DESIGN, ANALYSIS, AND TESTING OF HIGH PERFORMANCE BEARINGS IN A HIGH SPEED INTEGRALLY GEARED COMPRESSOR

by

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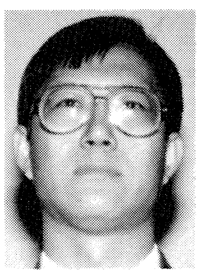

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ABSTRACT

Analysis, design, and optimization of high performance journal and thrust bearings in a high speed integrally geared compressor are described. The high speed and relatively small bearing size stretches the applicable limits of conventional style tilt pad bearings. The variation in the load magnitude, as is the case with most gear loaded bearings, requires careful analysis of the stability and unbalance response characteristics. An optimized new flexible pivot bearing was used in this application to provide stable operation throughout the wide ranges of load and inlet temperature. The bearing incorporated a directed lubrication feature to reduce the hot oil carry over which is known to be a major contributor to high bearing temperatures in high speed applications. The precision inherent in the manufacturing process used to produce this style of bearing allowed further optimization of the preload and pivot offset. The use of an offset

pivot configuration demonstrated lower sensitivity to bearing clearance variation.

The necessity for high rotational speeds in integrally geared compressors increases the frictional power losses in the journal and thrust bearings. The parasitic losses constitute a significant percentage of the overall bearing power losses. Bearing designs that were utilized to reduce both the frictional and parasitic losses in the thrust bearings are discussed. The new thrust bearings were designed to provide an optimum crown-to-film thickness ratio allowing the bearing to carry the design thrust load with fewer pads. The analytical results for the conventional and deflection journal and thrust bearings are presented and compared to the test data. The new bearing resulted in lower steady state vibrations. At surge conditions, the synchronous and subsynchronous vibrations were about 25 percent of the conventional tilt pad bearing. The frictional and parasitic power losses with the new thrust bearing were about 20 percent lower than those of the conventional bearings. The temperature rise was also lower which indicated that the inlet temperature can be further increased with the new bearing allowing a wider operating temperature range and lower frictional losses.

INTRODUCTION

The trend in integrally geared compressors is towards high performance and efficiency, high reliability, and lower maintenance. In order to achieve the high performance and aerodynamic efficiency in radial flow impellers, high rotating speeds are essential. The higher speeds and lighter shafts tend to make the machines more rotordynamically sensitive. The problems are compounded in the case of integrally geared compressors, where the gear drive arrangement results in an overhung impeller with its inherently higher aerodynamic cross coupling. This subsequently presses the bounds of the stability limits.

The services demanded from these compressors mandates robust and reliable operation at a relatively wide range of bearing loads and oil inlet temperatures. The aerodynamic stage efficiency requirements limits the axial float to a maximum of five mil. Therefore, reducing the power loss in the bearings is essential in order to maintain the high compressor stage efficiency. Higher speeds, increased rotor flexibility, and variations in the bearing gear load magnitude, coupled with the demand for lower frictional and parasitic losses necessitate an in depth bearing design and analysis process.

The two stage integrally geared air compressor shown in Figure 1 was the focus of this design investigation. The compressor was developed using agile engineering described by Japikse and Olsofka [1] and represents the state of the art in integrally geared compressors. The driver is a flange-mounted induction motor. The drive train is shown in Figure 2 and consists of a bull gear, an intermediate gear, and the high speed pinion shaft carrying two overhung impellers. The maximum rated pinion speed is 76,500 rpm and is expected, with further development, to reach even 20 to 30 percent higher speeds. The high speed pinion shaft shown in Figure 3 is supported by two combination radial thrust bearings. The bearings are of one piece (unsplit) configuration.

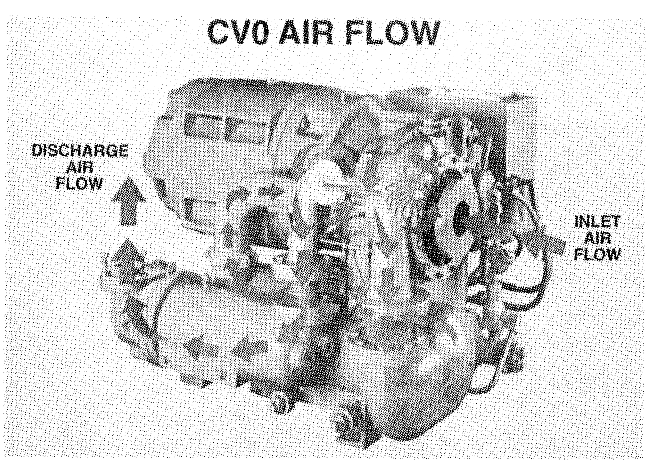


Figure 1. Two Stage Integrally Geared Air Compressor.

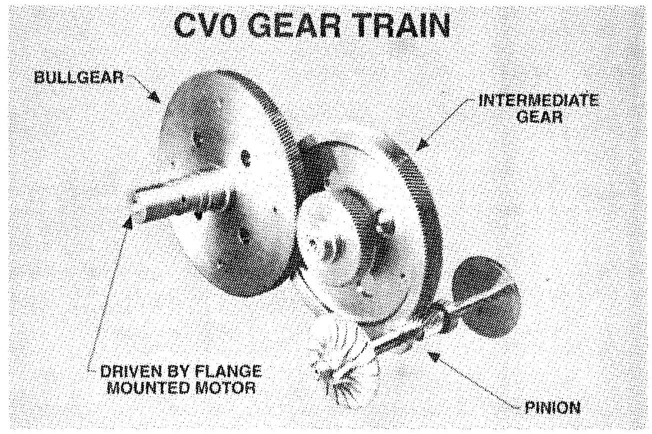


Figure 2. CV0 Gear Drive Train.

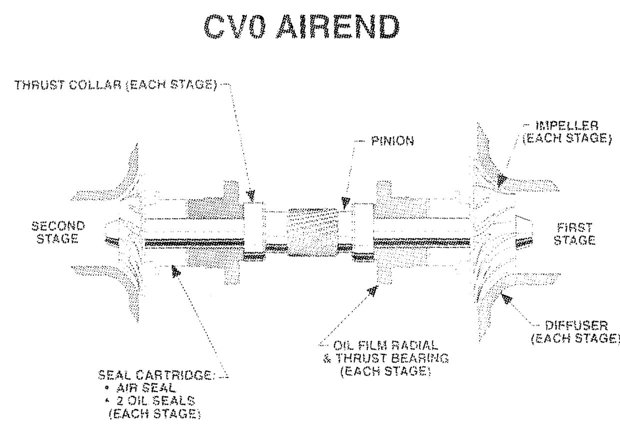


Figure 3. Schematic of Rotor Configuration.

ROTORDYNAMIC ANALYSIS AND BEARING OPTIMIZATION

Undamped Critical Speed Analysis

A computer generated rotor model for the high speed pinion shaft is shown in Figure 4. The undamped critical speed map as a function of support stiffness is shown in Figure 5. The equivalent dynamic stiffness coefficients for the conventional five pad tilting pad bearing (TPB) are shown as a function of speed and are superimposed on the undamped critical speed map. The coefficients intersect the first two rigid body critical speeds and the first flexible mode in the sloping section of the curves. This ensures that there is significant motion at the bearings to help dampen the shaft motion as the rotor traverses the critical speeds when accelerating to the operating speed. The coefficients for the optimized four pad flexible pivot bearing (FPB) are also superimposed on the critical speed map as shown in Figure 5.

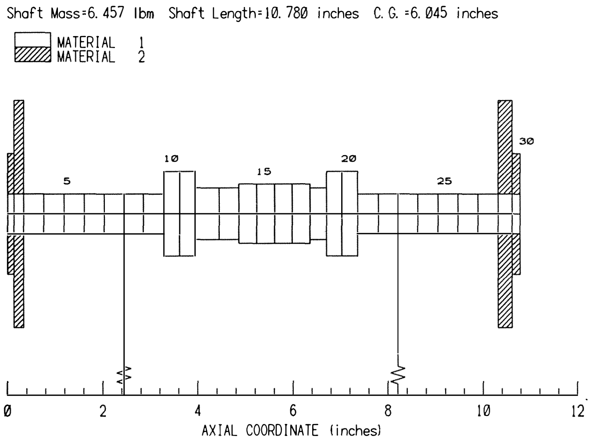


Figure 4. Computer Generated Rotordynamic Model.

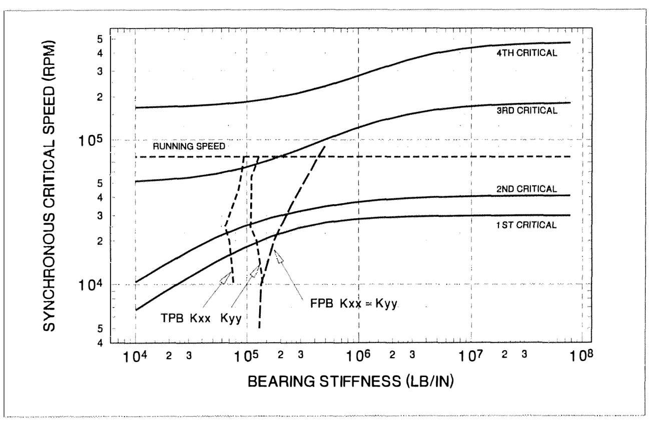


Figure 5. Undamped Critical Speed Map.

This bearing has an offset pivot which was chosen, as will be shown later, for its superior stability characteristics over a wider range of bearing clearances. The offset pivot configuration also provides higher stiffness in the bearing and helps shift the third mode further away from the running speed. This is later verified through the stability and unbalance response analysis.

Journal Bearing Design and Analysis

The calculations performed to determine the minimum and maximum clearances and preload at the extremes of the manufacturing tolerances are shown in Table 1. The conventional style tilt pad bearing, as shown, will have a rather wide range in clearance and preload. This is inherent in the design and assembly of these bearings that results from the stackup of the manufacturing tolerances for the individual parts making the bearing assembly. This is the reason why many of these bearings very often have a high preload value. The high preload value is required to counter the negative effects of the manufacturing tolerances that can otherwise result in a negative preload condition. Another complication that affects bearing design, and is characteristic of most gear loaded bearings, is the variation in the bearing load from partial load to full load conditions. This requires the bearing to have a wide range of stable operation under partial and full load conditions.

In order to accentuate some of the differences between the two bearing styles examined in this investigation, it is essential to describe the manufacturing method and techniques used in producing the flexible pivot bearings. These bearings are fabricated using wire electric discharge machining (EDM) while they are submerged in water during this process. The water is maintained at a constant temperature which is typically held within 1°F of the room temperature, to ensure thermal stability during the cutting process. The back of the pad and the web (pivot) are cut first, then the part is released to allow it to relax. The part is then reclamped and the final bore is cut in the last step. This ensures that the effect of any pad rotation is completely eliminated. Since the pad and bearing shell are one piece, tolerance stackup is also eliminated. As shown in Figure 6, the set bore and pad bore are cut in one operation, further reducing the manufacturing tolerance. The set bore and the pad machined bore are not independently generated, as is the case with the conventional style tilt pad bearings. This further reduces the resulting preload spread for the FPB as shown in the right column of Table 1. The radial and thrust bearing subassemblies are shown in Figure 7. The wire EDM machine used in the manufacture of the radial bearings is capable of cutting the set bore to within 0.0001 in, which is difficult to measure even with most electronic coordinate measuring machines (CMM). With this manufacturing method, there is no need for subsequent assembly and checking, which can be very time consuming with conventional style bearings. Note that in order to maintain a preload of 0.2, the drop off, as shown in Figure 8, from the center of the pad to the edge is about 0.000213 in to 0.000493 in, which would be impossible to achieve without the high precision and accuracy of state-of-the-art wire EDM machines. The material between the pads is utilized as integral directed lubrication nozzles, and the material left underneath the pads provides squeeze film damping.

