CASE STUDY
ANALYSIS OF PROCESS FAN FAILURE AND BEARING HOUSING/SHAFT DESIGN

Ken Singleton

Manager
KSC Consulting LLC, Bristol VA
ksingleton@vibrationconsulting.com

Alan Barbee
Reliability Engineer
Fortron Industries
Wilmington NC

Bob Bracher
ATK Maintenance Engineering Assistant
Radford VA

Bob McGinnis, P.E.
(Consulting to KSC)
McGinnis Engineering LLC, Kingsport TN

Abstract: An Air Movement and Control Association (AMCA) Arrangement 8 fan handling hot process gas experienced several bearing failures and ultimately a catastrophic failure requiring shutdown of the plant's production. The investigation found that the 1st critical of the fan rotor was too close to the operating speed resulting in excessive bearing loads and short bearing life. Rotor dynamic modeling was used to evaluate a new shaft design. A new shaft and one-piece bearing housing were designed, fabricated and installed, successfully moving the critical speed several hundred RPM above running speed. The new design also provided improved maintainability over commercially available designs, as well as lowering vibration and significantly improving reliability. This article discusses the analysis process, rotor modeling, bearing housing design process, vibration data comparison before and after the new shaft and bearing housing were installed.

Keywords: Bearing housing design; critical speed; failure analysis; overhung fan; shaft design; rotor dynamic analysis.

Background: The AMCA Arrangement 8 fan, shown in **Figure 1**, was designed for 450°F gas temperature. Operating speed was 1785 RPM, direct driven by a 200 HP motor using a Falk Grid type coupling with 7¼" spacer. The bearings were spherical roller 22218 CCK C3 fit lubricated with Mobilith AW-2 grease. The fan wheel was constructed of Avesta 2204, continuously welded with a 304L SS hub. The wheel hub fit to the shaft was with a tapered interference fit, set screws and an end cap. A Durametallic Triple Carbon Ring seal provided sealing at the shaft entry to the fan housing. The fan base was fabricated of A36 HRS (hot rolled steel). The fan housing was constructed of 304 SS. Fan performance was 34,200 CFM, SP 22.6", and operating inlet temperature of 365°F to 392°F.

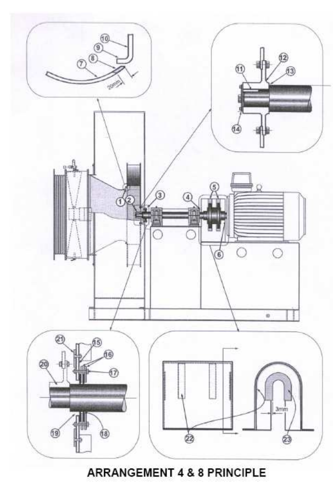

Figure 1. AMCA Arrangement 8 Fan Drawings.

Fan Reliability: Initial involvement on the project for KSC was providing assistance improving reliability of the fans. The plant had nine process fans and improving their reliability was a priority. One improvement was replacing the bearing support plates on some of the fans with 1-3/4" thick plates machined flat to 0.002 inch/ft.

Bearing reliability and elevated vibration levels continued to be the primary concerns for this particular fan. Analysis of the vibration data identified the need for higher resolution spectra to "see" the rotor critical. The fan rotor 1st critical is indicated at 1965 CPM in the data shown in **Figure 2**. The sidebands about 1X and 2X run speed are the difference frequency between rotor 1X and the rotor 1st critical. Amplitude modulation by the fan rotor 1st critical frequency is apparent in the data. The 1st critical frequency would shift higher and lower depending on the condition of the bearings. New bearings, having tighter clearance and higher direct stiffness, would shift the critical speed higher. It was found that by monitoring the frequency of the 1st critical that as the bearings wore the critical speed would move closer to the run speed frequency. Vibration levels would begin to rapidly increase as the critical speed envelope enclosed the running speed frequency range. Monitoring the frequency of the 1st critical <u>provided another indicator when to replace the bearings.</u>

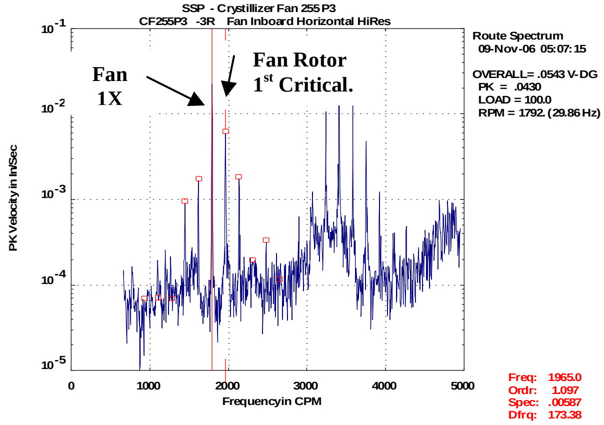

Figure 2. Spectrum at Fan Inboard Brg Horizontal Shows Fan Rotor 1st Critical at 1965 CPM, 173 CPM Above Running Speed. Sidebands About 1X and 2X Run Speed are Modulated by the Fan Rotor 1st Critical Frequency.

Fan Catastrophic Failure: On January 5th, 2006 the fan vibration levels were rapidly increasing. Vibration data was measured several times that day with the last data measured about 5:00PM. That evening the fan failed catastrophically. Fortunately no one was near the machine at that time.

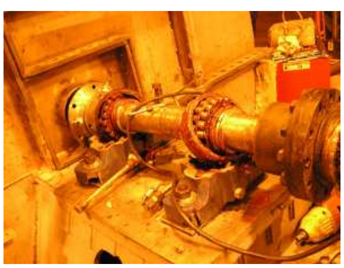

Figure 3. Image Taken Shortly After Fan Failure.

Figure 4. Fan Shaft and Wheel in Shop After Failure.

Figure 5. Mechanical Seal Housing.

One of the first images taken of the fan shortly after the failure is shown in **Figure 3**. Both bearing housings (cast iron) fractured. The fan shaft had a significant permanent deformation (bend) in the area of the mechanical seal, see **Figure 4**. The shaft at the mechanical seal area, see **Figure 5**, showed heavy rub part way around the circumference of the shaft.

Figure 6. Fan Shaft and Fan Hub. Shaft Heavy Rub Against Fan Wheel.

Figure 7. All Fan Wheel Hubs Failed.

Heaviest rubbing had occurred near the bottom of the seal housing. The shaft also exhibited heavy rubbing by the fan wheel, see **Figure 6**, after all of the wheel hub bolts failed, see **Figure 7**. The shaft rub locations and the bend in the shaft suggested a significant thermal bow in the shaft may have developed due to rubbing at the seal.

Two vibration spectra and time waveforms measured shortly before the failure are shown in **Figure 8 & 9**. The bearing housing vibration does not show a pure sinusoidal response but a 1X & 2X response. The shaft was likely contacting the seal housing when these data were taken. The bearing housing hold down bolts may also have loosened.

