TILTING PAD JOURNAL BEARING STARVATION EFFECTS

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ABSTRACT

Improved turbomachinery aerodynamic performance requirements have increased journal bearing operating speeds and loads well above traditionally acceptable values. For example, for high performance gearboxes, pinion bearing surface speed requirements are often over 325 f/s with bearing unit loadings in the 500 psi range. In order to meet the design challenges for these severe applications, evacuated bearing housings have been utilized as an effective means of reducing journal bearing operating temperatures.

Unfortunately, the use of evacuated housing designs has introduced a new and troubling phenomena—journal bearing starvation. This was never a problem with flooded designs with pressurized housings since any additional oil that may be required is simply drawn from the captured oil inside of the bearing housing. With the new evacuated housing designs, all required oil must be supplied by the oil inlet orifices. Often times, the amount of supply oil required to keep all pads from starving is well beyond reasonable. Thus, due to practicality, starvation in some form is allowed in almost all evacuated designs.

This paper discusses evacuated journal bearing starvation and its possible detrimental effects on rotordynamics. Specifically, the

effect of starvation on journal bearing stiffness and damping is investigated. A case history is presented showing the effect of increasing oil flow on the location and amplification of a gearbox pinion critical speed during near zero load mechanical testing. As flow increased and the bearing became less starved, the location of the critical increased while the amplification decreased indicating a strong dependency of bearing stiffness and damping on oil flow. Concurrently, a similar but smaller bearing was tested under zero load starvation conditions. Essentially no effect on stiffness and damping was evident. From these results, the authors conclude that although increasing the oil flow solved the problem, starvation in itself was not the cause.

INTRODUCTION

Improved turbomachinery aerodynamic performance requirements have increased journal bearing operating speeds and loads well above traditionally acceptable values. For example, for high performance gearboxes, pinion bearing surface speed requirements are often over 325 f/s with bearing unit loadings in the 500 psi range. In recent years, many gearbox applications have been above 350 f/s. Within the lead author’s experience, the fastest journal bearing surface velocity for an American Petroleum Institute (API) gearbox is 389 f/s. Again, within the lead author’s experience, faster surface velocities have been successfully achieved for high speed balancing applications with speeds up to 575 f/s.

Achieving these extremely high surface velocities would not be possible with a 1970s vintage tilting pad journal bearing (TPJB). Early journal bearing designs were almost exclusively flooded. That is, the exit area for the oil was less than the oil inlet area. This created a positive pressure inside the bearing housing, thereby flooding the bearing with oil (Nicholas, 1994).

In order to meet the design challenges for these severe applications with excessive surface velocities, evacuated bearing housings have been utilized as an effective means of reducing journal bearing operating temperatures. Tanaka (1991) presented experimental operating temperature data for a tilting pad journal bearing with bearing end seals (flooded and pressurized) and without bearing end seals (evacuated, nonpressurized). The bearing operated at lower temperatures without the end seals. Since then, many designs have been developed adopting the evacuated housing concept including Gardner (1994), Brockwell, et al. (1994), Ball and Byrne (1998), and Nicholas (2003).

Unfortunately, the use of evacuated housing designs has introduced a new and troubling phenomena—tilting pad journal bearing starvation. This was never a problem with flooded designs with pressurized housings since any additional oil that may be required is simply drawn from the captured oil inside of the bearing housing. With the new evacuated housing designs, all required oil must be supplied by the oil inlet orifices. Depending on the efficiency of the inlet oil supply mechanism, some oil escapes the bearing directly without participating in lubricating the pads. This certainly exacerbates the problem.

Another issue with evacuated housing tilting pad journal bearings is the unloaded pads. For heavy loads, the loaded pads, with a much smaller journal-to-pad leading edge entrance area, require much less oil compared to the unloaded pads that have a much larger entrance area. In most cases, the amount of supply oil required to keep all pads, including the unloaded pads, from starving is well beyond reasonable.

Finally, for high performance gearboxes, journal bearings are sized for peak performance at full load. The bearing is “oversized” for operation at near zero load during mechanical acceptance testing. Oil flow requirements for a full film on the loaded pads at full load results in starvation for all pads at near zero load. Again, the amount of supply oil required to keep all pads from starving at near zero load when the film thickness is larger is beyond reasonable. Thus, due to practicality, starvation in some form is allowed in almost all evacuated designs.

This paper discusses evacuated tilting pad journal bearing starvation and its possible detrimental effects on rotordynamics. Specifically, the effect of starvation on journal bearing stiffness and damping is investigated. A case history is presented showing the effect of increasing oil flow on the location and amplification of a gearbox pinion critical speed during no-load mechanical testing. Tilting pad journal bearing stiffness and damping test results will be presented for a zero load case with varying degrees of starvation (Harris and Childs, 2008). From these test results and the gearbox case history, the authors conclude that although increasing the oil flow solved the gear box problem, starvation in itself was not the cause.

OIL FLOW REQUIREMENTS

This author’s tilting pad journal bearing severe application experience plot is shown in Figure 1. The blue dots inside of the red box are API gearbox applications. Outside of the red box are test stand and high speed balance applications. Almost all applications shown on the plot are evacuated housing designs.

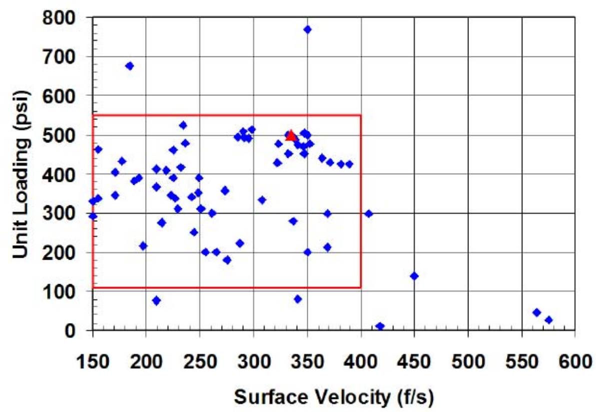

Figure 1. Tilting Pad Journal Bearing Severe Application Experience Plot.

The steps used to determine the oil flow requirements for all of these applications are summarized below:

1. Assume that the hot oil carryover from pad-to-pad is equal to the amount of oil that passes through the pads minimum film thickness (refer to ACKNOWLEDGEMENT section).

