts. Let it sit and soak in overnight if you can. Remove both machine bolts, and then try to tap the U-Joint (Universal Joint) down on the lower shaft. Once you have the top part of the U-Joint (Universal Joint) removed from the upper shaft, just tap the bottom so that you can slide it off. If the steering is stiff, then the problem is in the alignment of the steering column. If it is not, then your problem is within the steering rack. When reinstalling the U-Joint (Universal Joint), be sure to use antisieze compound on the threads of the machine bolts.

Upon reinstallation of the steering rack and the steering column, any imbalance in the number of turns of the steering wheel from straight ahead to full left lock as opposed to full right lock can only come from the adjustment of the tie rod ends, not from how the pinion is engaged with the rack. Because the cutout for the clamping bolt is just a notch, the column U-Joint (Universal Joint) only slides onto the rack shaft in one rotational position. On the steering column shaft the notch runs all the way around so that the two shafts can be assembled in any position within the number of splines on the column shaft. On early steering column shafts with the indicator reset peg screwed into the column, the two shafts must be correctly aligned for correct turn signal canceling. On later columns with the clamp-type reset cam (1968-1974) they can be assembled in any position and the cam slid around the shaft to suit. In all cases, the steering wheel then has to be correctly fitted to the column in order to be correctly aligned when the wheels are in the straight-ahead position.

Smear some anti-seize compound over the threads of the tie rods and install new tie rod ends, counting the number of turns as you go. Screw the jamb nuts back into position on the tie rods. Make sure that the small opening of the gaiters (boots) are positioned over the slight relief in the tie rods, and then secure their rubber necks around the tie rods with the cinch clamps, but do not tighten them fully just yet. In order to obtain a good seal of the gaiters (boots), smear some lithium grease onto the areas where they will be clamped. It is good practice to position the cinch clamps in such a position that the heads of the screws are going to be accessible for removal at a later date with the steering rack on the car. Tighten the cinch clamps in order to secure the boots to the tie rods.

Suspension Bushings

Obviously, there is no single "Magic Cure-all" for any car's handling. Only in fantasy or naive ignorance are things that simple. However, a simple modification involving nothing more than a change of parts can produce worthwhile results. The rubber front suspension bushings from an RV8 model produce significantly less longitudinal flex, endowing the steering with greater precision. The RV8 bushing is a one-piece assembly wherein the rubber bushing is bonded onto the steel sleeve in order to give more positive handling. With these it is important not to tighten the castellated nuts of the pivot pin until the weight of the car is on its suspension. This is due to the fact that the outer part of the rubber bushing is a tight fit into the A-arm, the steel sleeve acts as a spacer and is clamped tight by the nut, and so the action of the suspension tends to twist the rubber rather than slide it over the spacer. If the castellated nuts are fully tightened with the suspension in the extended position, then when the car is on its wheels there is already a considerable amount of twist imparted to the rubber bushing, and when the suspension is compressed over a bump, it becomes even more twisted. This can tear the rubber to the detriment of handling. Be aware that the steel sleeve is quite a snug fit over the pivot pin and can rust to it, so use antisieze compound when installing it. Be aware that these bushings are not a press-fit onto the inner wishbone fulcrum pin. Should the fit seem to be tight, clean the wishbone fulcrum pin to bare metal. It is crucial that they must be assembled correctly. The inner end of the stainless steel bushing sleeve should be mounted with its chamfered end matching the radiused inner end by the flange of the wishbone fulcrum pin. The large flat machine washers must fit over the Outside Diameter (O.D.) of the pin so that they will clamp the stainless steel bushing sleeve tight onto the lug of the wishbone fulcrum pin. The wishbone fulcrum pin has three diameters: the largest diameter and smooth shank that the bushing sits on, the somewhat smaller diameter shoulder that the retaining washer goes on (only about 3/32" long), and the threaded part. At the base of the threaded end of the wishbone pivot pin there is a small shoulder. The big retaining washer must fit onto this shoulder when the assembly is tightened. The aftermarket retaining washers supplied in today's front end rebuild kits have a hole that is too small to fit onto the intermediate shoulder, but instead rest against it. This prevents the correct compression from being applied to the bushings, which is also required for correct axial location. The correct compression is designed into the length of the large diameter of the shaft. Use either the Original Equipment retaining washers (BMC Part # AAA 1330), or machine the new retaining washers to fit. If the retaining washer is not properly seated upon the shoulder, the preload on the bushings will be inadequate, and rapid failure will be the result. If the hole is too large and goes over and past the shoulder, then the preload will be too great and both rapid failure of the bushings and rapid wear of the wishbone fulcrum pin will result. It is beneficial to first assemble both the retaining washer and nut onto the shaft, and then measure the amount of threaded part projecting out of the nut and the relative position of the split pin hole, and be sure that measurement is duplicated on final assembly. If the nut becomes tight when about 3/16" from that measurement, then you need to tap radially on the washer until it becomes located upon the shoulder, and then do the final tightening.

Firmer bushings, such as those made from nylon, will reduce compressibility in the suspension component mounting points and make small steering inputs result in correspondingly small reactions in the steering. In other words, the steering will become more precise, but the greater reactivity to steering input will also demand that you pay closer attention to what you are doing. Unfortunately, hard bushings are also not only unnecessary for either the mountings or the attachment points of stabilizer bars, Panhard rod ends, and Antitramp bars as they offer no benefit in handling, but are actually undesirable as they will fail to damp out vibration and road shock. Because they are harder, you will feel more vibration emanating from the suspension and steering wheel, hear more road noise emanating throughout the body of the car, and your hands will feel smaller pavement imperfections through the steering wheel. Hit a big pothole and you will know it! Even worse, the greater transmission of these forces means that associated load-bearing components (Steering rack and column components, tie rods, ball joints, kingpins, swivel axle bushings, dampers, etc.) will wear more quickly. The purpose of the original soft bushings at the top trunnion, the bottom of the swivel pin, and at both attachment points of the lower A-arms is to absorb energy. By absorbing energy, they prevent it from being transferred and transmitted elsewhere. In reality, there are better options for increasing steering response while avoiding most of these drawbacks. Sudden, outright failure of these components is highly unlikely unless they are of defective manufacture, but longevity is always desirable.

This is not to say that you should resign yourself to the use of rubber bushings. Whereas rubber bushings wear rapidly and rot, polyurethane bushings take a long time to wear and never rot. Sadly, almost all of the aftermarket suppliers in the USA offer only the harder varieties, being either of the "Racing and Competition" or of the "Fast Road and Rally" type. In terms of their quality, some of these bushings are real "bargain basement" items. In my opinion, the Australian company SuperFlex makes the best, and the price is quite reasonable for the quality of their product. They do not produce them in molds (a sure sign of an El Cheapo bushing), instead they start life as a solid rod that is actually precision machined to size and shape on computerized machines. As a result, they will slip-fit into place. This is not often the case with molded bushings. Sometimes you have to pound them into place with a mallet, which will result in their bores being distorted or compressed, which in turn will cause them to squeak. The SuperFlex bushings are self-lubricating once installed. They even include stainless steel sleeves so that rust cannot abrade them. If you want to purchase a softer set (that is, soft like rubber bushings) for use in a daily driver, go to http://www.racecar.co.uk/SuperFlex/ and specify 80 Shore-A bushing material for the A-arm bushings (BMC Part # AHH 7933*, BHH 1123 **, SuperFlex Part# SPF0012) and 90 Shore-A bushing material when you order the Trunnion (BMC Par t# 8G 621, SuperFlex Part# SPF0282), Leaf Spring (Front Spring Eye- BMC Part # AHH 644, SuperFlex Part # SPF0181; Rear Spring Eye (BMC Part # 2A 5176, SuperFlex Part # SPF0014), and Stabilizer Bar bushings. SuperFlex makes a product line that even includes stabilizer bar bushings for 1" (SuperFlex Part # SPF0063/25), 7/8" (SuperFlex Part # SPF0063/22), ¾" (Original Equipment Par # AHH 7927, SuperFlex Par # SPF0063/19), 11/16" (BMC Part # AHH 7921, SuperFlex Part # SPF0063/17.5), 5/8” (BMC Part # IB 4526, SuperFlex Part # SPF0063/16), and 9/16” (BMC Part # AHH 6541, SuperFlex Part # SPF0063/14) stabilizer bars. I would recommend 70 Shore-A material for both the lipped upper and flat lower front crossmember mounting pads (BMC Part # AHH 6204, AHH 6206, respectively; SuperFlex Part # SPF0015A, SPF0015, respectively) and the stabilizer bars.

*(MGB bushing) **(V8 bushing)

Prior to fitting any suspension bushing, remove all dust grit, previous paint, old grease, or bushing residue from all of the surfaces that can come into contact with the bushing. Be sure that any original outer shell is not inadvertently left in place. This is a common mistake whenever an old rubber bushing has unbonded from its shell. Do not fit new bushings to worn, rusty, or distorted fittings. All such worn components must be replaced. When preparing to install a bushing, lightly coat both it and the contact surfaces with CRC Sta-Lub Extreme Pressure Lubricant prior to fitting. In a very cold climate, immersing high-interference fit bushings into boiling water can facilitate fitting. Insert the stainless steel tubes (where applicable) after the bushings are installed into their housing. Before the final tightening, all of the suspension arms must be normally weighted at normal ride height and the car bounced up and down. Failure to observe this procedure will result in the bushings already being twisted when sitting at rest and can thus shorten their life. When replacing original components, ensure that all nuts and machine bolts are torqued to original manufacturer’s specifications. Note that polyurethane bushings must not come into contact with alcohol-based solvents such as MEK, methanol, or methylated spirit.

Body Roll

With the brakes properly sorted, the issue of improved handling can be safely addressed. The first thing that occurs when cornering is the vehicle rolls to its side. The degree of roll is a function of several variables: center of gravity, roll center height, track width, suspension roll stiffness, vehicle height, and most important: how hard the car is cornering. Being that the vintage bias-ply tires of the 1960’s possessed limited cornering capability, the amount of body roll was modest, and the handling balance of the car was very well maintained. With today’s higher-performance radial tires, body roll increases are proportional to the increased traction. However, only up to a point. The suspension geometry of the MGB lacks insufficient roll gain to deal with higher body roll angles and keep the tire vertically aligned to the road surface, thereby limiting grip and cornering speed.

Putting some negative camber in the front suspension would definitely help the outside tire grip the road. Yet, what about the grip of the inside tire? In addition, such a modification would compromise the straight line stopping ability, and while the car would respond more quickly to initial steering inputs, the total grip will be only marginally improved.

Therefore, we need to re-examine the issue of body roll. When cornering, inertia forces the body of the vehicle to roll to the outside of the turn. The resultant degree of body roll is a function of several variables; the height of the center of gravity, the height of the roll center, the roll stiffness of the suspension, the width of the track, and, most important, how hard the car is turning. You need to understand that the suspension of the MGB was designed for tires with about 0.65 G's of cornering capability. Being that the Original Equipment 1960's-vintage tires limited the cornering capability, the amount of body roll was modest and the handling balance was maintained. But by today's standards, the car is relatively slow. Today's high performance tires should let you approach or exceed 1.0 G. With the higher grip of modern tires, body roll increases proportionally to the increased grip. However, only up to a point. The suspension geometry of the MGB has insufficient Camber Gain to allow it to utilize higher body roll angles and keep the tire vertically aligned to the road surface, thereby limiting grip and commensurate cornering speed.

Obviously, reducing the amount of roll during vigorous cornering should be a high priority. The most involved procedure for reducing body roll is to lower the car. Lowering an MGB by 1 to 2 inches (or more on later Rubber Bumper cars) provides benefits in two ways: first, it reduces both body roll and lateral weight transfer. Why would reduced lateral weight transfer improve grip? Simply put, because the total normal load on the tires will remain be less effected by lateral weight transfer. As the tire normal force (vertical downward load, i.e., weight) increases, the tire grip also increases. However, this increasing grip is not equivalent to the increased normal load. Additionally, the load vs. grip curve is neither flat nor straight, so as the outside tires are increasingly loaded in a corner, we do not gain as much grip on the outside tires as we lose on the unloaded inside tries, hence the total amount of grip actually becomes less. Consequently, reducing lateral weight transfer actually improves the grip of the tires. Lowering the Center of Gravity reduces lateral weight transfer, and thus improves grip. On an MGB built for racing, ground clearance is the only limit to lowering.

Lowering The Body

Before performing any suspension modifications on a Rubber Bumper car it is best that it should be lowered the to the original height of the Chrome Bumper model. The use of shortened coil springs in the front suspension should be avoided as they will leave the front suspension travel with less upward travel, result in an extreme "Toe out" front wheel misalignment that will result in accelerated tire tread wear, as well as create a bad tendency toward bump steer as a result of the change of angle of the tie rods. Braking distances will also increase as a result of the deformation of the contact patch of the tires. In addition, these shorter springs will need to have a much stiffer rate to partially compensate for the resultant decrease in upward suspension travel and to prevent the suspension from crashing into its bump stops, thus creating very harsh ride qualities. The most comprehensive approach to lowering the front end is to install a Chrome Bumper front crossmember, complete with steering rack and steering column, and then use an engine front plate from an engine intended for installation in a Chrome Bumper model car.

Be advised that the steering rack mounting brackets of the two different front crossmembers are set at different angles, and that the steering racks have different length pinion shafts and use different length steering columns, and thus cannot be interchanged except as a complete steering system, provided that the appropriate corresponding steering rack mounts are welded onto the front crossmember. For the rear suspension, you need only cut off the front spring hangers and weld in new Chrome Bumper leaf spring mount assemblies. These are available from the British Motor Heritage Trust (BMIHT Part #'s BMH9002 for the right side, BMH9003 for the left side). These can be seen on the British Motor Heritage Trust webpage at: http://www.bmh-ltd.com/p61r.asp . You will also need to change the rear bump stops so that you will still have adequate suspension travel, and the limiting straps. Fortunately, the pedestals (BMC Part # AHH 7355) are the same on all four-cylinder models. If in the future you should decide to install a Rover V8 engine, a change of the angle of the steering rack mounts, plus the later steering rack and its steering column from the Rubber Bumper model, will be necessary in order to clear the rear of the exhaust manifold on the engine.

MGB's are easily lowered by modifying the stub axles and replacing or reworking the springs. The rear leaf springs can be lowered by means of installing special lower-arch rear leaf springs, or by taking the less expensive approach of removing a leaf and restacking the short leaves and / or adding lowering blocks. It is best to rework the rear leaf springs, as this will decrease the rate of the rear spring, which in an MGB roadster improves rear grip. Unfortunately, this can result in the rear axle crashing into its rubber bump stops on bad roads, so a certain amount of discretion on the part of the driver becomes mandatory.

On an MGB that is being built for street use, several other factors come into play. The first is the roll center of the front suspension. As the front springs are lowered, the roll center moves lower at least as fast as the Center of Gravity, thus reducing the benefits of the reduced vehicle roll rate. In addition, this further degrades the dynamic camber gain of the suspension, which will result in a condition referred to as Bump Steer.

If you look at the lower A-arm on a lowered MGB, you will notice that its outer end will be higher in relation to its inner end. With this condition, lateral wheel movement takes place during normal suspension travel - causing the car to lack directional stability and thus making it feel 'darty' on uneven road surfaces. The cure for this is to raise the lower A-arm's inner pivot point of the wishbone pivot by up to 2 inches, both maintaining the arm geometry parallel to the road and reducing body roll. However, the rotational axis of the swivel hub will be canted inward due the upward change of the angle of the suspension arms of the Armstrong lever arm damper, resulting in negative camber. This problem can be corrected by fabricating spacers to raise the Armstrong lever arm damper until the original suspension geometry is restored. This also helps to improve camber gain with resulting improvements in grip.

Adjusting Bump Steer is highly recommended to transform a car that has become a twitchy handful as a result of lowering into the proverbial "handles-like-its-on-rails" car. So exactly just what is Bump Steer? And what causes it? Bump Steer (often called "Roll Steer" by suspension designers) is the term used to describe how the steer wheels pivot in relation to the car's direction of travel as a steer wheel moves vertically. The steering arm, which is attached to the front suspension and to the steering rack by the tie rod, travels in an arc. When the connecting end of the tie rod travels in an arc that is different from that which is followed by the suspension, it changes the “Toe” of the suspension whenever the connected steer wheel rises and lowers. This is because the tie rod is a single point mount attached to the steering rack as well as to the steering arm for its particular wheel assembly. As a result, when the wheel assembly moves up and down, taking the attached end of the tie rod along with it, the effective length of the rod changes according to Pythagoras’ Theorem of right triangles (See, your old High School geometry teacher was right! There really is a real-world use for Plane Geometry!). This change in the effective length of the tie rod pulls on the steering arm as the wheel assembly moves up (and down), slightly changing the steering angle of the wheel. This is what pulls on the steering wheel and makes it seem as though someone other than the driver is steering whenever a bump is encountered by a single steer wheel, hence the term “Bump Steer”.

It is desirable to have the wheels Toe-in very slightly as the wheel is rises and Toe-out slightly as the wheel lowers. This is called “Roll Understeer” and makes for precise and stable handling. Roll Oversteer, however, is an opposite effect that causes difficult-to-drive, “squirrelly” handling. Do not confuse this description of oversteer / understeer with the same result that stems from limited tire grip at one or the other end of the car. Obviously, when the mounting points of the suspension arms are relocated upwards, the tie rod of the steering rack assumes a steeper angle to the steering arm. The original geometry can be restored by heating the steering arms with a torch and slowly bending them downwards until the original angle of the tie rod is established. Under no circumstances should the steering arm be hammered into position, as this may create fracturing.

For those who find this conversion to be too challenging or expensive, there is a much simpler approach to the issue. Instead, install a set of swivel hubs with vertically offset stub axles. Lowering the rear suspension with blocks will increase the fulcrum effect of axle torque on the rear springs as a result of moving the axis of the pinion gear further from the rear leaf spring. This in turn will lead to the dreaded “axle tramp” at the moment of a vigorous application of power when pulling away from a stoplight, a result that is unacceptable in an enhanced-performance car. The use of decreased-arch rear leaf springs is also inadvisable as their spring rate will have to be stiffer in order to prevent the suspension from crashing into its bump stops and will not match that of the front springs, thus damaging both the handling and the ride quality of the car. You will swiftly learn to avoid railroad crossings and speed bumps. Removal of the front muffler (silencer) from the exhaust system and its replacement by a section of straight exhaust tubing in order to avoid scraping will be mandatory. Even with the front muffler (silencer) removed, the car will still tend to scrape on speed bumps and some railroad crossings. Instead, change both the rear springs and their rear spring mounts to those of the Chrome Bumper model, and be sure to use the earlier model 8½" long axle straps (BMC Part # AHH 6355) and 8½" long damper linkages (BMC Part # 97H 2031) of the Chrome Bumper models. Although the front suspension of Chrome Bumper models can also be lowered using similar techniques and the rear suspension lowered by using decreased-arch rear leaf springs, such a modification is normally confined to cars that use very stiff springs and are normally driven only on a race track. As engineers are fond of saying, it is all a matter of finding a balance of priorities. In other words, you do not get something for nothing.

Camber Angle

Camber Angle is the angle between the vertical axis of the wheel and the vertical axis of the vehicle when viewed from either the front or the rear. It is used in the design of steering and suspension. If the top of the wheel is further outward than the bottom (that is, away from the axle), it is called positive (+) Camber; if the bottom of the wheel is further out than the top, it is called negative (-) Camber. Altering the Camber Angle alters the handling qualities of a suspension design; in particular, negative (-) Camber improves grip when cornering. This is because it places the tire at a more optimal angle to the road, transmitting the forces through the vertical plane of the tire, rather than through a shear force across it. Another reason for negative (-) Camber is that a rubber tire tends to roll on itself while cornering. If the tire had zero Camber, the inside edge of the contact patch would begin to lift off of the ground, thereby reducing the area of the contact patch. By applying negative (-) Camber to the outside tire during the turn, this effect is reduced, thereby maximizing the contact patch area. Conversely, the inside tire benefits most from positive (+) Camber. Unless you are lowering your car or modifying it for competition use, there is little point in attempting to alter the factory-specified Camber Angle.

For many, the steering ratio of the original equipment steering rack is simply too slow for hard driving on twisty roads. For these demanding souls, Cambridge Motorsport offers a QuickRack with a faster ratio (two turns lock-to-lock).

Most coil-over front suspension conversions use the Original Equipment knuckle-and-trunnion-plus-swivel-hub components from the Original Equipment front suspension system, so the steering geometry remains fundamentally unchanged. Why anyone would expect an improvement in handling from the same front suspension geometry is beyond me. Those that use a deactivated Armstrong lever arm damper and their upper arms cannot offer notably greater suspension travel, either. Of course, adjustable tubular dampers would have a potential advantage for racing on a racetrack, as their sealed, pressurized nitrogen atmosphere would offer reduced foaming. A better, though more expensive, option for those seeking dramatically improved handling and ride quality, would be to convert to an RV8 front crossmember and front suspension system, just as the factory found necessary. These are available from the British Motor Industry Heritage Trust.

Castor Reduction

Heavy steering has always been a noticeable feature of the MGB. Recently, a kit that makes use of needle-roller bearings has appeared on the market as a solution. Sadly, this disappointing kit is poorly thought out from an engineering standpoint. There is a consideration that must be made as regards the use of needle-roller bearings in such an application: since the needle-roller bearing is cylindrical in shape and radially oriented in this application, the inner end has to turn at a lower rate than the outer. This produces skidding of the bearings, and in fact, if loads are high enough and lubrication is lacking, you cannot rotate the assembly at all without shearing the bearings. This being the case, the reduction in steering effort may be much less than you would expect by thinking of them as “rolling element” bearings. In big thrust bearings, there are frequently multiple rows of short needles for this reason. The only proper solution is the use of tapered roller bearings, which works if the taper vs. length is such as to equalize the rotation rate at the opposite ends of the rollers. This adds space and also gives high end loading on the roller bearings, requiring strong cages, and increased friction.

The heavy steering characteristic is caused by the 6.5° of positive (+) Castor Angle that was required in order to produce self-centering of the steering action when used with the cross-ply tires that had a Slip Angle of from 10° to 11° that were available in 1962 when the design was first introduced. Since that era, radial tires have been developed that have a Slip Angle of 2° to 3° that produces far more directional stability and improved self-centering. This advance, coupled with improved rubber compounds that possess greatly improved traction, has the unfortunate effect of increasing the steering load, particularly under tight cornering or when cornering at speed. As modern tires have far more directional stability, less self-centering force is necessary, and as such, such a large Castor Angle is no longer required. Consequently, these tire improvements provide scope for reducing the Castor Angle and thereby obtaining the welcome benefit of lighter steering with the MGB.

There is a mistaken belief that the Castor Angle on the MG RV8 is the same as that of the MGB and MGB V8, and thus the castor reduction kit can also be fitted to the MG RV8. This is incorrect as the Castor Angle on the MG RV8 is 3° 48 minutes +/- 54 minutes (see the RV8 Repair Manual AKM7153ENG) so using a castor reduction kit that would remove 3° of Castor Angle would leave only 0° 48 minutes +/- 54 minutes which is an insufficient amount. In addition, the front crossmembers supplied by the British Motor Heritage plant at Witney, whether supplied individually or incorporated in new British Motor Heritage Trust body shells, already have a reduced Castor Angle. British Motor Heritage Trust has confirmed that the Castor Angle reduction was incorporated into the front crossmembers that they supply. Therefore, any MGB or MGB GT V8 fitted with a such a front crossmember, or even in some rare cases fitted with an MG RV8 crossmember, should not be fitted with a castor reduction kit.

Looking at the front suspension from the side, the Castor Angle is the angle, measured in degrees, that is formed between the axis of the kingpin and a perpendicular line to the ground. Since the angle is formed longitudinally relative to the vehicle, its more exact definition is “Longitudinal Castor Angle”. In practical terms it is known simply as the “Castor Angle”. The Castor Angle given to the kingpin creates three important characteristics for the ride and handling of the vehicle: first, stability in terms of maintaining the straight line of travel of the vehicle and, second, the extent to which the steering self centers after turning, and, third, the tilt of the wheel that occurs during turning. The stability characteristic is created on the basis of the distance between the point at which the kingpin axis extension falls (in relation to the direction of travel) and the point of contact between the tire and the ground. In the case of positive (+) Castor Angle (where the kingpin extension line falls ahead of the point of contact between the tires and the ground), the wheel is pulled, as it is the line of application of the force applied to the axis that passes in front of wheel's mid-point without taking the direction of travel into account, and each attempt made by the wheel to deviate from straight line travel will be counteracted by the Straightening Couple generated by the force and by the rolling resistance of the wheel. With negative (-) Castor Angle the wheel is pushed as it is the line of application of the force applied to the axis passes behind the mid point of the wheel. Consequently, the best stability condition for straight-line travel is obtained with a positive (+) Castor Angle. In this case, the problem of "wheel wobble" and the consequent effects on steering are avoided. These different behaviors of the wheels can be verified by driving the same vehicle in forward gear and then in reverse gear.

An improved Castor Angle reduction kit (Brown &Gammons Part # AHH6195 CASTOR) has been produced for both Chrome Bumper and Rubber Bumper MGB and MGBGTV8 models by Brown & Gammons, the MG specialists at Baldock in the UK. It is designed to accomplish two things: first, to reduce the Castor Angle by 3° from the original 7° to 4° and, second, to maintain the integrity of the mounting of the front crossmember to the leg of the chassis. It is worthwhile to understand how this new kit achieves this goal with well-thought-out, thorough engineering details that ensure that the mounting bolts continue to be positively located in the tapered seats of the chassis legs, and that the rubber mounting pads are not crushed, in order to achieve an accurate Castor Angle setting. This is an improvement over another kit currently available, which when fitted results in the taper of the machine bolt being held away from its seating and the rubber pad being crushed when the assembly is torqued down.

