A Tensioning System for Use with Synthetic Ropes

E. Johnson, Ph.D.
Advanced Design Consulting
126 Ridge Road,
Lansing, NY 13778
eric@adc9001.com

P. Petrina, Ph.D.
Department of Theoretical &
Applied Mechanics,
Cornell University,
Ithaca, NY
pp25@cornell.edu

Prof. S. L. Phoenix
Department of Theoretical &
Applied Mechanics,
Cornell University,
Ithaca, NY
slp6@cornell.edu

***Abstract* - Ropes made from Aramids, PBO and HMPE fibers have superior properties, in the longitudinal direction, when compared to those of wire ropes. A development program for the US Navy in the early 1990s sought to take advantage of these properties to design ropes that were lighter, more flexible, more resistant to corrosion and had a longer fatigue life than wire rope when used for the highline in STREAM rigs. Unfortunately the inherent weaknesses of synthetic ropes, which result from the weakness of the hydrogen bonds that connect the polymer molecules, were not considered and sample ropes failed quickly at the highline winch due to crushing and abrasion.**

Our approach to this problem has been to develop a tensioning system that takes advantage of the excellent longitudinal properties of synthetic ropes while avoiding the stresses that cause premature failure. Thus our emphasis has been on equipment design and the demonstration of its compatibility with synthetic ropes rather than on incremental improvements of rope properties. We have developed a system of powered sheaves that apply traction over a large area on the rope's surface and limit shear stress. The use of AC motors under flux vector control allows each sheave to be driven under torque control for precise application of tension through the system. Each sheave can run at a slightly different speed to accommodate the change in rope length as it passes through the tensioning system. N+1 design gives redundancy for greater safety and reliability. Our goal is to demonstrate that synthetics will perform better than wire rope when part of a complete, well-designed system. This system has applications in hoists, lifts and winches.

I. INTRODUCTION

During the period from 1993 to 1995 the U.S. Navy funded development of synthetic cables that were capable of withstanding large contact loads from sheaves [1-6]. This work followed from the successful development of a 2" Kevlar rope for the White Sands Missile Range that had been designed to sustain very large sheave loadings at high translational speeds. Perhaps the differences in the two applications were not fully understood by the Navy. As a result, when a Kevlar rope, that had been optimized for bend-over-sheave performance was tested as the highline in a STREAM rig (Standard Tensioned Replenishment Alongside Method) it failed within a few hours due to damage that occurred at the tensioning winch.

The winch used in the current STREAM rig typically holds about 800 feet of 1" diameter wire rope and can apply a tension of up to 19,500 lb_f. A winding device is used to provide even take-up of cable on the drum. Despite this, winding is not perfect and pinching and crushing occur as rope is wound or unwound to accommodate motion between ships. While this may not cause premature failure in wire ropes it is catastrophic for synthetics.

The Navy's plan to develop a high-capacity STREAM rig, with an operating tension in the region of 60 kips, poses an opportunity to re-examine the use of synthetic ropes in a system engineered specifically for their use. In keeping with the Navy's desire for an "all-electric ship" such a system would eliminate the hydraulic riser-tensioner that is currently in use. It would be advantageous to operate the system under computer control so that manpower requirements may also be reduced. If these goals are to be met the tensioning and storage of rope must be considered as part of a complete system design so that the advantages of synthetics can be exploited. Lateral loads that are applied to the rope on the take-up drum must be reduced. The design and initial testing of such a tensioning system is described in this paper.

II. WINCH SYSTEM DESIGN

The primary design criteria for a tensioning system, which includes the rope pathway and the rope itself, are its reliability, safety, power, cost and size. We have chosen to reduce the working tension in the rope to less than 10% at the take-up drum in order to eliminate crushing and abrasion that can result from poor spooling, leading to unexpected rope failure.