Table 1. Calculation of Minimum and Maximum Preload.

Conventional Tilt Pad Bearing	*Flexible Pivot Bearing*
Journal Diameter_{min} = 0.8198	Journal Diameter_{min} = 0.8198
Journal Diameter_{max} = 0.8200	Journal Diameter_{max} = 0.8200
Pad Set Bore_{min} = 0.8230	Pad Set Bore_{min} = 0.8224
Pad Set Bore_{max} = 0.8238	Pad Set Bore_{max} = 0.8228
Pad Machined Bore_{min} = 0.8234	Pad Machined Bore_{min} = 0.8232
Pad Machined Bore_{max} = 0.8243	Pad Machined Bore_{max} = 0.8236
$\text{Preload} = m = 1 - \frac{C_b}{C_p} = 1 - \frac{D_b - D_j}{D_b - D_j}$	
$\text{Preload}_{min} = 1 - \frac{D_{bmax} - D_{jmin}}{D_{pmin} - D_{jmin}}$	$\text{Preload}_{min} = \frac{D_{dmax} - D_{bmax}}{D_{pmax} - D_{jmin}}$
$\text{Preload}_{max} = 1 - \frac{D_{bmin} - D_{jmax}}{D_{pmax} - D_{jmax}}$	$\text{Preload}_{max} = \frac{D_{pmin} - D_{bmin}}{D_{pmin} - D_{jmax}}$
$m_{min} = 1 - \frac{0.8238-0.8198}{0.8234-0.8198}$	$m_{min} = \frac{0.8236-0.8228}{0.8236-0.8198}$
$m_{min} = \boxed{-0.1000}$	$m_{min} = \boxed{0.2105}$
$m_{max} = 1 - \frac{0.8230-0.8200}{0.8243-0.8200}$	$m_{max} = \frac{0.8232-0.8224}{0.8232-0.8198}$
$m_{max} = \boxed{0.3023}$	$m_{max} = \boxed{0.2352}$
$m_{nom} = 1 - \frac{0.8234-0.8199}{0.82385-0.8199}$	$m_{nom} = \frac{0.8234-0.8226}{0.8234-0.8199}$
$m_{nom} = \boxed{0.1139}$	$m_{nom} = \boxed{0.2285}$

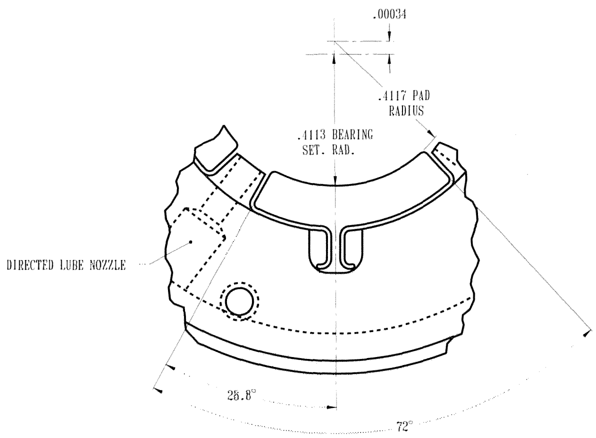


Figure 6. Schematic of the Flexure Pivot™ Bearing.

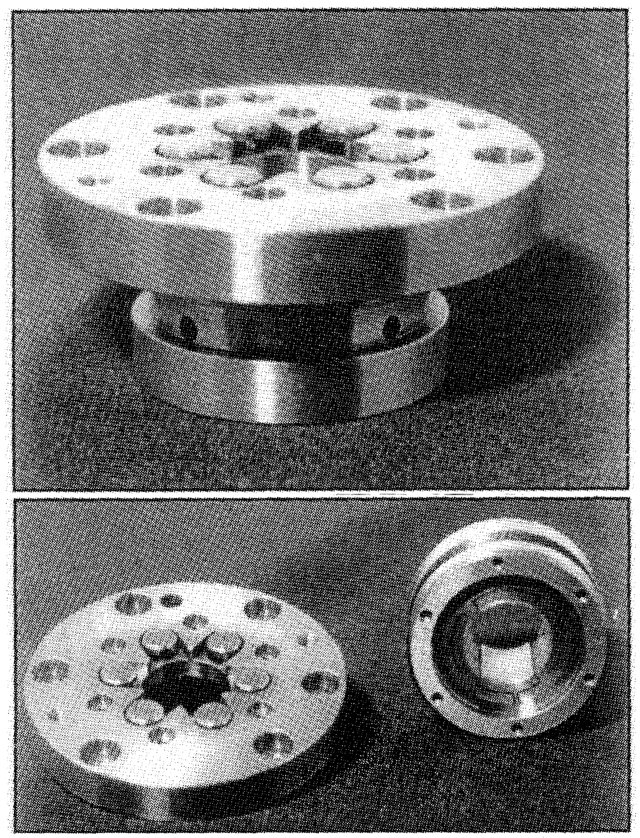


Figure 7. Radial and Thrust Bearing assemblies.

Stability Analysis

The first three forward modes with the conventional tilt pad bearings at the maximum preload conditions are shown in Figures 9, 10, and 11, respectively. Note that, as depicted in the undamped critical speed map, the third forward mode is around 60,000 rpm and is below the running speed. The first forward mode with the conventional tilt pad bearings at the minimum preload conditions is shown in Figure 12. Note that the logarithmic decrement (log dec) of the first forward mode drops by about 22 percent in comparison to the log dec at the maximum preload conditions. The minimum preload condition is also very susceptible to pad flutter due to the fact that the pads can possibly operate with a negative preload.

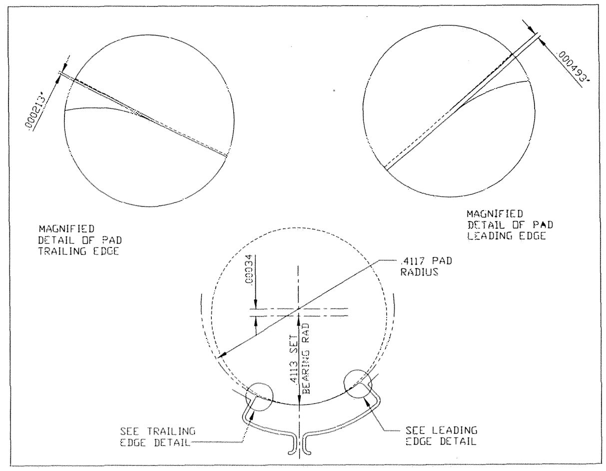


Figure 8. Precision Needed for Obtaining the Drop-off at the Leading and Trailing Edges.

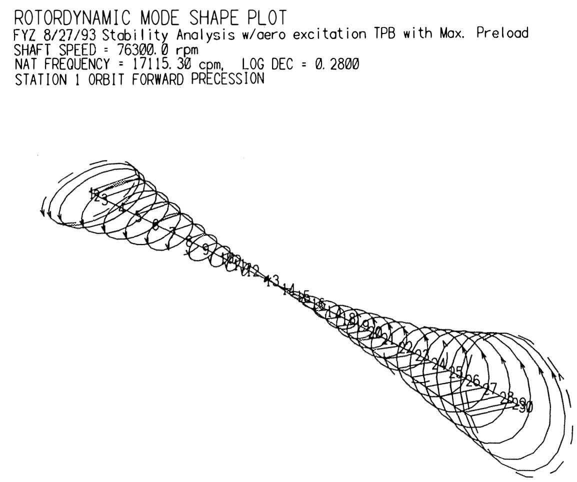


Figure 9. First Forward Mode with the Conventional Tilt Pad Bearings (Maximum Preload).

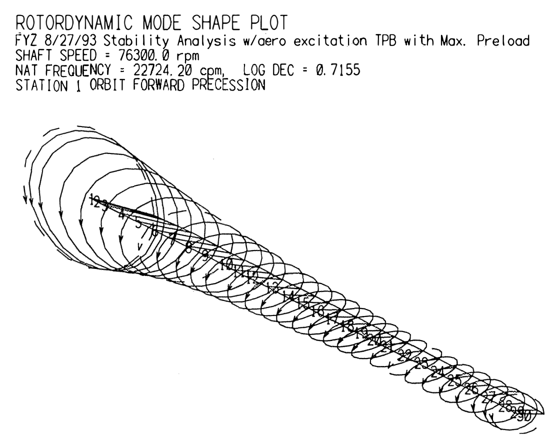


Figure 10. Second Forward Mode with the Conventional Tilt Pad Bearings (Maximum Preload).

The stability analysis was also performed with the flexible pivot bearings. The first three forward modes with a set of optimized four pad flexible pivot bearings are shown in Figures

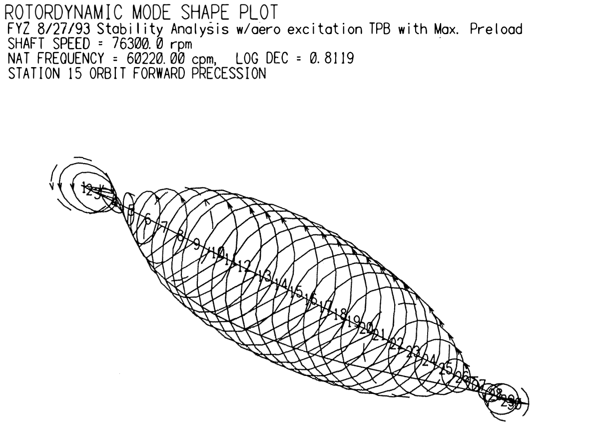


Figure 11. Third Forward Mode with the Conventional Tilt Pad Bearings (Maximum Preload).

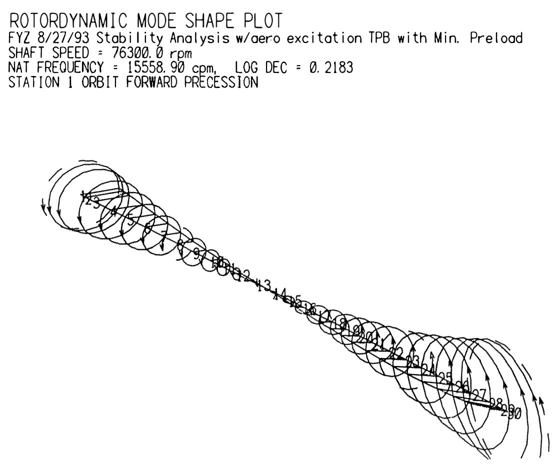


Figure 12. First Forward Mode with the Conventional Tilt Pad Bearings (Minimum Preload).