In the Original Equipment design, the front crossmember is mounted to the chassis leg in an orthodox manner. The MGB front crossmember is fabricated out of steel sheet metal that is pressed and welded, and is mounted on the underside of the chassis legs (which are box sections extending forwards from the monocoque body) with four high tensile steel mounting bolts which are positively located into the chassis leg on tapered seats. On either side that the topmost part is a platform with four holes on which the lever arm dampers are mounted. Just inboard of those platforms are the two large holes through which the front crossmember is bolted on either side to the chassis legs by the mounting bolts. The mounting bolts have screw threads at both their tops and their bottoms and a thicker plain shank in the middle, with a taper at the top. The intention of the design is that the taper locates to a corresponding taper seating in the bottom of the chassis leg. Hence, the mounting bolt is positively located in the center of the hole in the chassis leg when it is bolted on with a torque of 56 Ft-lbs. This leaves the bottom part of the mounting bolt protruding below the chassis leg with a plain section, and beneath that a narrower threaded section forming a shoulder at the end of its plain shank. A rubber pad that acts as a packing piece between the chassis leg and the mount on top of the front crossmember is fitted over the plain shank of the mounting bolt. This is held up by a rectangular washer with a smaller diameter hole so that the washer sits on the shoulder of the plain section of the mounting bolt but is held in place by the bottom locking nut. The pressure on the rubber pad between the chassis leg and the front crossmember is therefore limited so that crushing is avoided.

How does the kit reduce the Castor Angle? The method used is to simply rotate the front crossmember towards the front of the vehicle by inserting a precisely-machined stainless steel packing piece between the front crossmember mounting points and the underside of the chassis leg. Since the steel packing has used some of the length of the plain shank of the mounting bolt, a steel collar that has to be installed is supplied with the kit. Its effect is to extend the plain shank of the mounting bolt back to its original length. Without this collar, the rubber mounting pads would be overly compressed, thereby ruining both the rubber mounting pads and the ride quality. Of course, the crushing would give rise to variances in the Castor Angle, even between each side of the vehicle. New, slightly shallower high tensile steel locking nuts are provided in the kit in order to fit the reduction in useable thread length of the mounting bolts.

Because the angle of the front crossmember brackets upon which the steering rack is mounted will have changed slightly (3^o) in relation to the chassis legs, the body of the pinion shaft of the steering rack will no longer properly align with the universal joint of the steering column. The steering rack brackets will therefore have to be packed at the front in order to realign the rack with the universal joint. Six packing shims are included in the kit for this purpose. Brown & Gammons estimates that installation of the Castor Angle reduction kit requires approximately three hours of work. The kit includes comprehensive fitting instructions and detailed diagrams.

While you are working on installing the Castor Angle reduction kit in this area of the car, it is well worth checking the condition of the steering rack brackets for any hairline cracks or more serious fractures. Should you encounter such problems, a Steering Rack Mount Strengthening Gusset is also available from Brown & Gammons (Brown & Gammons Part # AHH6195 BRACKET).

Aligning the Front End

Now that you have the opportunity, all of those old inexpensive 1/4"-28 UNF grease zerks found on the suspension should be replaced in order to guarantee that you can get everything properly lubricated. Grease everything at their grease zerks, then lower the car to the floor and bounce the front end of the car in order to settle the suspension. With the steering wheel being held firmly in its centered position, step back from the front of the car, kneel down and sight down one side along the plane of the outside edges of one front tire toward the rear wheel. You should just be able to see the outside of the rear tire. If not, use slip joint pliers to turn the steering rod on that side into or out of the tie rod end in order to adjust the wheel direction as needed. Repeat this method on the other side.

Now, jack the car back up again until the tires clear the floor. Hold a piece of chalk firmly against something stable with the tip just touching the measured center of the tire tread, and then have an assistant rotate the wheel so that the chalk can make a mark on the rubber all the way around the tire. Repeat this process on the other front tire.

Lower the car onto the floor and bounce the front end of the car in order to settle the suspension again. With Original Equipment-specification suspension geometry, the Castor Angle should be within 6½$^{\circ}$ to 7°, the front Camber Angle should be -¼$^{\circ}$ to -½$^{\circ}$, while the optimal Toe-in can be anywhere from 1/8" and 3/32" (1.5mm and 2.3mm). Adjusting the Toe setting of the front suspension is a straightforward affair.

The car must be on a flat, level surface in order to take accurate measurements. If there is any doubt that the surface is level, use a carpenter's level to test it. Any variation in the car's stance after you are finished aligning the front end will change the angles that you set, so make sure the car is loaded as it will be when you are driving. Make sure that there are no heavy items in the trunk unless you plan to leave them there after the wheels are aligned.

With an assistant, use a flexible tape measure to gauge the distance between the lines on the center of the tire treads both on the front center of the tires and on the rear center of the tires. You might want to make the rear measurement first as you will have to thread the tape between the body and the down pipe of the exhaust system, stretching the tape as high up as you can without fouling anything. When you measure at the front center of the tires, try to take the measurement at about the same height off of the floor as you did on the rear measurement. Compare the readings. Remember to loosen the cinch clamp at the small end of the gaiters (boots) before you try to rotate the tie rod. Be aware that making all of the adjustment on one steering arm will move the steering wheel from being visibly central whenever the car is driving straight ahead. Making equal amounts of Toe adjustment on each tie rod will keep the steering wheel pointing to where it was before the adjustment is made. Your goal should be to end up with the front measurement being between 1/8" and 3/32" (1.5mm and 2.3mm) less than that of the rear measurement. If it is not, re-sight each wheel in turn and adjust the tie rod ends as carefully as you can to get the sight line to the rear wheel just right. Work with this measurement and sightings and adjustment process until the front wheels are "Toed in" between 1/8" and 3/32" (1.5mm and 2.3mm), and then tighten the jamb nuts on the tie rod end, and then tighten the small cinch clamps around the ends of the gaiters (boots) of the steering rack.

Now, take the car out for a test drive. If the car is lacking in directional stability, the odds are that the wheels are not Toed-in enough. If it feels a little imprecise, then the Toe setting is probably excessive. Work with the Toe adjustment as needed to get the correct sight lines, the correct Toe measurement, and the best "feel" when driving. The Castor Angle is more or less equal on both sides. This being the case, it ensures that when driving forwards on a flat straight surface the wheels will always take up an equal angle from straight ahead, i.e., half the total Toe-in (or out) on each wheel.

Dampers

Unlike more modern designs, the MGB does not make use of inexpensive disposable tubular shock absorbers in order to damp the movement of the suspension. Instead, the suspension is damped by Armstrong lever arm dampers. Unless the car is routinely operated on very bad roads, the Armstrong lever arm dampers are quite adequate for their purpose and need not be replaced. A significant design advantage of the Armstrong lever arm damper is that while the distance traveled by the piston rod inside of a tubular damper is linear, in the case of the Armstrong lever arm damper the distance traveled is exponential. That is, the further that the lever arm is moved, the less the distance traveled by the pistons per degree of lift. As a result, this causes an exponential change in internal pressure that is dictated by its connecting rod to stroke ratio. The dampers have a dual-damping feature that results in softer damping with small/slow movements and only stiffens up on large/rapid movements. Because the flow rate of the damping valves is limited by the bore size of their passages, an increase of internal pressure increases the damping rate whenever the lift / descent rate of the suspension increases and thus requires the lever arm to move rapidly. In addition, it has a two-stage damping effect whereby if the arms move faster than a normal rate, i.e., over a big bump or at high speed, a secondary damping valve closes in order to increase the damping effect. This gives a compliant ride under normal operating conditions, but more resistance to bottoming or rebounding under extreme conditions. Should you decide that an increase in their damping effect is desirable, this increase can be accomplished by simply replacing their damping valve units with heavy-duty ones which have had their rates uprated by 25%, as the factory race team did (Special Tuning Part # AHH 7217, Front; Special Tuning Part # AHH 7218, Rear). These uprated damping valves are available from Victoria British Ltd. at http://www.victoriabritish.com . If your present front lever arm dampers are worn out, newly-manufactured uprated units are available from Brit Tek (Part # GSA367UR). However, excellent uprated rebuilt units are available from World Wide Auto Parts of Madison. They have a website at www.nosimport.com/shoxcatalog.htm .

What fails in a lever arm damper? Almost all of the (non-traumatic) failures result from lack of oil in the damper. The manuals always recommend checking and topping-up your dampers every 3,000 miles or so. The shaft that protrudes from the body of the damper rotates in the body without a separate bearing. To ensure sufficient lubrication there is often a channel or groove in the shaft bore. At the outside there is a rubber packing retained by a thin metal washer. A packing needs some lubrication in order to function, and the weeping of oil acts as a deterrent to dirt getting in. Dirt getting in will score the shaft at the oil seal area, thus hastening the demise of the packing and wearing the bearing surface in the body.

The rebuilt lever arm dampers from World Wide Auto Parts of Madison, Wisconsin are the best available. They machine the body and install Delrin nylon bearings. Their competitors use bronze. Bronze requires oil for lubrication, but Delrin does not. World Wide Auto Parts then installs a radial double lip oil seal that incorporates a dust excluder. Their competitors use some variety of plain rubber washer retained by steel washers against the lever arm. World Wide Auto Parts installs a 3 micron finish stainless steel sleeve onto the shafts in order to repair the grooves and pitting usually present on the shaft. These sleeves are of their own design and guarantee a surface finish and concentricity required for the best performance from the oil seals. Apparently, their competitors either grind or sand, when necessary, the shaft to smooth it. They clean the units using a media tumbler, glass bead blasting, and tumble again, then ultrasonic cleaning. World Wide Auto Parts then finishes by painting with two coats of primer, followed by two coats of gloss black high-heat enamel. World Wide Auto Parts uses all-new, fine threaded hardware. Their competitors' units do not appear to. World Wide Auto Parts even goes to the extra trouble of using an anti-seize compound on those fasteners that the installer may need to loosen for installation.

It should be noted that while a few modern tubular dampers offer dual rate adjustability, it should be noted that any increase or decrease in compression and rebound damping is adjusted only in equal amounts for both. However, as regards comfort and performance, you may find modification of the present damping valve mechanisms of your Armstrong lever arm damper units to be a more preferable choice. When these units were new, there were alternate compression springs available. Certain dampers had a steel (grey) colored compression spring as standard, but other dampers had a bright copper colored spring as standard. The "competition" spring had a dull copper / bronze color. However, today these springs are a rare find, so adjusting the preload of the compression spring is the usual practice. It is usually easiest to use settings near the maximum and ease off from there if they are too harsh. Most drivers find that increasing the damping rate of the front dampers by 25% gives an impressive improvement in handling. In order to increase damping on the compression stroke, the larger spring must be compressed by inserting shims between the body of the damper and the spring. This will increase the preload on the larger spring and accordingly result in a stiffer damping rate. Some dampers already have shims in them, but more can be added. Remember to write down the original specifications of the damping valve mechanism (size and number of shims, and how many turns on the nut) prior to altering it so that you can reset it to an established baseline should your adjustments produce unsatisfactory results. You can start by tightening the nut on the rebound spring all the way down, and then install a 0.070" spacer under the compression spring at the bottom of the bore. An alternate method is to tighten the nut on the rebound spring two to four full turns, and then use a spacer of 0.040" to 0.080" in order to preload the compression spring. Select the amount of adjustment depending on how much damping you want. The damping valve mechanism also has a small spring on it that controls the damping rate on rebound. It has its preload increased by adjusting its compression with a threaded nut. The greater the preloading of the spring, the higher the rebound damping rate will be. If this is all too time-consuming for your taste, Cambridge Motorsport has available lever arm dampers whose damping rate can be readily altered by means of adjusting a knob. They have a website at http://www.cambridgemotorsport.com/index1.htm .

One of the most common causes of problems with lever arm dampers is overfilling them. The top dome on the units used on the MGB is present in order to act as an expansion chamber. If the damper is overfilled, i.e., all the way to the top of the threads, then no space will remain for expansion when the hydraulic fluid gets hot. This will result in excessive pressure and cause leakage. They should be filled to the bottom of the 5/16”-26 BSC threads as per the Workshop Manual. There is an air-space inside which absorbs pressure changes in use. As a side note, it should be recognized that 5/16”-26 BSC threads are of the Whitworth variety, and either a ¼” Whitworth or 17/32” American Standard wrench is appropriate for removing the screw-in filler cap.

Be aware that petroleum-based hydraulic fluids are not compatible with the natural rubber seals. Armstrong still makes its own specially-formulated hydraulic fluid available. It can be obtained through Brit Tek at http://www.brittek.com/ . The factory manual stated that, if necessary, 20W mineral oil could be substituted during warm weather. However, not all mineral oils are the same. During WWII the Germans made a lot of advances in this area, so mineral oils are much more common in Europe than on our side of the pond. Mineral oil with an anti-foaming agent was (and may still be) used in the power steering system of Audis. I suspect that the factory intended for mineral oil with an anti-foaming agent to be what they suggested, and just presumed that the customer would be knowledgeable enough to use the right stuff in a pinch (b-i-g presumption!)

Note that Armstrong conveniently stamped their part number on every damper (except for the Spridget front units that were cast). All front dampers are the same Part Number (8177/1), even though there was a change in the very earliest models. On all rear dampers, the number is stamped on the underside of one of the mounting ears. MGB rear dampers will have 8178LH or 8178RH, or 12012 (Left Hand Side) or 12075 (Right Hand Side). According to Armstrong's 1978 USA catalog, Armstrong Part # 8178 fits all MGB Roadsters and GT 4 cylinder models through 1974. The 1973 and 1974 BGT V8 models used Armstrong Part # 10801 (which I have never seen). All models 1975 through 5 / 1976 used Armstrong Part # 12012. Afterwards, all models from 6 / 1976 to end used Armstrong Part # 12075. While there appears to be no difference in the 8178, 12012, and 12075, if matching, check that the numbers are the same just to play it safe.

Panhard Rod

The single most significant improvement that you can make to your suspension system is the installation of a Panhard rod. People who tell you that a leaf spring rear suspension does not need a Panhard Rod are only partly right. Think of a rear leaf spring as a lever with its fulcrum at the mounts. When lateral forces are transferred through the springs during hard cornering, the axle and the springs attached to them tend to move laterally across the bottom of the chassis. The only thing restricting this lateral movement on most MGBs is the spring mounting bushings at each end of the leaf springs. As long as the bushings are not worn or degraded, this does not present a serious problem during normal, casual driving. However, if the bushings have become worn, perished, or ovaled (as in the case of old rubber bushings), the fulcrum effect makes it easier for the axle that is attached to those springs to move laterally. This results in the front and rear wheels being out of alignment. Panhard designed the Panhard rod to specifically address the problem of the insufficient lateral rigidity of leaf spring suspensions. In fact, the correct technical name for a leaf spring suspension is to call it a "Panhard Suspension".

Due to the Panhard leaf spring design that incorporates the use of rear spring shackles, as one spring flattens its arc upward and its opposite counterpart extends its arc downward, the leaf spring that is compressing effectively lengthens rearward on its shackle and the attached end of the axle moves to the rear along with it while the opposite leaf spring is that arcing downwards effectively shortens on its shackle, moving the other attached end of the axle forward. As the rear axle becomes diagonal to the chassis, the resulting thrust angle of the axle worsens both understeer and torque steer. When the lateral forces are high (as during hard cornering), both the lateral and the directional misalignment of the rear axle increase, resulting in an increasingly serious torque steer effect which, when amplified by the serious misalignment caused by lateral movement of the axle, results in what is called "Snap Oversteer" when the rear axle contacts its bump stop. This should not be confused with "Spin", which is usually caused by a loss of traction resulting from the lifting of a rear wheel. An Original Equipment trailing rear stabilizer bar will not only help to reduce body roll, but will also help to reduce this compound misalignment by virtue of its resistance to lateral movement. Adding a Panhard rod will all but eliminate it. A Panhard Rod with a center-to-centre length of 36" and a vertical movement of the suspension of +/- 3" will produce a lateral movement of 0.125" (1/8"), and a vertical movement of the suspension of +/- 4 will produce a lateral movement of 0.22" (less than 1/4"). This is a far greater restriction of lateral movement than that which is attainable with even the hardest bushings. With the Panhard Rod maintaining the lateral position of the axle, you need not install hard polyurethane bushings into the leaf springs in an attempt to limit the fulcrum effect of a swaying rear axle. All that harder bushings would do in such a case would be to transmit more wheel vibration, noise, and road shock. With a Panhard Rod, you can retain soft bushings and have the best of both worlds. Before you mount a Panhard Rod, you should do a four-wheel alignment. This will guarantee that the mounting on the body will be properly located.

Be aware that the height of the Panhard rod helps to determine the height of the rear roll center. The roll center is an imaginary point around which the rear of the car rolls. The height of the rear roll center, and the front as well, is critical to handling. When you lower the Panhard rod, the rear roll center will drop. A lowered rear roll center promotes side traction at the rear, which tends to tighten corner handling. However, an extremely low roll center can generate excessive body roll, which in turn can cause suspension geometry problems. In addition, excessive body roll can force a delay in acceleration when exiting corners. It should be noted that the roll center also moves downward whenever the angle of the Panhard rod is increased and thus moved downward. During cornering, the body of the car exerts a side force on the rear axle and tires through the Panhard rod. When the Panhard rod is level, it transmits a wholly lateral force to the rear tires. However, when the Panhard rod is angled downward, it transmits a partially downward force to the adjacent rear tire, and its traction is then enhanced. Conversely, when the Panhard rod is angled upward, it transmits a partially upward force to the adjacent rear tire and its traction is somewhat lessened. A good rule of thumb is to keep the difference in the height of the mounts of the Panhard rod to within 10% of the length of the Panhard rod. Therefore, the longer the Panhard rod is, the better.

The location of the mounting point of the Panhard rod, ahead of or behind the axle, determines whether the mount will move up or down during deceleration or acceleration. During deceleration, the mounting point drops if the mount is ahead of the axle, but raises if the mount is behind the axle. It should be noted that during cornering, whenever the Panhard rod is angled downward to the right, a front-mounted Panhard rod resists axle wrap-up under acceleration and enhances axle wrap-down during deceleration. A rear-mounted Panhard rod gives the opposite results. However, in practice, the effects upon handling appear to be minimal.

By configuring the Panhard rod with the body mount on the left-hand side of the car, the effects of engine torque on corner exit grip are minimized, allowing you to apply power earlier in the curve and thus get a jump on the competition down the straightaway, no matter what direction the corner is. If the Panhard rod were to be mounted with the body mount on the right there would be a much bigger disparity between left and right hand cornering ability. This would effectively make you choose between setting the car up to exit either left-hand corners or right-hand corners, and live with an under-performing car in the other direction.

One thing that you will notice after installation of the Panhard rod is that the steering response will seem to quicken and the rear end of the car will seem to be lighter on roads that offer reduced traction. This is due to the fact that there will no longer be any delay in response caused by the shifting of the body over the axle. You also may notice that the rear end of the car has a greater tendency to break loose during hard cornering. If this occurs, you will need to upgrade your tires as this type of suspension modification is intended for performance, and 60,000 mile treadwear family sedan tires just will not do! If you need a Panhard Rod whose length is adjustable in order to accommodate mounting onto either wire wheel or solid wheel Salisbury tube-type rear axles, you can get one from the MG Owners Club over in the UK. They have a website at http://www.mgocspares.com/

Stabilizer Bars

The suspension design of the MGB dates from back in the days when bias-ply tires were the norm. These tires could handle only moderate lateral forces before they would distort and warn the driver through their screeching that they were approaching the limit of their cornering ability. The geometry of the MGB's front and rear suspension systems were designed to handle well under those limits. However, today's modern radial tires can handle far greater lateral forces. This in turn will increase body roll when the car is pushed to the limit of the traction of the tires. When the higher lateral loads forces the body to roll to a greater degree than the designers had originally envisioned, the leaf spring on the outside of the car compresses more while the one on the inside extends more, causing the rear axle to assume a greater diagonal relationship to the longitudinal axis of the chassis. Should one of the rear wheels hit a large pavement undulation under this circumstance, its movement will increase this already diagonal relationship more radically than under moderate driving, thus increasing rear wheel steer, often causing the driver to experience "Snap" handling effects, especially when applying the brakes or increasing power. If you want to reduce these geometry-induced hazards, install stronger stabilizer bars along with a Panhard Rod.

When a car turns, it must be understood that the inside wheels of the car follow a smaller diameter circle than the outside wheels do. If both of the steer wheels were turned by the same amount, the steer wheel on the inside of the turn would scrub, i.e., effectively sliding sideways, thus lessening the effectiveness of the steering. This scrubbing of the tire, which also creates unwanted heat and wear in the tire, can be eliminated by turning the inside steer wheel to a greater angle than that of the outside one so that each steer wheel is perpendicular to the radius of its turning circle. To enable this to happen, the misalignment needs to progress from zero (both steer wheels pointing straight ahead) to a point where there is a sufficiently different angle between the steer wheels to create the desired misalignment of both steer wheels when they are turned.

In order to create the required mechanism for the angles of the steer wheels to change at different rates, the steering arms need to angle inwards so that a combination of both the angle and the length of the steering arms will achieve this desired misalignment and thus compensate for increasingly tighter turns. This is the basis of the Ackerman Steering Principle. A significant feature is that this unequal angular movement is exponential, that is, the more you turn the wheel, the greater the angular difference between the wheels. However, in order for this steering geometry to be effective, body roll must be restricted to within design limits, and this is where a combination of good stabilizer bars pays off.

It must be understood that when the body of the car rolls, the front suspension system on the outside of the curve compresses, while the front suspension system on the inside of the curve extends. This body roll causes the tie rod that controls the outer steer wheel to arc upwards, pushing its steer wheel to a more outward steering angle, while the tie rod that controls the inner steer wheel is caused to arc downwards, pushing its steer wheel to a more inward steering angle. This steering geometry achieves greater angular inequality of the turned wheels, which results in the inside wheel trying to follow a smaller diameter circle than it is actually doing. The net effect is what is described as "Oversteer". When first sold to the public in 1962, the MGB Roadster had no front stabilizer bar in order to restrict body roll, but one was offered as a factory option for owners who drove hard. As the lateral load capabilities of tires were rapidly improving during this era, oversteer increasingly became a problem, so in November of 1966 the factory wisely standardized the front stabilizer bar to become Original Equipment on all Roadsters. However, today's modern radial tires permit far greater lateral loads to be achieved, thus requiring stronger limitation of body roll when driving hard.

One of the best improvements that you can make to your suspension system is the installation of both front and rear stabilizer bars. Why install front and rear stabilizer bars? Stabilizer bars, sometimes known as sway bars and anti-roll bars (not to be confused with roll-over bars / cages), are devices which add suspension roll stiffness without significantly adding spring rate. The purpose of stabilizer bars is twofold: first, to reduce body roll, and second, to balance front vs. rear roll stiffness. Stabilizer bars do not reduce total vehicle weight transfer. When a stabilizer bar is added to one end of a car, the balance of the car is effected. This occurs because the end of the car with more roll stiffness also gets more of the total weight transfer, while the other end gets less. In most cases, MGs included, if a larger front stabilizer bar is installed, then a smaller rear bar is employed in order to re- balance the handling. A vehicle with a higher than normal Center of Gravity, such as a GT body or a heavy / tall roll cage, may require larger-diameter stabilizer bars. A reduction of Body Roll will correspondingly reduce the car's Roll Moment, the period during which weight is being transferred from one side of the car to the other. Directional reaction to steering input is slowed during a car's Roll Moment. By reducing Roll Moment, the period between a steering input by the driver and the car's directional response to it is reduced, so an increase in steering responsiveness should be expected. In addition, should the body of the car roll beyond the travel limits of the 8½"-long axle strap on the inside of the turn, the body will, via the axle strap, lift the inside rear wheel off the pavement, resulting in a quite exciting handling experience, mitigated only by the installation of a limited-slip differential.

A stabilizer bar is really nothing more than a torsion bar, which is a type of spring. However, when the car is moving straight ahead on a smooth road the force on the axle is nonexistent because the transverse axis of the body of the car and the attached stabilizer bar is parallel to the longitudinal axis of the axle. Only when the axis of the axle deflects out of parallel with that of the stabilizer bar does any springing effect occur. When the body of the car tries to lean over in a turn the longitudinal axis of the axle deflects out of parallel with that of the stabilizer bar whose arms are attached to each end of it. The stabilizer bar resists being twisted along its transverse axis under the car, attempting to return to its original untwisted shape. Thus the end of the axle on the outside of the curve is forced downward and the end of the axle on the inside of the curve is forced upwards by this torsional resistance. As long as the leaf spring on the inside of the curve exerts more downward force than the opposing lifting force of the torsion bar, the inner wheel will be held onto the ground. If the lifting force of the stabilizer bar is greater than the downward force of the leaf spring, as in the case of extreme body roll, then the wheel will be lifted off of the ground. Fortunately, such conditions are rarely encountered outside of a racetrack. However, a car not fitted with a rear stabilizer bar will tend to roll more, lifting its rear wheel when the axle strap extends to its full length. Racing legend Joe Huffacker developed the front / rear stabilizer bar combination to the point that this was not a problem.