The data in Table 1 demonstrate the reduction in life that can occur when a rope is spooled incorrectly on a drum at high tension. In this example two types of 3/8" diameter Kevlar ropes were cycled while wrapped around either a

Table 1: Fatigue life of 3/8" Kevlar ropes tested on traction sheave vs. drum.	
3-strand rope on drum @ 2 kips	219 cycles
3-strand rope on traction sheave @ 2.7 kips	3157 cycles
Single-jacketed rope on drum @ 1.8 kips	1240 cycles
Single-jacketed rope on traction sheave @ 3 kips	> 4000 cycles

Figure 1: 3/8" Kevlar rope wrapped around a 5.25" diameter drum to simulate poor spooling. An order of magnitude reduction in fatigue life was observed when tested in this configuration.

drum or a traction sheave. When a rope is wound so that it crosses itself, as shown in Figure 1, there can be an order of magnitude reduction in life.

The system's safety and reliability can be further improved by using several traction sheaves, independently driven by AC motors, in what is known as an n+1 system. In such a system, as illustrated in the figure within Table 2, there is sufficient power available to maintain normal operation when one of the motors driving the traction sheaves has failed. While using multiple motors leads to more frequent repairs these can be done at the operator's convenience and the system's safety and utilization are improved.

If the system comprises several sheaves, with large wrap angles, the traction can be applied to the rope over a very large area, reducing shear stress within the rope and minimizing internal wear due to relative motion of rope components. The winch drum and traction sheaves may be driven independently using AC induction motors under flux vector frequency control so that changes in the rope's length as it passes through the system can be accommodated without introducing slippage at the sheave surface. The use of these controllers can also give great flexibility to the system so that it may be used to maintain a constant tension, as required for the STREAM highline, or to run at a specified velocity profile, making it suitable for winches and lifts.

When used as a constant tension device torque is read directly at each sheave and fed back to its controller, which responds by increasing or decreasing its speed in order to maintain the correct tension in the rope sections immediately before and after the sheave. When a velocity profile is required one of the motors, for example the one controlling the take-up drum, is driven as a displacement device; the resulting torque at the drum can be measured, scaled and used as the input to the other motors.

The behavior of each motor and sheave is complex. For each pair a dynamic, electro-mechanical system is set up whose elements include the stiffness of each rope section, the inertia of the sheave and motor, the motor inductance, resistive losses and frictional losses in the gear train, rope and elsewhere. For a sheave in the middle of the traction system these sections of rope are very short and have an extremely high stiffness. The details of the feedback network are beyond the scope of this paper.

In order to achieve the stated reduction in tension of 10 at the take-up drum each of the sheaves in a 3+1 system must be capable of increasing the tension of the rope by a factor of $\sqrt[3]{10}$ (about 2.15). Assuming a wrap-angle θ of 1.25π radians (225°) and using the well-known expression $T_2 = T_1 e^{\theta\mu}$, the coefficient of friction μ must be at least 0.20. It may be necessary to use a surface treatment to the traction sheave if this minimum friction coefficient is to be met, depending on the material used in the rope.

The power required at each traction sheave for an operating tension of 60 kips is shown in Table 2. When all sheaves

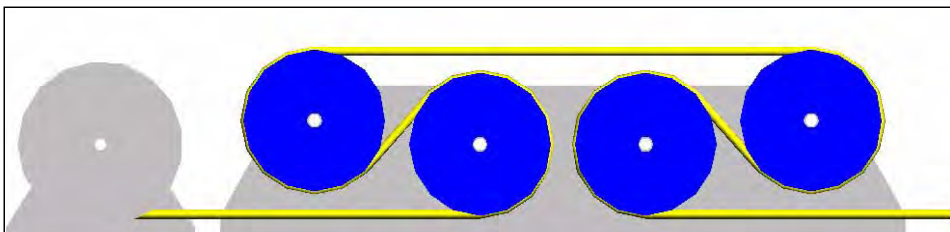


Table 2: Tension applied at each traction sheave in a 3+1, 60 kip system. Motor power, operating speeds and sheave sizes are chosen to meet the maximum traction requirements.