13, 14, and 15, respectively. Note that in this case the first forward mode has a logarithmic decrement 30 to 50 percent better than that achieved with the conventional tilt pad bearing. The third mode is also above the running speed and is well

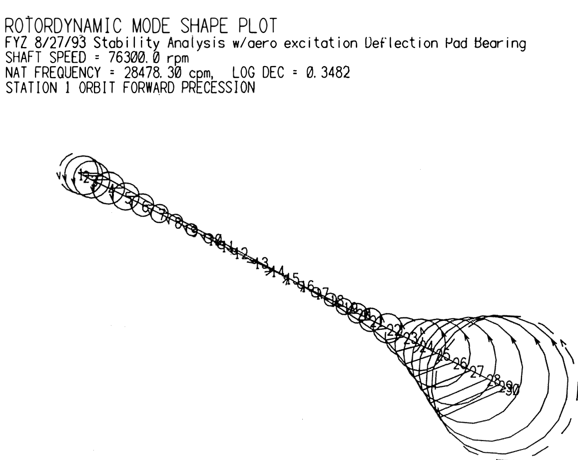


Figure 13. First Forward Mode with the Optimized Four Pad Flexure Pivot Bearing.

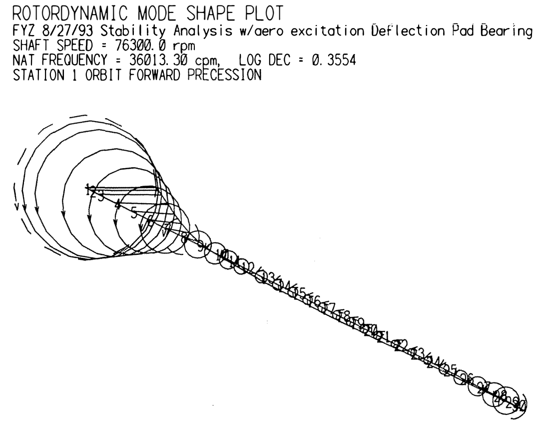


Figure 14. Second Forward Mode with the Optimized Four Pad Flexure Pivot Bearing.

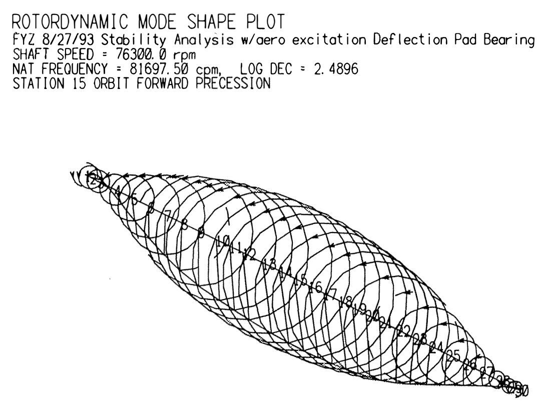


Figure 15. Third Forward Mode with the Optimized Four Pad Flexure Pivot Bearing.

damped with a log dec of 2.489. The ability to manufacture the bearing with a relatively low preload, and to maintain the minimum and maximum variations due to manufacturing tolerances very close to the target value is a great advantage in this application.

In order to demonstrate the high reliability over a long period of operation, the analysis was also conducted at a slightly higher set bore clearance (3.2 mil). This was performed to evaluate the effect of wear on the bearing in services that may require many starts and stops, and also to examine future applications where the speed may possibly increase from 20 to 30 percent over the current operating speed. The higher speeds will require the larger clearance to guard against bearing seizure. The results of this investigation are shown in Table 2. Note that the five pad configuration with the 0.5 pivot offset resulted in a negative logarithmic decrement for the first forward mode. Using a pad offset of 0.6 increased the stability margin (log dec = 0.186), but it continued to show sensitivity to bearing clearance with the five pad bearing. The four pad bearing with a 0.5 offset offered much better stability at 2.4 mils clearance (log dec = 0.38). However, there was a drastic reduction in stability (log dec = - 0.241) when the clearance was increased to 3.2 mils. A four pad bearing with an offset pivot of 0.6 provided better overall stability at both clearances as shown in Table 2. What is of importance in this optimization process is the fact that an offset pivot design reduced the sensitivity of the bearing to clearance variations. The offset pivot, as shown on the undamped critical

Table 2. Summary of the Stability Analysis Flexible Pivot Bearing.

# of Pads	Offset	Preload	Dia. Clr. (Mils)	First Forward Mode
5	0.5	0.2	3.2	-0.443
5	0.5	0.2	2.4	-0.290
4	0.5	0.2	3.2	-0.241
4	0.5	0.2	2.4	0.380
5	0.6	0.2	3.2	-0.007
5	0.6	0.2	2.4	0.186
4	0.6	0.2	3.2	0.249
4	0.6	0.2	2.4	0.348

speed map, also helped slant the equivalent dynamic bearing stiffness to the right, avoiding traversing the third critical speed.

There are other advantages that also can be realized with an offset pivot design. The maximum film temperature is one of the primary limiting parameters with gear loaded bearings running at high speeds. The offset pivot can significantly help reduce that temperature. This is shown in Figure 16 for both the five and four pad bearings at 0.6 pivot offset compared to the 0.5 offset values. There is a 20 to 30°F drop in the operating temperature with an offset pivot. The directed lubrication nozzles serve to further reduce the operating temperatures. Inhouse testing at the bearing manufacturer's test facilities showed that directed lubrication significantly reduces the hot oil carry over from one pad to the next. This can also result in reducing the maximum operating temperatures by about 25 to 30°F.

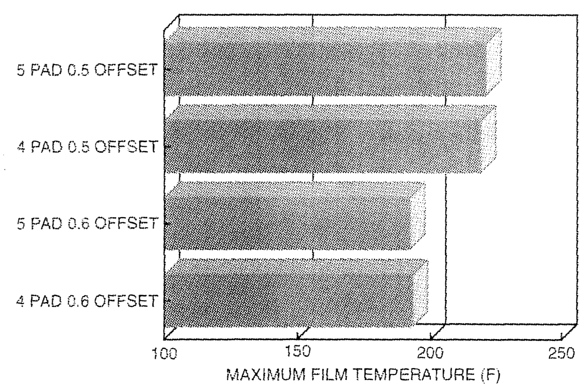


Figure 16. Maximum Film Temperature for Different Bearing Configurations.

Unbalance Response Analysis

Unbalance response analysis was computed with the conventional tilt pad bearing at both extremes of the manufacturing tolerances (minimum and maximum preload). The response with the optimized flexible pivot bearing is also shown on the same plot in Figure 17. The response at operating speed is about 50 percent lower with the flexible pivot bearing, compared to the conventional tilt pad bearings. Note that the response is still very low at the higher speeds projected for future machines. The peak response, due to the critical speed at 60,000 rpm with the conventional tilt pad bearing, is eliminated with the optimized flexible pivot bearing.

Design and Stress Analysis of the Radial FPB Bearing

An ANSYS® model of the pad geometry was generated for the purpose of determining the rotational stiffness of the pad and the

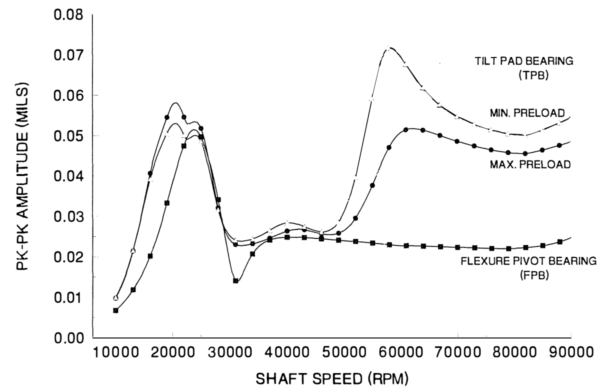


Figure 17. Unbalance Response Predictions with the Two Bearing Configurations.

location of the pivot in the structure. The bearing is then analyzed at the maximum static gear load to compute the static tilt or flexure. A stress contour plot of the stresses in the pad due to static loading is shown in Figure 18. The unbalance response analysis is then used to determine the dynamic loads transmitted from the rotor to the bearing. Using the stress results at static and dynamic conditions, one is able to compute the Modified Goodman diagram shown in Figure 19. The safety factor can then be determined from the information shown in the figure. Note that the dynamic load is calculated using eight times the maximum unbalance value specified in API 617 Fifth Edition. Even with this high unbalance value, the safety factor is over 12.5.

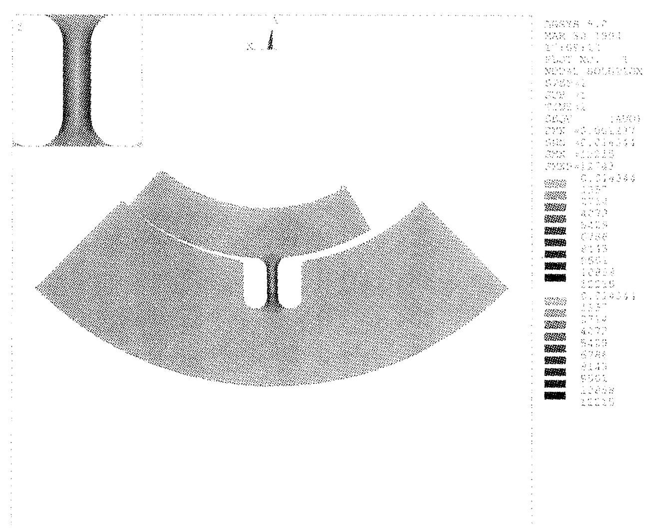


Figure 18. Finite Element Stress Analysis Contours for the Flexure Pivot Bearing.

An obstacle that has played a role in slowing acceptance of the FPB bearing is the relatively thin web feature. This thin structural member often leads to concerns about the stresses and the fatigue life of the web. These fears have prompted the detailed analysis of the stresses and fatigue limit, as shown in Figure 19. The modified Goodman diagram confirms that the bearing has a substantial margin of safety and that the stresses are well below the endurance limit of the structural material. In fact, the stresses are much lower than the contact stresses at the interface between the shell and the rocker back tilt pad in conventional tilt pad

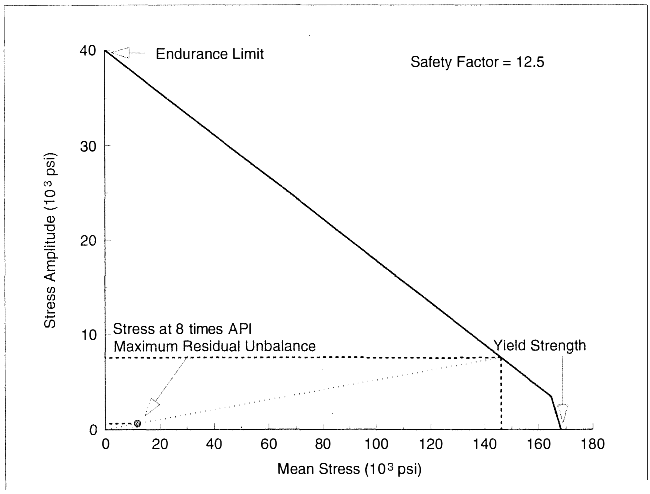


Figure 19. Modified Goodman Diagram for the Flexible Pivot Bearing.

bearings. Furthermore, there is no wear or fretting with the flexible pivot bearings. This ensures that the performance will be maintained regardless of how long the bearings are in service.