If the handling of a car is already well-balanced, then the addition of a stronger front stabilizer bar alone should result in more understeer, while simply installing a rear stabilizer bar without increasing the diameter of the front stabilizer bar will result in oversteer. On the other hand, there are other resulting factors, such as in the case of the issue of traction, that may not be apparent when such general rules are too broadly applied. The premise for installing a larger front stabilizer bar that is not balanced by the addition of a rear stabilizer bar is predicated on the increase in Camber that occurs during body roll (i.e., Camber Gain). The greater angle of body roll permitted by a smaller, weaker front stabilizer bar results in an increase of positive (+) Camber that is not negated by the increase of negative (-) Camber that occurs during suspension compression, thus reducing traction in the outside tire. The load on the outside tire is increased by increasing the diameter of the front stabilizer bar, but by reducing the angle of body roll, resulting in less total positive (+) Camber increase, thus yielding a net increase in front traction attained at the price of increased understeer. In seeking optimal handling, a balance of forces is essential. This is due to the fact that the first function of a stabilizer bar is the reduction of body roll. The reduction of body roll is dependent on the total roll stiffness of the vehicle. However, it must be understood that increasing roll stiffness does not change the weight transfer from the inside wheels to the outside wheels. It only reduces body roll. The total lateral load transfer is determined by both the height of the Center of Gravity and the car's track width. The second function of a stabilizer bar is to tune the high G-force / limit understeer behavior of the vehicle. This is accomplished by changing the proportion of the total roll stiffness that comes from the front and rear axles. Increasing the proportion of roll stiffness at the front will increase the proportion of the total weight transfer to which the front axle reacts and decrease the proportion to which the rear axle reacts. This will cause the outer front wheel to run at a higher slip angle, and the outer rear wheel to run at a lower slip angle, which results in an understeer effect. Increasing the proportion of roll stiffness at the rear axle will have the opposite effect and decrease understeer.

Increase in Roll Stiffness When Increasing The Diameter Of A Stabilizer Bar									
From/To	**1/2"**	**9/16"**	**5/8"**	**11/16"**	**3/4"**	**13/16"**	**7/8"**	**15/16"**	**1"**

1/2"	0%	60%	257%						
9/16"		0%	52%	123%	216%				
5/8"			0%	46%	107%	186%	284%		
11/16"				0%	42%	95%	162%	246%	
3/4"					0%	38%	85%	144%	216%
13/16"						0%	35%	77%	129%
7/8"							0%	32%	71%
15/16"								0%	29%
1"									0%

Many manufacturers of stabilizer bar kits offer their products in standard diameters such as 9/16”, 5/8”, 3/4”, 7/8”, and even 1”. However, this does not mean that all bars of the same diameter from different manufacturers have exactly the same torsion spring rate. This is dependent on what alloy they are made of and to what level of hardness they are taken to during the heat treating process. This is why it is so important to get them in sets from the same supplier, preferably one of established reputation. Naturally, these cost more than no-name imports from God-only-knows-where in Asia. Quality always costs more. Generally speaking, cheap stabilizer bars increase their resistance much more slowly as they twist, and have a shorter service life. If the quality control fails during the heat-treating process, they may even be prone to breakage.

Chrome Bumper models (1962-1974) and Rubber Bumper Models (1975-1980) each require different stabilizer bars. This is due to the differences in both weight and ride heights of the two versions of the car requiring different length arms and slightly different geometry for the stabilizer bars. It is important that both sets consist of a pair of front / rear mounted stabilizer bars with rates that are balanced against each other, otherwise the car will suffer from either understeer or oversteer. Perhaps the most practical combination is a 7/8" front stabilizer bar and a 5/8" rear stabilizer bar as used by racing legend Joe Huffacker.

Make sure that the pair that you purchase have the ends of the front stabilizer bar forged flat and have the mounting holes already in them, and that the rear stabilizer bar is both arched for clearance and that its ends are threaded for the installation of the Original Equipment end pivot bearings so that no modifications will be needed. I used a Salisbury tube-type rear axle housing and its stabilizer bar (BMC Part # BHH 2003) from a scrapped 1978 Rubber Bumper MGB. These rear axles of the 1977 through 1980 models have the brackets for the end pivot bearings of the rear stabilizer bar already welded onto them. All that I had to do was shorten the threaded ends of its arms a bit so that I could screw on the end pivot bearings. This was because mine is a Chrome Bumper car that sits lower than the Rubber Bumper cars, thus shortening the distance from the axle brackets to the pivot mounts. Couple these with a set of new Original Equipment-rate springs and rebushed suspension components and you will be pleasantly surprised at the difference. It is a time-proven formula that is better than spending many hours of labor (not to mention money) experimenting with home-brew combinations. As for comfort, you will notice very little difference when driving in a straight line on smooth roads. Only when pushing very hard through turns and curves will you notice that the ride is marginally stiffer. Add a Panhard Rod and you will be amazed! An Original Equipment rear stabilizer bar will also have an additional benefit: it will function as an antitramp bar under all but the most aggressive acceleration.

It should be noted that an Original Equipment rear stabilizer bar is mounted is mounted to the underside of the body of the car in a trailing configuration, thus permitting its arms to be connected directly to the rear axle without causing the rearward movement of the axle to result in spring binding as the leaf springs compress. All of the aftermarket rear stabilizer bars that I have seen are mounted under the luggage compartment in a leading configuration that requires the use of rod links in order to connect the stabilizer bar to the rear axle with the purpose of preventing spring binding as the leaf springs compress. This particular configuration will in no way limit the lateral movement of the rear axle. The deflection of the rods also allows a degree of free movement before the resistance of the stabilizer bar is applied, resulting in initial body roll when entering a curve or altering direction.

The new front and rear stabilizer bars greatly improved the handling of my 1972 Roadster, but attendant to the ability to corner harder I found that with the consequent lateral forces I had to install a Panhard Rod in order to properly control the rear wheel tracking. Once that was done, the car became what I had been seeking. The steering became neutral: No oversteer, no understeer. Just point it and it remains flat (almost no body roll) and goes where you want it to go, regardless of cornering forces, with no snap oversteer near the limit, tracking true just as a great sports car should. MG used this combination on the original prototype development cars, but the bean counters in the Cost Accounting Department had their say in the matter and the car entered mass production with no stabilizer bars at all. A front stabilizer bar was, however, available for the more discerning drivers as an extra-cost option. Although they succeeded in getting approval for the GT model to be equipped with a stabilizer bar from its inception in 1965, it was not until November of 1966 that the engineers convinced management to correct the omission on the Roadster models, and even then all that they could get approval for was the standardization of the front stabilizer bar. Pity!

The advent of the Rubber Bumper MGB was accompanied by handling problems that partly had their origins in its increased ride height. Of course, the extra 175+ lbs of weight of the Rubber Bumpers and their attendant mounting hardware added to the nose and tail of the car, compounded by the higher center of gravity and the corresponding altered roll center, only made things worse. In addition, the increased weight on the nose and tail ends of the car resulted in a Pendulum Effect, influencing the car's responsiveness. As if that was not bad enough, the Cost Accountants at British Leyland made sure that the 1975 and 1976 models had no stabilizer bars at all. Not surprisingly, these cars tended to roll and lurch through curves. The management of British Leyland finally realized their error and managed some damage control by permitting MG to install both front and rear stabilizer bars on the 1976 and all subsequent models. Now, if only they had put in the Panhard Rod..... (sigh).

Unsprung Weight

Of course, the primary function of any suspension system is to keep the wheels in contact with the pavement, which brings up the matter of unsprung weight. Unsprung weight is that portion of the car's mass which is not supported by the springs, i.e., that of both the suspension system components and the wheels. When a wheel strikes a bump in the road, this weight rises upward. The greater the weight of the wheel, the more it resists this movement, thus the shock of the impact is more greatly transferred through the suspension components to the body of the car. This rising weight also has mass, and therefore inertia. The greater the inertia of the unsprung weight, the more difficult it is for the suspension system to keep the movement of the wheels firmly under control and in contact with the ground. Decreasing unsprung weight reduces inertia in the reciprocating mass of the suspension, so it would be less necessary to uprate the springs and consequently the dampers. Thus, an increase in both roadholding and comfort can be obtained if unsprung weight can be reduced.

As it is, little can be done to reduce the weight of the components of the suspension system itself. The Hardy-Spicer banjo-type axle of the Mk1 MGB Roadsters weighs 115 lbs, sixty pounds less than that of the later 50% heavier Salisbury tube-type axle, which weighs in at a relatively ponderous 175 lbs. This being the case, the subject of the of the weight of the wheels must be addressed.

Wheels

The wheels originally were of two types: Rubery Owen steel disc wheels and Dunlop wire spoke wheels. The wire spoke wheels were 4.5JX14 (4.5" rim width). The Rubery Owen steel disc wheels for the Roadster model were 4JX14 (4" rim width) while those of the GT model were 5JX14 (5" rim width), both weighing in at about 14 lbs. These steel disk wheels were replaced with 5JX14 (5" rim width) Rostyle steel wheels in 1969. These weighed a heavier 17 pounds.

Cars originally equipped with steel wheels can have their unsprung weight reduced by switching to alloy wheels. MG used 5.5JX14 (5.5" rim width) alloy wheels on their Jubilee, Limited Edition, and V8 models. In addition, many MG dealers offered aftermarket alloy wheels as accessories. Of these, the 14" X 6'' Minilite (16 lbs, 12 oz., popular in the UK) and the 14" X 6" with 15mm offset Panasport (12 lbs, 8 oz., popular in the USA) seemed to be the most desirable. Panasport also offers a 15" X 6" (15 lbs, 7 oz.) wheel with 22mm offset. Minilite also makes a 15" X 5.5" wheel with 15mm offset that is 30% lighter at 10lbs, 13 oz., but it is a magnesium wheel that is not suitable for street use. In addition, both 14" X 5.5" with 23mm offset and 15" X 5.5" with 19mm offset alloy reproduction Minilite wheels are available from Minotaur at 14 lbs, 4 oz. and 17 lbs, 8 oz., respectively. All of these alloy wheels have become quite popular with those who wish to restore their cars to period-correct standards.

From the standpoint of both unsprung weight and strength, wire wheels are the heaviest and least desirable of all. A wire spoke wheel is weaker than a modern steel disk or alloy wheel, requiring 72 spokes in order to attain an acceptable level of strength for hard driving, and typically weighs in the neighborhood of 25 lbs. Two basic versions of the 4.5J Dunlop wire wheel were offered for the MGB in either a painted (BMC Part # AHH 6487) or a chrome-plated finish (BMC Part # AHH 6396) . The factory's Competition Department also made available a 70 spoke 5.5J Dunlop wire wheel in a painted finish (BMC Part # AHH 2350) . There are at present two principle manufacturers of wire spoke wheels: Dunlop and Dayton. Of the two, the standard Dayton wheel is definitely of superior strength and overall quality with its .203" (5.1562mm) diameter spokes. Their wire wheel is also available with optional .225" (5.715mm) diameter heavy-duty spokes for those who drive really hard. In addition, Dayton offers the unique option of sealed spokes so that tires may be mounted without tubes. Unlike Dunlop, they do not use the old pattern of rim edge, but instead use a modern design that will permit the beads of modern tires to seal against the side if the rim so that tubeless tires may be fitted without the necessity of using tubes once the spokes have been sealed. Also unlike Dunlop. Dayton wheels have a fully machined hub that allows them to be mounted correctly on tire balancing machines using standard cones.

For those who wish to reduce unsprung weight but still want to retain the use of wire spoke wheels, Dayton has also recently introduced a wire wheel that uses an aluminum rim that weighs in at a mere 15 lbs, 7 oz. However, there is a third manufacturer of wire spoke wheels: Borrani, of Aston-Martin fame. Their wheels use an aluminum alloy rim and are appreciably lighter than the other equivalent steel rimmed wire wheels, weighing 15 lbs, 7 oz.

They are, consequently, quite expensive, costing $1,090 each vs. $245 for the Dayton steel rim wheel.

Unless you have Dayton wire wheels and have paid extra to have the spokes sealed, you will have no choice but to use tubes inside of your tires. There are a few secrets to making sure that a tubed tire will give trouble-free service. Make sure that the threaded ends of the spokes do not protrude into the interior of the wheel. Be sure to put in new band on the rim. Beware of tubes that are labeled “radial / bias ply”. These one-size-fits-all tubes that are available at most modern tire shops simply will not work in our cars. By all means, use tubes that are intended solely for use in radial tires. Be sure to check the inside of your tire for a little square information block. Some tires have this and it will scrape on the tube, so it should be removed. Run a nylon stocking around the inside of the tire in order to assure there are no fine burrs or anything that might cause wear of the tube. Here is an ol’-timey-mechanic's tip: dust the inside of the tire and the exterior of the tube heavily with talcum powder. The stuff will act as a dry lubricant for reducing chafing. If you can, find an unscented product as the perfumes might have some effect on the rubber. Perfume in talcum powder rots rubber, which is why divers never use it on their wet suits. Unscented talcum powder is sometimes called “French chalk,” so you might find it under that name as well. It also helps to “burp” the tire after it has been inflated to seat the tire bead. Once the inner tube has been inflated and the tire bead is seated, remove the valve stem core and let the pressure out of the tube. Move the valve stem around (gently) while it is deflating. This lets any air out that is trapped between the tube and tire, causing a bubble. Repeat this process until no more air escapes around the valve stem.

All wire wheels require maintenance that solid wheels do not. They should always be retrued on the occasion of their tires being replaced. However, despite their drawbacks, wire spoke wheels do seem to have two advantages: they allow more airflow to the brakes, facilitating quicker cooling and thus less brake fade, and, to some people, they are the most beautiful type of wheel that can be fitted onto an MGB, giving the car what they perceive as a defining “Classic” appearance. For owners of wire wheeled cars there does exist the option of mounting alloy wheels with splined hubs. These are available from Minotaur and weigh 20 lbs.

The Dayton 15" X 6" 72-spoke wheel has a backspace of 4.25" (107.95mm), while the original Dunlop 15" X 5" had 3.69" (93.726mm) of backspace. This means that .56" (14.224mm) of the additional width is added to the inside of the wheel and .44" (11.176mm) is added to the outside. The center of the tire will be .060" (1.524mm) inwards from the Original Equipment specification position. Fortunately, this is not a great enough difference to be significant.

Control is at its best when the tires are solidly held on the road. If the wheels are not properly balanced, this is not possible. Rostyle wheels are notorious for being difficult to balance. This is usually the result of the balancing equipment being used. Always get your Rostyle wheels balanced by a company that has stud mounts for the balancer. The stud mounts use the actual mounting holes on the wheels themselves and not the big hole in what one would assume to be the center. People have had extreme problems with balancing when using the center of the Rostyle wheels as the mounting reference point. It is not the actual center of the wheel. Unlike modern wheels, the center hole of the wheel was never intended to be used for balancing, as computerized balancers did not exist at the time. They are present merely to provide clearance when mounting. At the very minimum find a balancer with a four-stud adapter, but on-car dynamic balancing is best (if you can still find one). Be aware that proper balancing can be performed only on undistorted wheels. After years of DPOs overtorquing the lug nuts, the soft grade of steel of which the Rostyle wheels are fabricated will allow distortion in the middle, dishing in, and causing the studs to bend slightly toward the wheel center when they are tightened. This makes it quite the chore to remove the brake drums. Sadly, if you merely replace the bent studs, the nut tapers will only contact a portion of the countersinks of the wheel, and not seat all the way around until the new studs have bent inward to match the old ones.

The steel (disc) wheels of the MGB Roadster originally used 4Jx14" rims while the steel (disc) wheels of the GT model originally used 5Jx14" rims, and the wire spoke wheels of both models used 4.5Jx14" rims, both models being fitted with 5.60x14 bias-ply tires. MG made both 5.5Jx14" slotted steel (disc) wheels and 5.5Jx14" wire spoke wheels available through their Special Tuning Department (Part #'s AHH 8112 and AHH8530), but today these wheels have become quite rare. However, any shop that specializes in the repair of damaged rims should be able to remove the rims from your steel (disc) wheels and weld on rims of appropriate width with the correct amount of offset.

Tires

Once the subject of controlling unsprung mass has been resolved, the complex issue of tires can be examined. When trying to improve handling, many people fail to include their tires in the equation. Putting stronger front and rear stabilizer bars onto a car, and then using tires meant for a family sedan is dangerous and simply asking for problems, as such tires have hard compounds that are meant for high mileage at the expense of traction. You get what you pay for, and economy tires have no business being on any sports car. Those economy tires may not seem to be such a good deal if you pull up 20 yards further down the road during an emergency stop, or slide sideways into a ditch on a surprise decreasing-radius turn.

The tires that originally shod the earliest MGBs were of the now-long-obsolete bias-ply design. Later, in 1965, radial tires became available as optional equipment, SR155/14 tires being used on Roadsters and SR165/14 tires on the GT. The SR 155/14 tire is 23.78" (604.012mm) tall, while the SR165/14 tire is 24.4" (619.76mm) tall. Interestingly, both of these tire sizes were used in conjunction with the same speedometer and speedometer angle drive unit on the transmission. Now obsolete, these tires have been largely superseded by tires of lower profile.

Today, with a rolling radius of 23.78" (604.012mm), P175/70R14 tires equate closely to the SR155/14 in terms of overall rolling radius. Equating to the SR165/14 that was found as Original Equipment on the MGB GT, the P185/70R14 appears at this time to be the most popular size in use by MGB owners. This size also retains (as close as is needed) the rolling radius of the original SR165/14 tire so that a reasonable degree of speedometer accuracy is retained. These tires provide a larger "footprint" of tread on the pavement, thus allowing better grip. However, a larger footprint increases low speed steering effort. A lower profile increases relative sidewall stiffness and thus steering response becomes sharper. P195/70R14 tires have been mounted by some owners, but with a 2.7% larger difference in rolling radius, they will cause the speedometer to give a pessimistically inaccurate reading. Clearance of the rear wheel arch will also become a problem, especially when there is lateral rear axle movement during cornering.

Sadly, the 70 Series tire is now disappearing from the product lines of the major tire manufacturers, having been largely discontinued by Dunlop, Pirelli, Michelin, Continental, as well as many others. Even worse, high performance tires in this size have long been discontinued as well. Consequently, the P185/65R14 is now becoming an increasingly more frequent fit as it is more commonly available. The 65 Series profile tire also has stiffer sidewalls than a 70 Series profile tire, so you may anticipate a further sharpening of the steering response and a corresponding increase in low speed steering effort as compared to that of the P185/70R14. If you go with a P195/60 tire you will need to purchase a set of 15" wheels with 5.5J or 6.0J rims in order to keep the profile of the tire within acceptable limits, plus the rolling radius of the assembled wheel will be smaller than that of the Original Equipment wheels and your speedometer will be resultantly optimistic. If you prefer to stay with 14" wheels, take a good look at P185/65R14 Series tires. These have a rolling radius very close to that of the Original Equipment tires (ever so slightly smaller by 3/8"), will fit into the wheelwells a little better, and will actually be closer to the original rolling radius than P185/70R14 tires. Being a more modern (read: advanced) design, not only will you end up with a smoother ride, but also P185/65R14 tires offer a wider range of possibilities for traction / handling / wear combinations.

Generally speaking, you should be able to come up with something that is superior in all categories to the old P185/70R14 designs. Why do not more people continue to use them on their MGBs? Because when the original size tires became harder and harder to find, people switched to the P185/70R14 because they were more modern and they were what was available at the time that was closest to the original rolling radius. Since then it has become something of an Urban Myth that they are the best way to go. Actually, the more modern P185/65R14 is superior on almost every count and is in no way inferior to the older design. Unless you are substantially uprating the power output of your engine or modifying the suspension so you can drive very, very hard on curves, you do not need to go much larger. For most people, the P185/65R14 will do fine. The P195/65R14 is also a popular choice since it offers the widest practical width that can be used within the wheelwell. Because the reinforcing lip of the fender is located at the widest part of the sidewall of the tire, rubbing of the inside of the fender during hard cornering is not unusual with such wide tires, so installation of a Panhard Rod is also advisable in such cases. Some people try grinding away the reinforcing lip of the arch that is rolled into the wheelwell aperture, but this is a poor idea as the purpose of the lip is to reinforce the edge of the wheelwell aperture and thus prevent cracking of the sheet metal panel. However, a skillful rolling of the lip may be helpful in some cases. Switching to wheels that have increased positive (+) offset can merely result in clearances issues with the inner bumpstop structures and the front stabilizer bar, as well as result in handling issues.

Be mindful of the fact that a 5” rim is too narrow for a 195-width tire. The lateral tread profile would be distorted outward and upward at proper inflation pressure. You could underinflate the tire to get the lateral tread profile to the correct contour, but that would result in the sidewalls flexing beyond their intended design limits and the tread squirming to the point that directional stability could be adversely effected, plus the tire would actually ride worse, not better, because you would be forcing it to do something that it was not designed to do. The heat that would be caused by the flexure would ultimately ruin the tire and could possibly result in sidewall failure or delamination of the tread from the body of the tire. In should be noted at this point that the heat that warms up the tread compound is not the result of the tire’s width; it is the result of flexure. Wider tires have a greater lateral surface area and thus they dissipate heat better. Therefore, if all other factors are equal, they build up heat more slowly. If you want a stickier tread compound, get a higher-performance tire with a higher speed rating in a proper size that will correctly fit the rim that it is being mounted onto.

I personally use the P195/60R15 Michelin Exalto high performance tire, but my car has had the aforementioned suspension improvements in order to make exploitation of the advantages of such a tire possible. Without those modifications, the use of such a tire would be pointless. However, the steering at low speeds is heavier, and the ride is as stiff as any performance-oriented driver would be willing to tolerate on long drives. The reduced Roll Moment coupled with the stiffer sidewalls of the high performance 60 Series tires makes the handling highly responsive, almost darty. The car will require that you pay attention to what you are doing, making for an involving driving experience, just as a true sports car should.

The following chart may prove to be useful in deciding which tires will best meet your needs:

Wheel	**Tire**	**Profile**	**Dynamic**	**Speedometer Reading**

Size	**Width**		**Rolling Radius**	@ 60 MPH @ 96.6 KPH	@ 60 MPH @ 96.6 KPH
14”	155	80	293 mm	60.0 MPH 96.6 KPH	100.0 MPH 160.9 KPH
14”	165	80	301 mm	58.5 MPH 94.1 KPH	97.4 MPH 156.7 KPH
14	175	80	309 mm	57.0 MPH 91.7 KPH	95.0 MPH 152.9 KPH
14”	185	80	316 mm	55.6 MPH 89.5 KPH	92.6 MPH 149 KPH
14”	165	70	285 mm	61.7 MPH 99.3 KPH	102.9 MPH 165.6 KPH
14”	175	70	292 mm	60.3 MPH 97.0 KPH	100.5 MPH 161.7 KPH
14”	185	70	298 mm	58.9 MPH 94.8 KPH	98.2 MPH 158.0 KPH
14”	195	70	305 mm	57.6 MPH 92.7 KPH	96.0 MPH 154.5 KPH
14”	185	65	289 mm	60.8 MPH 97.8 KPH	101.3 MPH 163.0 KPH
14”	195	65	296 mm	59.5 MPH 95.8 KPH	99.1 MPH 159.5 KPH
14”	205	65	302 mm	58.2 MPH 93.7 KPH	97.0 MPH 156.1 KPH

15"	165	65	289 mm	60.8 MPH 97.8 KPH	101.4 MPH 163.2 KPH
15"	175	65	295 mm	59.5 MPH 95.8 KPH	101.2 MPH 162.9 KPH
15"	185	65	302 mm	58.3 MPH 93.8 KPH	97.1 MPH 156.3 KPH
15	195	65	308 mm	57.1 MPH 91.9 KPH	95.1 MPH 153.0 KPH
15"	185	60	293 mm	60.1 MPH 96.7 KPH	100.1 MPH 161.1 KPH
15"	195	60	299 mm	58.9 MPH 94.8 KPH	98.1 MPH 157.9 KPH
15"	205	60	304 mm	57.8 MPH 93.0 KPH	96.3 MPH 155.0 KPH

Of course, when selecting a tire you should be aware of its Speed Rating in order to choose something that is appropriate to your application. All tires are rated with a speed rating letter. This indicates the maximum speed that the tire can sustain for a ten minute endurance period without coming to pieces and destroying itself, your car, the car next to you and anyone else within a suitable radius at the time.

Speed Rating	**Maximum Speed**	
L	75 MPH	120 KPH

M	81 MPH	130 KPH
N	87 MPH	140 KPH
P	93 MPH	150 KPH
Q	99 MPH	160 KPH
R	106 MPH	170 KPH
S	112 MPH	180 KPH
T	118 MPH	190 KPH
U	124 MPH	200 KPH
H	130 MPH	210 KPH
V	149 MPH	240 KPH
Z	150+ MPH	240+ KPH

W	168 MPH	270 KPH
Y	186 MPH	300 KPH

If you find that the internal splines on the wheel hub have knurled a pattern into the wheel adaptor at the base of the taper, then be aware that this is not from overtightening, but rather from running loose. The wheel wears the taper itself, and the splines of the wheel's hub cut into the taper of the axle. There will also be corresponding wear on the hub of the wheel, in particular a high ridge around its periphery where the axle hub taper ends, although this depends on whether or not the wheels have been changed around. The tapers have 2 functions: first, to hold the wheel, and, second (and most important), is they centralize the wheel onto the hub in much the same way as do the wheel/lug nuts do on conventional wheels. Either wheels or hubs that have wear should have these high spots taken off, since fitting parts that have not worn in together will leave the wheel sitting on little high spots, which will flatten out, leaving the wheel loose again. Then it wears more, until the wheel falls off! The wear on the tapers is also a result of long-term use without adequate lubrication. The wear mechanism involved is called "fretting corrosion". This happens when there is slight motion between close fitting parts that are not lubricated. It is what produces the characteristic red staining and the peculiar dark but high polished look of the worn surfaces. Both colors represent forms of iron oxide, and both are excellent abrasives, so when you get them powdered in a joint, you have trouble. It consumes a lot of metal, and is deadly to splines as well as tapers. The same thing can be seen around loose lug nuts, where it additionally results in wheel failure from fatigue. The tapers should be well greased, as grease on the tapers also acts as a seal to keep water and dirt out of the joint. Do not neglect to apply an RTV silicone sealant on top of the spoke heads inside of the wheel hub in order to seal out water and prevent grease or anti-seize lubricant from being spun out onto your spokes and wheels.