2. Using this hot oil carryover amount, determine the minimum lubricating flow requirement for a full film on the loaded pad at full load and at full speed (i.e., no loaded pad starvation) and then multiply by the number of pads. This is the minimum per bearing lubricating oil flow requirement.

3. Increase the flow as necessary from the calculated minimum to meet the bearing operating temperature requirements.

This results in a full film on the loaded pads at full load. It also results in starvation for the unloaded pads at any load condition and starvation for all pads during the no-load mechanical test. This design methodology worked successfully for all of the indicated Figure 1 applications except the one shown with the red triangle. Notice that it is safely within a batch of other successful applications as opposed to standing alone near the edge of the red box. Indeed, it is not the fastest application nor the most heavily loaded.

THE PROBLEM GEARBOX

The problem gearbox indicated by the red triangle in Figure 1 is a 24 MW, double helical, speed increaser driving three centrifugal compressors in offshore gas reinjection service. A photo of the box during mechanical testing is shown in Figure 2. The maximum continuous pinion speed is 12,700 rpm. The 6.0 inch diameter tilting pad pinion bearing's surface velocity is 333 f/s with a full load bearing unit load of 489 psi. The bearing's geometric properties are summarized in Table 1. The actual bearing is shown in Figure 3. Note that this design does not use end seals. Also of note is the huge discharge opening between pads to enable the oil to easily exit the bearing. This bearing design is described in detail in Nicholas (2003).

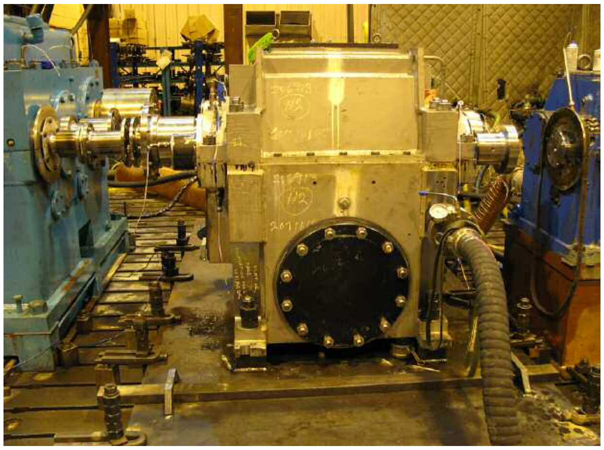

Figure 2. Problem Gearbox During Mechanical Testing.

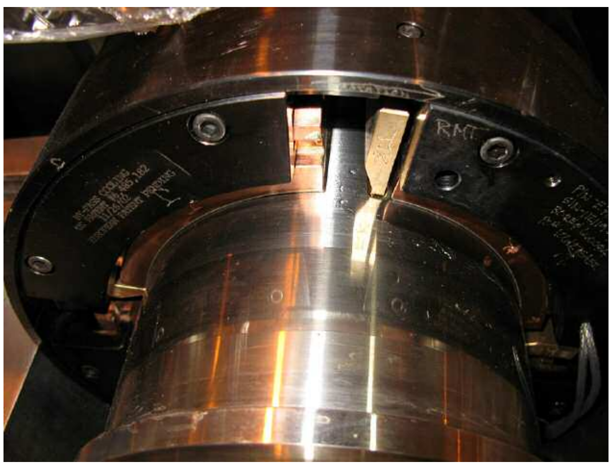

Figure 3. Tilting Pad Journal Pinion Bearing—Evacuated Housing Design.

Table 1. Pinion and Test Tilting Pad Bearing Geometric Properties.

	Pinion Bearing	Test Bearing
Journal Diameter, D_j (in)	6.0	4.0
Pad Axial Length, L (in)	6.0	4.0
Number of Pads	4	4
Pad Pivot Loading	Between	Between
Nominal Dimetral C_b (mils)	9.0	7.0
Nominal Preload, m	0.41	0.39
Pad Pivot Offset	65%	65%
Maximum Speed (rpm)	12,700	12,000
Surface Velocity (f/s)	333	209
Maximum Load (lbf)	17,947	4,640
Unit Load (psi)	489	290
Pivot Type	Spherical	Spherical
Pivot Diameter (in)	2.00	1.25
Oil Inlet Pressure (psi)	25.0	18.0

Using the design steps outlined in the previous section, the minimum lubricating oil flow requirement for a full film on the loaded pads at full load and full speed is 26 gpm. The oil flow was increased to 34 gpm of lubricating oil plus 5 gpm of by-pass cooling oil (Nicholas, 2003) to properly cool the bearing for a total of 39 gpm. However, due to pressure from the customer to reduce oil flow, the bearings were shipped to the gear manufacturer with a total oil flow rate of 34 gpm (includes lubricating oil plus by-pass cooling oil).

For reference, at zero load (i.e., gravity load only), the calculated full film lubricating oil flow requirement is 44 gpm. Adding in the 5 gpm of by-pass cooling flow, the total flow requirement for a full film at zero load is 49 gpm. Thus, it was anticipated that the bearing would be partially starved during the no-load mechanical test. Table 2 summarizes these results.

Table 2. Pinion Bearing Oil Flow.

Condition	Total Q (gpm)	ByPass Q (gpm)	Lubricating Q (gpm)	Percent Lub Q Full Film @ Full Load (%)	Percent Lub Q Full Film @ Zero Load (%)
Minimum Required - Full Film, Full Load	31	5	26	100	59
Minimum Required - Full Film, Zero Load	49	5	44	169	100
As-Designed	39	5	34	131	77
As-Shipped - Figure 4, First Run	34	5	29	112	66
Interim, P_{in} = 25 psi - Figure 5	45	5	40	154	91
Interim, P_{in} = 45 psi - Figure 6	60	5	55	212	125
Final - Figure 7	55	5	50	192	114

Running a pinion bearing partially starved during mechanical testing should not be a problem from the standpoint of load capacity. Since the bearing is sized for full load, it is obviously oversized at no-load. Partial starvation would, in effect, reduce the size of the bearing. At the leading edge of a partially starved bearing pad, there is not enough oil to fill the pad-to-journal gap. Thus, air is drawn into the pad and the initial section of pad is lubricated with an air-oil mixture. As this mixture moves farther into the pad, the film thickness decreases to the point where there is enough oil to fill the gap and a full film results. Thus, part of the pad's leading edge is ineffective in providing load capacity during partial starvation. Since the bearing is operating in a zero load condition, this reduction in load capacity will not be a problem. However, the starved part of the pad's leading edge will also not participate fully in providing the bearing's stiffness and damping properties. Thus, some degradation in the bearing's stiffness and damping is expected for partial starvation.