There is a striking similarity between the slow acceptance of the flexible pivot bearings and dry flexible element couplings. The analogy centers around concerns raised about the stresses, fatigue life, and endurance limit of the thin web section. However, the amount of pad tilt that actually takes place once the shaft is positioned in the bearing is very small. This is because once the shaft is in place within the bearing clearance, it will limit the maximum amount of pad tilt. As a result of this physical limit, the stresses will be very low. On the other hand, a clear distinction between flexible element couplings and flexible pad bearings lies in the mode of bending that occurs. Unlike the flexible element coupling, the flexible pad bearing does not undergo reverse bending. The pad is tilted in one direction due to the static load, and experiences only small oscillations about the steady state tilt position, due to the dynamic unbalance load.

Another advantage the flexible pivot bearings have is the relatively high radial stiffness. The pivot stiffness of this bearing and conventional style bearings are shown in Figure 20. Note that in the case of the flexible pivot bearing, the stiffness remains constant as the load changes. This is very attractive in the case

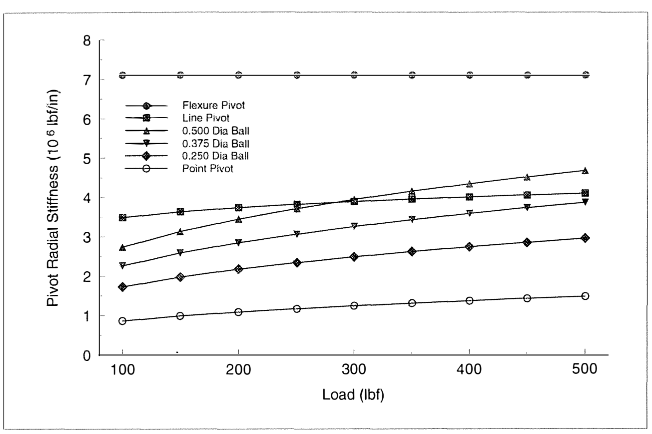


Figure 20. Comparison of Pivot Radial Stiffness for Various Pivot Geometries.

of integrally geared compressors where the bearing gear load changes with compressor load. Finally, bearings used in gear loaded applications represent a class of bearings that are subjected to the most demanding duty. The bearing unit loading typically ranges from 280 to 400 psi compared to 50 to 200 psi for conventional gravity loaded compressor applications. The flexible pivot bearing has proven to be better capable of handling these high loads (stresses) and speeds (30,000 to 80,000 rpm). This bearing design, unlike conventional tilt pad bearings, does not shift the weak link in the bearing from the babbitt to the pivot.

THRUST BEARING ANALYSIS

Fixed Geometry Bearings

The most common type of thrust bearings used in high speed compressors are taper land, Rayleigh step, and pocket bearings. These bearings are simple to manufacture and do not require a wide axial space. The requirement for a narrow axial space is very critical with high speed compressors in order to maintain high shaft rigidity. However, fixed geometry bearings are not capable of handling misalignment conditions. Consequently, this limitation with the fixed geometry bearings demands an over-designed bearing in order to account for the lack of misalignment capability. As a result, the bearing design is far from optimum and a significant frictional power loss takes place.

Variable Geometry Bearings

Self-leveling tilting pad thrust bearings can accommodate misalignment, but this is done at the expense of a relatively large axial space. Therefore, they are not considered as a viable option with high speed integrally geared compressors. Nonequalized tilt pad thrust bearings will require high precision due to the variation in manufacturing tolerance, and at high speeds, the inactive bearing will often experience pad flutter.

Flexible pivot thrust bearings offer a valuable alternative which is very well suited for these types of applications. The tilting motion and convergent wedge is achieved through the flexure of the pad top and of the post. The capability of including the structural finite element model of the pad into the finite element solution of the Reynolds lubrication equation, provides an accurate means of evaluating the performance characteristics of the thrust bearing. Including the crowning of the pad top under load allows optimization of the crown-to-minimum film thickness ratio. This will allow the pad to carry higher loads without sacrificing performance or reliability. The APPENDIX includes a derivation of the analysis routine used to analyze flexible pivot thrust bearings.

FLEXIBLE PIVOT THRUST BEARING DESIGN

The flexible pivot thrust bearings being discussed utilize the elastic deformations of the bearing structure to provide the converging wedge. In most oil lubricated bearings, the minimum film thickness is on the order of one mil. Since the optimum wedge ratio for a tilt pad bearing is around 2.5, this translates to a leading edge tilt or deflection on the order of 2.5 mils. Therefore, the thrust pad does not have to deflect much in order to provide optimum performance. Consequently, the stresses are very low and small compared to the contact stresses in a conventional tilting pad pivot bearing. Adequate safety margins exist in this bearing design to maintain a high degree of reliability.

The bearing performance characteristics are arrived at by solving the Reynolds and energy equations using a finite element formulation [2] that accounts for the structural properties of the pad. The mechanical and thermal deformations of the thrust pads are included in the analytical model [3]. Including mechanical and thermal deformations is critical to an accurate determination of the film thickness, wedge ratio, and power loss, etc. These effects are often ignored in conventional bearing analysis and, as a result, the information on the pad crown profile is not known. In thin section bearings and in large pads, the ability to model the pad structural and thermal deformations is of paramount importance, since this has a significant influence on the crown profile. An excessive crown profile, as shown to the right of the shaded region in Figure 21, results in lower load carrying capability. The shaded area represents a region of optimum performance where the crown height-to-film thickness ratio can be optimized for higher load carrying capability and lower power loss. The stress and temperature distribution along the surface and throughout the pad are also provided by the analysis and allow an accurate determination of the stress levels and safety factors.

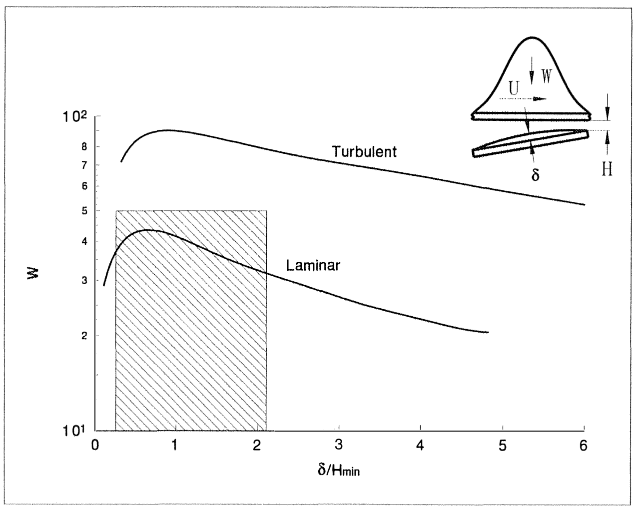


Figure 21. Effect of Film Thickness to Crown Ratio on the Load Carrying Capacity.

The thrust bearing in this configuration is designed to carry a load of 200 lb at 76,500 rpm when only one impeller stage is used. When two impellers or two stages are utilized, the maximum thrust load is around 100 lb. As mentioned earlier, the desired maximum speed in the future will be increased by 20 to 30 percent over the current speed. The thrust pads are made of a bearing grade copper alloy that has excellent thermal conductivity. This will aid in carrying the heat away from the oil film.

A finite element model of the thrust pad is shown in Figure 22. The pad has an offset pivot to provide lower power loss and higher load carrying capability. The circular pad geometry is selected for ease of manufacturing (turning), and because the surface area did not have to be maximized. The power loss and minimum film thickness as a function of load at 76,500 rpm are shown in Figure 23. The stress contours in the pad under load are shown in Figure 24. The maximum stress is 17,800 psi which results in a safety factor of 7.9.

A six pad thrust bearing was optimized to meet the design requirement of 200 lb thrust load at 76,500 rpm. The pads are assembled in the carrier with an additional height stock allowance. The pads are then ground and lapped as an assembly, thus eliminating the stackup of manufacturing tolerances associated with individual components. The carrier was designed to reduce the parasitic losses which are difficult to calculate but can be deduced from the test measurements.

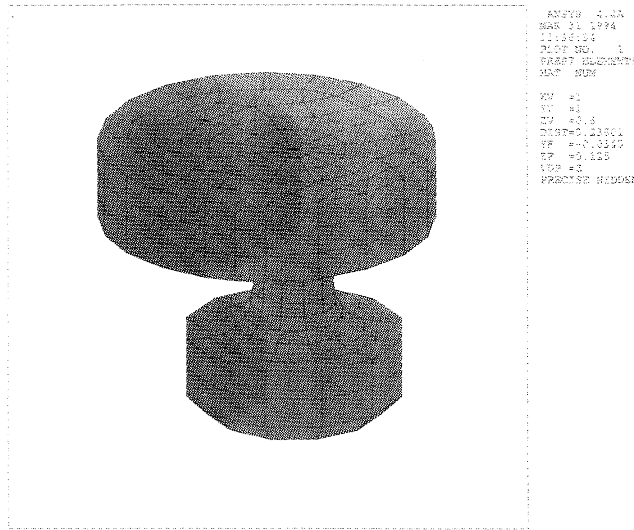


Figure 22. Finite Element Model of the Thrust Pad.

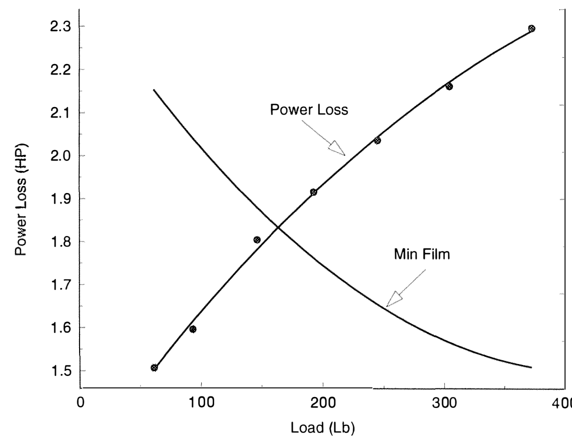


Figure 23. Flexible Pivot Thrust Bearing Power Loss and Film Thickness as a Function of the Load.

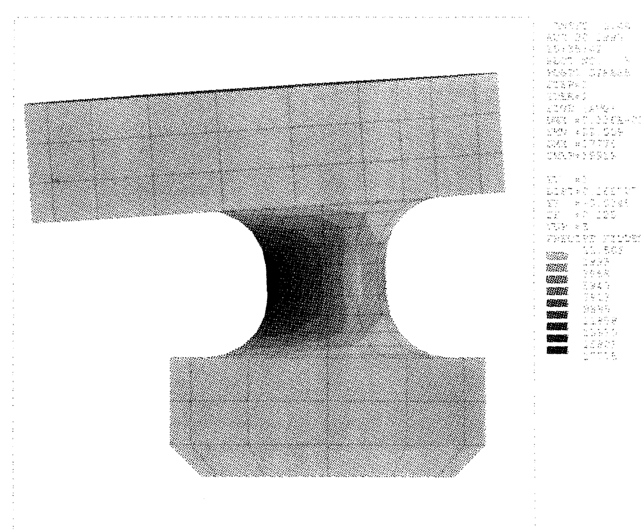


Figure 24. Finite Element Stress Contour Plot of the Thrust Pad.

Table 3. Comparison of Thrust Bearing Predicted Power Loss at Test Conditions. Speed = 76000 rpm; load = 100 lb; oil inlet temp = 160°F; and flowrate = 2 gpm.