Be aware that when installing a wire wheel, you should always tighten the knock-off nuts with the wheel off of the ground and free to rotate. BMC issued a Technical Service Bulletin on this matter. This will decrease the force transmitted into the spokes from each hammer blow. If the wheel is not free to turn, then the spokes are forced to take the full brunt of each hammer blow.

Springs

If you are inclined to try improving the handling of your car by switching from Original Equipment specification springs to something different, there are certain things that you need to be aware of. The first is the matter of spring rate. Spring rate refers to the amount of weight needed to compress a spring an inch (Example: 540 pounds per inch). In order to understand and properly check a spring for rate, you first need to know the factors that determine the rate of the spring. Fortunately, there are only three things that effect the rate of a coil spring. The first is the diameter of the wire from which the coil spring has been made. This effects rate since a large diameter wire is stronger and thus more resistant to twisting than a smaller diameter wire. Therefore, when the wire diameter is increased, the spring rate is also increased. The second is the Mean Diameter of the coil spring. The Mean Diameter is the overall Outside Diameter (O.D.) of the coil spring minus the diameter of one wire. When Mean Diameter increases, the spring rate decreases. If the rate of a spring is linear, then its rate is not effected by the load applied to the spring. For example, a linear rate spring that is rated at 540 Inch-lbs will compress one inch when a 540-pound weight is applied to the spring. If another 540-pound weight is placed onto the spring, then the spring will compress another inch. At this point, the load on the spring has increased to 1080 pounds. The rate of the spring, however, remains constant at 540 In-lbs.

If the load put onto a spring increases the rate of the spring, then the spring is said to have a progressive rate. Progressive rate springs are sometimes used on torque arms in order to absorb the torque of a Rover V8 engine. Keep in mind that the load (or preload) applied onto a progressive rate spring can greatly increase the rate of the spring. Typically, progressive rate springs are made by varying the spacing between the active coils of the springs. During compression, the close coils bottom out, bind, and deaden. This reduces the number of active coils available and the result is an increase in spring rate. Coil springs that are designed to include coils of different diameter or are wound using a tapered wire will also produce a progressive rate.

There are basically three different coil spring designs presently used in cars. They are: First, the closed and ground on both ends type (Coil-overs and rear conventional coil springs are of this type). Second, the closed at both ends but ground one end only type (Conventional front coil springs are normally this type). Third, the closed and ground on one end and open on the other end type (Similar to a conventional coil spring that has been cut). These three types of coil spring are used in different applications and provide different effects to rate. Since the designs are so varied, it only follows that the dynamics of each design are also varied. You must remember, however, that the only factors that affect spring rate are the wire diameter, the mean diameter, and the number of active coils.

Keep in mind that as a coil spring compresses, the inactive (dead) end coils gradually contact adjacent, active coils. This contact causes the active coils to deaden, thus increasing the rate of the spring. This is referred to as "Rate Creep". The rate creep that results usually stops after the first inch of spring travel and does not appear again until spring travel begins to approach coil bind. Accordingly, the rate marked on a spring can differ from the rate as seen by the chassis. This is especially true whenever a spring manufacturer rates a spring based upon the first inch of compression.

Rate creep can become even more complex and more difficult to monitor for owners using conventional type front coil springs that are designed with an open ended coil. The lower suspension arms used with conventional springs typically incorporate a stepped helix spring seat built to an SAE specification (.720" of step). The helix seat for the coil spring was designed into lower suspension arms in order to insure consistent installation of the spring. Keep in mind that any rotation of the spring affects the actual installed rate of the spring.

Unless springs used for this type of application are designed with one end coil that closely matches the seat of the spring in the lower suspension arm, a serious amount of rate creep can result. In order to minimize this type of rate creep, a conventional front spring should be wound with its bottom end closed so that it sits squarely in its helix spring seat. No active coil should touch the helix spring seat, just as in the case of the Original Equipment spring for which the suspension arm was designed : the second type of spring.

When built in this manner, the only contact of a coil spring with the lower suspension arm is through an inactive (dead) coil . As a result, as the spring compresses, the number of active coils in the spring is not affected by the lower suspension arm. Therefore, the spring's rate remains constant throughout normal suspension travel. Some rate creep will still occur due to contact between the dead end coils and the adjacent active coils, but the amount of rate creep is miniscule compared to the rate creep produced by an open-end coil spring.

If a spring has an open end coil, then the open end coil is active but gradually deadens as the lower suspension arm moves against the spring. A considerable increase in spring rate occurs until the open end coil is completely seated in the helix.

For example, during a test, a 1500 In-lbs open end coil spring gained 464 In-lbs. of rate after two inches of spring travel. By comparison, a 1300 In-lbs closed end coil spring gained only 48 In-lbs. of rate for the same amount of travel.

Keep in mind that any load change to an open end coil spring (via static weight, wedge, chassis roll, bumps, etc.) usually causes the spring's rate to change and, consequently, handling to change. If you are using open end coil springs you should chart their rates from static loaded height to fully loaded height weight(in one inch increments). You should compare this information before making spring changes. At this point, you should realize the importance of using springs that are designed to keep rate creep to a minimum.

Use springs that do not lean excessively (when positioned on a flat surface). This indicates that the ends are ground parallel to each other. This reduces the tendency for a spring to bow. You should check both ends. Use springs that are wound straight. You can roll the spring on a flat surface to check for straightness. Determination of the number of active coils varies according to spring design. Count the total coils minus two for springs with both ends closed. Count the total number of coils minus one for springs with one end closed and one end open. As the number of active coils increases, the spring rate decreases.

Coil bind occurs whenever a spring is compressed and one or more active coils of the spring contacts another coil. The rate of the spring increases whenever a coil binds since the bound coil or coils are no longer active (this changes one of the three rate-determining factors). Of course, handling is effected whenever a coil binds. If the spring is compressed to solid height (all coils touching) during suspension movement, the suspension will cease to compress. You can, and should, check for evidence of coil bind by examining the finish between the active coils. If a coil has bound, then the finish between them will show contact marks that appear as though they were drawn with a lead pencil. Normally any spring that is binding should be replaced with a taller spring. Be aware, however, there are springs on the market that are built with wire that is of a heavier gauge than that which is required. These springs will coil bind before others that are made of the proper gauge wire.

Under very extreme conditions, coil binding can cause a spring to unwind slightly. This can cause the mean diameter of the spring to increase and reduce rate of the spring. You should realize that the potential for coil bind is increased whenever short springs are used. Always match the spring to the job.

When a spring takes a set it will normally stabilize at its new height. The rate effectively remains the same since no appreciable changes have been made to any of the three factors that determine the spring's rate. Other than creating a need to readjust the chassis (in order to restore the original geometries and ride heights) the spring should provide satisfactory performance. It is not uncommon for even well designed and properly manufactured springs to settle up to 1% of their free height. It needs to be pointed out, however, that in cases where a poorly designed spring is subject to extreme over-stressing, the spring's height may not stabilize at all. The spring may continue to change height (both shortening and lengthening) as the spring is worked. As a result, the set-up on the car changes every time the spring's height changes. This can cause major chassis tuning headaches!

Many owners mistakenly believe that extra spacing between the coils of a spring indicates a preferable spring. While a spring must have sufficient stroke capacity, it also must have sufficient material in order to absorb the load put onto it. If the spring's material is not sufficient for the load put onto the spring, the material will become over-stressed and the spring will lose height and take a set. Handling, of course, is affected and the reason is not always apparent to the owner unless he pays close attention to his springs.

In choosing new springs for your suspension, be aware that the MGB used several different springs as the years progressed. Choose the wrong springs and the handling will definitely be adversely effected. To help guide you in making the right choices, the following information should prove useful-

Front Spring Specifications:

Model	Free Height	Coil Diameter	Free Coils	Loaded Height	Load Weight	BMC Part #
Pre-1972 Chrome Bumper Roadster	9.9	3.238	7.5	7	1,030 lbs	AHH 6451
1972 Chrome Bumper Roadster	10.2	?	7.5	7.24	1,030 lbs	?
1973-on Roadster	10.2	?	9	7.44	1,030 lbs	BHH 1225
Pre-1972 Chrome Bumper GT	9.1	3.28	7.2	6.6	1,193 lbs	AHH 5789
1972-on Chrome Bumper GT	9.32	?	7.2	6.84	1,193 lbs	BHH 1077
1975-on Rubber Bumper GT	10.2	?	9	7.44	1,030 lbs	BHH 1225

Front Spring Rates:

Chrome Bumper GT (circa 1973)	100 Inch-lbs	BHH 1077
Competition	348 Inch-lbs	AHH 6451
Competition	480 Inch-lbs	AHH 5789

Competition (lowered)	480 Inch-lbs	AHT 21

Rear Spring Specifications:

Model	Leaves	Interleavin g	Width	Gauge	Load	BMC Part #
Pre-May 1963 Chrome Bumper Roadster	5 + bottom plate	None	1 ¾"	7/32in	400 lbs	AHH 6453
May 1963-onward Chrome Bumper Roadster	5 + bottom plate	½, 2/3, ¾	1 ¾"	3 @ 7/32" 3 @ 3/16"	450 lbs	AHH 7080
Rubber Bumper Roadster to September 1975	6 + bottom plate	½, 2/3	1 ¾"	3 @ 7/32" 3 @ 3/16"	510 lbs	BHH 1767
Post-September 1975 Rubber Bumper Roadster	5 + bottom plate	?	?	?	?	BHH 1779
Chrome Bumper GT	6 + bottom plate	½, 2/3	1 ¾"	3 @ 7/32" 3 @ 3/16"	510 lbs-540 lbs	AHC 31
Rubber Bumper GT	6 + bottom plate	½, 2/3	1 ¾"	3 @ 7/32" 3 @ 3/16"	510 lbs	BHH 1767

Competition Rear Spring Rates:

	Rate	BMC Part #
Competition	93 Inch-lbs	AHH 7080
Competition	99 Inch-lbs	AHH 6453
Competition	99 Inch-lbs	AHC 31
Competition	100 Inch-lbs	AHH 8343
Competition	124 Inch-lbs	AHH 7346
Competition (lowered)	124 Inch-lbs	AHT 20

Vehicle Weights:

	Total Weight		Weight Distribution			
			Roadster		GT	
1963 - 1971	**Roadster**	**GT**	**Front**	**Rear**	**Front**	**Rear**
Kerbside Weight (Includes full fuel tank all optional extras and accessories)	2,303 lbs	2,401lbs	1,127 lbs	1,176 lbs	1,162 lbs	1,239 lbs

Normal Weight (Includes Kerbside weight plus 150 lb driver and 150 lb passenger and 50 lb luggage)	2,653 lbs	2,751lbs	1,235 lbs	1,418 lbs	1,269 lbs	1,482 lbs
Maximum Weight (Includes Normal weight plus towbar hitch load)	2,753lbs	2,851lbs	1,193 lbs	1,560 lbs	1,231lbs	1,620 lbs
1971 – 1974	**Roadste r**	**GT**	**Roadster**		**GT**	
			Front	**Rear**	**Front**	**Rear**
Kerbside Weight (Includes full fuel tank all optional extras)	2,394 lbs	2,446 lbs	1,216 lbs	1,178 lbs	1,198 lbs	1,248 lbs
Normal Weight (Includes Kerbside weight plus 150 lb driver and 150 lb passenger and 50 lb luggage)	2,694 lbs	2,746 lbs	1,332 lbs	1,362 lbs	1,314 lbs	1,432 lbs
Gross Weight (Includes	2,814 lbs	2,866 lbs	1,285 lbs	1,529 lbs	1,267 lbs	1,599 lbs

Maximum weight condition, including tow hitch and roof rack (GT)						

Adding some type of plastic or nylon strips between the leaves of the leaf spring makes the spring movement more progressive and less prone to the sudden, quick movement that is experienced when the static friction between inner leaves is abruptly overcome. To some extent, the same applies when the rear leak spring returns to a "normal" ride position. Without the reduction of interleaf friction the leaves of the spring do not fully rebound and usually comes up to a point then creeps up a small amount at a time as the leaf spring works over normal road irregularities. Suitable material is available from suppliers of street rod suspension parts such as Speedway Motors. Although it is a bit wide for the leaf springs of an MGB, it can easily be trimmed to fit. Speedway Motors has a website at: http://www.speedwaymotors.com/p/2079,323_Spring-Liners-with-Lip.html

One of the most recent developments is that of rear leaf springs made of composite materials. While interesting, it should be noted that what some people perceive as "softness" with the 135 lb composite springs is actually the result of their flexure characteristics. When settled under the weight of the car they have just enough resistance to leave the car at its proper height, yet because they are made of a different material and are of single leaf design, they have less resistance to additional flexure than the multileaf steel Original Equipment springs have. This reduced resistance to flexure, along with their elimination of interleaf friction and their 30 lb reduction in unsprung weight, makes them very supple and thus fine for relaxed cruising. However, their lesser resistance to additional flexure loadings makes them inappropriate for hard, sporty driving on a winding road. Even if a rear stabilizer bar is mounted in order to compensate for their decreased contribution to the control of rear axle sway, plus their reduced contribution to roll resistance, fast whoop-dee-doo pavement undulations can have the rear axle banging against its bump stops. This more rapid movement of the rear axle forces the fitting of tubular shock absorbers with coil-over-shock "helper" springs. In addition, they require the installation of a Panhard rod in order to prevent them from delaminating and even fracturing as a result of their tendency to twist along their length that is consequent to the heavy lateral loads that are induced by hard cornering.

It should be noted that the MGOC is now marketing a parabolic leaf spring of steel construction with a spring rate that is almost identical to that of the Original Equipment multileaf units. They also have the advantage of reduced unsprung weight, each being about ten pounds lighter than multileaf springs. Like the composite spring, it also has the virtues of no interleaf friction so that the suspension will be more supple than an Original Equipment rear suspension, but without the unfortunate tendency to delaminate and / or fracture under high lateral loads as composite springs are prone to do when not constrained by a Panhard rod. It should be noted that because they lack interleaf friction, they are more prone to torque-induced spring wrap and resultant axle tramp under high torque loadings. Multileaf springs have less of a tendency to twist along their longitudinal axis because their multiple leaves reinforce the upper spring along most of its length, but parabolic springs lack this reinforcement. Whenever a car equipped with parabolic springs tends to jiggle around on bad pavement, it is due to the axle's increased lateral movement due to their greater tendency toward deflection under high lateral loads. This deflection also results in a lengthwise twisting of the springs. Any lengthwise twisting of the leaf of a spring will also cause an increase in spring rate. The increase is proportional to the degree of the torsional twist. This increase in spring rate results in the rear end of the car feeling "twitchy" to the driver, and on bad roads can result in difficulty in maintaining control. This being the case, it would be wise to consider mounting a Panhard Rod in order to better stabilize the lateral tracking of the rear axle. It should be understood that the MGOC states that " these springs are only for use on standard cars to improve ride comfort, not for modified cars/engines." They can also become deformed along their arc profile due to the high levels of torque associated with modified engines or medium to high levels of torque and overzealous standing starts, allowing full torque to the rear wheels when setting off.

Replacing The Rear Leaf Springs

Many MGB owners approach the task of replacing the rear leaf springs on their car with trepidation. Actually, it is all very straightforward. Just work steadily and methodically and the work will go quickly. Try to rush the job and it will take forever.

Lubricate and preinstall the bushings and rear shackle links into the new springs before trying to remove either of the old springs. Never use petroleum-based grease as it will damage the bushings. Make sure that you have the special lubricant on the outside of the tubular stainless steel bushing sleeves that go inside of the polyurethane bushings. Push them into the bushings. Smear antisieze compound on the mounting bolts and inside of the tubular stainless steel bushing sleeve so that the mounting bolt will slide in as easily as possible. Slide the mounting bolt through the stainless steel bushing sleeve and twist it in order to be sure that the antisieze compound is smeared evenly inside of the stainless steel bushing sleeve.

Now, think in terms of safety. Chock the front wheels (always!), jack up and support the rear of the car with axle stands using the reinforcing plate immediately in front of the front eyes of the rear leaf spring, and then remove the rear wheels. Take care to adequately support the axle from beneath with either a hydraulic bottle jack placed on a hefty block of wood or, better yet, with a floor jack (trolley jack) under the differential casing, and then undo the damper drop-links links from the lever arm dampers for future inspection.

Removal of the rear leaf springs is a bit more involved than simply undoing the shackle bolts. The axle straps prevent the axle, and hence the springs, from lowering far enough to permit the shackle pins to be easily moved in and out. If you disconnect the axle straps, then you cannot lower the rear axle much more before the brake hydraulic hose (flexible pipe) comes under tension, which is definitely not a good thing. If you know that your axle straps are sound, then you can just let the axle hang on them, but otherwise you should support the rear axle as well and change the straps while you are under there. Next, for stability and safety, you need to support the springs with a jack while you remove the nuts of the U-bolts from one side of the axle, which will allow the spring to separate from the axle. However, the travel limit of the lever-arm damper will then stop it. If you have lever-arm dampers, then the bottom plate with its attached damper drop-link can be maneuvered out from under the spring, which should still be supported by the jack. Inspect the damper drop-link to ensure that it works freely. At that point, you can jack the spring down until it is hanging from its shackle, but is still supported by the jack, then you can proceed with the removal of the shackle nuts and the removal of the shackle and the rear leaf springs.

Having made those proper preparations, remove the rearmost spring mounting bolt first. Hopefully, the bushing sleeve on the front mounting bolt will not be rusted onto it. If it is, do not bother trying to pound the mounting bolt out because you will risk deforming the hanger bracket. If you have a large C-clamp, you can try pressing it out. If you can, you are lucky. If you cannot press it out, cut it off with a hacksaw or, better yet, with a Dremel tool fitted with a cutting wheel. When doing so, take care not to damage the mounting bracket. Cut between the flange and the bushing, on both sides, starting on the nut side so that you can hold the head of the mounting bolt with a pair of vice grips or a wrench so that it will not be able to spin as you cut it.

Once you get the spring off, examine the hanger brackets and the areas around them. It is not at all unusual to find rust there, especially in the area around the front bracket. Also, take a moment to examine the rear brake hoses. Be aware that they can collapse internally while showing no outward signs. Fragments from their interior lining can break free and cause clogs. If any of them are bulged, swollen, or, even worse, cracked, you should replace them as a matched set. This would also be an excellent occasion to flush the hydraulic system with denatured alcohol. If you do, you will be amazed at the crud that will come flushing out of the system. Denatured alcohol can be obtained at any paint store. Since the wheels have been removed in order to grant easy access to everything, pull the brake drums off and look to see if the brake slave cylinders or axle seals exhibit any signs of leakage.

As you are putting it all back together, be sure to use antisieze compound on all of the threads, and especially on the steel bushing sleeve that goes on the front mounting bolt. Whatever you do, do not reuse any of the old mounting bolts, nuts, or U-bolts as their threads are almost certainly corroded beyond redemption. Remount the rear end of the spring first (Yeah, I know that the manual says to do the front first, but let us do this the easy way, shall we?). Swivel the shackle links as far forward (toward the front of the car) as they will go and use the hydraulic jack under the axle in order to compress the spring so that it will extend forward into the front spring bracket until things align. As the spring extends you will need to tap the block of wood under the hydraulic jack with a heavy hammer in order to move the mounts of the axle into proper alignment with its locating holes in the leaf spring. An alternative method is to place a hydraulic jack under the spring with the axle unattached and tap the wooden block and jack forward as the spring extends. From the front side of the bracket, slide the tip of a tire iron above the eye of the spring so that you will be able to use it as a wedge lever in order to align the height of the eye of the spring inside of the bracket. When the eye of the spring is longitudinally even with the mounting hole of the bracket, work the tire iron slowly to lower the eye of the spring into alignment with the mounting hole of the bracket.

When replacing the mounting bolt of the shackle, note that the Original Equipment specification mounting bolt is pressed into a splined hole in the shackle plate and that the mounting bolt has a double shoulder at the threaded end, the smaller of which fits inside of the hole in the closing plate. This not only serves to protect the threads by keeping them away from the edge of the hole, but more importantly makes the tightened shackle into a rigid parallelogram that aids spring location, and hence in turn, axle location. Should you lose one of these special-purpose mounting bolts, do not attempt to use a plain machine bolt instead. Plain machine bolts will permit the rectangle of the closed shackle to be distorted into a rhomboid during cornering, which will give more lateral movement of the spring and hence of the axle. Over time, this will wear the holes in the plates into ovals as well as wearing grooves into the machine bolts, thus weakening them. There is also the issue of tightening the shackles. Even when the original shackle is tightened to 30 Ft-lbs, the bushings are only lightly nipped and clearance remains for the spring eye to pivot on them. However, without some form of spacer tube, a plain machine bolt is going to tighten the shackles onto both the bushings and the spring eye, thus restricting movement and damaging the bushings in a short time, as well as causing wear of the shackles.

Test the alignment by pushing in the mounting bolt. If it will not go through, do not attempt to pound it in with a hammer or you will damage the threads. Instead, patiently peer in there with a flashlight and adjust the alignment by compressing or decompressing the spring with the bottle jack (horizontal alignment), or by moving the tire iron (vertical alignment). Sometimes it helps to hold the flashlight against the outside of the bracket on the opposite side so that the necessary concentricity can be confirmed. When it is aligned, install the mounting bolt using hand pressure, and then spin on the nut. Be sure to use antisieze compound in order to protect the threads. Use the floor jack (trolley jack) to maneuver the axle so that the U-bolts can be installed into the spring plates and the damper arm plate. Also, check to be sure that the pads have their cutouts aligned properly. Misalignment of these can result in all sorts of strange handling effects. Once all of the U-bolts are in place on both ends of the axle, be aware that a static tension in the leaves is present whenever the leaves and the spring mounting plates are not longitudinally or laterally parallel. Any static tension will cause an increase in spring rate. If necessary, reposition the U-bolts on the axle tube in order to eliminate any twist. Bolt up the new springs loosely, push the axle over to center, and then lower the car onto the ground. Never tighten the front or rear spring mounting bolts until the car is back on the ground and the rear has been bounced up and down a few times in order to settle the suspension! Finally, torque the nuts on the U-bolts to 25-30 Ft-lbs. The axle should be about mid-way between hitting the bump-rubber stop and taking up all the slack in the axle strap, with a bit more distance to the bump rubber than the axle strap. The shackle should be pointing down and back slightly.

Drive the car in the driveway in a straight line in order to make sure that the front wheels are pointing straight ahead, and then measure the distance between the front and rear hubs. It is supposed be equal. If it is not, put the rear end up on stands again, loosen the U-bolts, place a wood block against the rear hub that has the longest measurement, and use a big hammer to give the rear axle a shove. It may take a few tries to get the measurements equal. When you have them equal the rear wheels will be properly aligned, so tighten the nuts on the U-bolts until the rubber pads bulge.

Prepare yourself to be almost shocked at the improvement in the ride and handling. That, of course, will give you all the incentive that you will need to rebuild the front suspension system. Once that has been accomplished you will know what an MGB is supposed to handle like and why so many people came back to the dealership right after their first test drive with a big grin on their faces and stopped their search for the right sports car! S-w-e-e-t!

Rebuilding the Front Suspension

Rebuilding the front suspension is very straightforward affair once you know the proper procedures: Always think in terms of safety before starting any project. Chock the rear wheels, set the hand brake, and then place two jackstands under the sill / frame rails.

Because you are working on the underside of the car, expect several, if not all, of the bolts and nuts to be rusted in place. Stock up on penetrating fluid and keep a large 4-lb hammer close at hand. A Dremel tool with some cutting wheels will also prove to be useful. Remove the bump stops and their distance pieces for replacement, and then disconnect the stabilizer bar link from the lower A-frame.

Before jacking the car up, remove the top machine bolt on the drop link that connects the front stabilizer bar to the lower front arm of the front suspension. Remove the wheels, detach the brake calipers by removing the two studs that secure the brake caliper to the swivel hub, and then disconnect the front brake hoses from the brake calipers.

Next, begin the removal of the front hubs by pulling out the front hub grease caps. The grease cap of the center-lock wire wheel, unlike the stud-wheel type, is recessed into the splined part of the hub. There is a 5/16" UNF threaded stud on the end of the grease cap. A slide hammer with a threaded adapter is normally used to attach to it, but what if you do not have a slide hammer, much less the needed adapter? True, the stud can fitted with nuts in order to protect its threads, then be clamped by a pair of vice-grip pliers and pulled and wiggled, but that leads to damage of the threads and distortion of the sealing rim, so a better solution is necessary. There is an easy way to remove the front hub grease caps. Take a 6" (152.4mm) long piece of ½" rod, drill and tap one end to 5/16" X 24 (this end will go into the threaded stud of the grease cap). Get a flat piece of steel with a hole in it for the rod to go through, and three nuts to fit onto the threads. Screw the tapped end onto the stud, then with the flat plate against the hub, while using the other two nuts to keep the stud from turning, tighten the nut against the plate. The grease cap will gradually be extracted. Now, remove the cotter pins, and then remove the hub nuts. Pull off the front hubs, and then remove the brake rotor assemblies and their splashguards (dust covers).

When preparing to remove the front coil springs, be sure to get the car high enough off the ground so that the spring pans will be able to end up vertical after both the swivel hubs and the springs have been removed. If you do not take the precaution of getting the car up high enough off of the ground, you will run into problems when trying to install the springs.