Although the problem described in this paper would not mechanically harm the pinion or bull gear, it was significant logistically and commercially as it did cause this gearbox to fail the API mechanical acceptance test. Everyone involved may believe that this problem is only an artifact of the test conditions and that it would not manifest itself under real operating conditions. Nevertheless, to meet the API test specifications (API 613, 2003), and ship the gearbox, it was necessary to make the changes described in this paper.

It is important to note that this gearbox ran flawlessly during the loaded string test. There was no indication of the problem described herein.

MECHANICAL TESTING

A speed-amplitude plot for the pinion from the initial no-load mechanical test run is shown in Figure 4 with a pinion bearing total oil flow of 34 gpm per bearing. For this plot and all other test results presented herein, an unbalance weight was placed on the coupling hub in order to excite and locate the pinion's first critical speed. From Figure 4, note that the pinion critical speed is evident at about N_1 = 13,500 rpm with an amplification factor of A_1 = 13.5. This does not meet the API 613 (2003) acceptance criteria. Note that the terms "no-load" and "zero load" are used herein to describe the mechanical test load, which, in actuality, was close to 5 percent of full load.

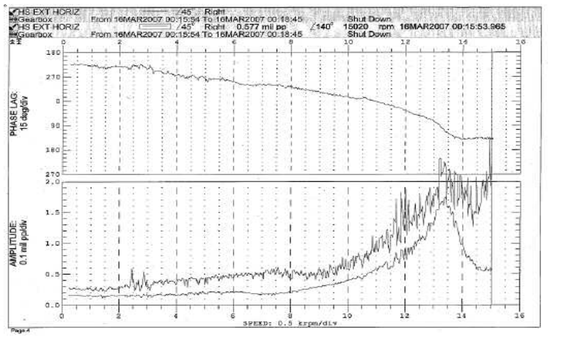

Figure 4. Initial Pinion Response, Q = 34 gpm, C_b = 9.0 mils, Fabricated Baseplate.

Increasing the inlet oil pressure appeared to push the critical up somewhat so the total bearing oil flow was increased to 45 gpm per bearing and the gearbox was retested. At the same time, since gearbox support stiffness was believed to be a significant factor, a stiffer solid baseplate replaced the original hollow, fabricated baseplate. The resulting speed-amplitude plot is shown in Figure 5. Now the pinion critical speed is at 14,200 rpm with a corresponding amplification factor of 14.2.

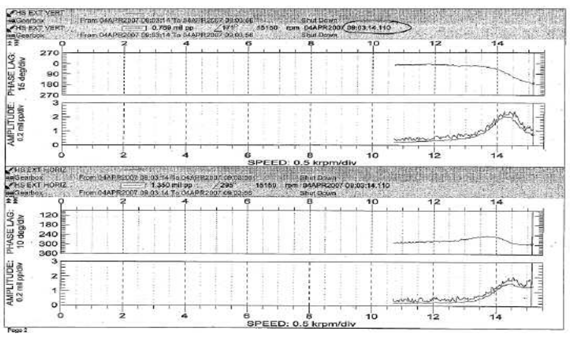

Figure 5. Interim Pinion Response, P_{in} = 25 psi, Q = 45 gpm, C_b = 9.0 mils, Solid Baseplate.

Since this still does not meet the API 613 (2003) acceptance criteria, the inlet pressure was increased from the design value of 25 psi to 45 psi in an attempt to easily check the effects of increasing oil flow. The corresponding per bearing total oil inlet flow increase was from 45 to 60 gpm. The resulting response run is shown in Figure 6 with N_1 greater than 15,200 rpm.

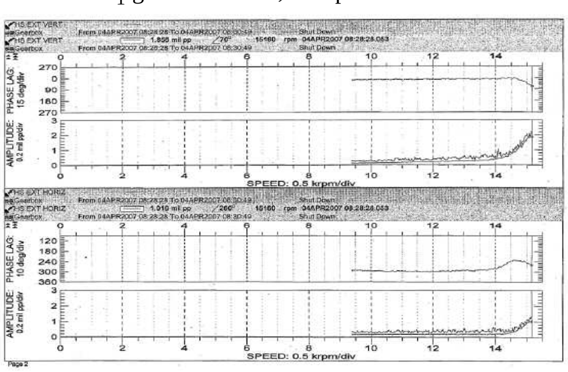

Figure 6. Interim Pinion Response, P_{in} = 45 psi, Q = 60 gpm, C_b = 9.0 mils, Solid Baseplate.

Based on the favorable Figure 6 results, another increase in oil inlet flow seemed appropriate. Further increasing the total oil flow to 55 gpm per bearing (50 gpm lubricating plus 5 gpm by-pass) and, at the same time, decreasing the bearing clearance by 1.5 mils diametral (from 9.0 to 7.5 mils diametral, nominal), results in the speed-amplitude plot shown in Figure 7. The pinion critical appears to be at 15,250 rpm minimum with A_1 = 10.9. This condition finally meets the API 613 (2003) acceptance criteria. These results are summarized in Tables 2 and 3.

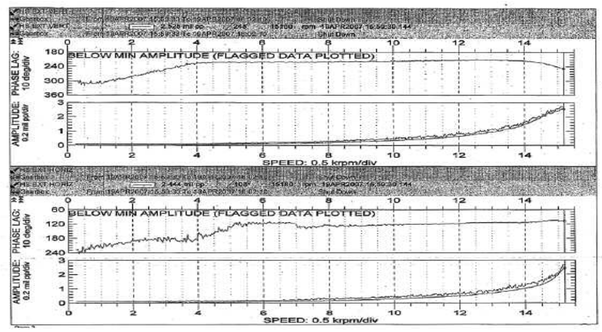

Figure 7. Final Pinion Response, Q = 55 gpm, C_b = 7.5 mils, Solid Baseplate.

Table 3. Summary of Mechanical Test Results.