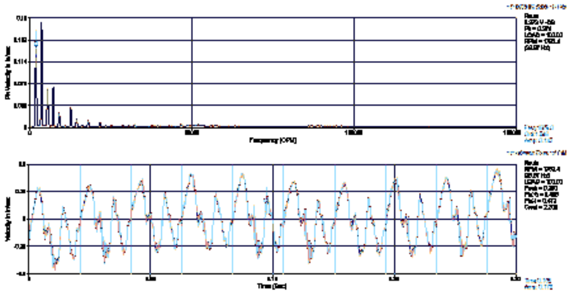

Figure 8. Fan Coupling End Bearing Housing, Horizontal.

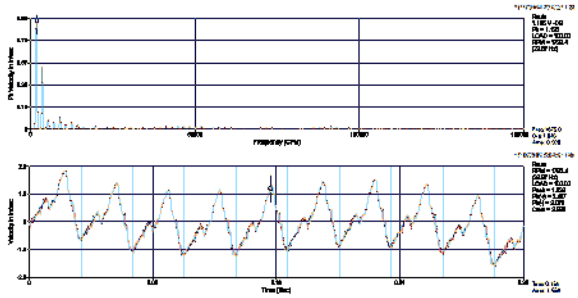

Figure 9. Fan Wheel End Bearing Housing, Horizontal.

Near Catastrophic Failure: The fan was returned to service after replacing the bearings, shaft, fan wheel and mechanical seal. Reliability of the bearings was a continuing problem. A near catastrophic failure occurred November 6, 2006. The fan was disassembled and physical inspection was made of the rotor and bearings. To include all stake holders in the investigation and resolution of the fan reliability problems, the client's reliability engineer had assembled a fan reliability improvement team that included the reliability engineer (team leader), mechanics, operators, maintenance planner and the vibration analyst.

Figure 10. Fan End Bearing Nov 6, 2006.

The wheel end bearing was an FAG 22218 ESK C4 with a steel double cage. The grease was very dark indicating contamination, as shown in **Figure 10**. The bearing outer race and housing bore exhibited severe fretting corrosion. The bearing cage at the wheel end exhibited axial wear from the rollers rubbing. The cage bore exhibited rubbing on the inner race. The inner race had guide grooves machined for the seals.

The coupling end bearing was also an FAG 22218 ESK C4 with a steel double cage. This bearing was the fixed bearing taking the thrust load. The thrust should have been carried by the higher radially loaded wheel end bearing. The grease was very dark indicating contamination. The bearing outer race and housing bore exhibited severe fretting corrosion primarily in the top half of the housing. Static load on the bearing would have been vertically up due to the overhung load of the wheel. The bearing cage exhibited axial wear from roller rubbing. Rubbing had also occurred in the bore of the cage against the bearing inner race. The inner race also had guide grooves machined for the seals.

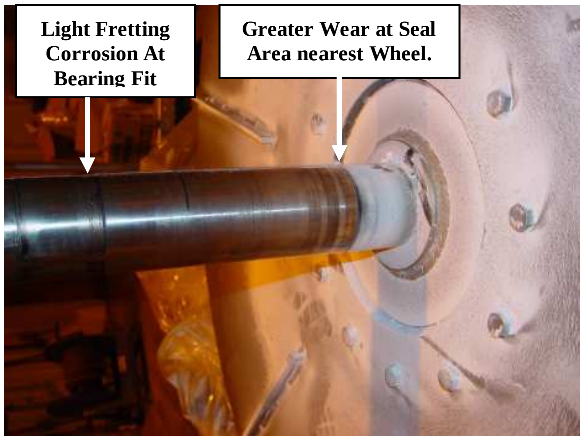

Figure 11. Fan Rotor Removed From Service.

The fan shaft exhibited lite fretting corrosion at the heavier loaded wheel end bearing fit, as shown in **Figure 11**. There was also lite wear at the mechanical seal area and the seal area exhibited melted material (product in the air stream). The reported melt temperature was 230 Deg C ~ 446 Deg F. There were no indications of wheel rub or shaft cracks.

The bearings were sent to SKF for analysis and their comments were as follows:

- Classic description of high unbalance load.
- The cage acts like a hula-hoop orbiting in the bearing about the inner race. The cage bore is very difficult to lubricate and thus begins to wear.
- Consider a bearing with a hardened steel cage as an option – 222218 EK.
- The fretting corrosion problem is caused by relative motion of the outer race and housing bore. If a groove is worn only a few thousands deep in the housing, the outer race can hang and not float axially. If this happens the bearing temperature will rise.
- Consider a four bolt housing FSAF 22518.
- Pay attention to color of the grease expelled from the seals. Dark color indicates contamination.

Vibration data was measured on the fan inboard bearing housing just prior to the November 5, 2006 event, see **Figure 12**, and Nov 11, 2006 after replacing the bearings, see **Figure 13**. The data shows the rotor 1st critical shifting about 120 CPM higher due to new bearings with no wear having higher direct stiffness.

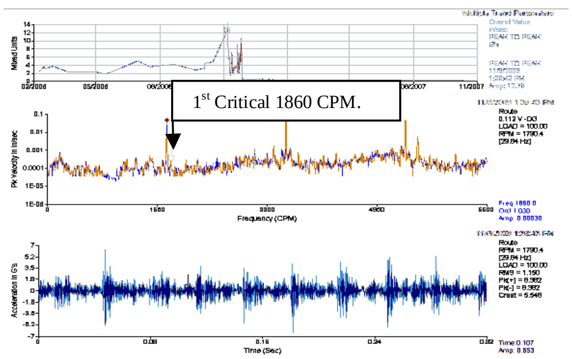

Figure 12. Fan Inboard Bearing Horizontal, High Resolution Spectrum, Just Before Shutdown November 5, 2006.

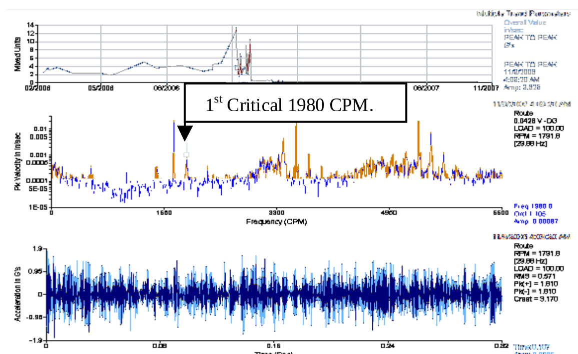

Figure 13. Fan Inboard Bearing Horizontal, High Resolution Spectrum, Rotor & Bearings Changed After Nov 5, 2006 Outage. Rotor 1st Critical Shifted to 1980 CPM.