Bearing Type			*Power Loss (HP)*
TPB (Baseline)	Active	Pocket	1.72[1]
	Inactive	Taper/Land	2.86
	Total		4.58
FPB	Active	Flex Pivot	1.67
	Inactive	Flex Pivot	1.05
	Total		2.72
Difference			1.86

A predicted power loss comparison of the TPB (baseline) thrust bearing to the FPB thrust bearing under the test conditions is presented in Table 3. The power loss of a FPB bearing is 40 percent lower than the power loss expected of the existing TPB (active-pocket and inactive-taper/land) thrust bearing. Since there is no improvement in the power loss due to the FPB radial bearing, and the power loss in the gear system is expected to be the same for the TPB and FPB system, any power loss saving can be attributed to the FPB thrust bearings. The experimental observations, which are discussed in the next section, show that the average power loss saving of the FPB compared to the TPB is 2 to 2.5 HP. This correlates well with the analytical predictions.

BEARING TESTING AND EVALUATION

Test Procedure and Layout

The testing of the bearing was performed at the OEM's test facility using a production compressor. A schematic layout of the instrumentation plan is shown in Figure 25. Eddy current proximity probes were used to measure shaft vibrations. Temperature sensors and flow meters were used to measure the lubricant temperature to and out of each bearing and the corresponding oil flow. The temperature rise and the oil flow rate were used to calculate the power loss in the journal and thrust bearings. The compressor uses MIL-7808, a synthetic oil for lubricating the gears and bearings. For the purpose of this investigation, the oil piping and controls have been slightly modified. The temperature regulating valve located upstream of the oil pump was disabled, and a control valve was installed in the hot oil line parallel to the oil cooler line. This allowed the oil inlet temperature to be varied. The oil pressure relief valve was completely closed to maintain the same oil flow to the compressor as indicated by the dotted line in Figure 25. A constant oil flow of 8 GPM was maintained throughout the test. The auxiliary driven prelube pump was on only during startup, shutdown, and idle time after a shutdown for cooling purposes. When the compressor reached operating speed, the main oil pump driven by the bull gear supplied oil to the bearings. Two separate flow meters were located at the feed to each bearing. Numerous thermocouples were installed at various locations for temperature measurements. This arrangement allowed determination of the power losses in each combination radial/thrust bearing, and across the entire compressor train. The oil inlet temperature to the compressor was varied by adjusting the hot oil flow control valve parallel to the cold oil through the oil cooler. The total power loss (frictional plus parasitic) was calculated using Equation (1).

$$hp = 0.1967 \cdot gpm \cdot density \cdot C_p \cdot \Delta T \qquad (1)$$

where,

hp	power loss (hp)
0.1967	a units conversion constant
gpm	flowrate (gpm)
density	fluid density (g/cc)
C_p	fluid specific heat (BTU/lb/°F)
ΔT	average temperature rise (°F)

The configurations of the three bearings tested are listed in Table 4. The third configuration is the same as the second with some minor changes to the thrust bearing carrier in order to reduce the parasitic losses.

Table 4. Bearing Configurations Tested.

Design Cases	Stage 1		Stage 2	
	Radial	Thrust	Radial	Thrust
Design 1 - TPB	Tilt Pad	Taper Land	Tilt Pad	Pressure Pocket
Design 2 - FPB	Flex Pad	Flex Pad	Flex Pad	Flex Pad
Design 3 - FPB	Flex Pad	Flex Pad	Flex Pad	Flex Pad

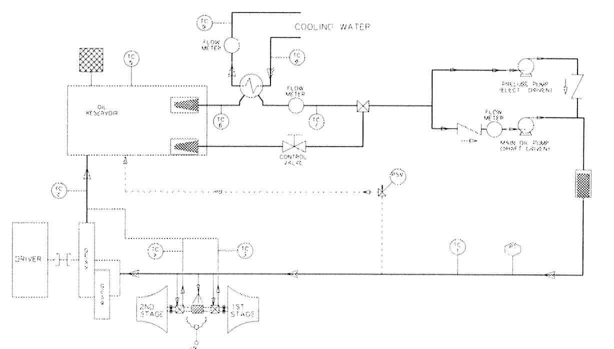


Figure 25. Schematic Layout of the Test Instrumentation.

Vibration Measurements

The vibrations were measured using an eddy current proximity probe system. The data was then processed through a frequency spectrum analyzer. The steady state vibration frequency spectra for the tilt pad and flexible pivot bearings, superimposed on the same plot, are shown in Figure 26. As predicted in the unbalance response analysis shown in Figure 17, the response with the flexible pivot bearing is about 50 percent lower than with the conventional tilt pad bearing. Furthermore, the tilt pad bearing had lower damping as evidenced by the subsynchronous components present in the frequency spectra.

Vibration data were also recorded during a surge cycle of the compressor. The subsequent vibrations spectra for both bearing types, superimposed on the same plot, are shown in Figure 27. Note that in this case too, the vibrations with the conventional style tilt pad bearing were higher and increased much more than the vibrations with the flexible pivot bearings. These bearings did not show any change or increase in subsynchronous vibrations, whereas there was a significant increase with the tilt pad bearings.

Measurements of Bearing Power Loss

The temperature rise was measured for the bearings corresponding to each of the two compressor stages. These measure-

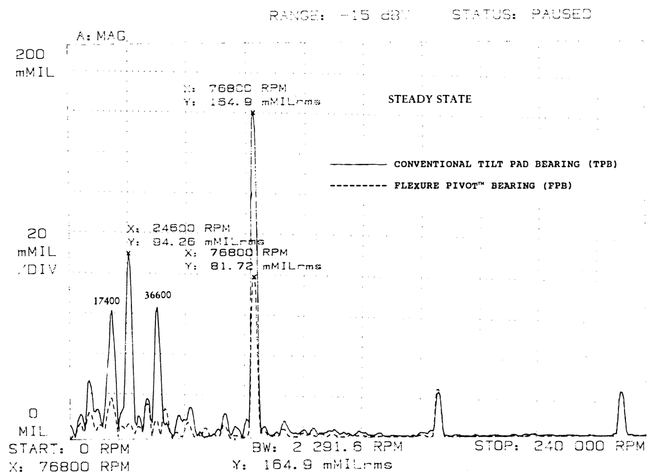


Figure 26. Steady State Vibrations for the Tilt Pad and Flexible Pivot Bearings.

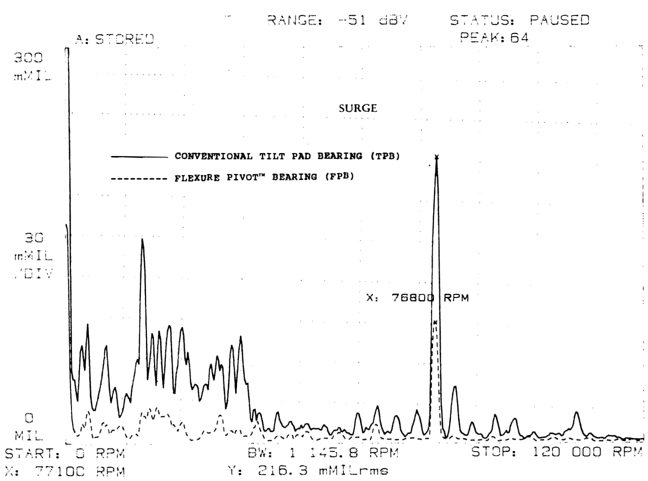


Figure 27. Vibrations Measured Under Surge Conditions for the Tilt Pad and Flexible Pivot Bearings.

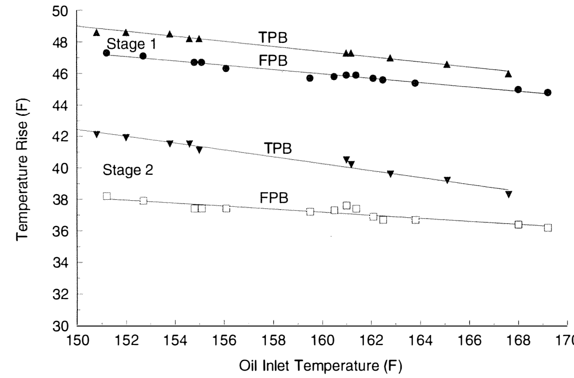


Figure 28. Measured Oil Temperature Rise as a Function of Oil Inlet Temperature.

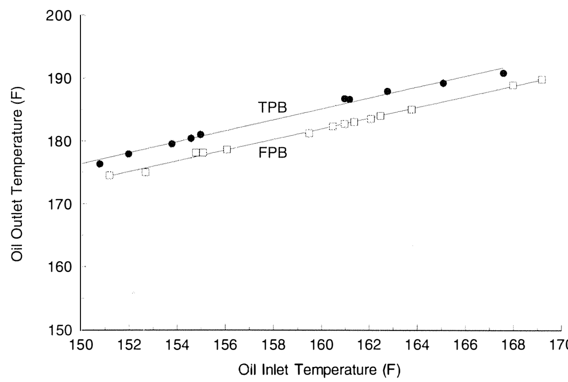


Figure 29. Oil Outlet Temperature as a Function of Oil Inlet Temperature.

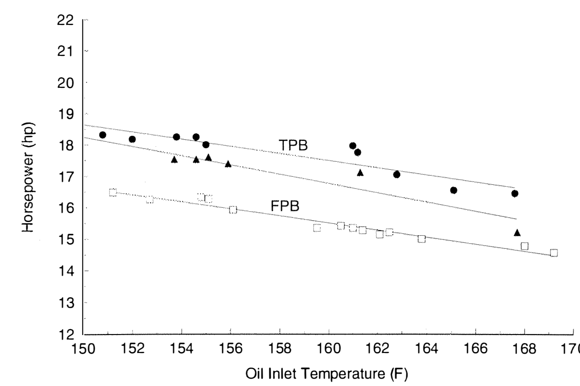


Figure 30. Power Loss as a Function of Oil Inlet Temperature.

ments were made while the oil inlet temperature was varied. Each data point was obtained after allowing all the measurement variables to reach steady state conditions. The results with both bearings are shown in Figure 28. The first stage bearing was subjected to the higher thrust load and, therefore, had the higher temperature rise. In both stages, the flexible pivot bearings experience lower temperature rise in comparison to the pocket and taper land thrust bearings. The oil outlet temperature as a function of the oil inlet temperature is shown in Figure 29. This confirms that the oil inlet temperature can still be increased to induce a further reduction in the power loss without reaching the maximum outlet temperature limit. The total power loss which includes the first and second stage journal and thrust bearings is shown in Figure 30. There is about a 2.5 HP savings with the FPB bearings in comparison to the conventional tilt pad bearings.

The first flexible pad thrust bearing carrier, shown in Figure 7, had sharp edges between pads. This configuration did not lend itself to reducing the parasitic losses. Parasitic losses, as reported by Booser and Missana [4], can account for 25 to 50 percent of the total bearing fluid power losses. The second flexible pivot thrust bearing eliminated some of the sharp edges and provided a more directed lubrication configuration. The results for all three bearing configurations can be seen in Figure 31. Since the flexible pivot thrust and journal pads were identical to those in the first configuration, the reduction in the power loss noted between the two flexible pivot bearings can be totally attributed to the effect of the parasitic losses.