Reference Figure #	Total Q (gpm)	P_{in} (psi)	Nom. C_b (mils)	Nom. m	Baseplate	N_1 (rpm)	A_1
4	34	25	9.0	0.23	Fabricated	13,700	13.5
5	45	25	9.0	0.23	Solid	14,200	14.2
6	60	45	9.0	0.23	Solid	>15,200	-
7	55	25	7.5	0.51	Solid	15,250	10.9

It is important to note that a similar problem also occurred on the low speed shaft with a maximum continuous speed of 6865 rpm. The low speed shaft problem was solved in a similar manner as described above. Due to space constraints, only the high speed pinion problem will be discussed herein. Additionally, the problem also occurred on the spare gearbox.

Finally, subsynchronous vibration for both the pinion and the bull gear shafts was not an issue. Spectrum plots show very low subsynchronous vibration levels throughout the no-load mechanical and full load string tests.

ANALYTICAL CORRELATION

In an attempt to match the no-load test results of Figure 4 (N_1 = 13,600 rpm, A_1 = 13.5), a rotordynamics analysis was performed on the pinion. Initially, the tilting pad bearing analysis assumed a full film as the bearing code used in the analysis did not have the capability to calculate any starvation effects explicitly (Nicholas, et al., 1979). A reasonable gearbox case support stiffness value of K_s = 7.0×10^6 was assumed. The nominal as-shipped bearing clearance of C_b = 9.0 mils diametral was also used. The resulting speed-amplitude plot is presented in Figure 8. The pinion critical is predicted at 13,700 rpm with an amplification factor of 7.6. The frequency is a good match but the amplification is predicted to be about 50 percent of the actual value indicating far less damping in the system than anticipated.

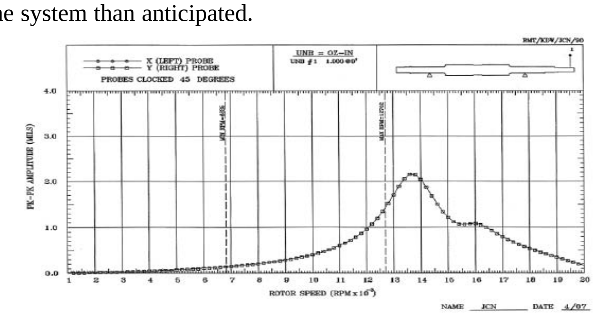

Figure 8. Predicted Pinion Response, Full Film, C_b = 9.0 mils, K_s = 7.0×10^6 lbf/in.

Figure 5 test results (N_1 = 14,200 rpm, A_1 = 14.2) were obtained with 45 gpm of total per bearing oil flow plus the stiffer, solid baseplate. The original fabricated, hollow baseplate is shown in Figure 9. In an attempt to increase the pinion's critical speed by increasing the gearbox support stiffness, a new solid baseplate, Figure 10, replaced the original prior to the Figure 5 test. A rap test was performed on the gearbox with the new, solid baseplate. Results indicated a dynamic support stiffness of 20.0×10^6 lbf/in at a frequency of 14,000 cpm. Using K_s = 20.0×10^6 lbs/in and the nominal as-shipped bearing clearance of C_b = 9.0 mils diametral, results in the speed-amplitude plot shown in Figure 11. The pinion critical is now predicted at 17,200 rpm with an amplification factor of 5.5. The frequency is over predicted by 3,000 rpm and the amplification factor is under predicted by a factor of 2.6. These results are summarized in Table 4.

Figure 9. Original Fabricated Hollow Baseplate.

Figure 10. New Solid Baseplate.

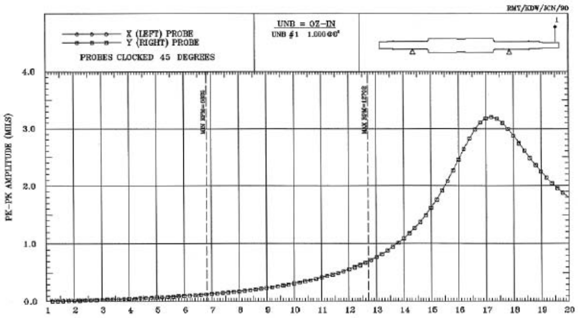

Figure 11. Predicted Pinion Response, Full Film, C_b = 9.0 mils, K_s = 20.0×10^6 lbf/in.

Table 4. Summary of Analytical Results.

Reference Figure #	Q Model	C_b (mils)	K_s (Mlbf/in)	N_1 (rpm)	A_1
8	Full Film	9.0	7.0	13,700	7.6
11	Full Film	9.0	20.0	17,200	5.5
12	Starvation w/ 52° Arclength, 50% Offset	9.0	20.0	18,000	1.4
13	Starvation w/ K = 70%, C = 15% of Full Film	9.0	20.0	14,700	11.3
19	Full Film w/ Integral Coupling Flange	9.0	20.0	16,200	9.0

STARVATION MODELING

In an attempt to better match the no-load test results with analytical predictions, starvation is included in the tilting pad journal bearing analysis. A tilting pad journal bearing computer code developed by He (2003) was used to predict the angle from the pad's leading edge where a full film would occur. This predicted angle is 20 degrees. A simplistic approach would be to assume that the pad arc length is effectively reduced by 20 degrees, from 70 degrees to 52 degrees. This also reduces the pad pivot offset from the as-machined 65 percent to an effective value of 50 percent. Using these effective values in the original bearing code, Nicholas, et al. (1979), along with C_b = 9.0 mils diametral and K_s = 20.0×10^6 lbs/in results in the speed-amplitude curves shown in Figure 12. Now the pinion critical is critically damped. Clearly, this model predicts too much damping.

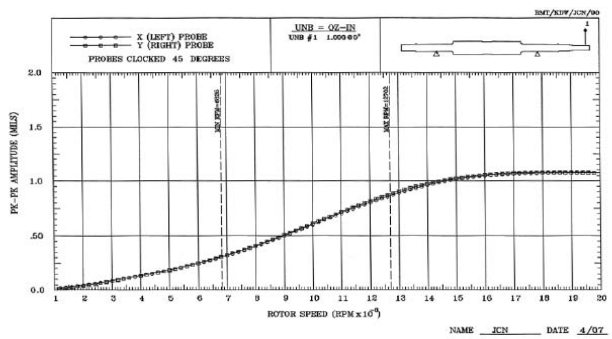

Figure 12. Predicted Pinion Response, Starvation Model with 52 Degree Pad Arc Length, 50 Percent Pad Pivot Offset, C_b = 9.0 mils, K_s = 20.0×10^6 lbf/in.