During this time period several changes were made including replacing the bearing hold down bolts with grade 8 bolts and washers, different grease lubrication was tried and bearing clearance was increased

Fan Rotor-Bearing Dynamic Model: The rotor was initially modeled in November 2006 using finite element rotor modeling software. The OEM shaft was 3.187 inch straight shafting. The 1st critical calculated for the original shaft material of 304SS to just above the fan run speed range as shown by the undamped critical speed map in **Figure 14**. Bearing stiffness was estimated to be about 235,000 lb_f/in.

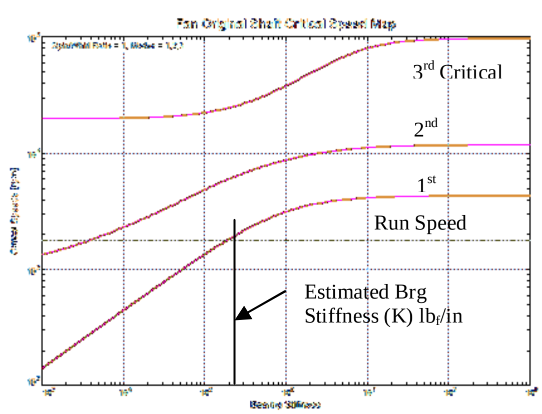

Figure 14. Undamped Critical Speed Map for The Original 3-3/16" Fan Rotor.

During discussion about machining a replacement shaft, there were questions about the shaft material to use. The rotor model was run for shaft material properties of 304, 316L, 17-4 H 1025 and 4140. The wheel weight was determined by measurement and the wheel inertial properties were provided by the fan OEM. The coupling weight, bearing locations and shaft dimensions were obtained from an OEM drawing. Calculations were made to determine the following:

- Rotor 1st critical speed.
- Rotor static deflection.
- Bearing reaction loads.
- Rotor response to unbalance at the fan wheel center of gravity.
- Undamped critical speed map of rotor versus bearing stiffness.
- Maximum static shaft stress in bending.
- Effect on rotor critical speed with 17-4 shaft diameter increased from 3.187 to 3.25 inch.
- Effect on rotor critical speed with 17-4 shaft diameter increased from 3.187 to 4.00 inch.
- Residual static and couple unbalance tolerances for Balance Quality Grade G2.5 calculated per ISO 1940-1.

As expected, the different modulus of elasticity for each shaft material had little effect on the calculated shaft 1st critical speed. Material 4140 had the highest modulus of elasticity and calculated to have a critical speed about 50 RPM higher than the stainless steel materials.

Shaft diameter changes were evaluated for increasing the 1st critical speed. Shifting the wheel end bearing closer to the fan wheel was also considered. Increasing the shaft diameter from 3.187 to 4.00 inch at all locations except the coupling hub and the tapered wheel hub fit and using 17-4 SS was predicted to raise the 1st critical speed 30%. This appeared to be an effective option to shift the critical speed and increase shaft stiffness but would require lowering the bearing support plate to accommodate larger bearings and housings.

New Rotor and Bearing Housing Design Development: After evaluating the options to increase the shaft size to move the 1st critical speed higher and the bearing and support plate changes, a decision was made to use a one piece bearing housing. Quotes were requested from a well known bearing supplier for a tunnel type bearing and shaft assembly. Delivery was several months since these bearing housings are not a stock item and are manufactured in Europe. Additionally, the shaft options offered by the bearing supplier did not provide the larger diameter at the fan end that was needed to shift the critical speed higher. In addition, the tunnel bearing housing centerline height was different than the SAF Pillow Block housings which would require lowering the bearing support plate.

The client's reliability engineer and the fan reliability improvement team evaluated all these options. After following a very methodical process, a decision was made for KSC to design the bearing housing and shaft assembly and to use a local machine shop for fabrication of the shaft and bearing housing assembly. The client would review and approve the design and drawings.

The team wanted the bearing housing to be designed for direct swap out of the existing pillow block bearings, something the off the shelf bearing housing design would not accommodate. This meant that the shaft centerline height and bolt pattern would match the existing SAF 218 bearing housings and the fan's position axially within the fan housing had to be maintained. The design would provide a backup plan if the new bearing housing and shaft design did not work and it was necessary to reinstall the original shaft and bearing housings. The team wanted all of the components to be assembled from one end, so that the fan wheel did not have to be removed from the shaft

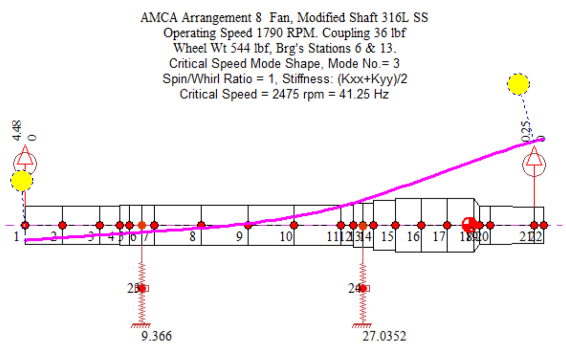

Figure 15. Model of Final Shaft Design.

during bearing replacement. Removing the fan wheel from the shaft was a major, time consuming job. These details were incorporated into the bearing housing and shaft design as well as other features to optimize lubrication, vibration analysis, and temperature monitoring.

Over several weeks the design was developed and refined per input from the client's reliability team. The rotor final design model with the 1st critical mode shape is shown in **Figure 15**. The 1st critical was predicted at about 2475 RPM providing about 27.7% separation from run speed. Rotor weight and bearing reaction loading calculated as follows:

Shaft wt:	115.35 lb_f
Total Rotor wt:	694.35 lb_f
Coupling end bearing reaction:	-333 lb_f
Fan end bearing reaction:	1,027 lb_f

The damped unbalanced response for the wheel end bearing is shown in **Figure 16** and the 3D plot of the rotor displacement in mils pk-pk at run speed is shown in **Figure 17**.

Rotor Bending Stresses: The team leader asked for the maximum bending stresses in the fan rotor and the location of those stresses. The stresses were calculated using DyRoBeS [3,4] rotor modeling software for the static bending stresses in the original rotor, see **Figure 18**, and the redesigned rotor **Figure 19**. However, additional calculations must be made to evaluate if these stresses are acceptable.

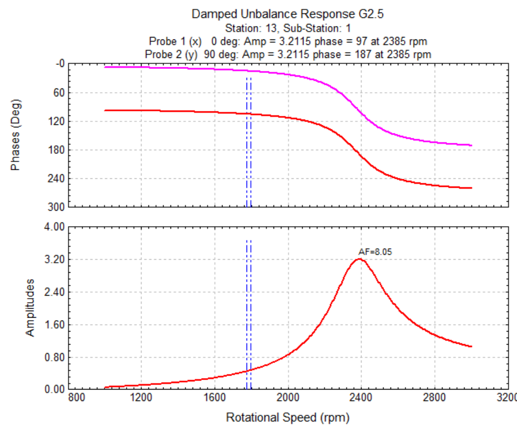

Figure 16. Undamped Critical Speed Map for The Original 3-3/16" Fan Rotor.