In order to remove the springs of the front suspension, place a hydraulic bottle jack under the spring pan in order to contain the pressure of the coil spring. Loop a strong rope through a coil of the spring and tie it to the upper suspension arm in order to prevent the coil spring from jumping out as this event can occur quite violently. Remove the cotter pins from both the top of the king pin and the fulcrum pin, and then loosen both of the castle nuts. Make sure that you do not remove the upper fulcrum pin before you have loosened the castle nut that secures the upper trunnion to the kingpin as the upper fulcrum pin secures the upper trunnion to the kingpin and thus prevents it from turning. Next, unscrew the nuts on both of the fulcrum pins until they are flush with the end of the machine bolt and then strike them with a hammer in order to determine if either of the fulcrum pins is rusted in place. Note that the lower fulcrum pin has a steel bushing on its shank inside of the trunnion. If the bushing has rusted onto the fulcrum pin, the fulcrum pin will have to be cut off by means of a Dremel tool that is fitted with a cutting wheel. When doing so, take care not to damage either the trunnion or the wishbone arm of the lever arm damper. Next, unscrew the steering rack tie-rod ends from their ball joints. Remove the machine bolt of the center arm of the lever arm damper and the upper fulcrum pin, and then allow the swivel hub / kingpin assembly to swing away. Remove the lower fulcrum pin from the bottom of the king pin. Remove the swivel hub / kingpin assembly and place it in a pan of solvent to soak. Note that the center portion of the kingpin is protected by upper and lower spring-loaded dust shields. Finally, slowly lower the jack until the coil spring falls free.

Once the coil spring has been removed, remove the A-arm machine bolts that secure the spring pan to the arms, and then separate the arms of the A-arms from the lower pivot. This will also separate the A-arms from the spring pan. Next, inspect all of the A-arms and the spring pan for ovaling of the holes, and replace if you find any. Finally, remove the steering arm from the swivel hub. Clean and degrease everything, and then repaint the components with POR-15.

Make sure that you use some emery cloth to clean up the lower pivot shafts of the A-arms before you attempt to install the A-arms with their new bushings. If they must be replaced, note that the lower A-arm pivot shafts (BMC Part # AAH 4003) that bolt to cross member are symmetrical and thus are not “handed”. In other words, there is no left-side version or right-side version, and no front or back to them. Use a flap sander in order to clean up and polish the inside of the bushing mounting bosses of the A-arms. This will allow the bushings to rotate freely and keep them from galling and “winding up”. Crud on these parts will play a major role in tearing up your nice, new bushings. Once you have their mounting surfaces nice and slick, do not install the bushings in a dry state. Instead, get some of that wonderful silicone grease from a Honda dealer and smear it all over the sides of the bushings. If you do not have a Honda dealership within a reasonable distance, instead lubricate them with a liquid soap solution prior to fitting. The bushings should then slide right in. If you choose to install rubber A-arm bushings, use the A-arm bushings of the RV8 model, not the Original Equipment rubber ones of the MGB, as these have a longer service life and produce a more positive steering response. These will have to be pressed into place. Be sure to use antisieze compound on all of the mounting nuts.

Do not attempt to reuse the fulcrum pin thrust washers if they are either grooved or ridged. Their endplay (endfloat) upon reassembly should be between .008” and .013” (.2032mm and .3302mm). When you reinstall the swivel hub assembly, be sure that the trunnion on the bottom of the kingpin is turned inward, facing towards the car. The mounting lug of the upper trunnion should be turned inward, facing toward the car.

You will probably need to replace all of the associated fulcrum pin parts. If possible, purchase these parts in a kit because you are probably going to need all of it. The kit should contain all of the parts surrounding the bolt / pin and the brass bushing that goes into the king pin fulcrum pivot. When you reassemble the mechanism, be sure to use plenty of antisieze compound on all of the threads and on the distance tubes.

You will need to make an honest appraisal of your own shop equipment as well as your own machining skills in order to decide if you can independently install the brass bushings of the swivel hubs. The old ones will have to be either pressed out or, in rare cases, cut out and the new ones subsequently pressed in. Once that has been done, the bushings will have to be line-reamed with a special reamer that has been designed expressly for the task ($$$). This expense usually prompts most owners to opt for a pair of professionally rebuilt swivel axles.

Be sure to use antisieze compound on all of the threads when you reassemble everything. With the exception of the inner pivot, do not fully tighten anything until the suspension is back at riding height. Keep the machine bolts of the A-arm loose until the car is back on all four wheels and “bounced” up and down a few times. Have someone sit in the driver’s seat in order to realistically load the suspension before you crawl under the car in order to tighten up all of the machine bolts. If the machine bolts are tightened before the car is weighted, then you should expect the bushings to wear out very quickly.

The proper torque settings for the front suspension are as follows-

A-arm to spring pan bolts: 22 Ft-lbs

Brake caliper mounting bolts: 40 to 45 Ft-lbs

Disc brake dust covers: 19 Ft-lbs

Disc Brake rotor to hub: 40 to 45 Ft-lbs

Front crossmember to body nuts: 54 to 56 Ft-lbs

Front lever arm damper (shock absorber) machine bolts: 43 to 45 Ft-lbs

Hub nut, align to next hole: 40 Ft-lbs

Lower fulcrum: 45 Ft-lbs

Lower suspension arm nuts, align to next hole

Lower suspension arm / spring pan nuts: 22 Ft-lbs

Road wheel lug nuts (Bolt on steel wheels): 60 to 65 Ft-lbs

Shock absorber pinch bolt: 28 Ft-lbs

Stabilizer bar link nut: 60 Ft-lbs

Steering arm machine bolts: 60 to 65 Ft-lbs

Steering tie rod locknuts: 33.3 to 37.5 Ft-lbs

Steering rack to front crossmember: 30 Ft-lbs

Steering U-Joint (Universal Joint) Machine bolt: 20 to 22 Ft-lbs

Swivel pin nut, align to next flat

Upper fulcrum: 40 Ft-lbs

Nuts And Bolts

Since different grade fasteners require different torque values, if you do not have access to, or do not have enough space to use a stretch gauge, the following torque values may prove to be of use to you –

	Unlubricated* or Unplated Threads**	**Unlubricated* or Unplated** Threads**
	Regular Hex	**Regular Hex**
	Grade 5 Bolt **Grade 5 Nut or Grade B Nut**	**Grade 8 or Grade 8.2 Bolt** **Grade 8 or Grade C Nut**
	Torque: Ft-lb (Nm)	**Torque: Ft-lb (Nm)**
¼"-20 UNC	8 Ft-lb (11 Nm)	10 Ft-lb (14 Nm)
¼"-28 UNF	9 Ft-lb (12 Nm)	12 Ft-lb (16 Nm)
5/16"-18 UNC	15 Ft-lb (20 Nm)	22 Ft-lb (30 Nm)
5/16"-24 UNF	17 Ft-lb (23 Nm)	25 Ft-lb (34 Nm)
3/8"-16 UNC	28 Ft-lb (38 Nm)	40 Ft-lb (54 Nm)

3/8”- 24 UNF	31 Ft-lb (42 Nm)	45 Ft-lb (61 Nm)
7/16”-14 UNC	46 Ft-lb (61 Nm)	65 Ft-lb (88 Nm)
7/16”-20 UNF	50 Ft-lb (68 Nm)	70 Ft-lb (95 Nm)
½”-13 UNC	70 Ft-lb (95 Nm)	95 Ft-lb (129 Nm)
½”-20 UNF	75 Ft-lb (102 Nm)	110 Ft-lb (149 Nm)
9/16”-12 UNC	100 Ft-lb (136 Nm)	140 Ft-lb (190 Nm)
9/16”-18 UNF	110 Ft-lb (149 Nm)	155 Ft-lb (210 Nm)
5/8”-11 UNC	135 Ft-lb (183 Nm)	190 Ft-lb (258 Nm)
5/8”-18 UNF	155 Ft-lb (210 Nm)	215 Ft-lb (292 Nm)
¾”-10 UNC	240 Ft-lb (325 Nm)	340 Ft-lb (461 Nm)
¾”-16 UNF	270 Ft-lb (366 Nm)	380 Ft-lb (515 Nm)
7/8”-9 UNC	385 Ft-lb (522 Nm)	540 Ft-lb (732 Nm)
7/8”-14 UNF	425 Ft-lb (576 Nm)	600 Ft-lb (813 Nm)

Cadmium-plated; It is recommended that all plated and unplated threads be coated with oil before installation.

** Use these torque values if either the bolt or the nut is lubricated or plated (Zinc-Phosphate coated or waxed).

	Lubricated* or Plated** Threads	Lubricated* or Plated** Threads
	Regular Hex	**Regular Hex**
	Grade 5 Bolt **Grade 5 Nut or Grade B Nut**	**Grade 8 or Grade 8.2 Bolt** **Grade 8 Nut or Grade C Nut**
	Torque: Ft-lb (Nm)	**Torque: Ft-lb (Nm)**
¼”-20 UNC	7 Ft-lb (9 Nm)	8 Ft-lb (11 Nm)
¼”-28 UNF	8 Ft-lb (11 Nm)	9 Ft-lb (12 Nm)
5/16”-18 UNC	15 Ft-lb (20 Nm)	16 Ft-lb (22 Nm)
5/16”-24 UNF	16 Ft-lb (22 Nm)	17 Ft-lb (23 Nm)
3/8”-16 UNC	26 Ft-lb (35 Nm)	28 Ft-lb (38 Nm)
3/8”- 24 UNF	30 Ft-lb (41 Nm)	32 Ft-lb (43 Nm)
7/16”-14 UNC	42 Ft-lb (57 Nm)	45 Ft-lb (61 Nm)
7/16”-20 UNF	47 Ft-lb (64 Nm)	50 Ft-lb (68 Nm)
½”-13 UNC	64 Ft-lb (87 Nm)	68 Ft-lb (92 Nm)

½”-20 UNF	72 Ft-lb (98 Nm)	77 Ft-lb (104 Nm)
9/16”-12 UNC	92 Ft-lb (125 Nm)	98 Ft-lb (133 Nm)
9/16”-18 UNF	103 Ft-lb (140 Nm)	110 Ft-lb (149 Nm)
5/8”-11 UNC	128 Ft-lb (173 Nm)	136 Ft-lb (184 Nm)
5/8”-18 UNF	145 Ft-lb (197 Nm)	154 Ft-lb (209 Nm)
¾”-10 UNC	226 Ft-lb (306 Nm)	241 Ft-lb (327 Nm)
¾”-16 UNF	253 Ft-lb (343 Nm)	269 Ft-lb (365 Nm)
7/8”-9 UNC	365 Ft-lb (495 Nm)	388 Ft-lb (526 Nm)
7/8”-14 UNF	402 Ft-lb (545 Nm)	427 Ft-lb (579 Nm)

It is recommended that all plated and unplated threads be coated with oil before installation.

** Use these torque values if either the bolt or the nut is lubricated or plated (Zinc-Phosphate coated, Cadmium-plated, or Waxed.

Unlubricated, Unplated	**Lubricated* or Plated** Threads**	**Lubricated* or Plated** Threads**
Flanged	**Flanged**	**Flanged**
Grade 8 or 8.2 Bolt Grade G	**Grade 5 Bolt, Grade 5 or Grade B**	**Grade 8 or 8.2 Bolt Grade**

	Nut	Nut	G Nut
	Torque: Ft-lb (Nm)	**Torque: Ft-lb (Nm)**	**Torque: Ft-lb (Nm)**
¼”-20 UNC	---	6 Ft-lb (8 Nm)	10 Ft-lb (14 Nm)
¼”-28 UNF	---	7 Ft-lb (9 Nm)	12 Ft-lb (16 Nm)
5/16”-18 UNC	22 Ft-lb (30 Nm)	13 Ft-lb (18 Nm)	21 Ft-lb (28 Nm)
5/16”-24 UNF	---	14 Ft-lb (19 Nm)	23 Ft-lb (31 Nm)
3/8”-16 UNC	40 Ft-lb (54 Nm)	23 Ft-lb (31 Nm)	37 Ft-lb (50 Nm)
3/8”- 24 UNF	---	25 Ft-lb (34 Nm)	42 Ft-lb (57 Nm)
7/16”-14 UNC	65 Ft-lb (88 Nm)	35 Ft-lb (47 Nm)	60 Ft-lb (81 Nm)
7/16”-20 UNF	---	40 Ft-lb (54 Nm)	66 Ft-lb (89 Nm)
½”-13 UNC	95 Ft-lb (129 Nm)	55 Ft-lb (75 Nm)	91 Ft-lb (123 Nm)
½”-20 UNF	---	65 Ft-lb (88 Nm)	102 Ft-lb (138 Nm)
9/16”-12 UNC	140 Ft-lb (190 Nm)	80 Ft-lb (108 Nm)	130 Ft-lb (176 Nm)

9/16"-18 UNF	---	90 Ft-lb (122 Nm)	146 Ft-lb (198 Nm)
5/8"-11 UNC	190 Ft-lb (258 Nm)	110 Ft-lb (149 Nm)	180 Ft-lb (244 Nm)
5/8"-18 UNF	---	130 Ft-lb (176 Nm)	204 Ft-lb (277 Nm)
¾"-10 UNC	340 Ft-lb (461 Nm)	200 Ft-lb (271 Nm)	320 Ft-lb (434 Nm)
¾"-16 UNF	---	220 Ft-lb (298 Nm)	357 Ft-lb (484 Nm)
7/8"-9 UNC	---	320 Ft-lb (434 Nm)	515 Ft-lb (698 Nm)
7/8"-14 UNF	---	350 Ft-lb (475 Nm)	568 Ft-lb (770 Nm)

The threads may have residual oil, but will be dry to the touch.

** Both male and female threads (bolt and nut) must be unlubricated and unplated; if either is plated or lubricated, use the first two tables above.

Note that that bolts smaller than 1/4" are measured in machine screw sizes, running from Zero (0) to 14, with 0 being the smallest and 14 the largest. They run O to 6, then 8, 10, 12 and 14. Each has a standard UNC/UNF thread specification. These are the most commonly encountered sizes. There are, however, a number of specialty machine screws either dating from before there were standardized sizes, or to meet special requirements. These will seldom be encountered in automotive work.

Be aware that the vast majority of the nuts and bolts that are used on the MGB are Grade 5 machine bolts. Unmarked bolts that are readily available from home improvement stores are simply not up to the strains of automotive use. It should be noted that all bolts used for machinery purposes on your MGB should always be genuine machine bolts. Original bolts on the MGB and the earlier MGs carried markings on the bolt head, the most common of which were “R”, “S”, or “T”. R is a basic grade of bolt. S is high tensile, used for most engine and suspension applications, 120 ksi. T is extra high tensile, used in particularly critical areas, 150 ksi. The strength and type of steel used in a machine bolt is indicated by a raised mark on the head of the bolt. The type of mark depends on the standard to which the machine bolt was manufactured. Most often, bolts that are used in machinery are made according to SAE standard J429, while bolts used in structures are made according to various ASTM standards. Bolts made to SAE standard J429 Grade 5 will have small linear markings radiating outwards at the twelve o’clock, four o’clock, and eight o’clock positions on their heads, and have a tensile strength of 105 to 120 ksi. These are made of medium carbon steel that has been both quenched and tempered. These are the machine bolts that are used as Original Equipment for machinery purposes in the MGB. Higher grade machine bolts require more torque in order to achieve the necessary amount of stretch in order to make them secure, and thus should not be used for machinery purposes in any mechanical part of the MGB, otherwise thread damage is likely to result. Machine bolts made to SAE standard J429 Grade 8 will have small linear markings radiating outwards at the twelve o’clock, two o’clock, four o’clock, six o’clock, eight o’clock, and ten o’clock positions on their heads. These are made of medium carbon alloy steel that has been both quenched and tempered and have a tensile strength of 150 ksi. Machine bolts made to SAE standard J429 Grade 8.2 will have small linear markings radiating outwards at the nine, o’clock, ten o’clock, eleven o’clock, one o’clock, two o’clock, and three o’clock positions on their heads. These are made of low carbon martensite steel that has been both quenched and tempered and have a tensile strength of 150 ksi. Bolts lacking markings on their heads should never be used for machinery purposes as they are made of either low or medium carbon steel. Their tensile strength, depending upon their size, is only 60 to 74 ksi. Often, in addition to those described above, you may find extra marks on a bolt head. Usually these marks indicate the manufacturer of the bolt. In any case, it would be a worthwhile investment to purchase a set of sizing dies so that you can check the threads and sizes of the bolts. If a bolt seems to be tight in the die, it may jam partway into its hole, or even tear up the threads. If a bolt seems to be loose in the die, a torque reading will be irrelevant, and loading may produce distortion of the threads that it is screwed into. An undersize bolt is useless for machine purposes.

Locking Compounds

When it comes down to having to use a locking product, I often prefer to employ Loctite products, as they are of consistent quality and ultra-reliable. Genuine Loctite products are not that expensive. Used correctly, a small 6 ml squeeze bottle will do many fixings, plus they have an indefinite shelf life since they only cure when deprived of air. Although Loctite makes a large number of chemical compounds for automotive use, I will confine myself to those that I have personal experience with.

Loctite 222 is a low-strength, purple acrylic liquid that dries into a vinyl-like material that is best used when there is a great deal of thread engagement or where regular and / or easy removal is needed.

Loctite 222MS is a low-strength, purple acrylic liquid that dries into a vinyl-like material that is best used on small threads of ¼" (6.5mm) and smaller threads are involved. It is what I prefer for use on fine components such as carburetors. Like Loctite 222, it is for where regular and / or easy removal is needed.

Loctite 243 is the successor to old Loctite 242. It is a medium-strength, oil-resistant blue liquid that is especially well suited for fasteners between ¼" and ¾" diameter (6 to 20mm). It is used when a semi-permanent bond is needed, but can be undone with standard hand tools. Be aware that it dries to a cement-like consistency and frequently damages threads when they are unscrewed.

Loctite 270 dries to a green colored, very-high-strength adhesive for use when near-permanent fixing or high-yield strength is required, such as in the case of flywheel and harmonic balancer pulley bolts.

Loctite 271 is a red acrylic that dries into a particularly tough vinyl-like material that will not powderize under vibration. It is intended for the specific purpose of permanent installation. However, it can be removed with heat and hand tools. It is formulated to perform best with fasteners from 3/8" to 1" (9.5mm to 25mm).

Loctite 272 is a red acrylic that dries into a particularly tough vinyl-like material that will not powderize under vibration. It is formulated to withstand temperatures up to 450°F (232.2° Celsius). It performs best with fasteners up to 1 ½" (36mm). Be aware that in rare cases localized heating to 250° Fahrenheit (121.1° Celsius) will be needed for disassembly.

Loctite 290 is a medium-strength green acrylic that dries into a vinyl-like material and is ideal for fasteners from #2 to ½", although it can be used up to ¾". It is a low viscosity wicking grade, liquid that wicks along the threads of pre-assembled fasteners to secure them in place. Since it is applied after assembly, preventive maintenance procedures are simplified. Be aware that in rare cases localized heating to 250° Fahrenheit (121.1° Celsius) will be needed for disassembly.

Loctite 545 is a low-strength, purple acrylic liquid that dries into a vinyl-like material and is designed for locking and sealing high-pressure pneumatic and hydraulic systems. If your brake system has a seemingly incurable leak, this stuff just might end your misery when all other compounds fail.

Loctite 567 is a white acrylic paste that forms into a low strength threadlocking adhesive. It is notable because it will withstand temperatures of up to 400° Fahrenheit (204.4° Celsius). It is designed for the locking and sealing of metal tapered threads and fittings. The high lubricating properties of this compound prevent galling on stainless steel, aluminum and all other metal pipe threads and fittings. I strongly recommend it for use on fuel system connections as it is highly resistant to oil and solvents.

Loctite 609 is high-strength green retaining compound that is intended for use in the mounting of close fitting bearings, especially in slip-fit applications. It augments the strength of press-fit assemblies or slip-fit assemblies up to a 0.005" diametrical clearance. It has a shear strength of up to 3,000 psi.

Loctite 680 is a high-strength green acrylic adhesive used in slip-fit, press-fit, and shrink-fit applications, particularly when low viscosity is desired. It has a shear strength of up to 3,500 psi. I have found that it is particularly useful for sealing the bottom joint of the dipstick tube.

Whichever Loctite product you select for a particular application, there are two important factors that you need to be aware of. First, make sure that all of the threads are clean, dry, and free from contaminants, especially oil and / or grease. Second, it is never a case of "the more, the better". You only need to use a very small amount. Too much Loctite can get under and in-between things, causing either a great deal of swearing when attempting removal, or powdering in service as a result of vibration, allowing them to loosen. The results can be disastrous, such as when used on internal engine components. When they become loose, they will lose the clamping-load that was developed by torquing them down, break, and can then proceed to quickly destroy an engine or a transmission. Use one small drop a third of the way up the engaged area, or two drops spaced one as previously described and another a few threads further up when a longer threaded area is used. Never simply put it on the end of the bolt. If you do, it will have been spread too thin over too large an area to be properly effective.

Converting from Positive Ground (Earth) to Negative Ground (Earth)

While one does not normally think of the generator (dynamo) or the alternator as being an item that influences a car's performance, it must be remembered that if the ignition system does not have a sufficient supply of electricity, then the engine cannot perform up to its potential. Today, in the interest of safety, it is common for owners to install more powerful headlights and brighter bulbs into their cars, and these items can easily consume more electricity than the Original Equipment items that they replace, so much so that relays are often installed in order to prevent the greater amount of current being drawn from overloading and burning out the driver's switches. In such cars that are equipped with the Lucas C40/1 generator (dynamo) of the MGB Mark I, or earlier model versions of the Lucas alternator, this circumstance can tax the system to the point that high-energy ignition systems are receiving a supply of electricity that is borderline at best, leading to otherwise inexplicable poor system performance and contributing to a shortened component lifespan.

Probably the main reason for converting from a generator (dynamo) to an alternator is to get a higher charging capacity as the Original Equipment Lucas C40/1 generator (dynamo) is limited to a rather puny output of 22 Amps. Although this is usually adequate for most standard and particularly 'classic' use, when stuck in traffic with the headlights, the heater fan, the wipers, the radio, etc. consuming electrical power, its charging rate will almost certainly be inadequate. This means that you will be discharging the battery, and this can rapidly discharge the battery to the point where it will no longer be able to start the car. In addition, bad connections will inhibit the flow of the electrical current, and will contribute to a drop in both system and the charging voltage. It is often this which people experience, as much as it is inadequate power output.

Having been originally equipped with a generator (dynamo), the 18G and 18GA (3 main bearing) and 18GB (5 main bearing) engine blocks had no provisions for the mounting of an alternator. Beginning in 1968, the use of Lucas generators (dynamos) was discontinued in favor of the more efficient Lucas alternators. These later 18GD through 18V engine blocks had provisions for mounting either an alternator or a generator.

The first thing to be aware of concerning the process of converting from a generator (dynamo) to an alternator is that unless it has already been done, you will have to convert the electrical system of a MKI MGB from positive (+) ground (earth) to negative (-) ground (earth). Positive (+) ground (earth) alternators will probably be very difficult if not impossible to find, a negative (-) ground (earth) system will be tricky to convert to a positive (+) ground (earth) system, and the availability of the variety of used, rebuilt and new negative (-) ground (earth) alternators is almost infinite. Also, it might be safer to take things one step at a time and do the polarity conversion first, then check everything to be sure that it works properly, and only then perform the alternator conversion. Getting the polarity wrong with an alternator connected will probably destroy the alternator.

Be aware that this process only covers the use of a Lucas alternator, as there are too many variations in Bosch and GM Delco alternator connections, although the alternators themselves are quite suitable for use.

In converting an automotive electrical system from positive (+) ground (earth) to negative (-) ground (earth), the first consideration must always be the batteries. Before doing anything else, make sure that the battery ground (earth) connection is the first thing that you remove. It is also the last thing to reconnect at the end. Because batteries have different-sized positive (+) and negative (-) posts so that, in theory at least, you cannot connect them the wrong way round, the connectors will have to be either swapped over or replaced. The Original Equipment 'helmet' type that completely cover the post are secured with a small screw that goes into the post expand and become loose with age and repeated removal and replacement, resulting in poor connections. Since you are changing the polarity, “originality” is not an issue, so if you have not already, replace these Original Equipment 'helmet' type connectors with the modern bolt-on type which give a much better connection. The other anachronistic thing with the helmet type connectors is that they are usually molded onto their leads, so these have to be cut off and replaced with the clamp on type, which usually have two large screws which secure the cable. While this results in shortening each cable by about an inch, that should not become a problem. If it does, then you will have to replace the cable(s). If you already have clamp-up type connectors, remove these from the hot and ground (earth) connections and swap them over. Unless you have already replaced the twin-6 Volt batteries with a single-12 Volt Group 26 battery, you will also have to deal with the interconnecting cable in the same way. Unless it can be physically removed from the car and reversed, you will have to cut off and replace the molded-on helmet-type connectors, or remove and swap over the clamp-up type.

All MGBs after the 1964 model year had electronic tachometers, and this also has to be converted. You have to get into the case, find the supply and ground (earth) wires from the case to the circuit board, and reverse the connections. However, be aware that some cars have the circuit board screwed to the case and pick up the ground (earth) connection this way. In this case, you will have to isolate the circuit board from the case, relocate the original 12 Volt supply wire from the terminal on the case to the body of the case, and provide a new wire from the ground (earth) connection on the circuit board to the 12 Volt supply terminal on the case. In all cases you have to reverse the direction of the current pulse through the pickup and this also varies. On early models, the pickup is external and a continuous white wire comes out of the wiring loom, through the pickup twice (i.e. one turn) then back into the wiring loom. With these, carefully note the route of the wire, remove it, and then reverse the direction of the wire through the pickup, but keeping everything else the same (e.g., the position of the loop). On later tachometers the pickup is external with two flying leads with male and female bullet connectors to match up with their opposite numbers on wires from the wiring loom. The tachometer will have to be changed internally to operate in a negative ground environment as follows: In order to reverse the power wire and earth wire inside of the tachometer, it is necessary to remove the chrome ring, the glass face, and the glare shroud. The chrome ring is usually removed with great difficulty by prying the tabs with a small screwdriver, and then by rotating until the tabs can fit through the slots in the case. Remove the two screws on the back of the unit that hold the internals to the case (not the two whose heads fit in holes in the case). Taking care to not bend the needle, allow the internals to drop carefully into your hand. The spade terminal is the power connection. Just next to this is the earth connection. A resistor is soldered to one of these connections, and a green wire to the other. Unsolder these ends of the green wire and the resistor from their current positions. Re-solder the green wire to where the resistor was connected, and then re-solder the resistor to where the green wire was connected. While you are doing wiring changes inside the tachometer, it would be neater to reverse either the ignition wire as it passes through the pickup, or the output of the pickup to the circuit board, whichever you find to be easier. An alternative is to reverse the bullet connectors where the wiring loom joins the tachometer. Since it is the car that has changed polarity, it might make sense to change the bullets on the wiring loom wires. However, since you have to change the polarity of the power supply at the tachometer, it might make sense to change the bullets on the tachometer wires as well. The only advantage of doing it the other way is that should you have to replace the tachometer in the future, the pickup part of the polarity change will not have to be done again.