Artificially decreasing bearing stiffness and damping independent of each other until the results match Figure 5 (N_1 = 14,200 rpm, A_1 = 14.2) with K_s = 20.0×10^6 lbs/in produces the speed-amplitude curve shown in Figure 13. Now N_1 = 14,700 rpm and A_1 = 11.3, a reasonable match to test results. To obtain this match, the bearing stiffness was decreased to 70 percent of the full film value, which is reasonable, but the bearing damping was decreased to 15 percent of the full film value, an 85 percent decrease, which is quite unreasonable.

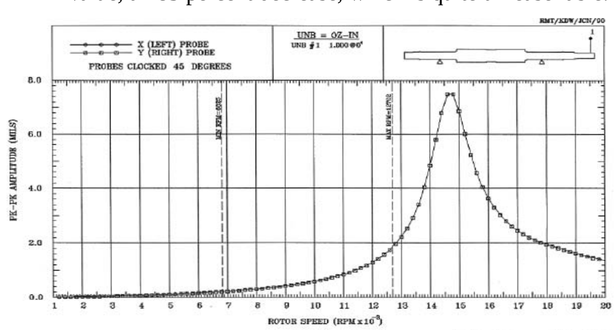

Figure 13. Predicted Pinion Response, Starvation Model with K = 70 Percent and C = 15 Percent of Full Film Values, C_b = 9.0 mils, K_s = 20.0×10^6 lbf/in.

Returning to the code by He (2003), plotting the normalized bearing stiffness, K, and damping, C, as a function of lubricating oil flow results in the plot shown in Figure 14. The code does predict a drastic decline in K and C, but they decrease at about the same rate. The dots shown on the plot indicate a stiffness value that is 70 percent of full film and a damping value that is 15 percent of full film. The K value that is 70 percent of full film occurs at a predicted lubricating flow rate of 65 gpm while the C value that is 15 percent of full film extrapolates out to a lubricating flow rate of 36 gpm. Since both flow rates cannot occur at the same time, 70 percent of full film K and 15 percent of full film C also cannot occur at the same time. But, they would have to occur together in order to match the test results. Thus, it is difficult to envision a starvation model that matches the test results. These results are summarized in Table 4.

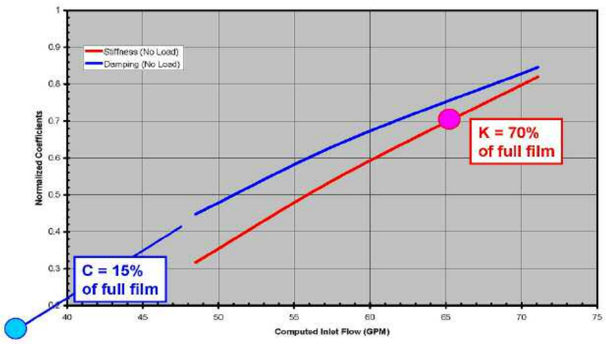

Figure 14. Starvation Model, Normalized K and C Versus Oil Flow Showing K = 70 Percent and C = 15 Percent of Full Film Values.

STARVATION TESTING

Coincidently, as the gearbox was experiencing problems with a 6.0 inch tilting pad journal bearing with an evacuated housing, a 4.0 inch diameter evacuated housing bearing was undergoing laboratory testing (Harris and Childs, 2008). Except for the test bearing being geometrically smaller, the designs were identical. Some of the geometric parameters are four pads, load between pivots, 65 percent pad pivot offset and L/D = 1.0 (Table 1). A special test was requested at 12,000 rpm and at zero load with the bearing flow rates varying from a full film to a starved condition (Table 5). These results are shown in Figures 15, direct stiffness, and 16, direct damping.

Table 5. Test Bearing Oil Flow.

Total Q (gpm)	ByPass Q (gpm)	Lubricating Q (gpm)	Percent Lub Q Full Film @ Zero Load (%)
		17	100
16	3	13	76
14	3	11	65
12	3	9	53
10	3	7	41

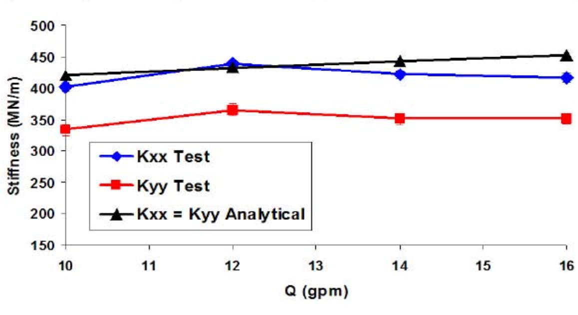

Figure 15. 4×4 Inch TPJB Zero Load Test Data—Principal Stiffness Versus Oil Flow.

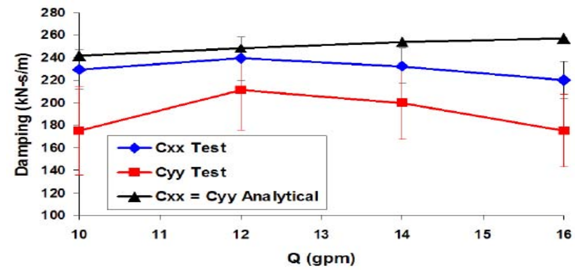

Figure 16. 4×4 Inch TPJB Zero Load Test Data—Principal Damping Versus Oil Flow.

From Figure 15, the test curves show a barely perceivable decline in bearing stiffness as the total oil flow decreases from a full film value of 16 gpm to a starved value of 10 gpm.

From Figure 16, the test results indicate that the direct damping increases slightly and then declines as flow decreases. Certainly, no decrease in damping is evident from Figure 16 that approaches the 15 percent of full film value discussed previously.

Also notice that full film analytical predictions from Nicholas, et al. (1979), are included on both plots. The nominal as-built bearing clearance was used for the analysis. Pivot stiffness was included and calculated by the Hertzian method from Kirk and Reedy (1988) and Nicholas and Wygant (1995).