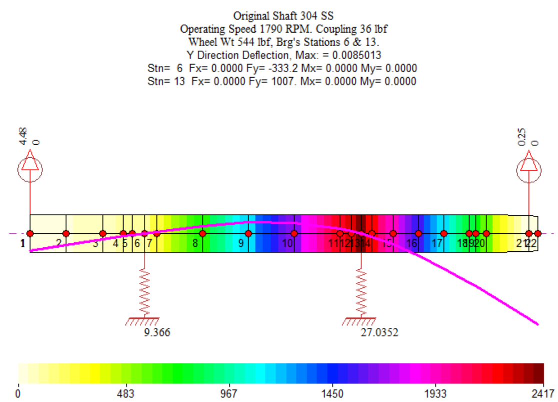

Figure 18. Static Bending Stresses, OEM Shaft 304SS, Maximum 2417 psi at Wheel End Bearing.

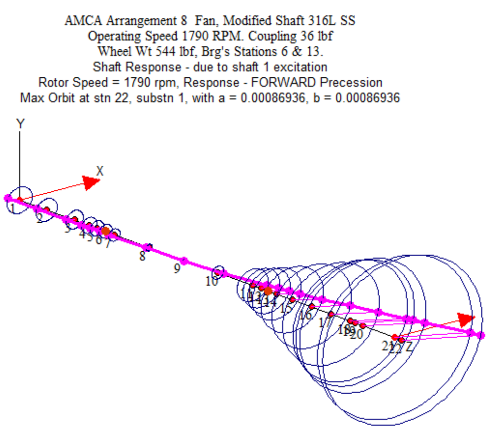

Figure 17. Shaft Damped Unbalanced Response to G2.5 Balance Quality Grade at Run Speed.

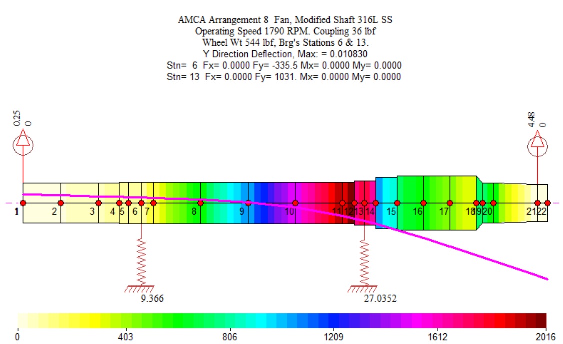

Figure 19. Static Bending Stresses, New Shaft Design, 316l SS, Maximum 2016 psi Inboard Wheel End Brg.

According to [2], to allow for lack of complete design information, design texts, handbooks, and engineering codes recommend use of safety factors and service factors. The design of machine members is dominated by the concept of allowable working stress. This stress level which should not be exceeded is a fraction of either the ultimate, yield or fatigue strength of the material. For static loading, yield or ultimate strength is needed. Cyclic loading is called material fatigue. The transition in strength from static to cyclic loading strength drops to 90% of the initial static value after about 1000 cycles and to about 50% after 1 million cycles then the curve is fairly flat thereafter. The derating factors [2] used to determine the allowable working stress in bending for the redesigned shaft using 316 SS are as follows:

- S_e= Endurance limit/fatigue limit of machine component
- S'_e=Endurance limit/fatigue limit of a rotating beam specimen of the same material.
- k_a = Surface finish derating factor. Polished surfaces have lower derating factor than rough machine surface.
- k_b = Size finish derating factor.
- k_c = Reliability derating factor.
- k_d = Temperature derating factor.
- k_e = Stress concentration derating factor.
- k_f = Impact derating factor.

$$k_a = \frac{k_a S'_e}{S'_e} = \frac{55{,}000}{0.5 S_{ut}} = \frac{35{,}000}{0.5 \bullet 200{,}000} = 0.35 \qquad k_b = 0.75 \quad \text{Where } d > 2.0\,\text{in}$$

$$k_c = 0.702 \text{ Where Reliability} = 99.99\% \qquad S'_e = \frac{35{,}000}{0.6} = 58{,}333$$

$$k_d = \frac{620}{460 + T} = \frac{620}{460 + 100} = 1.1 \qquad k_a S'_e = 35{,}000\,\text{psi} \quad \text{From Chart Pg 3.22 Ref 2}$$

$$\text{Where } k_f = \text{Notch Sensitivity}$$

$$k_e = \frac{1}{k_f} = \frac{1}{1.86} = 0.54 \qquad \begin{aligned} g &= 60 \text{ From Chart 3-15, Pg 60 Ref 2} \\ k_f &= 1 + g(k_t - 1) = 1 + 0.86(1-1) = 1.86 \end{aligned}$$

$$S_e = k_a k_b k_c k_d k_e k_f S'_e \qquad S_e = 0.35 \bullet 0.75 \bullet 0.702 \bullet 1.1 \bullet 0.54 \bullet 1.86 \bullet 35{,}0000 = 7{,}126\,\text{psi}$$

The calculated allowable working stress of 7,126 psi was greater than the values allowed by MIL STD 167 [6]. The factor of safety is the allowable working stress divided by the stresses in bending which calculated to 3.5. A factor of 2 it typically recommended.

$$\text{Factor of Safety} = \frac{7{,}126}{2{,}016} = 3.5$$

Bearing Housing Design Process: The bearing housing and shaft assembly, shown in **Figure 20**, were the end result of the design process. The bearing housing body was fabricated from mechanical tubing. The mounting feet were fabricated of mild steel and welded to the main body. The bottoms of the feet were machined flat to within 0.001” and coplanar to within 0.0015”. Dowel holes were provided in the feet for accurate location of the bearing housing when mounted to the fan pedestal. The bearing bores were held to guidelines published by SKF for size, concentricity, run-out and perpendicularity. Two grease purge holes were drilled and tapped ½”-14 NPT, centered between bearings and positioned 45° off axis below center. Three ¼”-28 tapped holes with spot facing were provided (1) top vertically and (2) horizontally each side for the mounting of accelerometers directly over each bearing. One ¼”-18 NPT tapped hole was provided at each bearing for a temperature probe.

A labyrinth ring was attached to the fan end of the bearing housing by bolts, as shown in **Figure 21**. The labyrinth provided a non-contact seal. The design maintained 0.010” radial clearance between the minor diameter of the ring and the outside diameter of the shaft. The labyrinth ring bore was profiled to form three recessed grooves. These recessed grooves provided a non-contacting grease seal between the outside contaminates and the bearing.

The seal ring was made from 6061 aluminum to prevent damage to the shaft if a rub occurred against the shaft. A V-Ring Dirt Excluder was added as a supplemental seal to repel dust. The V-Ring fit snuggly over the shaft and sealed against the outer face of the static labyrinth ring.