	in	Sheave 1	out / in	Sheave 2	out / in	Sheave 3	out / in	Sheave 4	out
All four sheaves	6	+4.7	10.7	+8.3	19.0	+14.8	33.8	+26.2	60
First sheave failed	6	+0	6	+6.9	12.9	+14.9	27.8	+ 32.2	60
Last sheave failed	6	+6.9	12.9	+14.9	27.8	+32.2	60	+0	60
Maximum traction applied (lb_f)		6.9		14.9		32.2		32.2	

are operating the traction that must be applied varies from 4.7 kips at sheave 1 to 26.2 kips at sheave 4. When one of the sheaves fails the traction on the remaining sheaves varies from 6.9 to 32.2 kips. Sheaves 3 and 4 must be designed to apply 32.2 kips while sheaves 1 and 2 must be capable of applying 6.9 and 14.9 kips respectively. For a system that is designed to operate at 8 feet/second, with a power loss of 15% within the transmission at each sheave, sheaves three and four would require 550 HP. Assuming that an 1800 RPM motor is to be selected this would require a 3-foot diameter sheave and an approximately 35:1 gear reducing unit.

An example of a motor that meets these requirements is the RPM AC Inverter Duty motor available from Reliance Electric. The 600 HP version of this motor supplies 1800 ft-lbs of torque and occupies about 8 ft^2 of deck space. Gear reducers for this motor are available from Dodge Power Transmission as well as other vendors.

There is some confusion in the way that the industry states the torque characteristics of variable frequency motors. The industry standard is to call this capability “1000:1 Constant Torque” although some manufacturers identify this capability as 2000:1, 3000:1, infinity:1, etc. All of these terms describe the same capability of the AC motor, which is designed to operate continuously from zero to its rated speed with full load current and torque and without exceeding the specified temperature rise. This ability to maintain full torque continuously at zero speed is essential for the highline or any hoist applications as is the ability to operate with equal efficiency in both rotational directions.

II. TESTER DESIGN

The rope tester was designed to simulate the loads that are applied to the rope by the traction sheaves of the winch system described above. Concerns about relative motion between the jacket and the rope, about shear-transfer to the rope’s core and about the onset of dynamic friction were addressed using this device. It also presented an opportunity to experiment with the variable frequency AC motor and to develop different feed-back schemes.

The tester has two sheaves that are mounted on a frame made from ½” aluminum plate assembled without welds. This construction allows the frame to be disassembled and modified so that the positions of the sheaves may be changed to accommodate wrap angles of from 90° to 225°. Sheave diameters of up to about 6.5” may be used.

Torque is applied to one sheave by an AC induction motor operating through a 64:1 gear reducer and controlled by Sensorless Flux Vector Control (SFVC). The gear-box is mounted to the tester’s base plate with its drive shaft keyed to the traction sheave. APEC-3 ball bearings are pressed into the precision-bored holes in the tester, supporting the 5/8” diameter drive shaft. A keyway in the traction sheave is used to transmit torque. A 1.5 HP, 208V 3-phase AC induction motor with a NEMA-145TC frame is mounted directly to and supported by the gear-box.

The tester, which is shown in Figure 2, has been mounted in an Instron load-frame that is used to move the free end of the rope. The Instron’s cross-head movement can be controlled to move at speeds of up to 20 inches per second and at loads of up to 5000 lb_f.

The rope is attached to the cross-head by a load-cell whose output is calibrated and amplified by the strain-conditioning unit in the Instron electronics rack. The scaled DC voltage from the strain conditioner is used to monitor the rope’s tension and to provide an analogue feedback signal to the motor controller. A computer is connected to this variable frequency controller over an RS232 bus so that commands can be sent and data recorded.

The other end of the rope, which in a winch system would be attached to the take-up drum, is attached to an air cylinder that can provide up to 2000 lb_f. This air cylinder is attached to a large accumulator, providing a constant pressure at all test speeds. A bearing can be added to the piston to prevent it from rotating when unbalanced ropes are tested.

A portion of the rope path may be seen in Figure 1. From the cross-head the rope runs directly downwards before passing beneath and around the traction sheave with a wrap angle of 180°. Running directly upwards it then passes over and around the idler sheave for just less than 180° before running downwards to the socket that is attached to the air cylinder. In the figure the idler sheave has been replaced by a drum so that poorly controlled spooling may be studied.

Figure 2: Rope tester mounted on Instron.