CONCLUSIONS

The vibration characteristics with the flexible pivot bearings showed improvements over the conventional style tilt pad bearings during normal steady state operation and under surge conditions. The precision with which the flexible pivot bearing is manufactured, gives it a distinct advantage over conventional style tilt pad bearings. This is because the stackup in tolerances

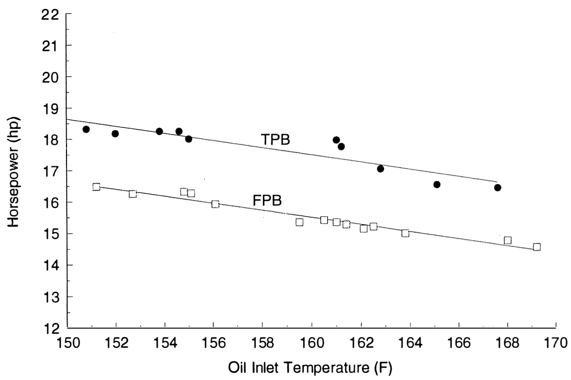


Figure 31. Power Loss as a Function of Oil Inlet Temperature for All Three Tested Bearings.

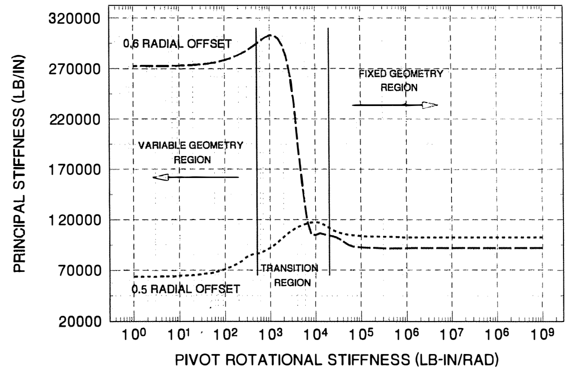


Figure 33. Principal Stiffness (Kxx, Kyy) as a Function of FPB Rotational Stiffness.

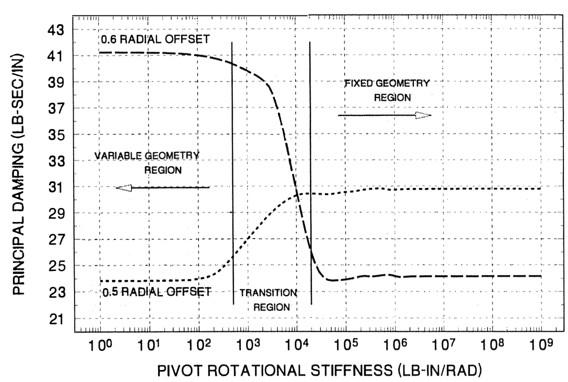


Figure 34. Principal Damping (Cxx, Cyy) as a Function of FPB Rotational Stiffness.

with conventional style tilt pad bearings results in a very wide range for the clearances and preload.

The use of an offset pivot in this application showed that the journal bearing can run 20 to 30°F cooler. The offset pivot design also showed advantages that were not clearly identified in the literature. The bearing sensitivity to clearance variations with an offset pivot was demonstrated to have a limited influence on the stability. The cross coupled stiffness for the 0.5 and 0.6 offset bearing as a function of the flexure pivot rotational stiffness is shown in Figure 32. The difference in the cross coupled stiffness between the 0.5 and 0.6 offset is negligible. The distinct difference, however, can be seen when the principle stiffness is plotted for both offset positions as a function of rotational stiffness, as shown in Figure 33. As the rotational stiffness in the pads approach the variable geometry (tilting) region, there is a dramatic increase in the direct stiffness with the 0.6 offset pad configuration. What is of even more significance is the increase in damping for the 0.6 offset pads, as shown in Figure 34. The increase in stiffness and damping is responsible for the reduction in sensitivity with variation in bearing clearance and for the higher logarithmic decrement predicted with the offset pivots.

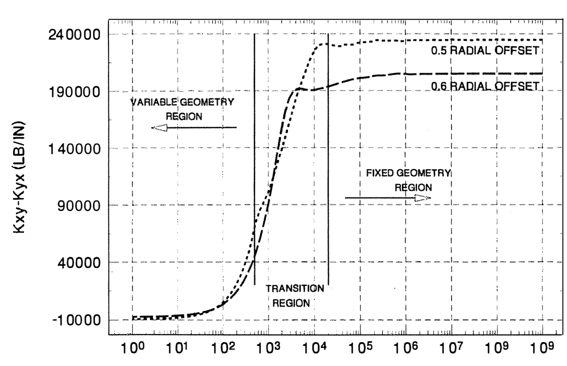


Figure 32. CrossCoupled Stiffness (Kxy - Kyx) as a Function of FPB Rotational Stiffness.

This investigation also confirmed that parasitic losses form a substantial part of the overall power losses in a bearing. The FPB bearing design reduced the frictional and parasitic losses by 2.0 to 2.5 hp. More importantly, the design of the flexible pivot bearing enables incorporating features in the thrust bearing, which can reduce the parasitic losses. Subsequently, the parasitic losses were reduced by an additional 1.0 to 1.5 hp.

APPENDIX—DESIGN AND ANALYSIS OF FPB THRUST BEARINGS

In a conventional fixed geometry fluid film bearing design, such as a taper land or a Rayleigh step bearing, the converging geometry is generated by machining the bearing surface to provide a converging wedge. A tilting pad bearing on the other hand, through its ability to tilt and form a convergent wedge geometry is classified as a variable geometry bearing. The flexible pivot bearing falls in the same category as the tilting pad bearing due to its ability to flex and tilt, thus forming a variable geometry wedge. In the flexible pivot bearing, the hydrodynamic pressure developed in the fluid induces the pad top and the flexible pivot to elastically deform. This deformation forms the converging wedge-shaped gap between the bearing surfaces.

The primary objective in the design of a flexible pivot bearing is to optimize the deformation of the bearing components, within the structural limits of the material. The differential equation governing the pressure in the fluid film is the Reynolds lubrication equation. Solving the Reynolds equation and accounting for all the factors which influence the performance characteristics is a very involved procedure. In many of the conventional bearing designs, factors such as the changes in the fluid temperature and viscosity, pad mechanical and thermal deformations, etc., are neglected. Flexible pivot bearings, on the other hand, address these effects and use them to advantage in order to generate an optimum pad profile and develop a higher load carrying capability. In a flexible pivot bearing design, a numerical solution to the Reynolds equation is obtained, followed by an energy balance calculation, and a structural elastic deformation analysis. These

neer in the State of Pennsylvania and a member of ASME, STLE, ASM and the Vibration Institute.

ABSTRACT

Experimental results are presented showing that the leading edge groove (LEG) tilting pad journal bearing has significantly lower pad operating temperatures than the conventional designs. This decrease in the operating temperature of the bearing is attributed to a reduction in hot oil carryover. The lower temperatures are then discussed in terms of the safety and reliability of the leading edge groove bearing. Results from tests to determine minimum critical flows are also presented, and these data are investigated in terms of using the reduction in operating temperature to increase the efficiency of the leading edge groove bearing. Finally, examples are given on how the leading edge groove tilting pad journal bearing may be used to improve the safety, reliability, and efficiency of high performance turbomachinery.

INTRODUCTION

For cost effective production, today's world economy demands process machines that run at their full capacity. Maximum efficiency is also demanded, since this increases throughput and profitability. When assessing turbomachinery for the power generation industry, the efficiency of the machine is valued at thousands of dollars per kilowatt, and fines for not meeting performance criteria can be of the same order as the cost of the project. Upgrading machinery may include increasing the size or running speed of the shaft, and in new machine designs, the trend is towards higher output and higher operating speeds. These upgrades and trends have a negative impact on the bearings since they all increase one or more of the following: load, speed, power loss, oil flowrate requirement and operating temperature.

Bearing power loss, which increases exponentially as the surface speed extends into the turbulent region, is of concern since these high speed losses greatly diminish the efficiency and cost effectiveness of the machine. Correspondingly, an increase in oil flowrate increases the size of the machine and the lubrication system, resulting in a higher system cost that is of the order of hundreds of dollars per gpm. Also, there are concerns about oil leakage and its impact on the environment, and bearing temperature, which is a measure of the safety and reliability of the machine. Any gain in efficiency cannot be obtained at the expense of safety and reliability. Furthermore, the cost of lost production as a result of unplanned shutdowns can be large in comparison to the cost of the original equipment.

Expectations of bearing improvements for upgraded and new machine designs are clear: increased load capacity, reduced losses, and reduced oil flow requirements, without increasing operating temperatures. In regard to thrust bearings, this technology has been available for the last decade [1], although it is only recently that these same technologies have been applied to the tilting pad journal bearing [2, 3, 4, 5]. This is because machine speeds are now nearing the limits of conventional tilting pad journal bearings.

Tests are reported of a journal bearing that specifically addresses the concerns of power loss, oil flow, bearing temperature, efficiency and safety. The bearing is the LEG (leading edge groove) tilting pad journal bearing, which is constructed so that the cool inlet oil flows directly into the leading edge of each pad. The idea behind the principle of this bearing is that, by introducing cool oil directly into the oil film wedge, the face of the bearing is isolated from the hot oil carryover that adheres to the moving surface of the journal. An earlier theoretical and experimental study [2] confirmed this hypothesis, and showed that there were substantial reductions in pad temperature. This reduction in temperature can be used to accommodate higher loads and/or higher speeds in either existing or new designs of turbomachinery. Brockwell, et al. [3], discussed how the lower operating temperature of the leading edge groove bearing can be used to reduce power loss, by increasing the oil inlet temperature. This was accomplished without exceeding the maximum temperature of the conventional bearing, which had a normal oil inlet temperature.

In this study, the authors used the reduction in pad temperature to increase machine efficiency by reducing the flowrate of oil through the bearing. Data are presented that confirm that the leading edge groove tilting pad journal bearing can significantly reduce both oil flow and power loss without exceeding the maximum temperature of the conventional bearing. This effectively increases machine efficiency, reduces the oil quantity and size of both the machine and the lubrication system, and maintains the same level of safety as that of the conventional bearing. These data are used to compare the effects of the leading edge groove design when applied to a process machine, and to a large turbogenerator set.