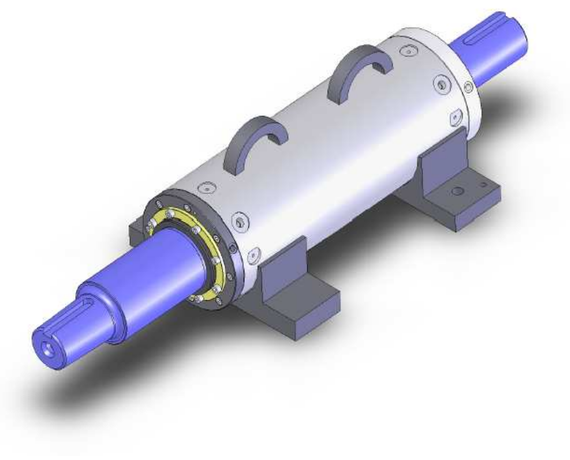

Figure 20. 3D Model of Bearing Housing and Shaft.

Two grooves were machined in the inner diameter of the end cap to serve as a type of labyrinth seal, as shown in **Figure 22**. A V-Ring Dirt Excluder (14) was also provided as a supplemental seal. The grooves in the end cap were sized so that the bore could be machined if required, removing the grooves, to receive a 90 x 110 lip seal. This provided for use of a lip seal if the labyrinth seal and grease proved inadequate. A seal path or sleeve was installed with a shrink fit to the O.D. of the shaft to provide a wear surface for the lip seal. A seal path provides a means of renewing the shaft seal contacting surface without having to reclaim the shaft to remove grooving. Adequate hardening along with <u>plunge grinding the surface</u> of the seal path offers a superior surface, extending the life of the seal.

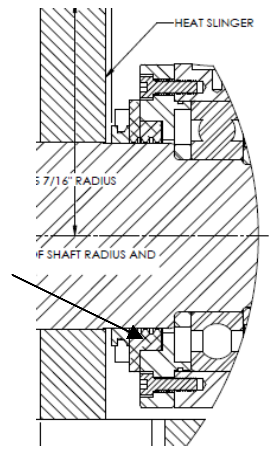

Figure 21. Seal at Fan End of Brg Housing.

End Caps: The end caps were tapped for 1/8”-27 npt for grease lines, see **Figure 23**. Also provided were ¼”-28 holes on the face of the end cap for the mounting of accelerometers to measure axial vibration.

Grease Dams: Grease dams were added inboard of each bearing to retain or hold grease close to the rolling elements due to the open space in the housing between each bearing, **Figures 24 & 25**.

Sleeve: In order to facilitate removable of the shaft assembly from one end as requested by the fan reliability team, a spacer sleeve was used between the bearings, as shown in **Figure 26**. A close fit between the I.D. of the sleeve and the O.D. of the shaft was used to minimize residual unbalance affects.

The stack-up tolerance between bearings was maintained by an SKF lock nut, KMFE style with locking screw.

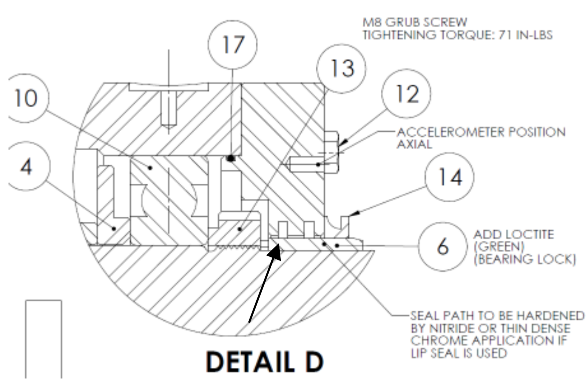

Figure 22. Detail of Coupling End of Bearing Housing Showing Seal and Seal Path.

Drawings of the bearing housing and shaft assembly are shown in **Figures 27**.

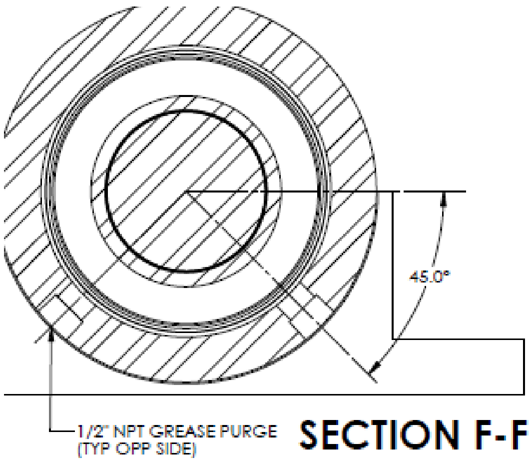

Figure 23. End Cap Grease Purge Holes.

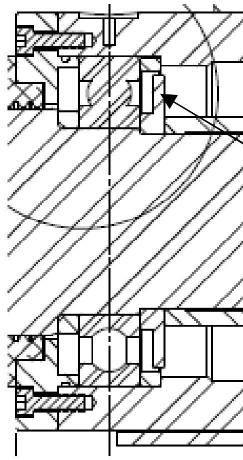

Figure 24. Grease Dam at Fan end of Brg Housing.

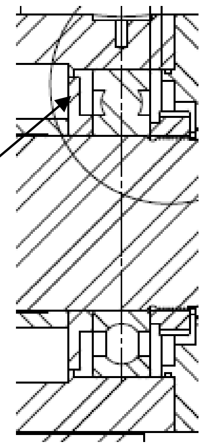

Figure 25. Grease Dam at Fan End of Brg Housing.

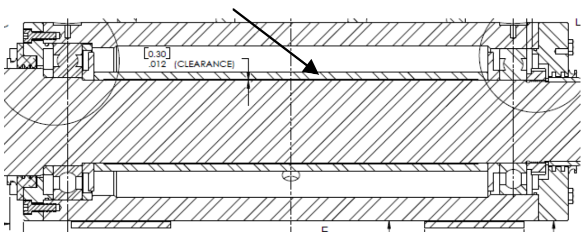

Figure 26. Bearing Spacer Sleeve.

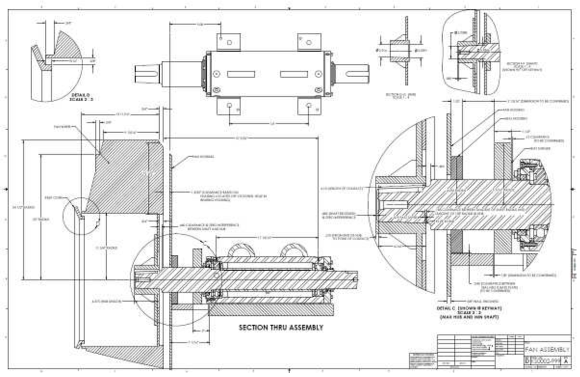

Figure 27. Bearing Housing, Shaft, Fan and Bearing Support Plate Drawing.