ULTRA HIGH SPEED APPLICATION

Soon after the problem gearboxes were shipped, a similar evacuated tilting pad bearing design with a 6.65 inch journal diameter was used for a high speed balance of a magnetic bearing rotor. This application is shown on the plot of Figure 1 in the extreme lower right-hand corner, 575 f/s surface velocity, 26 psi unit loading. The tilting pad bearings, used for the high speed balance only, supported the rotor on the magnetic bearing laminations. Since the laminations had a relatively large outside diameter, the tilting pad bearing surface velocity was extremely high with the evacuated bearing running up to 435 f/s. The actual bearing is shown in Figure 17. The journal surface velocity versus metal temperature plot from one of the initial test runs using the evacuated design is shown in Figure 18, blue line with solid blue circles (refer to ACKNOWLEDGEMENT section).

Figure 17. Ultra High Speed Application—Evacuated TPJ Bearing.

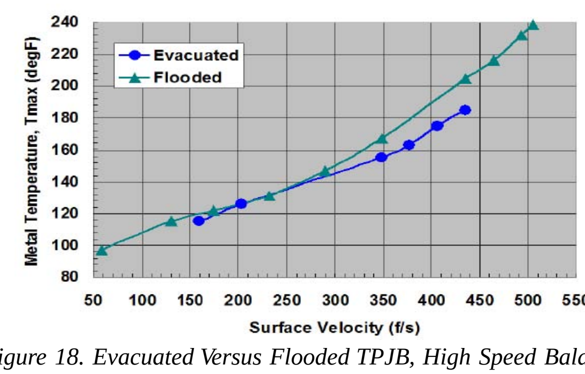

Figure 18. Evacuated Versus Flooded TPJB, High Speed Balance on Laminations.

The minimum oil lubricating flow requirement for operation at 435 f/s with the evacuated design is 36 gpm calculated as described in the OIL FLOW REQUIREMENTS section. Because of flow restrictions in the high speed balance facility, the evacuated bearings were designed for 28 gpm of lubricating flow. After the test resulting in the evacuated bearing data shown in Figure 18, it was determined that one of the oil pumps was not operational. Thus, the resulting lubricating flow was 20 gpm and the bearing ran 56 percent starved.

Furthermore, because of a misunderstanding, it was believed during the run that the oil drains were pressurized. This was not the case and the evacuated bearing operated in a vacuum, further starving the bearing due to oil atomization. Even in this extreme starvation condition, the rotor critical was located as anticipated with a reasonable amplification factor.

POSSIBLE CAUSES

As stated previously, this gearbox pinion rotor experienced a vibration problem during no-load mechanical testing. The location of the pinion's critical speed was well below predicted and the amplification factor was well above predicted. Increasing oil flow increased the critical speed location but had a minor effect on reducing the amplification factor. A similar problem occurred on the low speed shaft. It was solved in a similar manner. In the author's experience with dozens of gearboxes built in a similar manner with essentially the same evacuated tilting pad bearing, this was the first gearbox to exhibit this problem. However, it must be noted that unbalance testing was not conducted on all of the gearboxes and similar problems of this type may have been missed.

Clearly, increasing oil flow had a large influence on the critical speed location. This seems to indicate that starvation was the cause of the problem. However, there are contrary issues that seem to negate this conclusion:

- With all of the experience shown in Figure 1, it would seem likely that this problem would have manifested itself previously since all of the bearings were similar in design, have evacuated housings, and were designed to operate partially starved at no-load.
- A similar problem did not occur during the magnetic bearing rotor high speed balance with a similar evacuated tilting pad bearing design, at very light bearing loads, and in an extreme starvation condition.
- It is a highly unlikely coincidence that the problem finally showed itself on both rotors at the same time on the same gearbox.
- Increasing the total per bearing oil flow from 34 to 55 gpm, from partial starvation to a full film, increased the critical speed frequency but the amplification factor remained unreasonably high.
- An 85 percent decrease in full film bearing damping is necessary to match the test data.
    - Laboratory test results at no-load for a similar but smaller bearing did not show a dramatic decrease in bearing damping or stiffness as starvation increased.
    - Analytical starvation modeling does not predict this dramatic decrease in bearing damping necessary to match the test stand results.

The conclusion is that increasing oil flow helped to solve the problem but was not the cause. It may have been a contributor, but not the sole cause of the problem. Other possible causes or contributors are outlined below.

Air Entrainment

Air entrainment is a condition where air bubbles are trapped or entrained in the lubricating oil. It is a well-known phenomenon for squeeze film dampers. It has been shown that air entrainment can drastically decrease the damping provided by a squeeze film damper (refer to Figures 4 and 6, Tao, et al., 2000).

For an evacuated housing journal bearing, this type of entrainment may occur in the starvation region at the leading edge of a tilting pad. However, if there is inadequate dwell time for the oil in the reservoir, air bubbles will be present in the oil when it is reintroduced to the bearing. Several weeks after the gearbox was shipped, the amount of air entrainment present in the oil cooler was measured at less than 5 percent. From Figures 4 and 6 (from Tao, et al., 2000), this is not nearly enough to account for an 85 percent reduction in full film damping.

Mesh Oil and Air Impingement

It may be possible that the oil that exits the gear mesh may jet into the bearing and interfere with the inlet oil or the lubricating oil film. Another factor that may affect the bearing is the windage from the gear mesh. Since the housing is evacuated, there are no end seals on the bearing. Thus, the journal-to-pad interface and the lubricating film are exposed to the oil that is forced out of the gear mesh and from mesh windage. On this gearbox, the oil exits the gear mesh at a velocity of around 1000 f/s. This may be sufficient to interfere with the lubricating film thereby affecting the bearing's stiffness and damping properties. Whether this effect is sufficient to cause an 85 percent damping decrease from full film levels is unknown.

To prevent these phenomena from occurring, some gearboxes have a shield at the end of the gear mesh. A mesh shield was not present on the problem gearbox nor has it been used on most of the applications shown in Figure 1.

Another way to eliminate mesh oil or air impingement is to place an end shield on the mesh side of the journal bearings. No shields were present on the bearings for this gearbox. However, the inlet oil is protected from mesh impingement by side shields on the spray-bar blocker as shown in Figure 19.