EXPERIMENTAL APPARATUS

Test Rig

A partially sectioned view of the test rig is shown in Figure 1. The test head consists of a shaft, with a 3.875 in diameter

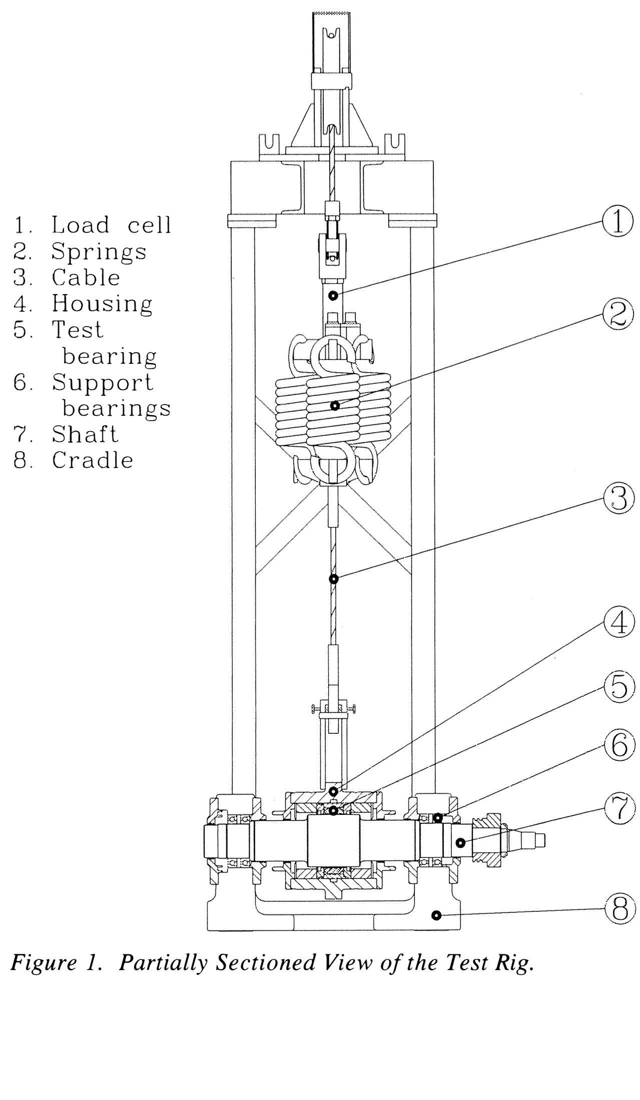


Figure 1. Partially Sectioned View of the Test Rig.

journal, that is supported on two pairs of high precision, preloaded, angular contact ball bearings. A variable speed electric motor, driving through a system of belts and pulleys, provides an operating speed range of zero through to 16,500 rpm. Positioned midway between the two support bearings is a specially constructed housing that encloses the test bearing. A vertical load is applied to this housing by means of a tensioned cable. Eddy current probes mounted on the ends of the housing measure horizontal and vertical displacements of the bearing, and provide a check of the alignment of the bearing with respect to the shaft. Spring steel flexure wires attached to the sides of the housing axially locate the test bearing and reduce nonparallel movement of the bearing that may otherwise occur as a result of misalignment of the bearing loading system. These wires are flexible in directions perpendicular to the bearing axis and do not influence the radial movement of the bearing housing.

The lubricant used in this study was an ISO VG32 turbine oil with a viscosity of 32.8 centistokes at 104°F and 5.4 centistokes at 212°F. The oil supply temperature to the test bearing was regulated between 118°F and 122°F by means of a feedback control device and a shell and tube heat exchanger.

Test Bearings

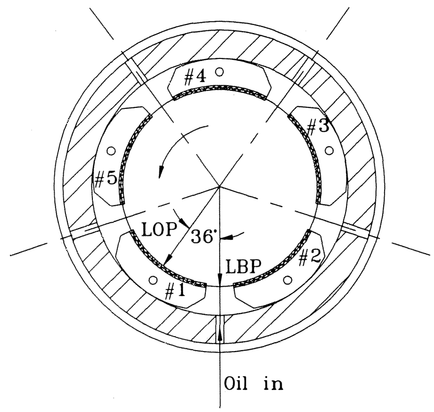
Side views of the test bearings, which have a nominal diameter of 3.875 in and a length/diameter ratio of 0.387, are shown in Figure 2, The journal pads are supported within a precisely machined aligning ring, which is split axially to permit easy assembly of the bearing around the shaft. In the conventional bearing, five radial holes direct oil from an annulus machined on the outside of the ring to the inter-pad spaces. The leading edge groove bearing uses oil feed tubes to direct the oil to the center of the leading edge groove of each pad. This groove is an extension of the pad, occupying the space between two adjacent pads. The effective pad angle of all test bearings is 56.1 degrees. The back surface of the pads of both the conventional and leading edge groove bearings is contoured axially and circumferentially to accommodate shaft misalignment. Axial pins and retaining plates locate the pads and hold them against rotation. Details of pad and bearing radial clearances, and pad preload values, are given in Table 1. Both groups comprised a 0.6 offset pivot leading edge groove bearing and a 0.5 center pivot conventional bearing. In addition, group 1 included a 0.6 offset pivot conventional bearing.

Table 1. Bearing and Pad Clearances.

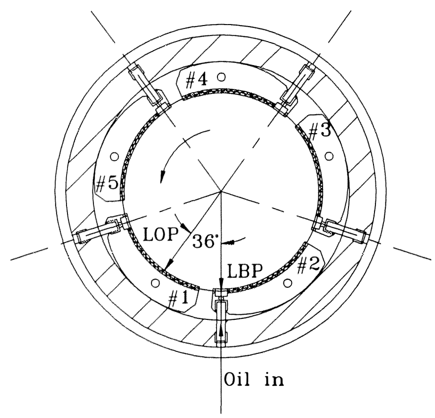
Group #	Bearing clearance, in	Pad clearance, in	Preload
1	0.003	0.004	0.25
2	0.004	0.004	0.0

Instrumentation

Copper-constantan thermocouples, mounted to within 0.02 in of the surface of the pads, measured the circumferential temperature distribution of each pad. The location of these thermocouples, defined as a percentage of the pad's arc length as measured from the leading edge, are shown in Figure 3. Other thermocouples measured oil inlet and outlet temperatures. To measure the power consumed by the main drive motor, a power cell fitted with three balanced Hall Effect devices, each with a flux concentrator, is used. The signals from each of the three phases of the power supply is summed to give an analogue output signal from the power cell, which is proportional to the three phase power. Oil flowrate is measured with a turbine flow sensor, the signal



Conventional Bearing
(Center Pivot)



LEG Bearing
(Offset Pivot)

Figure 2. Conventional and Leading Edge Groove Tilting Pad Journal Bearings.

from this sensor being transmitted to a calibrated readout device. As mentioned before, the displacement of the bearing/housing assembly with respect to the shaft is measured by eddy current probes mounted to the sides of the housing. Data from this instrumentation are processed by a host PC computer.

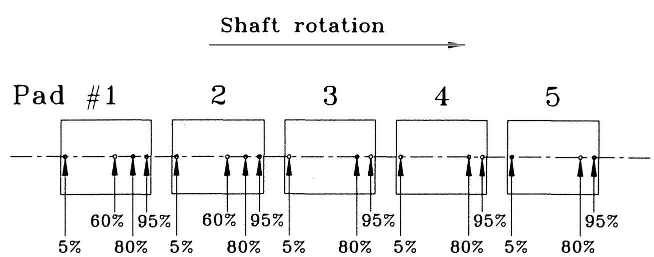


Figure 3. Thermocouple Locations in the Test Bearings.

The errors associated with the measurements are as follows:

Temperature	±2°F
Load	±0.9 percent
Shaft rotational speed	±0.6 percent
Thermocouple location	±0.02 in
Flowrate	±1.0 percent
Power	±0.25 percent

EXPERIMENTAL PROGRAM

In this study, both the conventional and the leading edge groove bearings were tested for two directions of load, viz., load between pads (LBP) and load on pad (LOP). The performance of the bearings was monitored for loads of 290 lb, 1,200 lb, and 2,500 lb, and shaft speeds that ranged between 1,800 rpm and 16,500 rpm. Nominal oil flowrates are dependent on load and speed and are given in Table 2. In some tests on the group 2 bearings, the oil flowrate was reduced below the recommended nominal values given in Table 2. In such cases, the actual flowrate is presented as a percentage of the nominal flowrate.

Table 2. Nominal Volume Oil Flowrates (US gpm).

	Shaft speed, rpm (ft/sec)					
Load, lb (lb/in^2)*	1800 (30)	3600 (60)	5000 (83)	9000 (150)	12000 (200)	16500 (275)
290 (50)	0.23	0.62	0.96	2.17	3.35	5.71
1200 (207)	0.34	0.83	1.21	2.47	3.67	6.04
2500 (430)	0.46	1.07	1.51	2.94	4.24	6.77

*Unit load - based on the projected area of the bearing

EXPERIMENTAL RESULTS

Bearing Temperatures

The 0.25 preload bearings (group 1) with center and offset pivot conventional pads and offset pivot leading edge groove pads were tested for the purpose of assessing the difference in operating temperature between the different pad designs. A typical set of results showing pad temperature distribution in the circumferential direction, for LOP and LBP, are shown in Figures 4 and 5, respectively. The shaft speed is 16,500 rpm, the load is 1,200 lb, and the flowrate is 100 percent of nominal. With LOP (Figure 4), the applied load is supported predominantly by the pad at the bottom-dead-center location of the bearing, i.e., pad number 1. It is this pad that experiences the highest operating temperatures, which in the case of the conventional center pivoted bearing is approximately 217°F. It is of interest to note that this maximum temperature occurs in the vicinity of the 80 percent location, and that there is a reduction in temperature between this location and the pad's trailing edge. Also, at the 80 percent location, the temperature of the loaded conventional offset pivot pad is approximately the same as that of the loaded center pivot pad. However, unlike the center pivot bearing, the temperature of the offset pivot pad continues to climb to a maximum of 232°F at the trailing edge. Thus, the conventional offset pivot bearing has a higher maximum temperature than that of the center pivot bearing. The leading edge groove bearing also has an offset pivot pad, so its maximum temperature also occurs at the trailing edge, although the overall operating temperatures are substantially lower than those of the conventional bearings. For the case presented in Figure 4, the maximum recorded temperature of the leading edge groove bearing is 203°F. For LBP orientation (Figure 5), where the load is shared between the two bottom pads number 1 and number 2 (one each side of the load line), the shape of the temperature distributions on the loaded pads is similar to that obtained from the LOP test. Also, the conventional offset pivot pad exhibits the highest temperature (240°F), while the leading edge groove pad is the coolest at 198°F.

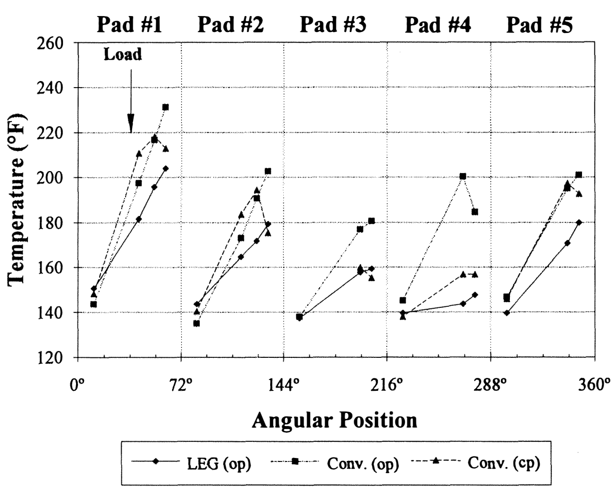


Figure 4. Pad Temperature Profiles; Conventional and Leading Edge Groove Bearings 0.25 Preload, LOP, 100 Percent Nominal Flowrate 16,500 RPM, 1,200 LB.

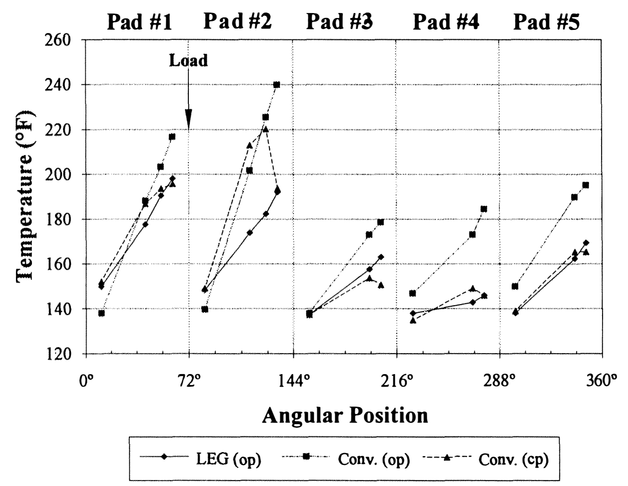


Figure 5. Pad Temperature Profiles; Conventional and Leading Edge Groove Bearings 0.25 Preload, LBP, 100 Percent Nominal Flowrate 16,500 RPM, 1,200 LB.