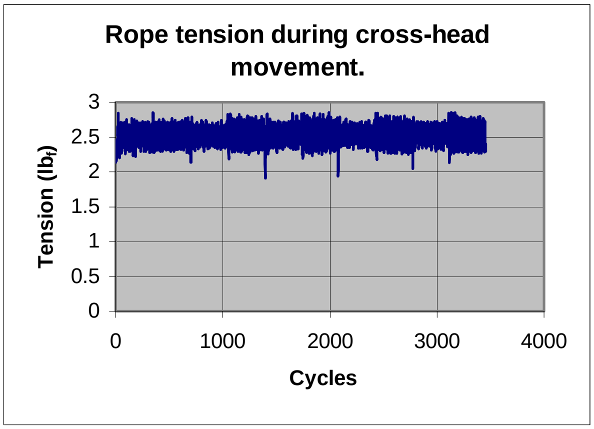


Figure 3: Variation of rope tension during cross-head movement.

Sockets were made from mild-steel with a 3" long taper of about 3°. A cone with an included angle of about 5° was used so that the gap was maintained at a near-constant volume per unit length. A West Systems, two-part epoxy was used. No socket failures have occurred.

A 3/8" rope diameter was chosen for testing and tensions were scaled to the STREAM high-line requirements, corresponding to a final tension of about 2800 lb$_f$.

Torque applied at the traction sheave was found to vary by about !10% as shown in Figure 3. This was due to the limitations of the sensorless FVC motor controller. SFVC controllers use a sophisticated model of the motor to estimate torque and speed from the strength and rotational speed of the magnetic field, rather than use a tachometer and rotary position encoder. While this is accurate at high speeds it is less accurate as the motor nears zero speed. The controller is, therefore, designed with a minimum rotational speed of 0.5 Hz and 0.1 Hz speed increments. With the gear-reducer and a 5.25" traction sheave this minimum speed was about eight linear inches per minute with increments of 1.5"/min. The relatively imprecise speed control was the principle cause of tension variations. In true FVC systems, designed to operate at zero frequency using velocity and position feedback, torque control should be greatly improved.

Although the tension control was less accurate than expected the tester nevertheless demonstrated that a constant tension could be applied to the rope regardless of the movement at its free end. It also allowed the interaction between the rope and the traction sheave to be studied.

III. TEST RESULTS

A. Fatigue Tests on Traction Sheaves

Initial tests have been conducted in order to determine the extent to which traction sheaves may reduce the life of a rope in comparison to bend-over-sheave fatigue. Of the nine rope constructions that have been obtained two types have been tested on the traction sheave and on a drum. This latter was included to demonstrate the reduction in life that occurs when a synthetic rope is tensioned on a winch drum.

Both rope constructions were 3/8" diameter, Kevlar 29 with polyester jackets. Their average breaking strength was about 12 kips. The single-jacketed rope had been developed specifically for bend-over-sheave applications and was expected to have excellent fatigue life, however, there was concern that the jacket would move as the rope was cycled across the traction sheave. The second rope consisted of three jacketed strands. It was expected that the jackets would not move on this rope. In all testing conducted so far no movement of the jacket has been observed on either type of rope.

Initial fatigue life tests were made with a tension of 1800 lbf at the air cylinder at the air cylinder and 2700 lbf at the cross-head. The three-strand rope failed, on average, at 3157 cycles while there were no failures of the single-jacketed ropes during the test duration of 4000 cycles. Longer-term fatigue tests are currently underway.

Tension tests of individual yarns were used to look for damage within ropes tested to 1000 and 4000 cycles. A capstan fixture was used to clamp yarns that had been removed from various components within the rope. Breaking strengths of untested rope samples were compared to those subjected to cycling by the traction sheave. Substantial damage could be identified in the three-strand rope as early as 1000 cycles but there was only a very limited reduction in breaking strength within the single-jacketed rope, even at 4000 cycles. These data are collected in Table 3. Fibers from yarns that were adjacent to those tested were examined using SEM. Some of these were found to be crushed or to exhibit fibrillation. An example is shown in Figure 4.

It still remains to determined whether the fatigue life of a rope differs when it is cycled over a traction sheave or over an idler at the same nominal tension. Early data suggest that there is little, if any, difference.