If you have the heater fan motor with black and green/brown wires, these may have to be reversed at the connectors by the motor. If in doubt, try them both ways (you can't do any harm), and if one way blows more air than the other, then the fan is rotating in the right direction and that is the correct way to connect it.

If the vehicle is equipped with an Ammeter, reverse the leads to the ammeter.

Originally the fuel pumps had capacitor protection in order to reduce the burning of the points, and these are not polarity conscious. However, if you have installed one of the later fuel pumps with an 'X' in the part number, e.g. AZX137, then these have diode protection for improved points protection and will need to be modified. Open the electrical end-cap and locate the black cube with two wires, one black and one red, both coming out of the same end of the cube. These wires must be removed from the pump and connected the other way round. There is no immediate risk of damage powering-up without having done this change, but it will largely disable the protection of the points, resulting in the points burning out faster than usual. Note that if you have fitted one of the 'pointless' electronic pumps, then you will probably have to replace the pump as converting the electronics in these is much more involved, and changing the battery polarity without doing anything about the pump will result in either at best the pump not working, or at worst in the destruction of the electronics, as well as possibly burning the wiring loom.

It is frequently stated that when changing the car's polarity, you should also reverse the coil connections in order to keep the polarity of both it and the High Tension spark the same. I usually mention it when the subject comes up, but I am not totally convinced that it is necessary, and it may even be undesirable. One reason sometimes given is that spark plugs need to have a given polarity, negative (-) at the spark plug relative to the body, as the other way results in more energy needed in order to jump the gap (due to the different temperatures of the electrode and the body). I've seen figures of anything from 15% more to 40% more. Some manufacturers apparently fit different spark plugs for positive (+) High Tension than to negative(-) High Tension, but this is more about saving money in terms of the amount of platinum on each electrode than it is about spark plug performance. What I do wonder about is the internal connections of the ignition coil. The grounded (earthed) end of the High Tension winding, which it needs to make a complete circuit with the spark plugs grounded (earthed) in the cylinder head, is not connected to the casing of the ignition coil and hence to the body of the car as many assume, but is internally connected to one of the Low Tension spades. On a diagram of a positive (+) ground (earth) coil from the Leyland Workshop Manual you can see that it is connected to the CB terminal, so when the High Tension fires it either has to ground (earth) through the condenser (the contact breaker points being open at the time), or it grounds (earths) through the Low Tension winding, ignition switch and battery. I rather suspect that it takes the latter route, as it is of lower Impedance and striking the condenser with High Tension voltage may destroy it, and one source claims that having the two windings in series increases High Tension voltage. I know from personal experience that 200 to 300 Volts is momentarily developed at the coil's Low Tension terminal as the contact breaker points open. If you reverse the Low Tension winding in order to restore the polarity of the High Tension current, the High Tension winding will then bypass the Low Tension winding on its way to ground (earth), which may have an impact on spark energy. So, rather than reverse the windings if the ignition coil, **or** keep them the same, it may be better to fit a later negative (-) ground (earth) ignition coil with '+' and '-' markings instead of 'CB' and 'SW' as originally. However, I suspect that none of these factors are likely to be an issue on our cars except under very marginal conditions of ignition coil voltage, state of tune and atmospheric conditions, it is more likely to be a hangover from early spark ignition systems where High Tension voltage and other factors were marginal anyway.

That completes the polarity conversion, unless you have any additional electronic devices, which will be aftermarket items, and so their conversion will have to be up to you. The only possible other thing might be that the wipers might now park in a slightly different place. If it bugs you, then simply relocate the arms on the spindles. Start the car, check to be sure that the tachometer is working, and measure the voltage on the brown wire at 3,000 RPM with minimal load. With a generator (dynamo) you should see in the order of 14.3 Volts to 15.5 Volts depending on ambient temperature (with the lower voltages at higher temperatures), and with an alternator you should see 14.3 Volts to 14.7 Volts.

Converting From A Generator (Dynamo) to an Alternator

Now, before starting any of the following work, first disconnect the battery ground (earth) strap, and only replace it as the final step.

The dynamo should have brown / green (field) and brown / yellow (output) wires. At a pinch the old brown / green wire could be used for the alternator output, but this will start to decrease voltage at higher currents. If you use this, it must be removed from the F terminal of the control box and connected to the three brown wires instead. However, it is better to run a new brown wire of suitable gauge for the current to be carried from the output terminal of the alternator to the battery cable stud on the solenoid of the electric starter, taping back both ends of the old brown / green wire. The brown/yellow wire connects to the smaller indicator spade on the new alternator. The two brown / yellow wires at the control box must be removed from it but connected together, insulated and taped back. This circuit supplies the priming voltage to start the alternator charging, and indicates charge failure. That should leave three brown wires and one black wire at the control box. If you are leaving the dynamo control box in-situ, the brown wires and the black wire can be left connected. If you choose to remove the control box, the brown wires at the very minimum must be connected together securely (this connection carries the load of all the cars electrics, with the exception of the starter), insulate them, and tape them back as before. Similarly, insulate and tape back the black ground (earth) wire. However, the higher alternator current will be flowing through these older brown wires and their connections and it will be beneficial to move them to the solenoid, which will remove one of the 'choke points' for current, as well as the spade connectors which are not good with high currents. To do this, remove the three brown wires from the control box, and identify which one goes back to the solenoid. This wire is no longer required, so its connectors can be insulated and the tails taped back out of the way to the harness. With the two remaining brown wires (one to the fusebox and the other to the ignition switch, lighting switch etc.), replace the spades with bolt-through connectors and put them on the battery cable stud of the solenoid.

If converting the polarity at the same time, leave the alternator unplugged when connecting the batteries the new way round for the first time, Should you get it wrong and the alternator is connected, you will then blow its diodes and burn its wiring. Confirm the that polarity is correct before continuing by connecting a voltmeter between a brown in the alternator plug with the meter set on positive(+) and an engine ground (earth) with the meter set on negative (-).

After confirming that the polarity is correct, connect an analogue voltmeter on its 12 Volt scale in place of the battery ground (earth) strap. There should no voltage registered. If there is, it will probably be a full 12 Volts, and means that some circuit on the car is switched on (Courtesy lights? Boot light?), which should be found and switched off before proceeding. When no voltage is shown, plug in the alternator. You may now see a few volts registered, which will be the normal microscopic leakage current of the diodes and can be ignored. If a full 12 Volts is shown, then the diodes in the alternator are faulty. If the reading is correct, replace the battery ground (earth) strap. With the ignition turned off there should be no glow from the ignition light. A glow now indicates either faulty diodes inside the alternator or a faulty voltage regulator, or incorrect connections to it. With engine turned off, you should see 12.5 to 13.0 volts at the battery terminals. You should also see the same reading on the outside of the battery clamps if the connections are clean and tight.

With the ignition switched on, the alternator warning light (ignition warning light) should glow. If it does not glow, then remove the plug from the alternator and connect a ground (earth) to the brown/yellow terminal. If the ignition light glows, then the alternator is faulty. If it does not glow, then either the bulb has blown or the circuit is broken back towards the ignition light, possibly where the brown/yellow wires are joined where the control box was, There should be 12 Volts on the white wire at the bulb holder and a ground (earth) on the brown/yellow wire to light the bulb.

With the alternator warning light (ignition warning light) glowing, start the car, and with the engine speed above 1,000 RPM, the ignition light should go out. If the light remains on, then the alternator itself is faulty. Only if the engine speed drops below about 600 RPM should the light come back on, stay off until about 1,000 RPM, then go out again as before. Early cars had an idle speed of 500 RPM and if the light comes on at idle, particularly with lots of load switched on, then you would be wise to increase the idle speed to, say, 700 RPM in order to keep the light out at all times. When the light is on the alternator isn't charging and the battery is discharging, which largely negates the effort of converting!

With the engine speed at about 1,000 RPM, and all loads switched off and the ignition light off, measure the voltage between the brown wire circuit either at the front side of the fusebox or off of the battery terminal clamps and the ground (earth). You should see about 13.5 Volts to 14.5 Volts. If you see voltage readings above 14.5 volts or below 13.5 volts there is a problem with the alternator which needs to be examined. Next, turn on the headlights, the brake lights, the heater fan etc. The voltage will probably drop, possibly to less than 13 Volts with one of the smaller output Lucas alternators. Increase the engine speed to about 3,500 RPM and the voltage should rise to above 13 Volts again, indicating the battery is still being charged even with everything switched on. If the voltage doesn't rise above 12.8 Volts, then check the voltage at the alternator output terminal(s), and if similarly low, this indicates that the alternator has a low output current fault. However, note that the smaller Lucas alternators will probably not be able to supply anything above the standard factory loads at best. If the voltage is closer to 14 Volts at the alternator, then there is a bad connection somewhere between the alternator and the brown wire at the fusebox, so check the voltage on each brown wire and the battery cable at the solenoid.

Lucas Alternators

One widespread mistaken belief is that fitting a higher-output alternator will automatically force more current through the wiring. The truth is that the maximum amount of current that will flow is dependant up on the amount of electrical load being drawn, not the maximum capacity of the alternator fitted. If you only have 40 Amps of load being drawn, then only 40 Amps of current will flow, even with a 100 Amp alternator fitted.

An alternator produces an Alternating Current (AC) that is converted to Direct Current (DC) prior to being connected to the electrical system of the vehicle. In this respect the generator (dynamo) and the alternator are similar, since the current generated in the armature windings of the generator (dynamo) is also an Alternating Current (AC) that has to be converted into Direct Current (DC) before it can be used by the electrical system of the vehicle. In the case of the generator (dynamo), the Alternating Current (AC) is rectified by means of a commutator and a brush-gear. On the other hand, the Alternating Current (AC) of an alternator is rectified by semiconductor devices that allow current to flow in one direction only, thus supplying a unidirectional (Direct) current (DC) to the electrical system of the vehicle. The electrical output of an alternator is controlled by a voltage regulator that is completely electronic, having no vibrating contacts. The use of printed circuits and semiconductor devices allows this type of regulator to be both more reliable and more stable than the conventional type of mechanical regulator used with generators (dynamos). No cut-out device is required with this type of control since a semiconductor device prevents reverse currents from flowing through the electrical system.

Before you decide you need to remove an alternator and replace it, make sure that you have checked the tension of the V-belt. If the belt is worn or loose, the alternator will not function properly. A bad V- belt is easy to replace and will not cost much money. Some people have a difficult time installing the V-belt. The trick is to first get the belt over both the crankshaft and alternator pulleys, and then over the pulley for the coolant pump. When the belt is in contact with each pulley, the installation technique becomes obvious. Oddly, this technique is not mentioned in any manual that I have read. Another trick is to remove the adjuster bolt from the slider bracket of the alternator altogether in order to give more movement of the alternator.

As a rule, a three-phase alternator can operate with only one of the stator windings operational, although it is only one-third as efficient. To test whether your car has an issue with either one or more of its stator windings, you will need to perform a Load Test by using a Voltmeter in order to check the voltage. Since the battery produces Direct Current (DC) power, set the Voltmeter to "DC" rather than to "AC". Connect its red lead (wire) to the positive (+) terminal and the black lead (wire) to the negative (-) terminal. With no accessories on, start the car and increase the engine speed to around 1,000 RPM. The voltage should register at around 14 Volts. Anything less than 12 Volts may indicate a problem. Next, turn on the headlights, radio, and anything else that draws electrical power. Increase the engine speed and recheck the Voltmeter. Again, the voltage should register at around 14 Volts. If you have a failing alternator, the voltage will be well below 14 Volts. If so, then it is time to rebuild or replace your alternator.

However, you do have options. If you happen to own an alternator that has a repair kit available, you can really save some money. Again, you need the proper tools and a little know-how, but if you know what you are doing, you can rebuild an alternator for a fraction of the cost of even a remanufactured unit.

So, what does that alternator warning light (ignition warning light) that is labeled “ALT” on the dashboard indicate when it comes on? Quite simply, it means that the alternator output voltage is either higher or lower than the battery voltage. The warning light circuit also supplies a small current, via the "IND" terminal, to the field coil when the ignition is switched "on". That in turn sets up an electromagnetic field, or "flux", within the field coil. Then, by rotating that electromagnetic field inside the stator windings, the alternator generates electricity. Once the car starts, the field becomes "self excited" and no longer needs current from the warning light circuit. If the alternator produces voltage that is less than that of the battery (or none at all), current continues to flow from the battery through the bulb and it continues to glow. If, after the car starts, the output voltage of the alternator equals the voltage at the battery terminals, then no current can flow because both terminals of the bulb have the same "potential". If the light becomes dimmer as you increase the engine speed, then you have a problem with the alternator. If it gets brighter, then the battery is most likely bad and the problem can be cured by simply replacing the battery.

If your battery goes flat and the alternator warning light (ignition warning light) never comes on, you can check the operation of both the bulb and its wiring. After recharging the battery, and with the engine off, disconnect the plug that attaches the brown/yellow wire to the back of the alternator and turn the ignition switch to the "On" position. Using a jumper wire, ground the brown/yellow wire. The bulb should light. If it doesn't, find out why. The bulb is either burned out or there is a bad connection.

When your alternator warning light (ignition warning light) starts glowing or the dashboard gauge shows a no-charge condition, see if charging current is reaching the battery. With the engine running at high idle (1500 rpm), measure the voltage at the battery terminals using a Volt/Ohm meter (VOM); a digital model is more accurate, but for this test an analog needle will work fine. On high idle, the voltage at the terminals should be between 13.6 and 14.4 volts. Outside that range indicates either bad / dirty connections or a defective alternator.

Just exactly what does all of that mean? In order to get a good idea, it is first necessary to understand how an alternator works. You do not need a degree in electrical engineering, just a basic understanding of the general principles that are involved.

We will start our tour of the alternator at the rotor. The rotor is driven by the alternator pulley wheel, rotating as the engine runs, hence the name “rotor”. The rotor consists of a coil of wire that is called a “field winding” which is wound around an iron core. The current through this coil of wire is called “field” current , so called because it produces an electromagnetic around the iron core. It is the strength of the field current that determines the strength of the electromagnetic field. This field current is Direct Current (DC). In other words, the current flows in one direction only, and is supplied to the wire coil (field winding) by means of a set of two stationary carbon brushes that ride upon two rotating slip rings. The two slip rings are located on one end of the rotor assembly. Each end of the rotor field winding is attached to a slip ring, thereby allowing current to flow through the rotor field winding. The electromagnetic field saturates the iron finger poles. One finger pole becomes a north pole and its opposite becomes a south pole. The rotor spins, creating an alternating electromagnetic field, North, South, North, South, etc.

Surrounding the rotor is another set of coil windings, three in number, called the stator. The stator is a stationary field winding that is attached to the inside of the shell of the alternator, and hence does not turn. As the rotor turns within the field windings of the stator, the electromagnetic field of the rotor sweeps through the field windings of the stator, producing an electrical current within its field windings. Because of the rotation of the rotor, an Alternating Current (AC) is produced. As, for example, the north pole of the electromagnetic field approaches one of the stator windings, there is little coupling taking place, and a weak current is produced. As the rotation continues, the electromagnetic field moves to the center of the winding, where maximum coupling takes place, and the induced current is at its peak. As the rotation continues to the point that the electromagnetic field is leaving the stator winding, the induced current is again small. When the south pole is approaching the winding, it produces a weak current in the opposite direction. A laminated iron frame concentrates the electromagnetic field. The three stator coil windings are spaced 120° apart inside of the alternator, producing three separate sets, or "phases", of output voltages, each being spaced 120° apart.

Alternating Current (AC) voltage is of little use in a Direct Current (DC) system, so it has to be converted (rectified) into Direct Current (DC) before it can be useful. This rectification to Direct Current (DC) takes place in the output diodes of the rectifier bridge. Diodes have the property of allowing current to flow in only one direction, while blocking current flow in the other direction. The rectifier bridge consist of six diodes, one pair for each winding. One of the pair is for the negative half-cycle and the other of the pair is for the positive half-cycle. The output diodes are mounted in a heat sink in order to dissipate the heat that is produced by their function.

The rectified output of an alternator is not a pure, steady Direct Current (DC), but a pulsating Direct Current (DC). Because there are three windings, each with a positive (+) half and a negative (-) half, by the time that the voltage is passed through the six output diodes of the rectifier bridge, there are six pulsations for each rotation of the rotor. This is close enough to Direct Current (DC) for most automotive components. Critical components, such as radios, have their own internal filtering circuits to further smooth out the waveform to a purer Direct Current (DC).

The regulator of the alternator has two inputs and one output. The inputs are the field current supply and the control voltage input, and the output is the field current to the rotor. The regulator does just what its name implies: it regulates the output of the alternator to the proper voltage and current by using the voltage input from the battery in order to control the amount of field current input that is allowed to pass through to the rotor windings. If the battery voltage decreases, the regulator then senses this, by means of its sensor connection to the battery, and allows more of the field current input to reach the rotor, which increases the strength of the electromagnetic field, which in turn increases the voltage output of the alternator to the battery. Conversely, if the battery voltage increases, then less field current goes through the rotor windings, and production of the output voltage is reduced.

Field current supply is provided from two different sources – either from the alternator itself via its diode trio and its regulator, or from the battery via the alternator warning light (ignition warning light). The amount of field current that is allowed to pass through the regulator to the field coil of the rotor is controlled by the voltage feedback that is supplied from the battery via the sensor connection. When you turn the ignition key on, the field current is sourced from the battery, through the ignition switch, and on through the ignition warning light. You need 12 Volts through the alternator warning light (ignition warning light) in order to power the alternator. This energizes the stator coils, in order to provide the electromagnetic field that the rotor turns within in order to provide electricity. After the engine starts and the alternator is at operating speed, the output of the diode trio is routed to the regulator, and then is fed back to serve as the source of field current. The diode trio consists, as the name suggests, of three diodes, one per phase, which provides field current to the regulator of the alternator. At this time, the alternator becomes self-sustaining, and the battery is no longer needed in order to power the electrical system. However, be warned that this is theoretical only - in actual practice, the voltage surge that results from disconnecting the battery can seriously damage the circuitry of the regulator. All alternator manufacturers strongly advise NOT doing this! This test will not prove the functionality of the alternator, as the engine may still run with a weak alternator output.

This brings us back full circle to the starting point - the alternator warning light (ignition warning light). There is a path to ground (earth) from the field current supply input to the regulator. As a result, when the key is turned on, current flows through the alternator warning light (ignition warning light), through the resisters, transistors, and field coil, and then to ground (earth), causing the alternator warning light (ignition warning light) to illuminate. Once the alternator is at full output, voltage from the diode trio equals the battery voltage. At this point, with 12 Volts on both sides, the alternator warning light (ignition warning light) does not illuminate.

The alternator warning light (ignition warning light) acts like a pair of balance scales in that it glows when there are different voltages on each side, and going out when there is the same voltage on each side. In theory, the alternator puts out 14.5 Volts to the brown / yellow wire to one side of the alternator warning light (ignition warning light) and the same voltage on the brown wire to the battery and the rest of the electrical system, including via the ignition switch and the white wire to the other side of the ignition warning light, hence the alternator warning light (ignition warning light) does not glow. However, if there is a bad connection somewhere on the white wire side, then its voltage will be lower than at the brown wire and at the brown / yellow at the alternator. If there is a problem inside of the alternator, then the reading for the brown / yellow wire could be either higher or lower than the brown and the white wires.

Whereas the alternator only starts charging when the engine speed is raised to approximately 900 RPM, once it has started charging, it will continue to do so down to about 600 RPM or so under all “normal” electrical loads. Once started and blipped over 900 RPM it should charge at its idle speed of 750 RPM or whatever without the alternator warning light (ignition warning light)glowing. The alternator warning light (ignition warning light) should switch on and off at those speeds, and not flicker or glow dimly at any speed. If it does either, then you have a problem. A dim alternator warning light (ignition warning light) when running a load (e.g., lights, brakes, etc.) is a sure sign of partial diode failure, which frequently causes a low charging voltage. If the problem occurs when not running under a load, then you will need to measure the voltage on each side of the alternator warning light (ignition warning light) while it is glowing dimly.

If the alternator warning light (ignition warning light) becomes dimmer as engine speed is increased, it is because the voltage output of alternator the is rising with the engine speed, producing more voltage on the alternator side of the alternator warning light (ignition warning light). The closer the output voltage gets to the battery voltage, the dimmer the alternator warning light (ignition warning light) will become. By the same way, if the light gets brighter with increasing engine speed, it is because as the alternator voltage increases, it is getting higher than the battery voltage. The higher the alternator voltage in relation to the battery voltage becomes, the greater the voltage difference across the alternator warning light (ignition warning light) becomes, and the brighter the alternator warning light (ignition warning light) becomes. If the alternator should fail, voltage from the diode trio would drop, and once again the alternator warning light (ignition warning light) will illuminate from the battery voltage being the greater of the two. If the alternator output is only a little low, then the alternator warning light (ignition warning light) will be dimly illuminated. If the alternator fails completely, and the output voltage goes to zero, the alternator warning light (ignition warning light) will be illuminated at full brilliance. Conversely, if the battery should fail, and the battery voltage drops, with the output voltage of the alternator on one side and the low battery voltage on the other, the alternator warning light (ignition warning light) will also illuminate.

In summary, it can be said that a field current through the rotor coils produces an electromagnetic field, which is coupled over to the stator coils, producing an Alternating Current (AC) voltage that is converted by the output diodes of the rectifier bridge into a pulsating Direct Current (DC) voltage which charges the battery.

Troubleshooting the Alternator

Barring physical damage (engine fire, broken casing, etc.), there are only a couple things in an alternator that can fail. Bearings can fail from over tightened belts or age -- usually the one at the pulley end goes first. A medium sized gear puller and an $8 SKF brand bearing (SKF Part # 6202-2Z or equivalent) can it put right. Brushes, the carbon type that conducts "flux" current to the field coil through the slip rings, will last for 75-100 thousand miles and are cheap -- about $4 for the pair -- to replace. Replace them if they're shorter that 1/4".

Electrically, the stator and field windings are pretty durable, again, barring physical damage. That leaves the diode pack or regulator for likely failure. The stator, field windings, and diode pack can all be checked with any reasonable digital VOM (Radio Shack has a pocket sized unit, Part # 22- 171, that sells for about $25, $18 on sale) or a self-powered 12 Volt test lamp.

Get the alternator off the car, over to the work bench, and remove the rear plastic cover. Support the unit with the electronic component end up. See those three soldered connections near the grounding bolt? Those stator leads need to be desoldered from the diode pack. You'll probably need a 100-125 watt soldering gun. In order to prevent diode damage, avoid using any more heat that necessary. Using needle-nose pliers, gently pull them away from the diode leads. Next, note how many and what color wires are coming out of the voltage regulator (that 1 1/2" square metal-clad box), and where they attach. Note also any other wires and where they go. There will be at least one more jumper running from the outer field brush to the "IND" terminal. MAKE A DRAWING!

Remove the spade connectors that attach the regulator (and brush box) to the diode pack. Remove the two screws that retain the brush box/regulator assembly; you need to do this to check the brushes and get to the slip rings of the field windings.

Using a digital VOM, in the "Ohm" setting, or the 12 Volt powered test lamp, begin with those unsoldered stator leads. Check resistance between all three leads. On the VOM you should get a reading of around 0.2 to 0.3 ohms, virtually no resistance. With the 12 Volt lamp it should glow at full brightness. Next measure between those same three wires and that grounding bolt that holds the diode pack in place. There should be no reading, or the bulb should not light. Any other reactions here and you've found a bad stator.

Now that the brush box has been removed, check the windings of the field coil. With the VOM still in the “Ohm” setting, probe the surfaces of the inner and outer slip rings. The copper should be nice and shiny. The reading should be around 4-7 Ohms, or the bulb should light. Next, test the insulation of the rotor windings. Get one probe tip down onto a clean metal area (scrape a little if necessary) of the rotor body, then probe both slip rings, one at a time. There should be no reading or the bulb shouldn't light. Any other reactions indicate a bad field winding, slip ring, or solder connection between the two.

All nine diodes of the rectifier need to be checked individually, but that can be done with it still in place. Remember good diodes only flow current in one direction. The diodes are tied together electrically onto three plates which serve as heat sinks and as attachment points for the alternator's plug for the wiring loom and the voltage regulator. Testing can become a little confusing to explain, so I'll break it down into simple steps. The two outer plates have the same polarity. The inner plate, closest to the brush box, is reversed. With that in mind, set your digital VOM to the "diode" setting (the “Ohm” setting is no good here), or power up the test lamp.

Attach the black negative (-) lead to one of the spade terminals of the outermost "IND" plate. Touch the red positive (+) lead to each of the three diode leads (where you desoldered the stator wires). You should get a meter reading at each terminal. Using a test lamp, the bulb should light.

Move the black lead onto the center plate and repeat the process with the red lead.

For the third, innermost plate, reverse the leads; i.e. red lead to the plate, black lead to the diode leads.

If you got a reading, or if the bulb lit in each of the tests, so far, so good. The diodes are all conducting in the forward direction.

Now, we need to reverse the diode test procedure, and the current flow.