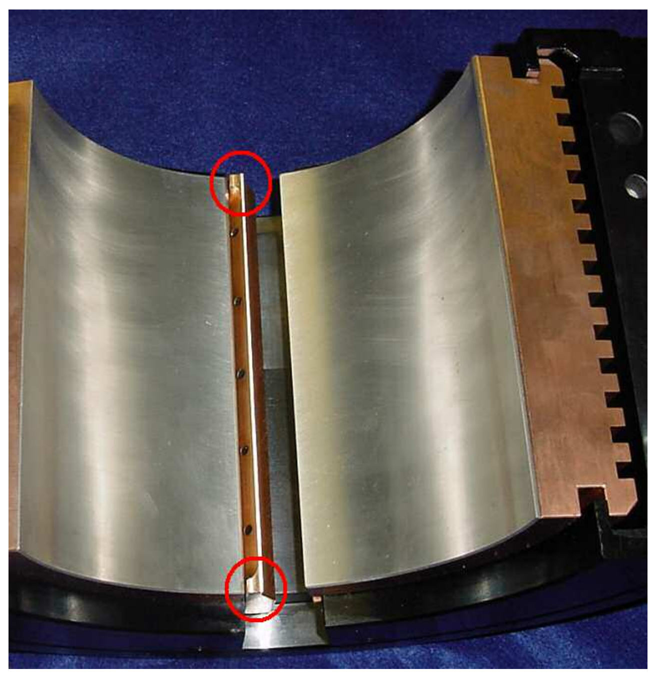

Figure 19. Spray-bar blocker Side Shields Protecting the Inlet Oil.

Support Stiffness

From Figure 8, with a reasonable gear case support stiffness of 7.0×10^6 lbf/in, the pinion critical is predicted at 13,700 rpm. However, the predicted amplification factor remains well below actual. Regardless, with the solid baseplate, the support stiffness was measured at 20×10^6 lbf/in. This is a relatively high value and, therefore, the problem cause is not likely to be a soft gear case support.

Pivot Stiffness

To obtain all of the analytical results, the spherical pivot stiffness was included in the calculations. Using the method outlined in Kirk and Reedy (1988) and Nicholas and Wygant (1995), the calculated Hertzian spherical pivot stiffness is K_p = 24.7×10^6 lbf/in. The bearing's spherical pivot was reconstructed and its stiffness measured using a hydraulic press to simulate the pivot load. The measured pivot stiffness was 17.4×10^6 lbf/in. Using the measured value for K_p, all of the analytical results show virtually no change. Thus, the problem cause is not likely to be a soft pivot.

Pad Flutter

Pad flutter is a tilting pad bearing phenomenon that may occur on the pads that are located opposite of the load vector. If these pads become unloaded (Nicholas, 1994), they may not be able to find an equilibrium position. Pad flutter may cause babbitt damage but rarely leads to high synchronous vibration. Furthermore, pad flutter occurs under heavy loads. Since this problem occurs under light loads, pad flutter is not a consideration.

Increasing Gearbox Performance Requirements

Gearbox performance requirements have been steadily increasing. As gearbox transmitted torques have increased, gear mesh face widths have also increased to carry the load while keeping gearbox shaft center distances small and gear pitch line velocities down. Wider face widths lead to longer rotors. Additionally, carburized gearing is often used to meet these higher power and torque levels. The higher tooth surface hardness achieved by carburizing permits considerably more torque to be transmitted than would be possible with a through-hardened gear of the same physical size. Bearing technology advancements are permitting more load to be carried by smaller bearings with reduced oil flow and power loss. These factors combine to result in heavier couplings for torque transmittal without a corresponding increase in the rotor diameter. Longer rotors and heavier couplings decrease the rotor's critical speeds. At the same time, operating speeds are increasing. All of these factors apply to this gearbox. Further compounding the problem, the flexible couplings on this gearbox utilize hydraulic taper fits, resulting in even heavier couplings and higher overhung moments, further reducing the critical speeds.

PROPOSED SOLUTIONS AND RECOMMENDATIONS

While the exact cause of the problem is unclear, some solutions and recommendations are suggested below. Keep in mind that, while gearbox mechanical tests are run at loads in the range of 5 to 10 percent of full load, this light load condition for centrifugal compressor trains is not realistic in the field. Specifically, operation at 10 percent of full load and at maximum speed is unrealistic for centrifugal compressor trains as the minimum load at maximum speed would be much greater than 10 percent. The solutions and recommendations follow:

- The use of an integrally flanged coupling with a low overhung moment is highly recommended. From the pinion mode shape illustrated in Figure 20, the pinion critical is clearly controlled by the coupling end overhang moment. This pinion coupling employed a shrunk-on hub. An integral flange attachment would greatly reduce the overhung moment. From Figure 13, N_1 = 14,700 rpm and A_1 = 11.3 with a coupling hub attachment. This analysis used the artificially degraded bearing stiffness and damping properties to match the test results. Replacing the hub attachment with an integral flange attachment with a lower overhung moment, results in Figure 21. Now, N_1 = 16,200 rpm and A_1 = 9.0 and this problem would not have materialized. For this gearbox, there were no engineering reasons to use a hub attachment instead of an integral flange.

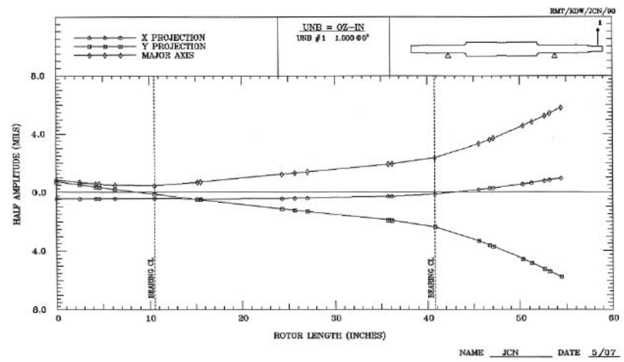

Figure 20. Pinion Mode Shape with Coupling Hub Attachment.

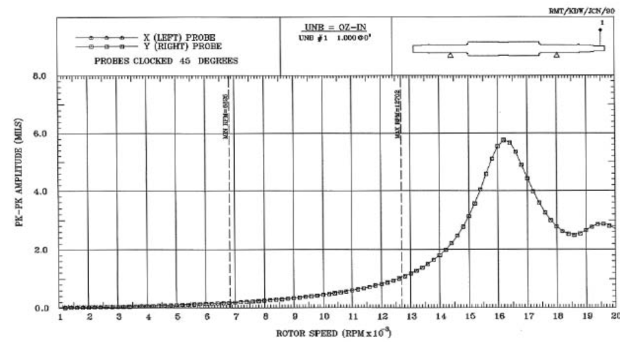

Figure 21. Predicted Pinion Response, Starvation Model with K = 70 Percent and C = 15 Percent of Full Film Values, C_b = 9.0 mils, K_s = 20.0×10^6 lbf/in, with Integral Flange.