B. Friction Measurements

A control program was used to conduct measurements of

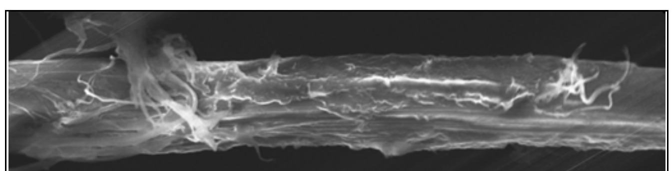


Figure 4: SEM image of a fiber showing crushing near its center.

Table 3: Results of yarn tests on rope samples. Rope types: A = Three, polyester-jacketed Kevlar strands. B = Single polyester-jacketed Kevlar rope.

Rope	Strand location	Cycles	Residual strength, % of initial		
			avg.	⑨	samples
A	N/A	1k	37.7	16.5	3
A	N/A	4k	No surviving ropes		
B	outer	1k	89.1	8.0	7
B	outer	4k	85.3	7.5	6
B	core	1k	No statistical difference		
B	core	4k	97.3	2.1	5

static and sliding friction coefficients on all rope samples. The cross-head was set to move at a different speed than the traction sheave so that tension at the cross-head was gradually increased. Rope tension was recorded until static friction was overcome and slippage occurred. Under some conditions a slip-stick mechanism could be established during rope cycling. During initial tests on the three-strand rope at 2700 lbf variations in tension caused the static coefficient of friction to be exceeded, inducing slip-stick, which then continued until the rope broke. These ropes failed, on average, in less than 300 cycles. Optimization of coefficients within the motor feedback network eliminated this phenomenon and increased the average fatigue life of the rope to over 3000 cycles.

It is obvious that any system using traction sheaves for rope tensioning must be designed to accommodate the friction between the rope and sheave surface at all times. The effects of age, wear, temperature and the presence of contaminants such as dirt, grease or water must all be exhaustively tested before such a system can be deployed.

C. Free vs. Fixed End Conditions

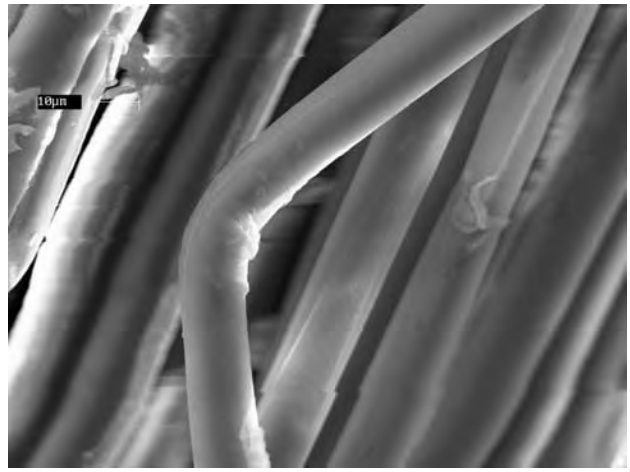
For ropes used in hoists, where one end is free to rotate, any imbalance in the construction can lead to early failures in bend-over-sheave applications. This was demonstrated by allowing the piston in the air cylinder to rotate during cycling of a 3/8" diameter, right-hand-lay, unbalanced rope at 2000 lb_f tension. Each rope was cycled 300 times. It was then dissected and yarns were removed from the region that had traversed the 5.25" diameter idler sheave. Residual strength was compared to yarns from identical ropes that had been tested with the rotation constrained.

Results from this testing are shown in Table 4 where the residual strengths are presented as a percent of the strength of uncycled specimens. There has been much more damage to the fibers of the outer strands of the rope that is free to rotate than in the rope that has had its end fixed. Conversely, the fibers within the core of the rope that has been allowed to rotate are much less damaged than those in the rope that has had its end fixed.

The damage that occurs in the outer fibers when rotation is allowed may be easily explained. Tension on the unbalanced rope causes it to rotate, relieving the strain in the outer strands, and transferring tension to the core. As the rope traverses the sheave the portions of the outer strands that are in contact with its surface are compressed. This local strain is superimposed on the global, average tensile strain carried by the strand. Normally this results in only a diminution of the global tensile strain, however, the rotation of the rope has reduced the global, average strain to the point where the local strain puts some fibers into compression. Kevlar and other synthetic fibers fail at a much lower strain in compression than in tension. In addition, buckling of fibers can create extremely high local strains. This is evidenced by the SEM image in Figure 5 showing buckling and compression bands within a fiber taken from one of the samples.