Starting with the inner plate, connect the black lead to the plate, and then probe the diode leads with the red lead. There should be no reading or the bulb should not light.

Moving back out to the middle plate, once again swap the leads, putting the red lead on the plate's spade connector and probe the diode leads with the black lead.

Finally, back to the outer plate, attach red lead onto the spade terminal, black lead on the diode leads.

During the reverse part of the diode test there should be no reaction by either the meter or the bulb.

If you find one or two bad diodes in the pack you may be able to replace the offending parts (30 amp, 100 Volt), but the whole diode pack can be replaced for between $12 to $30; by the time you find the parts, try to cobble them in . . .why bother? If there are two or more diodes out, consider the regulator faulty also. Replace both units.

There are involved ways to check the regulator, but if the brushes, stator, field windings, and diode pack check out, chances are real good that replacing the regulator will get the alternator back in service. Replacement with a regulator with one that has the same wire count as that of the original is the quickest and the most desirable method.

Regulators

Over the years, as voltage regulators have been superseded, there have been two, three, and four-wire hook-ups inside of Lucas alternators. Some are battery sensing, some are machine (alternator) sensing, some can do both (dual sensing). This is where some of the confusion starts.

The four-wire (Black, Red, Yellow, White) regulator has a dual sensing, "fail-safe" feature; it can monitor both the battery and, if that connection fails, the machine (alternator). The two-wire (Black, Yellow) unit monitors available field "excitation" voltage, while the three-wire (Black, Yellow, White or Red) can monitor either the machine (alternator) or the battery, depending upon its connection at the diode pack. (Have your eyes started glazing over yet?) In all units the "field" connection is made via a small strap, or "link", between the case of the regulator and the center field brush. This has replaced a green wire found on earlier units.

Unless you know that your alternator is original to your car, it may have been replaced with an aftermarket rebuild using a non- Lucas regulator. Original Lucas regulators had a numerical code stamped into the case. Later units have a combination 3 letter with a 3 digit code. The I.D. will be something like “UCB 104”. If your regulator does not have those codes, then assume that it is a non-Lucas aftermarket regulator. As of mid-1994 Lucas was supplying two, three, and four-wire regulators.

There are a couple of other items that may or may not be encased inside of your alternator. Earlier units housed a 3 microfarad capacitor in order to suppress radio noise. It was attached between ground (earth) and the main output terminal. This was discontinued in 1979 and the part is no longer available. On newer units there is a surge protection diode that is wired between ground and the "IND" (indicator) conductor. The alternator will work without it, but it does serve a purpose. This device is a zener diode, installed to protect the regulator's main output transistor. It absorbs high transient voltage caused by a faulty connector or by removing a battery cable while the engine is running. This diode (Lucas Part # UZB 106) is still available from Lucas and, if you're missing it, you may want to get one. Allow me to drive this point into your head: Surge protective diode or no, *never, ever, remove either of the battery cables while the alternator is supplying current to the wiring loom*. In less than a heartbeat an overvoltage spike could destroy a perfectly good regulator, or worse.

The only difference between the three and four-wire regulators is the dual-sensing capability provided by the fourth wire; the white provides battery sensing while red provides machine sensing (alternator sensing). A three-wire will replace a four-wire unit by disregarding the fourth wire connection.

On three-wire units, the white wire will attach to the "S" terminal (where the smaller brown harness plug wire attaches); that makes it a battery sensing unit. A red wire from the regulator will attach to a positive (+) main output terminal of the diode pack (where the larger brown plug wire attaches); it's a machine (alternator) sensor. Black and yellow will go back to ground and the outer field brush respectively. A four-wire unit will replace a three-wire by simply connecting the additional red or white wire to the other appropriate terminal.

Two-wire units, with just a black and yellow lead, monitor the voltage values of the "IND" exciter circuit. No external sensor is required. Attach the black lead to ground and the yellow lead to the outer field brush terminal.

When reattaching the wires, don't forget that little jumper wire that runs from the outer field brush to the "IND" spade.

Replacing an Alternator

It is more important to purchase a remanufactured alternator rather than a rebuilt alternator. This is due to the fact that a company that rebuilds alternators replaces only the damaged or worn parts. A company that remanufactures alternators replaces all of the typical parts that tend to go bad, whether they immediately need replacement or not. Parts that are typically replaced are the bendix or gear drive clutch, the diode trio, the regulator (if installed), the bearings, and the solenoid (if so equipped). Such alternators meet or exceed original design parameters. They are, for all practical purposes, essentially new alternators. On the other hand, a rebuilder may merely clean it, slap in a new diode bridge, and spray it with paint. A lifetime warrantee is meaningless if you have to remove it and take it back a half a dozen times. Personally, I would rather take my alternator to a remanufacturer and have them rework my own unit. After all, remember that these things were built 30 or more years ago! Remanufacturing does cost a bit more than rebuilding, but at least you get what you pay for!

The MGB Mark I (1962-1967) was equipped with the 12 Volt Lucas C40/1 generator (dynamo), which produced a rather puny output of a mere 22 Amps. This usually prompts owners to install a Lucas alternator. However, it should be noted by those who desire to retain an “original” appearance that MGB MKI models supplied for police service were equipped with the higher output (30 Amps) 12 Volt Lucas C42 generator (dynamo) that also equipped Land Rovers and Jaguars of the same era. While both generators (dynamos) make use of the same Lucas RB340 control box, the Lucas C42 generator (dynamo) is 3cm longer in length than the Original Equipment Lucas C40/1 model and was driven by a ½” (12.7mm) wide V-belt by means of thicker pulley wheels, thus requiring the installation of non-original pulleys and the fabrication of a 3 cm long tubular mounting spacer, as well as a 3 cm longer mounting bolt. Unfortunately, this higher power output came at the price of an increased parasitic power loss from the engine (thus requiring the wider V-belt to drive it), so the more efficient 12 Volt Lucas 16AC alternator was chosen as a better alternative for the MGB MKII models. An alternator typically requires about 1 BHP in order to produce 25 Amps of electrical power. Therefore, a 34 Amp alternator will require about 1.36 BHP at full output, while a 43 Amp alternator will require about 1.72 BHP at full output.

Over the course of its production, the MGB was equipped with various models of the Lucas alternator, each having its own power output capacity. While these 12 Volt alternators have many parts in common, there were progressive changes that were made to them :

Model	**16AC**	**16ACR**	**16ACR**	**16ACR**	**17ACR**	**17ACR**	**18ACR**	**18ACR**
Introduced	November 1967	October 1968	January 1971	March 1972	February 1973	? 1974	June 1976	May 1978
BMC Part Number	37H 2245	37H 4194	37H 6938	37H 7503	37H 7959	37H 8208	37H 7961	AAU 1013
Lucas Part Number	23548	23748	?	23716	23756	23804	23737	?
Note	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**
Amperage	34 AMPS	34 AMPS	34 AMPS	34 AMPS	36 AMPS	38 AMPS	43 AMPS	43 AMPS

Rotor	37H 2249	37H 4196	37H 4196	37H 4196	37H 6024	37H 6024	37H 7253	37H 7253
Stator	37H 2257	37H 2257	37H 2257	37H 2257	37H 7966	37H 6203	37H 7254	37H 7254
Rectifier	37H 6924	37H 6594	37H 6594	37H 6913	37H 7965	37H 7965	37H 7962	37H 7962
Integrated Regulator	(Remote) 4TR Control Unit	(Internal) 8TR Control Unit 37H 6911	(Internal) 8TR Control Unit 37H 6911	(Internal) 37H 7516	(Internal) 37H 7964	(Internal) 37H 7964	(Internal) 37H 7964	(Internal) 37H 7961
Surge Protector	None	None	None	37H 7517	37H 7517	37H 7517	37H 7517	37H 7517
Suppression Capacitor	None	None	None	None	None	None	37H 7963	37H 7963
Brush set	37H 2255	37H 4199	37H 4199	37H 4199	37H 4199	37H 4199	GGB504	GGB504
Brush Box	37H 2254	37H 2254	37H 2254	37H 2254	37H 2254	37H 2254	37H 6915	37H 6915
Bracket, Slip Ring End	12H ?	12H 4330	12H 4330	12H 4330	12H 4330	12H 4330	12H 4330	12H 4330
Bracket, Drive End	37H 2246	37H 2246	37H 2246	37H 2246	37H 2246	37H 2246	37H 2246	37H 2246
Drive Bearing	97H 626	47H 5431	47H 5431	47H 5431	47H 5431	47H 5431	47H 5431	47H 5431
Fan	12H 2515	12H 2515	12H 2515	12H 2515	12H 2515	12H 2515	12H 2515	37H 7518

1) Five-wire Alternator

Engines 18GD/-/101 to 7,000

Engines 18GF/-/101 to 13,650

2) Five-wire Alternator

(Used prior to 37H 6938)

3) Three -wire Alternator

Engines 18GK/We/h 10,280 to (+)

Engines 18GK/Rwe/H 10,497 to (+)

Engines 18GG/We/H 21,285 to (+)

Engines 18GG/Rwe/H 21,804 to (+)

Engines 18GG/Rc/h 758 to (+)

Engines 18V 581/ H 101 to 2,101

Engines 18V 581/L 101 to (+)

Engines 18V 582/H 101 to 4,229

Engines 18V 582/L 101 to (+)

Engines 18V 583/H 101 to 303

Engines 18V 584Z/L 101 to 12,913

Engines 18V 585Z/L 101 to 1,829

4) Three-wire Alternator

Engines 18V 581/ H 2,102 to 4,418

Engines 18V 581/L (+) to 1,039

Engines 18V 582/H 4,230 to 13,177

Engines 18V 582/L (+) to 1,224

Engines 18V 583/H 304 to 604

Engines 18V 584Z/L 12,914 to (+)

Engines 18V 585Z/L 1,829 to (+)

Engines 18V 672Z/L 101 to 6,183

Engines 18V 673Z/L 101 to 777

5) Three-wire Alternator

6) Three-wire Alternator

Engines 18V 581/H 4,419 to End

Engines 18V 581/L 1,040 to End

Engines 18V 582/H 13,178 to End

Engines 18V 582/L 1,225 to End

Engines 18V 583/H 605 to (+)

Engines 18V 672Z/L 6,184 to End

Engines 18V 673Z/L 7,78 to End

7) Three-wire Alternator

8) Two-wire Alternator

Rationalized 16ACR with

modified, larger-diameter fan

(+) Change points unavailable

Note that on the five-wire alternators, four of the wires connect the alternator to the rest of the charging system. The “B” terminal is used for the alternator output wire that supplies current to the battery. The “IG” terminal is used for the ignition input that turns on the alternator / regulator assembly. The “S” terminal (sensor terminal) is used by the regulator to monitor charging voltage at the battery. The “L” terminal is used for the wire that the regulator uses to ground (earth) the charge warning lamp. The fifth terminal is the “F” terminal that is used for the full-field bypass of the regulator.

Be aware that on all Lucas 17ACR and 18ACR alternators there is spring clip (BMC Part # 13H 8163) that prevents the plug from accidentally detaching and falling out of the back of the unit. Running the alternator without the plug installed can cause damage as a result of voltage overload.

There were several different connection arrangements for Lucas alternators over the years ranging from 4-pin of the 16AC with its remote regulator (best avoided for a conversion), and then a 5-pin using two connectors on the early internally-regulated Lucas16ACR and finally a 3-pin single connector for other Lucas 16/17/18ACR variants. 5-pin/two plug systems have two Indicator spades in one of the connectors which are linked together by a loop of brown/yellow wire in the plug, possibly to protect the alternator if the engine is run with the IND/B+ plug removed. The 3-pin version has two variants, one with two large spades side-by-side and a single normal-sized spade to one side, and another with a single large spade and two standard-sized (or one standard and one medium) spades either side of it. With the first 3-pin type there seems to have been two variations of how the spades were used. On one variant the central large spade is the output spade and the other large spade is the battery sense terminal, with the normal-sized spade being the Indicator terminal, and on the other variant both large spades are outputs where either (or both together for more current carrying capacity) can be used, and the normal-sized spade is the Indicator terminal.

Where provided, the B+ (or BATT+) terminal is a battery voltage sensing terminal that is wired back to the solenoid with a standard gauge brown wire. This is used to sense the voltage at the solenoid rather than at the alternator for voltage regulation purposes, and would ensure that under high current conditions any voltage-drop occurring in the main output wires (the thick brown and black wires) between the alternator and the solenoid/body is ignored and that the voltage at the solenoid (and hence the battery) is maintained at the correct level. This was the case in the 5-pin 2-plug 16ACR from 1968 to 1971.

Initially, the 3-pin single-plug alternators used machine sensing (i.e. the correct voltage was maintained at the alternator terminals, but could be lower at the solenoid and hence the battery under high current conditions, with just a single thick brown wire and a standard gauge brown/yellow wire in the alternator plug. This is a '2-wire' alternator. Clausager states that a new version of the 16ACR with a modified regulator and surge protection was provided in March of 1972.

Possibly because of problems with low battery voltage, in 1973 the alternators seem to have reverted to battery sensing again in order to control the voltage regulator (Clausager states the 17ACR was fitted from February 73). The 17ACR is a bit different from the other Lucas alternators in that it used battery sensing to control the voltage regulator and not machine sensing as the previous and subsequent models. As well as the main heavy gauge brown output wire, this alternator uses an extra, standard gauge brown wire in the alternator plug which attaches to the solenoid, making for three wires at the alternator in total. If your new harness does not include this wire, you will then almost certainly have charging issues, This wiring seems to have remained the case up to and including the 1977 model year. This is a 3-wire alternator, but can be used on a 2-wire circuit by connecting the third spade to the output spade in the alternator plug. If your new harness doesn't include this wire, you will then almost certainly have charging issues, unless you link the two spades at the alternator together. This wiring system seems to have remained the case up to and including the 1977 model year when the Lucas 18ACR model was introduced.

The final model development was the 18ACR. Clausager says the exact change point is unavailable, but thinks that it was from June of 1976. This is borne out by the Parts Catalogue which shows the 18ACR being used before September of 1976, but not by the wiring schematics which indicate that it did not happen until the 1978 model year. This model of the alternator seems to have reverted to machine sensing again, with two large gauge brown wires in the alternator plug, and the two large spades on the alternator are both output terminals. This gave increased current carrying capacity and lower voltage drop now that cars had electric cooling fans, offsetting the loss in voltage that was caused by the regulator sense terminal moving from the solenoid to the alternator again. Once again, this is a '2-wire' alternator.

'Two browns' wiring will cope equally well with both battery sensing and machine sensing alternators, but battery sensing alternators must have the 2nd brown wire, or at least a link in the harness plug between the positive (+) and B+, in order to operate correctly. So when making changes some care needs to be taken to determine just which type of wiring, plug and alternator you have, even when swapping alternators which take the same plug. If by looking at the two large spades on the alternator you can see that they are clearly connected together, then you have a machine-sensing alternator and can use either or both large spades for the output. But if the two are clearly insulated from one another, then you have a battery sensing alternator. On these you must, at the very least, have a large gauge brown wire on the output spade and a smaller gauge wire at the least on the sense terminal. If in doubt as to which you have, it may be possible to determine by voltage measurement. Turn all the electrical loads on that you possibly can, alternator plugged in, engine running at a fast idle, and then connect a voltmeter between the two large spades. If you can measure any voltage between the two (may only be in the order of tenths of a volt) then you probably have a battery sensing alternator. If there is zero volts between the two large spades, then you probably have a machine sensing alternator. Or simply provide large gauge brown wires to both large spades in order to cover both eventualities, and get the benefit of a lower voltage-drop under high-current conditions if you have a machine sensing alternator.

If you carry an alternator as a spare, then it is a good idea to make sure that it already has an appropriate diameter pulley wheel fitted. The large nut is very tight and makes it very difficult, if not impossible, to remove the pulley wheel from a failed unit as there is no easy way of holding the rotor still. If your spare alternator has a pulley, then compare its diameter with what is on the car. If it is the same diameter, then all well and good. If it is of a different diameter, then check now by trial-fitting that it is compatible with your fan-belt! Be aware that if the pulley is of a smaller diameter, then the alternator will then rotate faster than normal, so you may want to limit your engine speed a little to avoid over-revving the alternator. If it is of a larger diameter, then it will rotate slower, so you may find that the engine speed needs to be increased before it starts charging, will stop charging sooner as the engine speed falls, and that it may not charge at idle. The charge voltage and current during normal driving will also be lower than usual, but if you keep the engine speed up and/or the electrical load down, then it should still charge well enough to get you where you are going.

The Tachometer

The tachometer gets its signal from the plain, unfused white wire that comes from the ignition switch. The 1968 to1972 cars used a current-pulse triggered tachometer (RVI on the faceplate), and these are known to have problems with after-market electronic ignition systems. On these tachometers, the coil wire runs through a transformer-type pickup. The pickup was external and a continuous loop of white wire from the main harness is routed through it. On later versions, there are male and female bullets on the back of the tachometer and corresponding bullets on two wires coming out of the main harness. The electrical circuit is ignition switch-tachometer-coil. In these cars, connect the white with black stripe wire that runs from the rear of the tachometer to the negative (-) terminal on the coil and the tachometer. Note that the type of tachometer has nothing to do with whether or not the ignition system is ballasted or unballasted. The successor to the current-pulse tachometer arrived when the 12 Volt unballasted ignition system was still in use. The ballasted ignition system did not arrive until the advent of the Rubber Bumper cars during the 1974 model year. 1973 and later tachometers (RVC on the face-plate) are voltage-triggered and should function properly with electronic ignition. The switch from current-pulse triggering to voltage-triggering was chosen so that the ignition current would no longer be routed from the tachometer, problems with which could effect the performance of the ignition system. These later tachometers have a single bullet connector that a white / black wire connects to the negative (-) post on the coil. It also has a 12 Volt supply from a green, fused ignition wire that feeds both the tachometer and the voltage stabilizer with system voltage, i.e., a full 12 Volts, and a black ground (earth) wire. The voltage stabilizer was never used to feed the tachometer, only the ancillary gauges. All of the variations have a standard-sized spade for the 12 Volt supply (another white wire on early cars, green after that) and a smaller spade for a black ground (earth) connection. These latter two are to power the tachometer electronics.

Of course, it is possible that you will have tachometer problems that do not originate with the electronic ignition system. A tachometer reading that jumps around when you accelerate could be the result of one of the wires inside of the distributor having an intermittent connection. The vacuum changes when you alter the throttle, and that will cause the vacuum advance control capsule to rotate the contact breaker points plate. There are two special highly flexible wires between the contact breaker points plate and the body of the distributor negative (-), the coil wire and the ground (earth) wire. Both of these wires are supposed to be more flexible than standard wires, but they do eventually fail. It is absolutely necessary that these be the special highly flexible wire, otherwise they also will fracture in a short time and give intermittent ignition. “Correct” ground (earth) wires for the Lucas distributor are not available as far as I know, but Moss does have an alternative, although I do not know how its flexibility compares.

If you want to test a later negative (-) ground (earth) RVI or a RVC tachometer while it is mounted in the car by using the distributor, then simply connect the three or four wires, as there should not be any way of getting the wires mixed up. For the earlier RVI tachometers with the external pickup, you also have to get the loop of white wire routed through the pickup in the correct direction in order to drive the tachometer. Getting that the wrong way round will not cause any damage, but getting the 12 Volt and ground (earth) supply the wrong way round might.

To perform a bench test on a tachometer, you need a way of generating pulses as well as a 12 Volt supply. The 12 Volt supply for the tachometer electronics needs to be "smoothed", i.e., from a battery, connected to the two spades with the correct polarity. The 1964-1967 tachometers use a positive (+) ground (earth), while 1968 and later tachometers use a negative (-) ground (earth). However, many 1964-1967 tachometers will have been converted to negative (-) ground (earth), so you will need to use the same ground (earth) polarity as the car that they came from. Many conventional battery chargers can simulate the trigger pulses. If you have a current-pulse RVI tachometer, you need to have a circuit from the positive (+) connection of the battery, a steady 12 Volt supply to the tachometer electronics, i.e., through the tachometer pickup, through a suitable load such as a 3 Ohms resistor or coil, then back to the negative (-) connection of the battery wired in series with the charger. Higher value resistances may not result in enough current through the pickup to trigger the tachometer electronics. For an RVC tachometer you need to have a load connected directly to the charger terminals, and as well as from the positive (+) connection of the charger to the bullet connector on the back of the tachometer, and from the negative (-) connection of the charger to the ground (earth) spade on the backside of the tachometer. You should be able to use a higher resistance load, such as a bulb, for these tachometers. It then depends on whether or not your charger uses a full-wave rectifier or a half-wave rectifier. A full-wave rectifier will deliver 100 pulses per second, which equates to 3,000rpm. A half-wave rectifier will deliver 50 pulses per second, which equates to 1,500 rpm, and in theory, this is what you should see on the tachometer.

There are three types of Smiths tachometers: external Current Sensing loop (Positive Ground (earth)), internal Current Sensing loop (Negative Ground (earth)), and Electronic Pulse Sensing (Voltage Sensing) (1973 and later models). Current Sensing tachometers have a wire loop with two turns that passes the ignition coil current through a pickup at the tachometer. This loop can be external or internal. Electronic Pulse Sensing (Voltage Sensing) tachometers have a trigger wire that connects to the negative (-) terminal of the ignition coil.

In the Current Sensing tachometer, ignition coil current flows via the inductive pick-up on the back of or inside the tachometer, so the tachometer responds to the current pulses through the ignition coil, as the points open and close. If that circuit breaks, then the engine stops, if it shorts to ground, then you fry the loom! Negative (-) ground (earth) 1964-67 types had the pickup mounted on the outside of the case and the later positive (+) ground (earth) one (1968-1972) had it mounted inside the case. For the earlier type, a continuous loop of white wire comes out of the main harness from the ignition coil SW terminal, through the pickup as indicated in the top picture on the left (click to expand), then back into the harness again towards the ignition switch. This is for positive (+) ground (earth) cars. If the car is converted to negative (-) ground the wire must be removed from the pickup and routed through in the other direction. If fitting a new harness I am not sure if the ends of the loop are marked to indicate which is the ignition coil end and which is the switch end. If not, you will just have to try first one way then the other. The tachometer will only work with the current going through the pickup in one direction - a bit fiddly to reverse with the continuous loop, but easy to reverse with the bullet connectors. It will not hurt the tachometer if you get it the wrong way round. For the later type with internal pickup (negative (-) ground cars) the ends of the loop of wire through the pickup are brought out to male and female bullet connectors mounted on the back of the case, see the lower picture on the left. The spade for the 12 Volt supply to the electronics of the tachometer is close by the bullet connectors, and the spade for the ground connection for both the electronics and the instrument lighting is spot-welded onto the back of the case. The harness now has separate white wires with female and male respectively bullet connectors, meaning incorrect connection is not possible.

With the Electronic Pulse Sensing (Voltage Sensing) type, the ignition current goes directly from the switch to the ignition coil, and a tapping off the ignition coil negative (-) / points connection that goes to the tachometer on a white/black wire, which responds to the voltage changes as the points open and close. If this circuit breaks the tachometer ceases to register and the engine continues to run. If it should happen to short to ground, then the engine will stop, but not fry the loom. (In fact this makes a nifty anti-theft device using a hidden normally open switch connected to the wire at the tachometer rather than in the engine compartment!) Since the current flowing through the ignition coil has a direct relationship with the voltage at the ignition coil CB or negative (-) terminal it follows that the two types indicate the same thing. Both types have a threshold above which any voltage or current pulse will register on the needle, and the relative sensitivities of the tachometer electronics and the engine are such that, under normal circumstances, weak ignition pulses will affect the engine before they affect the tachometer reading.

Typical problems are sticking, wavering, or simply not working at all. Sticking, where a rap with a knuckle on the glass fixes it, and it only occurs after being parked for a while or at certain times of year, is almost certainly a mechanical problem with the movement itself.

Wavering or flicking about, if accompanied by changes in the idle speed, have a good chance of being caused by bad connections in the ignition Low Tension (LT) circuit (that is ignition switch–to-ignition coil (via tachometer where appropriate) –to- points–to-ground.

Wavering or flicking about that is *not* accompanied by changes in idle speed, randomly dropping to zero for longer periods, or not working at all, could be either the 12 Volt supply to the tachometer, the connection between ignition coil and tachometer on the later Electronic Pulse Sensing (Voltage Sensing) types, or electronic problems inside of the tachometer itself. From 1964 to 1967 the tachometer was powered from a third white unfused ignition wire and a black ground (earth) wire, but after that it was from a green (fused ignition) wire and a black ground (earth) wire for both the Current Sensing and the Electronic Pulse Sensing (Voltage Sensing) types. In no case is the tachometer powered from the instrument voltage stabilizer as the output from this is 12 Volt switched on and off about once a second and so is unsuitable for the tachometer for obvious reasons. Get a multi-meter with an RPM range, connect it to the points-side of the ignition coil, and compare that with the car's tachometer. If it shows similar variations, then there is a problem in the ignition Low Tension (LT) circuit through the ignition switch, ignition coil and points. If it is steady when the car's varies, and you have the Electronic Pulse Sensing (Voltage Sensing) tachometer, then connect the multi-meter to the white/black at the tachometer. Variation here but not before would indicate problems with the white / black wire or connections between the tachometer and the ignition coil. If that is steady too, or if it was steady at the ignition coil and you have a Current Sensing tachometer, monitor the 12 Volt supply and ground at the tachometer. If these are steady too, then the problem must be inside the tachometer itself.

Installation of a Crane electronic ignition may cause erratic operation of a Current Sensing tachometer, due to the higher coil current. Modification of the current pickup in order to reduce the signal level will usually eliminate this problem. In order to accomplish this, remove the Smiths tachometer from the instrument panel. The tachometer has two threaded studs that are retained from the rear of the panel. Label all of the wires in order to avoid errors upon reinstallation. Be warned that improper connection may damage the tachometer. Locate the external current pickup on the rear of the tachometer. If your tachometer does not resemble this but has coil and ignition key wires going to a plug, it may have an internal current pickup. In this case, it will be necessary to disassemble the tachometer.