Fan Wheel/Hub Mounting: The fan wheel hub and shaft used a tapered fit. The amount of advance controlled the amount of interference fit and the compressive forces of the hub to shaft. Calculations were made for the following:

- Hub material 304 SS per the fan OEM drawings.
- Two shaft materials were considered, 316 SS and 17-4 PH Condition 1025

The fan hub and shaft used a tapered fit of 0.750 in/ft per the fan OEM drawings, **Figure 28**. The shaft, hub and wheel material properties are listed in **Table I**.

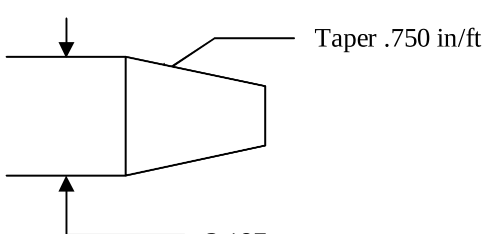

Figure 28. Shaft Taper Fit Dimensions

Table I: Material Properties Used For Calculations

		Modulus @ 400 Deg F	2% Yield Strength psi, min	Coefficient Thermal Expansion
Shaft Material	17-4 1025	27.0E6 psi	145,000	6.1E-6 /Deg F
Hub Material	304 SS	27.9E6 psi	30,000	9.0E-6 /Deg F
Shaft Material	316 SS	28.1E6 psi	30,000	9.0E-6 /Deg F
Wheel Material	RA2205	26.6E6 psi	65,000	7.81E-6 /Deg F

The team asked about the advantages and disadvantages of a straight fit versus a tapered fit. A comparison was provided as shown in **Table II**.

Table II: Advantages/Disadvantages Straight & Tapered Fits. [1]

Advantages of a taper fit:	• The contact area between bore and shaft can be checked • The interference between hub and shaft is controllable by the amount of advance of the hub on its shaft • The removal of hubs is easier
Disadvantage of a taper fit:	• It is more difficult to accurately machine the tapers of the hub bore and the shaft. • It is possible to overstress the hub if it is advanced too far on the shaft taper. • Dirt and surface imperfections restrain hub advance, and can give the false impression that the desired interference was reached.
Advantages of Straight fit:	• A straight fit is more easily machined and measured.
Disadvantages of a straight fit:	• Assembly without galling is difficult. • Disassembly without galling is even more difficult. • Maintaining zero clearance between shaft and hub is difficult. • Locking the hub to the shaft requires a step in the shaft and creates a stress riser.

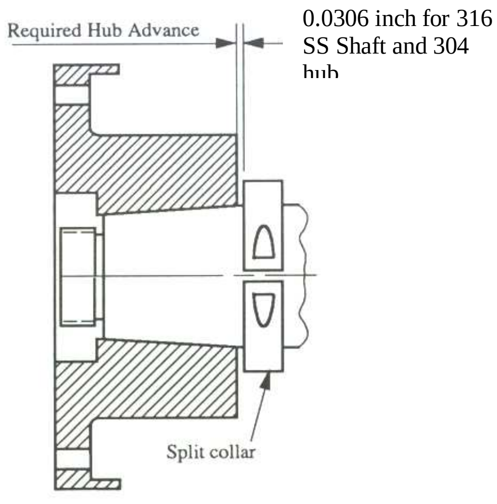

Figure 29. Illustration of Split Collar Installed To Act As Stop When Installing Heated Hub. [1]

The specific fan hub advance required to provide adequate interference fit at operating speed and temperature calculated to 0.0306 inch, **Figure 29**.

Equations: Equations are from [1]

$$A = 12 \bullet i \bullet \frac{d}{t}$$

Where:

A = Hub advance on the taper
t = Taper (inches/foot)
d = Shaft nominal diameter (inch)
i = Interference rate (inches/inch)

$$P = \frac{i \bullet E \bullet (d^2 - c^2) \bullet (D^2 - d^2)}{2 \bullet d^2 \bullet (D^2 - c^2)}$$

Where:

P = Contact pressure
i = Interference rate (inch/inch)
E = Modulus Elasticity Tension
d = Shaft diameter (large end)
D = Hub OD
c = Shaft hole diameter (if any)

Interference rate is the difference between bore and shaft diameter (diametrical clearance), divided by the shaft diameter. The interference rate is a ratio of quantities with the same units and thus has no unit of measurement.

$$i = \frac{D - d}{d}$$

Where:

i = Interference rate (inch/inch)
D = Hub bore (large end)
d = Shaft diameter (large end)

The combined hoop and compressive stress in hubs caused by interference is maximum at the bore:

$$\sigma_{psi} = \frac{P \bullet \sqrt{3 + k^4}}{1 - (\frac{d}{D})^2}$$

Where:

σ = Combined hoop and compressive stress in hub psi
P = Contact pressure psi
k = d (shaft diameter)/D (hub bore)
d = shaft diameter
D = hub bore
D = Hub bore
d = Shaft diameter (large end)

After completing the fan hub and shaft fit calculations, the findings were as follows:

1. Changing the shaft material to 17-4 PH Condition 1025 offered significant advantages in material properties over 304 SS. <u>But, the differential coefficient of expansion between the hub and shaft would not permit achieving an interference fit at the specified running speed and operating temperature without exceeding the yield strength of the hub</u>. The maximum interference rate at ambient temperature to avoid exceeding the yield strength calculated to 0.0006 inch/inch. <u>But, this would result in a loose fit of .0016 inch at operating speed and temperature.</u> In addition to the inability to maintain low residual unbalance, the loose fit would cause fretting corrosion of the fit.

2. For shaft material 316 SS and hub material 304 SS a maximum interference rate of 0.0006 inch/inch at ambient temperature calculated to not exceed the specified stress limits. At operating speed and temperature an interference of 0.001 inch was calculated to be maintained. <u>For the existing wheel and hub, 316 L SS shaft material appeared to be the optimum choice.</u>

3. Even with a larger shaft diameter of 4.000 inch and less flexure, there was still some concern about uneven rubbing of the seal material against the shaft causing thermal bowing. Thermal bowing would cause increased unbalance forces and vibration. Coating the seal contact area with a ceramic material would reduce the coefficient of friction significantly and reduce potential thermal bowing and shaft wear. Ceramic coating the seal area was recommended.

4. For assembly of the 304 SS hub and 316 SS shaft, use of a split collar clamped to the shaft was recommended to provide controlled 0.0306 inch advance of the hub, as shown in **Figure 30**. For ambient air temperature of 70 Deg F, the hub required heating to about 200 Deg F prior to installation. The hub would be installed on the tapered fit against the split collar and the draw bolt hand tightened to hold the hub in place while it cooled.

5. A minimum 70% contact is recommended per AGMA Standard 9002-A86 for keyed shafts with tapered fits. If the fit does not meet the 70% minimum, and the fit requires lapping, the shaft and hub should not be lapped together. A ring and plug type lap of soft material such as brass should be used.

Figure 30. Nov 6, 2006 Failure, All Fan Hub Bolts Failed.