Table 4: Effect of end rotation on an unbalanced rope in bend-over-sheave testing.

Strand location	End condition	Residual strength, % of initial avg.	C.O.V.	samples
outer	free	33	8.0	8
core	free	90	12	8
outer	fixed	59	7.9	3
core	fixed	77	22	3



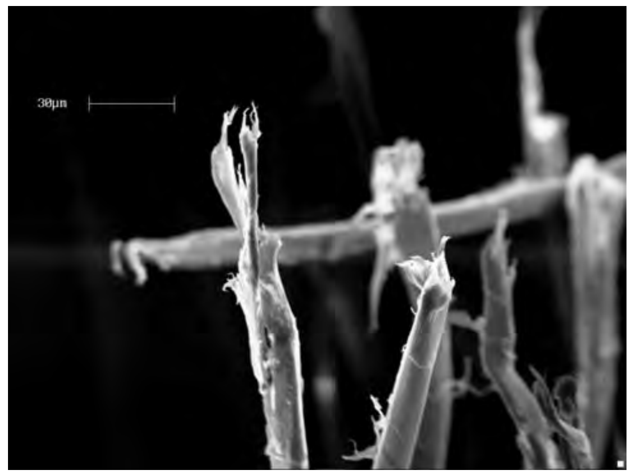


Figure 5: Bent fiber from an outer strand showing compression bands (above) and a fiber from the core, broken in tension, with no sign of fibrillation (below) during bend-over-sheave testing of an unbalanced rope with one end free to rotate.

Some rope specimens tested with rotation at the free end broke before 300 cycles had been completed. Those specimens were not included in the test results but were examined using SEM. The lower image in Figure 5 shows fibers from the core of one of these ropes. There is very little fibrillation, indicating that a sudden, tensile fracture has occurred. Our conclusion is that compression failures accumulate in the fibers of the outer strands as the rope traverses the sheave. These failures quickly reduce the ability of the outer strands to sustain any tensile load as the rope moves off the sheave, leading to abrupt tensile failure of the outer strands and, finally, the core. Inspection of the failed ropes supports this supposition. Outer strands failed at different positions that can be traced to a point at which each was in contact with the sheave and presumably in compression.

It is more difficult to understand the residual strength of the core with end rotation. Unloading of the outer fibers would be expected to significantly increase the tension in the core with the result that its strength would also be degraded. It is our contention that another mechanism explains the results: the unloading of the outer strands not only transfers tension into the core but also reduces transverse forces that, along with internal movement, cause fibrillation. Thus, the core appears to be undamaged until the outer strands fail, transferring all load to the core, which breaks suddenly in tension. When end-rotation is fixed there is no unloading, compressive failures do not occur in the outer strands and fibrillation of both the core and outer strands eventually leads to failure. Further testing is in progress to better understand this failure mechanism.

These results have some important implications for ropes used with traction sheaves and, most likely, for dual capstan devices as well. In developing ropes optimized for bend-over-sheave applications low-friction coatings have been used. If this allows the outer strands of a rope to slide along the core as it passes a traction sheave it may create compressive fiber failures. The traction applied to the rope's surface must, therefore, be limited and the rope's structure must be considered as an integral part of the overall system.

IV. CONCLUSION

A tensioning system has been designed specifically for use with synthetic ropes. It uses AC motors under Flux Vector Control to drive a series of traction sheaves, allowing complete flexibility in rope velocity or tension profiles. Slippage and shear stresses in the rope are minimized by applying traction over a large surface area. By using an n+1 design reliability and safety are enhanced. Rope tension on the storage drum can be reduced by an order of magnitude, eliminating the likelihood of rope failure due to poor spooling or the accumulation of crushing force.

In order to determine the performance of such a system a rope tester has been designed and built. Initial tests show that tension can be applied to the rope by the traction sheaves without inducing early failure. Tension control is currently maintained within !10% although a more sophisticated device, using a position encoder and FVC, will dramatically reduce this variation.

Static and dynamic friction coefficients have been measured between two sample ropes and the polished, stainless-steel traction sheave. The effect of contaminants, wear, temperature and other factors remain to be examined.

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