Once the tachometer is disassembled, modify the current pickup by removing one loop of wire. Note the direction that the wire passes through the pickup. If this direction is reversed, the tachometer will not function.

At this point, you should recalibrate the tachometer for best accuracy. Connect a test tachometer and have a helper rev the engine. Hold the tachometer in the same position that it is mounted in (orientation may affect calibration). Adjust the calibration screw on the back of Smiths tachometer until the reading matches that of the test tachometer. 4,000 RPM is a reasonable engine speed to use for calibration. Please note that older Smiths tachometers may vary as much as 500 RPM throughout their range. This variation is not the fault of the ignition system.

Finally, reinstall tachometer into the instrument panel, and then check all of the wire connections.

In some cases the tachometer will not read the correct engine speed after installation of an electronic ignition. A calibration screw on the back of the tachometer can usually be adjusted to give correct readings. If the tachometer still reads high, put a resistor in the tachometer wire to reduce the signal level. Start with a 10K Ohm ½ Watt. You can go as low as 1K ohm 1/2 watt. You can buy the resistors from Radio Shack or other electronic suppliers. Solder into tachometer wire and wrap with electrical tape for protection.

Wiring

Understand that if you switch to an alternator that produces more amperage, then you will need to check to be sure that your wiring loom is capable of handling the increased power output. When upgrading to a higher-output alternator, you should install a larger gauge size wire between the alternator and the battery. Even when using a standard-output alternator, you will typically obtain better performance and lengthen the service life of your alternator if you upgrade the main battery wiring. The original gauge size wire is simply not large enough for proper power transfer. If you are using your alternator to its maximum output, or when you upgrade to a higher output alternator, then you must increase the gauge size of the wire. An alternator's ability to send the power that it is producing to the battery is directly related to the wire gauge size and the quality of the connections between the alternator and battery. In addition, a wire gauge size that is too small when used on a high output alternator can cause the power to back up within the alternator, causing it to overheat and burn up. An insufficient gauge size ground (earth) wire can cause as much damage to the alternator as an insufficient gauge size wiring between the alternator and the battery, so check to make sure that the gauge size of the wires between the battery and alternator are properly matched for capacity. The following table should serve as a basic guide in this as well as in other matters:

American Wire Gauge (AWG)	Diameter (Inches)	Diameter (mm)	Resistance in Ohms (Per 1,000 Ft / Km)	Maximum Amperage for Chassis Wiring
5	0.1819”	4.62026mm	0.3133 / 1.027624	118
6	0.162”	4.1148mm	0.3951 / 1.295928	101

7	0.1443”	3.66522mm	0.4982 / 1.634096	89
8	0.1285”	3.2639mm	0.6282 / 2.060496	73
9	0.1144”	2.90576mm	0.7921 / 2.598088	64
10	0.1019”	2.58826mm	0.9989 / 3.276392	55
11	0.0907”	2.30378mm	1.26 / 4.1328	47
12	0.0808”	2.05232mm	1.588 / 5.20864	41
13	0.072”	1.8288mm	2.003 / 6.56984	35
14	0.0641”	1.62814mm	2.525 / 8.282	32
15	0.0571”	1.45034mm	3.184 / 10.44352	28
16	0.0508”	1.29032mm	4.016 / 13.17248	22
17	0.0453”	1.15062mm	5.064 / 16.60992	19
18	0.0403”	1.02362mm	6.385 / 20.9428	16
19	0.0359”	0.91186mm	8.051 / 26.40728	14
20	0.032”	0.8128mm	10.15 / 33.292	11
21	0.0285”	0.7239mm	12.8 / 41.984	9

22	0.0254”	0.64516mm	16.14 / 52.9392	7
23	0.0226”	0.57404mm	20.36 / 66.7808	4.7
24	0.0201”	0.51054mm	25.67 / 84.1976	3.5

Note that the original five-wire Lucas 16 AC alternator makes use of an external voltage regulator, and that all of the subsequent Lucas five-wire, three-wire, and two-wire alternators have an internal voltage regulator. Sometimes there are three wires on the alternator, and sometimes there are only two wires on the alternator. Even when there are only two factory wires, the alternators always have three terminals – two available for large output wires, and one available for a thinner, standard brown / yellow indicator wire from the alternator warning light (ignition warning light). When there are three wires, the third wire is always brown, and can either be the same thinner standard gauge wire as that of the brown / yellow indicator wire from the alternator warning light (ignition warning light), or it can be the same heavier gauge wire as that of the main brown output wire that carries the charging current to the battery. In the first case of the third wire being of the thinner standard gauge, be aware that the third wire was used with “remote sensing” alternators that controlled the voltage at the solenoid instead of at the alternator output. This was used in order to sense the voltage at the solenoid rather than at the alternator for voltage regulation purposes, and would ensure that under high current conditions any voltage-drop occurring in the thick brown and black main output wires between alternator and solenoid/body was ignored and the voltage at the solenoid (and hence the battery) was maintained at the correct level. These alternators can be identified by their having one large main output terminal and two standard smaller size terminals. However, in the second case, the heavier gauge brown third wire is an extra output wire that is utilized to increase current carrying capacity and reduce voltage-drop on later cars. These alternators can be readily identified by their having two large output terminals and one small terminal for the brown / yellow indicator wire from the ignition warning light.

In order to upgrade from a 34-Amp Lucas 16AC alternator to the higher-output Lucas 18 ACR alternator that produces a more useful 43 Amps, it becomes necessary to convert an original five-terminal wiring harness to a system that will accommodate this three-terminal alternator. In order to accomplish this task, disconnect the battery, and then cut off the wiring connectors of the small brown wire and the small black wire. Remove and discard the link wire, and then remove approximately one inch of the tape of the wiring harness. Slide a small insulator over the remaining brown / yellow indicator wire from the ignition warning light, and then solder the wire to the connector that is used for the small terminal of the alternator. The small brown wire and the small black wire are not used, so bend them back and tape them individually to the wiring harness. Connect the large brown / white positive (+) wire to either of the two large terminals on the alternator. In order to maximize electrical conductivity and prevent future corrosion of the connections, be sure to smear a thin coating of dielectric grease onto each terminal. Reconnect the battery and you are finished with the conversion.

Be aware that you do not need to remove all of your old wiring when performing an alternator upgrade. Instead, you can merely install a second wire of adequate gauge size between the alternator and battery. The main battery wire that is connected to the back of the alternator constantly has power to it, even when the engine is not running. Connect this wire as per normal, and then attach a second wire between the alternator and the battery. The power coming out of the alternator will follow the path of least resistance. On a safety note, when running the second wire, you should also install a fuse near the battery. The fuse will function as a "just-in-case" should the wire ever become pinched or short-circuited, the fuse blowing out instead of the wire burning up. You should use the largest fuse you can for the wire gauge size as fuses are always restrictive to current flow. Typically, you want the fuse value to be 80% of the load carrying capacity of the wire.

Another area that receives little attention is the ground (earth). A poor ground (earth) will not only hinder the ability of the alternator to supply power to the battery, it can also cause an alternator to burn up just as readily as can an alternator-to-battery wire of inadequate capacity. Although your ground (earth) may well be sufficient when you first install your alternator, over time corrosion and resistance builds up in the connections of its ground (earth). This is why experts recommend that it is best to run the ground (earth) directly from the rear of the alternator to the battery.

Of course, It goes without saying that if your car has recurrent electrical problems, then it is hard to enjoy ownership of it. All too often, the recurrent problems can be eliminated only by repairing or replacing the wiring loom. Whether or not it is advisable to fabricate a new wiring loom, as opposed to repairing the existing wiring loom, depends upon a number of factors. If you have no wiring loom at all, it would be much easier and cheaper to buy a ready-made wiring loom. Building a wiring loom from scratch, without a pattern to go by, is extremely tedious, to say the least. If, on the other hand, you have a fairly good wiring loom, with only a few damaged wires, then it would be both cheaper and easier to repair it rather than creating a new one. These two extremes make the choice fairly easy; it is when your situation is somewhere between that the decision is difficult.

A common approach is to have one or more wiring looms that have been salvaged from a junkyard. In this case, it may be desirable to save as much of the wiring looms as you can, re-using the connectors and wire to make a new one, or to repair the better of the wiring looms to make one good one out of those that you have.

As for re-using the connectors, I would advise caution. Of all of the parts of the wiring loom that are likely to give problems, the most trouble-prone are the connectors. If you do re-use them, make sure they are clean and corrosion free. The bullet connector sleeves, the black pieces that are used to connect two pieces of wire, are particularly trouble-prone. Often, the metal sleeve inside will crumble with age, and will no longer have sufficient tension to make a good connection.

Salvaging a wire is not a problem, as long as the wire has not been damaged. If you can, i.e., the wire is long enough, I recommend cutting the wire an inch or two from any connector. Often, moisture will have wicked up the strands and will have corroded the wire, making it difficult to get a good connection. In such as case, you should cut back to where the wire is clean and shiny.

The easiest way to repair a wiring loom is to clear out a large space on the garage floor, and lay the wiring loom down, stretched out, with each "leg" of the wiring loom laid out by itself. Strip back the tape surrounding the damaged area, being carefully not to displace the wires from their original positions. Lay the new wires down, one at a time, alongside the original wires, to make sure they are the same length, allowing a 3/4" (19 mm) overlap for the splice. Strip off 3/4" of the insulation from both the old wire and the new, and then slip a piece of heat-shrink tubing over one of the wires. Twist the wires together, and then solder. After the joint has cooled, slip the heat shrink tubing over the joint, and apply heat in order to shrink it. Attach whatever connector is appropriate to the free end of the wire. Continue with each wire, till that section is done, and then go on to the next section. Prior to re-wrapping, Use plenty of cable ties to hold the wiring in place as you work. During the re-wrapping process, as you come to each cable tie, cut it off, and discard it. Be prepared to use plenty of cable ties.

Do not make all of the repair splices in exactly the same place. If you make this mistake, then the wiring loom will be very thick at that point, and all the stress will be concentrated in that one area. If possible, space the splices out over the length of the wiring loom, at least six inches from each other.

Be careful to note any pre-formed bends in the wiring loom, and reproduce them when you do the repairs. If you bend the wires, and tie them tightly with the wrap, they will hold their shape surprisingly well.

When you do the re-wrapping, begin with the connector end and work back towards the main wiring loom. That way, you will have fewer loose ends of the wrapping tape to worry about, as the wrapping for each leg of the wiring loom gets wrapped again by the main wrap. Start by running a short piece of the wiring loom tape parallel to the wires, heading towards the connector end, and start wrapping back over this parallel piece, back towards the main wiring loom. This way, the wrapping secures that end of the tape. Use the special wiring loom tape supplied by British Wiring, which has no adhesive, but instead is self-adhering. When you get to the last piece, use electrical tape to secure the last wrapping.

Making a new wiring loom will be exactly the same as above, with the exception that you will run the wires from one end to the other, using the existing wiring loom as a guide. If you are making a new wiring loom, there will be places where you will have to splice, just as the factory did. Just make your splices in the same place and manner as original.

If you do not have an Original Equipment wiring loom to use as a pattern, you can still make your own, but the degree of difficulty significantly increases. In this case, you will need to route each wire individually, in place, in the car, connecting each end as you go. If you can, it is a good idea to look at a completed car, to see how the wiring loom was installed at the factory. The factory provided sufficient supports, and these should be used for the new wiring loom. The location of the supports provides a good indication of the correct wiring loom routing. Connect one end, then route the wire to the other end, and make that connection. As you lay the wires down, pay particular attention to the need to remove and replace both the wiring loom and the electrical components. There must be sufficient slack in the wire to allow for this. Also, great care must be given to the routing of the wire to avoid any possibility of damage from rubbing against sharp edges, and making sure that the wire is properly supported to eliminate strain on the connectors. As you run the wires, use cable ties to hold everything is place. When you have completed and tested the wiring, very carefully remove the wiring loom from the car, and then wrap as described above.

Whichever route you take, you will need to obtain the required supplies. There are multiple sizes of wire used in an MG, and over 50 different color codes on the later models (out of a possible 144, 106 of which are available), the exact number depending on the model year. An excellent source for accurate wiring diagrams for the different year/model of the MGB can be found at the website of Dan Masters at http://www.advanceautowire.com/mgb.pdf

In the United Kingdom the British Standard BS-AU7 determines color coding of automobile wiring. Lucas used a 7 color set in which plain colors - purple, green, blue, red, white, brown and green are supplemented by a further group using a base color with a thin line trace of a different color, as thus:

Black	Ground (earth) connections.
Green	Feeds to auxiliary devices controlled by the ignition switch, e.g., wipers, flashers, etc.
White	Base color for ignition circuits.
Red	Sidelights (parking lights) and rear lights.
Blue	With white trace main beam headlamp, With red trace - dip (meeting) beam headlamp.

Other colors are used, according to equipment specifications, e.g., light green, pink, slate. Handbooks are usually printed in black and white only, so the cable colors are identified by a lettering code, such as:

B	Black	**P**	Purple	**N**	Brown	**S**	Slate
U	Blue	**G**	Green	**R**	Red	**W**	White

When a cable has a base color and a second color spiral trace the code is, for example: WG = White with green trace

Main	**Tracer**	**Destination**
Black		All earth connections.
Black	Blue	Tachometer generator to tachometer.
Black	Brown	Tachometer generator to tachometer.
Black	Green	Windshield wiper switch to windshield wiper (single speed) relay to radiator fan motor
Black	L. Green	Vacuum brake switch to warning light and/or buzzer.
Black	Orange	Radiator fan motor to thermal switch.
Black	Red	Electric speedometer.
Black	White	Brake fluid level warning light to switch and handbrake switch.

Black	Yellow	Electric speedometer.
Blue		Lighting switch (head) to dipper switch.
Blue	L. Green	Windshield wiper motor to switch.
Blue	Pink	Headlamp dip beam fuse to left hand headlamp (when Independently fused).
Blue	Red	Dipper switch to headlamp dip beam. Headlamp dip beam. fuse to right-hand headlamp (when independently fused).
Blue	Slate	Headlamp main beam fuse to left hand headlamp or inboard headlamps (when independently fused).
Blue	White	Dipper switch to main beam (subsidiary circuit – headlamp flasher relay to headlamp). Headlamp main beam fuse to right-hand headlamp (when independently fused). Headlamp main beam fuse to outboard headlamps (when outboard headlamps independently fused). Dipper switch to main beam warning light.
Blue	Yellow	Long range driving switch to lamp..
Brown		Main battery feed.
Blue	Yellow	Long range driving switch to lamp..
Brown		Main battery feed.
Brown	Black	Alternator warning light, negative side.
Brown	Blue	Control box (compensated voltage control only) to ignition and ignition switch, e.g., wipers, flashers, etc lighting switch (feed).

Brown	Green	Dynamo ‘F’ to control box ‘F’ Alternator field ‘F’ to control box ‘F’.
Brown	L. Green	Windshield wiper motor to switch.
Brown	Purple	Alternator regulator feed.
Brown	Red	Compression ignition starting aid to switch. Main battery feed to double pole ignition switch (a.c. alt. system).
Brown	White	Ammeter to control box. Ammeter to main alternator terminal.
Brown	Yellow	Dynamo ‘D’ to control box ‘D’ and ignition warning light. Alternator neutral point.
Green		Accessories fused via ignition switch (subsidiary circuit fuse A4 to hazard switch (terminal 6)) .
Green	Black	Fuel gauge to fuel tank unit or changeover switch.
Green	Blue	Water temperature gauge to temperature unit.
Green	Brown	Reverse lamp to switch.
Green	L. Green	Hazard flasher unit to hazard pilot lamp.
Green	Orange	Low fuel level warning light.
Green	Pink	Choke solenoid to choke switch (when fused).
Green	Purple	Stop lamps to stop lamp switch.
Green	Red	Left-hand flasher lamps.
Green	Slate	Heater motor to switch (or to fast) (on 2-speed motor).

Green	White	Right-hand flasher lamps.
Green	Yellow	Heater motor to switch, single speed (or to ‘slow’ on two-speed motor).
L. Green		Instrument voltage stabilizer to instruments.
L. Green	Black	Screen jet switch to screen jet motor.
L. Green	Blue	Flasher switch to left-hand flasher warning light.
L. Green	Brown	Flasher switch to flasher unit ‘L’.
L. Green	Pink	Flasher unit ‘L’ to emergency switch (simultaneous flashing).
L. Green	Purple	Flasher unit ‘F’ to flasher warning light.
L. Green	Red	Fuel tank changeover switch to right-hand tank unit.
L. Green	Slate	Fuel tank changeover switch to left-hand tank unit.
L. Green	Yellow	Flasher switch to right-hand flasher warning light.
Purple		Accessories fused direct from battery.
Purple	Black	Horn or horn relay to horn push.
Purple	Brown	Horn fuse to horn relay (when horn is fused separately).
Purple	Orange	Aerial lift motor switch DOWN.
Purple	Red	Boot light switch to boot light.
Purple	Slate	Aerial lift motor to switch UP.
Purple	White	Interior light to switch (subsidiary circuit—door safety lights

		to switch).
Purple	Yellow	Horn to horn relay.
Red		Side and tail lamp feed.
Red	Black	Parking switch to left-hand side lamp.
Red	Brown	Variable intensity panel lights (when used in addition to normal panel lights).
Red	Green	Lighting switch to side and tail lamp fuse (when fused).
Red	L. Green	Windshield wiper motor to switch.
Red	Orange	Parking light switch to right-hand sidelamp.
Red	Purple	Map light switch to map light.
Red	White	Panel light switch to panel lights.
Red	Yellow	Fog lamp switch to fog lamp.
Slate		Window lift
Slate	Black	Window lift
Slate	Blue	Window lift
Slate	Brown	Window lift
Slate	Green	Window lift
Slate	L. Green	Window lift
Slate	Orange	Window lift

Slate	Pink	Window lift
Slate	Purple	Window lift
Slate	Red	Window lift
Slate	White	Window lift
Slate	Yellow	Window lift
White		Ignition control circuit (unfused) (ignition switch to ballast resistor).
White	Black	Ignition coil CB to distributor contact breaker. Rear heated window to switch or fuse TAC ignition.
White	Blue	Choke switch to choke solenoid (unfused). Rear heater fuse unit to switch. Electronic ignition TAC ignition unit to resistance.
White	Brown	Oil pressure switch to warning light or gauge.
White	Green	Fuel pump No. 2 or left-hand to change-over switch.
White	L. Green	Windshield wiper motor to switch.
White	Orange	Hazard warning feed (to switch).
White	Pink	Radio from ignition switch.
White	Purple	Fuel pump No. 1 or right-hand to change-over switch.
White	Red	Solenoid starter switch to starter push or inhibitor switch.
White	Slate	Tachometer to ignition coil.

White	Yellow	Starter inhibitor switch to starter push. Ballast resistor to coil. Starter solenoid to coil.
Yellow		Overdrive
Yellow	Blue	Overdrive
Yellow	Brown	Overdrive
Yellow	Green	Overdrive
Yellow	L. Green	Windshield wiper motor to switch.
Yellow	Purple	Overdrive
Yellow	Red	Overdrive

Be aware that British wire, as used in MG, is not sized by gauge as is American wire. The size of British wire is stated as the number of individual strands of 0.30 mm copper used in order to make the wire. The sizes available are listed below, along with the maximum current rating of each size:

British 9 strand wire = 22 gauge US = 5.75 amps

British 14 strand wire = 18 gauge US = 8.00 amps

British 28 strand wire = 14 gauge US = 17.50 amps

British 44 strand wire = 12 gauge US = 25.50 amps

British 65 strand wire = 10 gauge US = 35.00 amps

British 84 strand wire = 9 gauge US = 42.00 amps

British 120 strand wire = 8 gauge US = 60.00 amps

For any given wire in your existing wiring loom, just strip a short length and count the strands. This will tell you what size to buy for replacement. It is very unlikely that you will encounter the 120 strand wire, but if you should ever upgrade your alternator to a more powerful unit, this is one size wire that you should consider using. British wire, as well as the connectors that are used in British cars, is available from British Wiring. They have a website at http://www.britishwiring.com/

It goes without saying that British sports cars do not have a great reputation for their electrical systems. Whether that reputation is deserved or not, it often makes sense to remove the ancient wiring and replace it with fresh parts and more modern technology. Long-time MGB enthusiast Dan Masters of Advance Auto-Wire makes that feasible by providing a high quality kit and including good documentation. Perhaps the most important feature of Advance Auto-Wire's kit is the wiring that is used. The Advance Auto-Wire wiring kit comes with a great big bundle of fat wires. Frankly, every single lead is designed to be longer than it needs to be. You simply trim them to your desired length. Why are they so much thicker? Simple: On almost all of the circuits, the copper wire is made of a thicker gauge than that of the Original Equipment wire. The practical significance of the thicker wire is that it results in less electrical resistance and thus less voltage-drop, and consequently many devices work better. (The heater fan spins faster. The lights glow more brightly.) The wiring is also covered with a thick layer of cross-linked polyethylene insulation. SXL insulation has a significantly higher melting temperature than the polyvinylchloride (PVC) insulation that originally came on the wiring of the MGB. The higher temperature rating of SXL insulation provides a better safety net in the event of a mishap. With a one-off, hand-built car there is a very real possibility that a fuse will not always be there in order to protect against an installer's mistake. The SXL insulation can withstand a lot more heat, whether the heat is caused by either over-current or by accidentally having a wire fall against a hot exhaust system. SXL insulation is also quite noticeably thicker than the usual insulation. However, Dan Masters has gone to the extra effort of color-coding the wire insulation in order to match the familiar British automotive standards. Other wiring kits do not offer this convenient, and critical, feature.

Another significant advantage of the Advance Auto-Wire wiring loom is that it provides better circuit protection and more relays than the original wiring loom used. There are more fused circuits (eight, or twelve if you buy the optional accessory add-on) and also modern

ATC fuses in lieu of the antiquated Original Equipment glass fuses. In all cases the fuses are sized for the wires that they are intended to protect (length and gauge) – rather than being sized for the devices powered by those respective circuits. Take particular note of the use of relays. With relays added into the circuits, the electrical current loads that pass through their corresponding switches is reduced dramatically. Again, there is typically significantly less electrical resistance and thus less voltage-drop in the circuit, so the switched device performs better. Furthermore, the switch contacts themselves consequently endure less stress, prolonging the service life and the reliability of these sometimes hard-to-find Original Equipment items.

Typical Advance Auto-Wire Circuit Protection and Relay Scheme

Position	Circuits Protected	Fuse Rating
Fuse 1	Windshield Wipers / Gauges / Turn Signals	15 Amp
Fuse 2	Heater / Reverse Lights / Brake Lights	15 Amp
Fuse 3	Fuel Pump	15 Amp
Fuse 4	Parking Lights	15 Amp
Fuse 5	Courtesy Lights / Radio Memory	15 Amp
Fuse 6	Horn / Hazard Lights	20 Amp
Fuse 7	Cigar Lighter / Accessories	20 Amp

Fuse 8	Cooling Fan	20 Amp

Relays	
Position	**Circuit Switched / Device Type**
1	Main Power Relay (4 pin)
2	Fuel Pump Relay (5 pin)
3	Cooling Fan Relay (5 pin)
4	Horn Relay (4 pin)
5	Low Beam Relay (5 pin)
6	High Beam Relay (5 pin)
7	Hazard Lights Flasher
8	Starter Relay (4 pin)
9	Turn Signal Flasher

Perhaps one of the nicest things about Dan Master's work as an enthusiast-turned-entrepreneur is his Advance Auto-Wire website. It includes highly detailed, easily understandable, downloadable installation instructions in a PDF format so that you can determine if you wish to use any of his products without first. having to purchase it You can find his website at http://www.advanceautowire.com/ .

Conclusion

Well, that is just about it. I guess that could say a lot more, but Peter Burgess has already said most of it (such as the intricacies of camshaft lobe design and combustion chamber modification) in his books. Buy them and give them a thorough reading. Beyond this I assure you that if you build your engine as Peter Burgess recommends in his books, your engine will amaze you with how smooth, durable, and powerful it is. If you have any questions or feedback, drop me a line.

MG owners have been improving their cars since day one. In fact, the entire history of MGs goes back to the days when mechanics at Morris Garages would take a standard Morris automobile and "improve" it for discerning customers who wanted a little better performance, then affix an octagonal MG emblem to their radiators to advertise where the car was serviced (now you know where the name "MG" comes from). MGs have always been enthusiast's cars, and it is just in the nature of things for enthusiasts to improve their cars. Only the most rabid of purists would object to an owner doing period-correct modifications to it. What entails "period-correct" modifications, you ask? Quite simply, anything that was being done to the cars when they were still in production, including some really interesting work done by the factory race team. This includes, but is not limited to, changes such as: camshaft, headwork, valvetrain work, carburetors, intake manifolds, air filters, exhaust system work, distributor modifications, changing transmission and differential gear ratios, suspension modifications including different springs, damper rate modifications, stabilizer bars (both front and rear), lowering the chassis, adding a Panhard rod, wheels, tires, and just about anything else that the mind had conceived of in those days, which is a lot. I have never met an MG owner who has actually done all of these things to his car, but if I ever do, you can bet he will be wealthy. I can see no reason for any MG enthusiast to have a problem with pointless ignition, better headlights, better brake friction materials, radial tires, or anything else that is a reversible "improvement". To those enthusiasts who take pleasure and pride in tinkering with and improving their MGs I say: "You are the true keepers of the MG Heritage." To those who insist that an MG should always be exactly as it was when it left the factory at Abingdon, I can only say this: "You are missing the whole point of the Marque and its history."

Happy Motoring, Steve S., Virginia, USA