A question was raised by the fan reliability team about welding the fan wheel to the hub instead of using the bolts originally provided by the fan OEM. This question was asked since all of the bolts failed during the Nov 6, 2006 failure. See **Figure 30**. A recommendation was made to increase the hub-to-fan wheel bolts by 50% in diameter. High strength, corrosion resistant bolts, should be used and the welding process eliminated. Because of the difference in the thermal coefficient of expansion between 304 SS and RA 2205, welding could cause weld cracking problems over time if the hub and wheel are welded together.

Bearing Evaluation: Many AGMA Arrangement 8 fans are provided with spherical roller bearings. We have found that the bearings are often not properly selected for the speed, load and lubrication. This is especially true for the coupling end bearing which often does not have the minimum loading specified by the bearing manufacture. For this fan the spherical roller bearing limiting speed and required minimum loading (using 2% of the dynamic load rating for roller bearings and 1 % for ball bearings as recommended by SKF) calculated as shown in **Table II**.

Table II: Comparison of OEM Specified Spherical Roller Bearing and Ball Bearings Used in Custom Design.

.Bearing Location	Fan OEM Design	One Piece Bearing Housing Design	Bearing Location	Fan OEM Design	One Piece Bearing Housing Design
Coupling End	22218 EK	6217 Deep Groove Ball	Fan End	22218 EK	6320 Deep Groove Ball
Limiting Speed	5300 RPM	5600 RPM	Limiting Speed	5300 RPM	4300 RPM
Static Radial Load lb_f	333 lb_f	335.5 lb_f	Static Radial Load lb_f	1007 lb_f	1031 lb_f
Bearing Static Load Rating C_O	84,300 lb_f	14,400 lb_f	Bearing Static Load Rating C_O	84,300 lb_f	31,500 lb_f
Bearing Minimum Required Load lb_f	1461 lb_f	196 lb_f	Bearing Minimum Required Load lb_f	1461 lb_f	391 lb_f
	Did not meet Minimum load for Grease	Meets Minimum Load		Did not meet Minimum load for Grease	Meets Minimum Load

Problems Encountered During The Project: All projects have hiccups as did this one. Obtaining accurate information from the fan OEM, the rotor assembly, and rotor balancing presented some difficulties. During initial rotor balancing, there was damage to the shaft bearing fits. The images of the shaft bearing fits, **Figures 31, 32** show scoring damage to the journals by the balancing machine rollers. Scoring of the shaft journals is always a concern when using rollers in a balance machine to support the shaft. To minimize scoring some actions that can be taken are as follows:

- The balance machine pedestals and rollers should be aligned perpendicular to the shaft centerline. Misalignment of the rollers to the shaft can reduce the roller/shaft contact area and increases the stress level. Misalignment can also cause the rollers to generate an axial thrust increasing scoring damage.
- Lubricating oil should be applied to the shaft and rollers at regular intervals to maintain an oil film.
- Felt wipers in good condition should be installed. The felt should contact the rollers to clean the rollers and minimize metal particles being carried between the rollers and the shaft.

Figure 31. Journal Showing Balance Machine Roller Damage to Bearing Fit (Coupling End). Metal Shaving Embedded in Journal.

Figure 32. Journal Showing Balance Machine Roller Damage to Bearing Fit, Wheel End. This Roller Track Appeared Wider or the Shaft Shifted Axially During Balancing Operation Generating Two Tracks.

The shop performing the rotor assembly and balancing did not follow ISO 1940-1 process to determine the correct balance tolerance. The balance machine printout showed that the residual unbalance tolerance used was for a center hung rotor which was then divided by 2 to obtain the tolerance at each bearing journal. The residual unbalance tolerance should have been calculated per ISO 1940-1 to G2.5 for an overhung configuration, as shown in **Figure 33**. The calculations are as follows:

$$U_{static} = \frac{U_{per}}{2} \bullet \frac{d}{2C} \qquad U_{couple} = \frac{U_{per}}{2} \bullet \frac{3d}{4d}$$

U_{static} = Permissible residual static unbalance

U_{per} = Permissible residual unbalance

U_{couple} = Permissible residual couple unbalance

b = 9.625
c = 30.682
d = 14.187

$$U_{per} = 6.015 \bullet 2.5 \bullet \frac{671}{1800} = 5.61\, oz-in$$

$$U_{static} = \frac{5.61}{2} \bullet \frac{9.625}{2 \bullet 30.682} = 0.65\, oz-in$$

$$U_{couple} = \frac{5.61}{2} \bullet \frac{3 \bullet 9.625}{4 \bullet 14.187} = 2.1\, oz-in$$

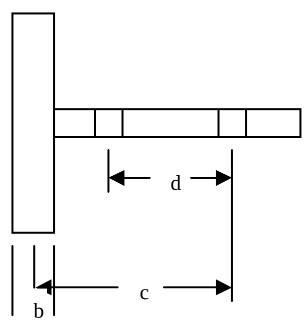

Figure 33. Schematic of Fan Rotor For Balance Calculations.

Note that the static residual unbalance for the overhung wheel is about 1/10 of the tolerance for the wheel if is center hung. Comparing the calculated residual unbalance tolerances to the shop's balance report showed the following:

- Balanced to 25.893 oz-in Right Side
- Balanced to 20.027 oz-in Left Side.

These values were far in excess of the calculated maximum residual unbalance values:

- Static unbalance = 0.65 oz-in
- Maximum couple residual unbalance = 2.1 oz-in.

Two balance facilities were used but the quality of work did not meet expectations at either facility.

Installation of New Bearing Housing and Shaft: The specially designed bearing housing and rotor were installed in Sept 2007. Images of the beefed up bearing support plate, access panel in the fan housing (added to permit in-place balancing and inspection of the wheel), and the bearing housing and shaft are shown in **Figures 34-37**.

Figure 34. Photos of Newly Installed One Piece Bearing Housing.

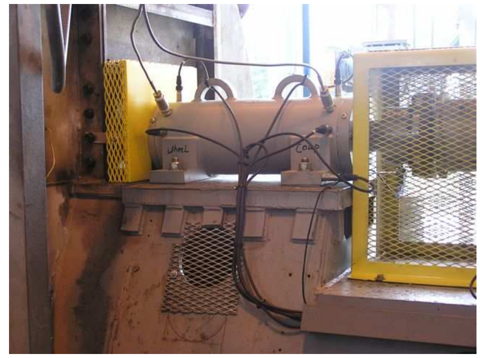

Figure 35. Photo of Newly Installed One Piece Bearing Housing With Accelerometers and Temperature Sensors Installed.

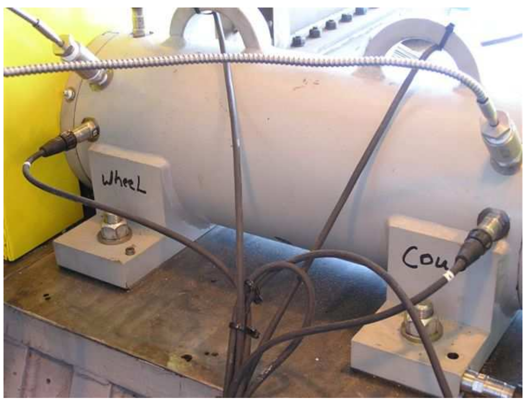

Figure 36. Close Up Photo of Newly Installed One Piece Bearing Housing With Accelerometers and Temperature Sensors Installed.