- Another possibility to eliminate this problem is to use a properly cooled flooded bearing. The solution for the low speed shaft on this gearbox included switching to a flooded bearing design. While the operating temperatures increased compared to the evacuated design, they were still within the customer's specification. Also, it was estimated that the gearbox power loss increased by roughly 15 percent. A flooded design can be successful in severe application if proper cooling features are employed. An example is shown in Figure 18, green line, solid green triangles. This was the magnetic bearing rotor high speed balance application discussed previously in the ULTRA HIGH SPEED APPLICATION section. After some initial runs up to 435 f/s with the evacuated design, the bearings were changed to a flooded design with special cooling features. The special flooded tilting pad bearings ran successfully up to a surface velocity of just over 500 f/s. As expected, the flooded design ran slightly hotter. For example, at 435 f/s, the flooded bearing ran 20°F hotter compared to the evacuated bearing.

- When using an evacuated housing design, properly size the bearing oil flow. The final configuration for this pinion bearing ended up with 55 gpm of total per bearing oil flow. While this is 114 percent of the minimum lubricating flow for a full film at zero load, it is a staggering 192 percent of the minimum flow for a full film at full load. The authors suggest sizing the flow for 150 percent of the minimum flow for a full film at full load.

- Use a realistic gear case support stiffness value in the forced response analysis.

- Include the pivot stiffness in the bearing dynamic analysis.

- When evacuated housing bearings are used, a conservative analytical separation margin should be employed.

CONCLUSIONS

- The subject gearbox pinion rotor experienced a vibration problem during no-load mechanical testing.

- The location of the pinion's critical speed was well below predicted and the amplification factor was well above predicted. Increasing oil flow increased the critical speed location but had a minor effect on reducing the amplification factor.

- Increasing oil flow had a large influence on the critical speed location. This seems to indicate that starvation was the cause of the problem. However, there are contrary issues that seem to negate this conclusion. The major issues are outlined below:

    - With all of the experience shown in Figure 1, it would seem likely that this problem would have manifested itself previously since all of the bearings were similar in design, had evacuated housings, and were designed to operate partially starved at no-load. However, not all of the gearboxes were unbalance tested, and this problem may have been overlooked on past gearboxes.

    - A similar problem did not occur during the magnetic bearing rotor high speed balance with a similar lightly loaded, evacuated tilting pad bearing design in an extreme starvation condition.

    - An 85 percent decrease in full film bearing damping is necessary to analytically match the test data.

        - Analytical starvation modeling does not predict this dramatic decrease in bearing damping necessary to match the test stand results.

    - Laboratory test results at no-load for a similar but smaller bearing did not show a dramatic decrease in bearing damping or stiffness as starvation increased during no-load testing.

- From the above, it is concluded that increasing oil flow helped to solve the problem but was not the cause. It may have been a contributor, but not the sole cause of the problem.

- Other possible causes or contributors include:

    - Mesh oil and air impingement on the oil film. Mesh oil impingement is more likely to be a cause for concern with single helical gearboxes whereas the problem gearbox has a double helical mesh. Air impingement from mesh windage is obviously present in all gearboxes.

    - Air entrainment in the lubricating oil.

- The use of an integrally flanged coupling with a low overhung moment is highly recommended. If one were used on this application, this problem would not have occurred.

- An unbalance test is recommended for all gearboxes with pinion speeds above 8000 rpm to locate both the bull gear and pinion rotor critical speeds.

- Consider the use of a properly cooled flooded bearing.

    - This was the resulting configuration for the low speed shaft bearings.

    - Operating temperatures will increase.

    - Power loss will increase.

    - Proven successful in severe applications if properly designed with appropriate cooling features.

- When using an evacuated housing design, properly size the bearing oil flow. A design flow equal to 150 percent of the minimum lubricating flow for a full film at full load is a suggested target.

- Use a realistic gear case support stiffness and include the pivot stiffness in the bearing and forced response analyses.

- When evacuated housing bearings are used, a conservative analytical separation margin should be employed.

- Bearing manufacturers, gear vendors, compressor manufacturers, and the end users all need to work together to help prevent similar problems and to provide the best possible system. To this end:

    - Acknowledge that operation at 10 percent of full load at maximum speed is unrealistic for field operation of centrifugal compressor trains.

    - Relax the no-load mechanical test acceptance criteria. This will allow the bearing designers to design for full load and not for the no-load mechanical test. It would permit the elimination of otherwise unnecessary oil flow and power loss, and thus reduce the operating cost of the equipment, without reducing reliability.

- This gearbox performed flawlessly during the full load string test. No vibration problems were experienced. The bearing temperatures were all well within specifications.

In summary, the subject gearbox pinion and bull gear rotors experienced vibration problems during no-load mechanical testing with evacuated housing tilting pad journal bearings. The bearings were operating at essentially zero load in a partially starved condition. Test results indicate that the location of both rotor critical speeds were well below predicted with the amplification factors well above predicted. Increasing the oil flow increased the location of both critical speeds thereby solving the vibration problem. From this, one may conclude that bearing starvation was the problem cause. However, independent bearing testing in a starved condition at zero load did not induce a significant bearing stiffness or damping decrease. Thus, it is concluded that starvation alone did not cause the problem. The most probable cause was starvation in conjunction with another gearbox and/or test stand related phenomena: either mesh air impingement, mesh oil impingement, or air entrainment in the lubricating oil.

NOMENCLATURE

A_1	= First critical speed amplification factor (rpm)
C_b	= Bearing diametral clearance (mils)
C	= Bearing principal damping (lbf-s/in)
C_{xx}, C_{yy}	= Horizontal, vertical principle bearing damping, kN-s/m (lbf-s/in)
D_j	= Journal diameter (in)
K	= Bearing principal stiffness (lbf/in)
K_p	= Pivot stiffness (lbf/in)
K_s	= Gear case support stiffness (lbf/in)
K_{xx}, K_{yy}	= Horizontal, vertical principle bearing principle stiffness, MN/m (lbf/in)
m	= Pad preload
N_1	= First critical speed (rpm)
P_{in}	= Oil inlet pressure (psi)
Q	= Oil flow (gpm)
T_{max}	= Maximum bearing operating metal temperature (°F)
W_p	= Pivot load (lbf)

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