Figure 37. Photo of Modified Fan Housing to Install Access Panel for In-Place Balancing & Inspection.

The rotor 1st critical speed, measured in the vibration data. was very close to the frequency predicted by the rotor model. The spectra plots with log magnitude scaling, see **Figure 38**, were measured on the fan outboard bearing housing horizontal direction and shows the rotor 1st critical speed of 2351 CPM shortly after startup.

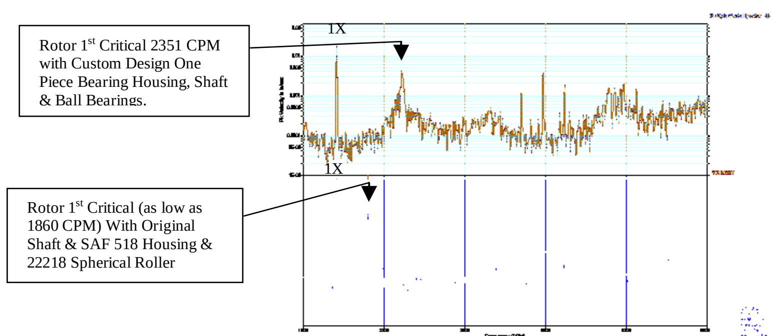

Figure 38. Frequency Spectra at Fan OB Brg Housing, Hor, Original Shaft & Bearings Top Plot, New Design Brg Housing and Rotor Lower Plot.

Before & After Vibration: The overall and 1X vibration in/sec pk are shown in **Figures 39 & 40**.

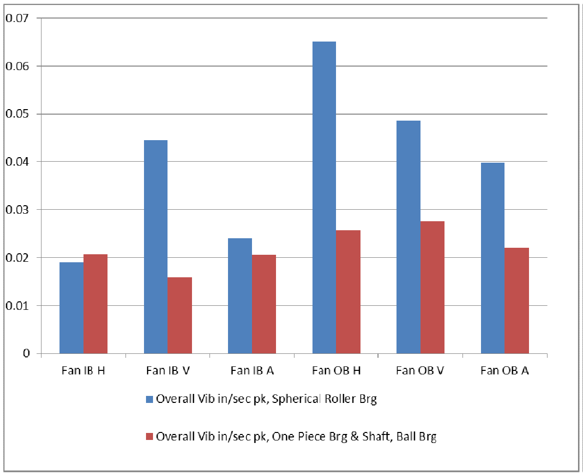

Figure 39. Chart Fan Bearing Overall Vibration Original Design and Modified Design.

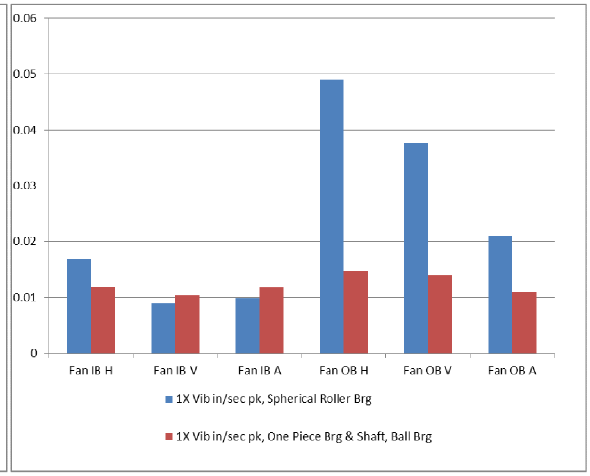

Figure 40. Chart Fan 1X Vibration in/sec pk Modified Bearing Housings & Shaft.

Conclusions:

1. The specially designed one piece bearing housing, shaft and bearing assembly successfully raised the fan rotor 1st critical from 1860 RPM to about 2350 RPM providing a separation margin of around 24% from run speed. The design provided several advantages over an off-the-shelf one piece bearing housing and shaft assembly. The major advantages were (1) direct bolt in-place with hold down bolt holes matching the base plate existing holes; (2) lower bending stresses generated by the overhung load, (3) improved maintainability (designed to permit bearing replacement without pulling the complete rotor or fan wheel removal), and (4) ball bearings met the minimum load guidelines specified by the bearing manufacture for grease lubrication.

2. Bearing housing vibration levels were reduced after installation of the one piece bearing housing and shaft and continued to remain low until the plant ceased operation in 2008.

3. Problems were encountered during the project. Some of these problems had to do with the quality of work performed at the shops which included excessive runout of the tapered shaft fit on the 1st shaft, unacceptable balance quality at two balancing facilities, and inaccuracy of some field measurements.

4. Important steps that should be considered for similar projects:
   - **a.** Detailed analysis of the fan should include but not limited to vibration, bearing calculations, bearing lubrication requirements, bearing housing mounting, bearing support flatness and rigidity, foundation, accurate rotor/bearing model, fan performance curves, review of maintenance history and OEM critical speed calculations.
   - **b.** Develop an accurate layout drawing of the fan base, bearing support plate, hold down bolt locations and bolt size, the fan housing, wheel dimensions, wheel axial position, the inlet cone position and overlap.
   - **c.** During fabrication and machining, witness dimensional inspections of the bearing housing and shaft at the machine shop. Shaft runout measurements should be done with the rotor supported in V-blocks at the journals per API 687. Don't trust a coordinate machine measurement machine to accurately measure shaft runout and fits.
   - **d.** If using a contract balance shop, conduct detailed inspection of the balancing facility to determine training level of personnel, condition of the balancing machine, most recent calibration date and capability to perform the requested balancing work.

References:

1. Michael M. Calistrat, **Flexible Couplings Their Design Selection And Use**, ISBN 0-9643099-3.
2. Hindhede, Uffe, Zimmerman, John R., Hopkins, R. Bruce, Erisman, Ralph J., Hull, Wendell C., Lang, John e., **Machine Design Fundamentals, A Practical Approach**, ISBN 0-471-04136-X.
3. Chen, W.J., Ph.D., P.E., Gunter, C.J., Ph.D., www.DyRoBeS.com
4. Chen, W.J., Ph.D., P.E., Gunter, C.J., Ph.D., **Introduction to Dynamics of Rotor-Bearing Systems**, ISBN: 978-1-4120-5190-3.
5. Gunter, Edgar J., Understating Amplitude And Phase in Rotating Machinery, RODYN Vibration Analysis, Inc., Charlottesville, Virginia, Vibration Institute 33 Annual Meeting June 24-26, 2009.
6. Szenasi, Fred R., P.E., Torsional Analysis of Variable Frequency Drives, Engineering Dynamics, Inc. San Antonio, Texas