Results of pad temperature for LOP and LBP are summarized in Figures 6, 7, 8, and 9. Here, temperatures are plotted against

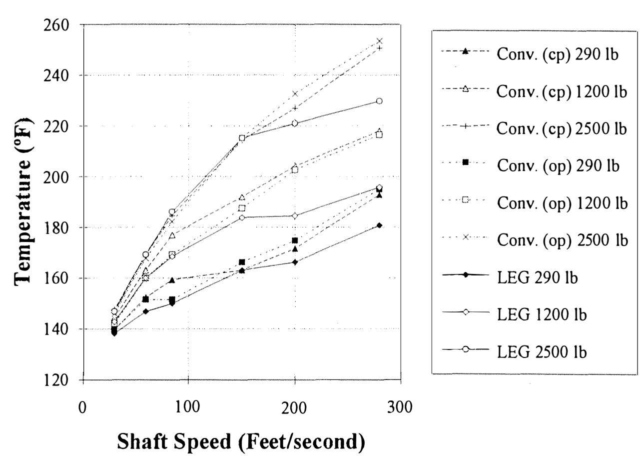


Figure 6. Variation of 80 Percent Pad Location Temperature with Shaft Speed 0.25 Preload Conventional and Leading Edge Groove Bearings LOP, 100 Percent Nominal Flowrate.

shaft speed for loads of 290 lb, 1,200 lb, and 2,500 lb. Temperatures recorded on the hottest pad at the 80 percent location are plotted in Figures 6 and 7 and maximum recorded pad temperatures are shown in Figures 8 and 9. As pointed out earlier, the location of the maximum pad temperature is dependent on the position of the pivot. Also, in the majority of cases, the conventional offset pivot bearing has the highest maximum temperature. A few exceptions are noted at lower speed with LOP when this bearing has the same, or a slightly cooler, temperature than the conventional center pivot bearing (Figures 6 and 8). However, the leading edge groove bearing runs significantly cooler at higher speeds, especially with LBP. Pad temperature profiles from the three pad designs, for both LOP and LBP, are shown in Figures 10, 11, 12, 13, 14, and 15. The shaft speed is 16,500 rpm. The profiles provide a different perspective for comparing bearing temperatures and a discussion of these results, in relation to API Standard 670 [6], is presented later in the paper.

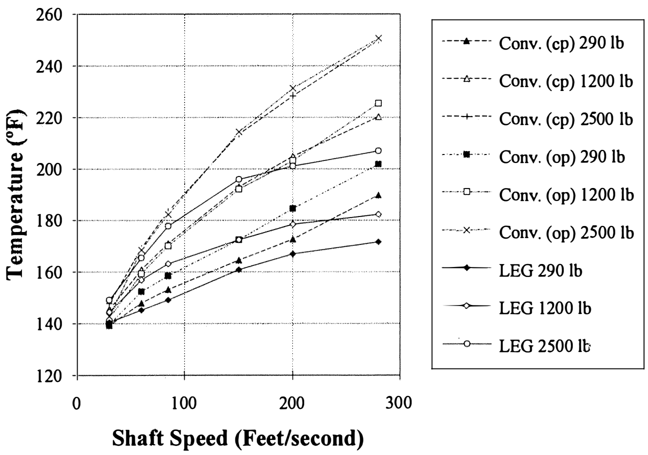


Figure 7. Variation of 80 Percent Pad Location Temperature with Shaft Speed 0.25 Preload Conventional and Leading Edge Groove Bearings LBP, 100 Percent Nominal Flowrate.

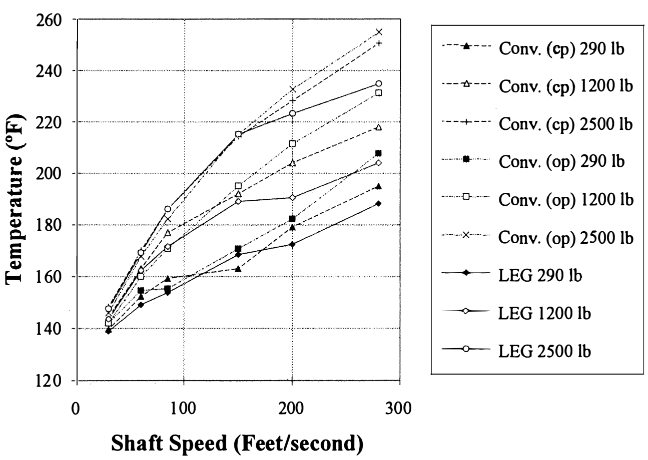


Figure 8. Variation of Maximum Pad Temperature with Shaft Speed 0.25 Preload Conventional and Leading Edge Groove Bearings LOP, 100 Percent Nominal Flowrate.

The study also featured tests on zero preload (group 2) conventional and leading edge groove bearings, although it should be noted that a comparison is drawn between only the conventional center pivot bearing and the offset pivot leading edge

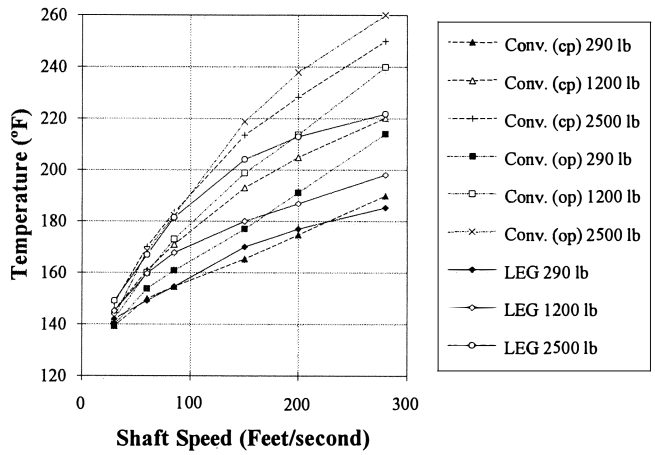


Figure 9. Variation of Maximum Pad Temperature with Shaft Speed 0.25 Preload Conventional and Leading Edge Groove Bearings LBP, 100 Percent Nominal Flowrate.

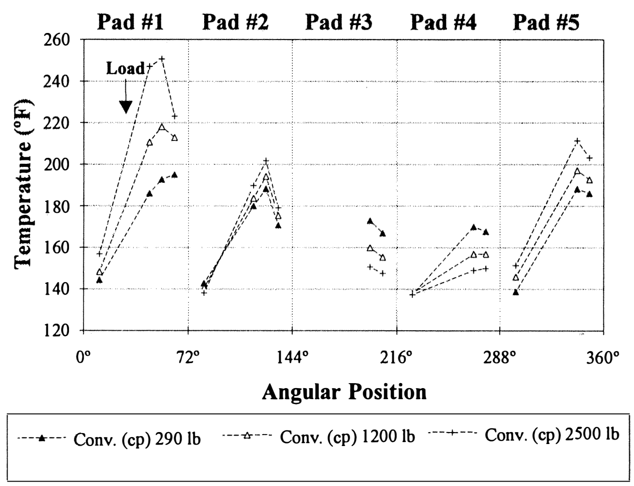


Figure 10. Pad Temperature Profiles; Conventional Center Pivot Bearing 0.25 Preload, LOP, 100 Percent Nominal Flowrate 16,500 RPM. 290, 1,200, and 2,500 LB Loads.

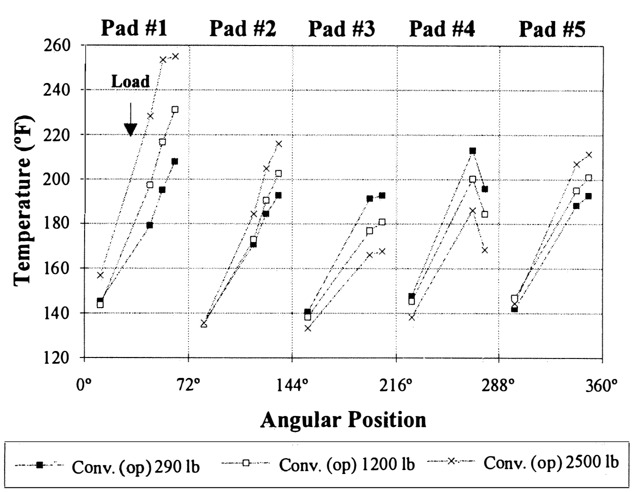


Figure 11. Pad Temperature Profiles; Conventional Offset Pivot Bearing 0.25 Preload, LOP, 100 Percent Nominal Flowrate 16,500 RPM. 290, 1,200, and 2,500 LB Loads.

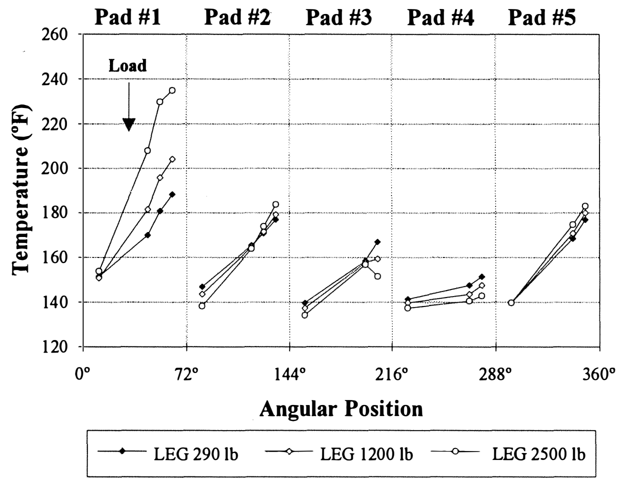


Figure 12. Pad Temperature Profiles; Leading Edge Groove Offset Pivot Bearing 0.25 Preload, LOP, 100 Percent Nominal Flowrate 16,500 RPM. 290, 1,200, and 2,500 LB Loads.

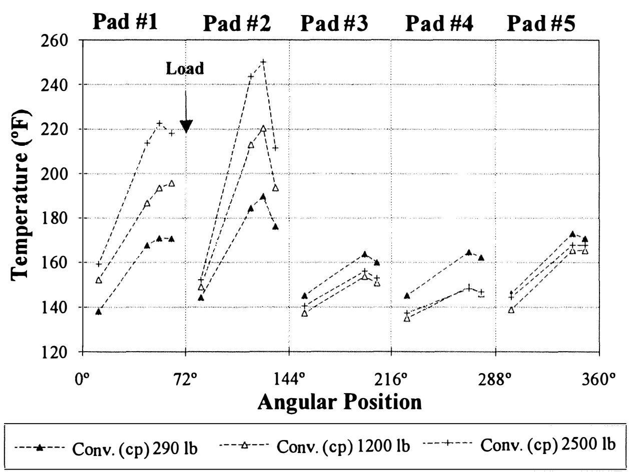


Figure 13. Pad Temperature Profiles; Conventional Center Pivot Bearing 0.25 Preload, LBP, 100 Percent Nominal Flowrate 16,500 RPM. 290, 1,200, and 2,500 LB Loads.

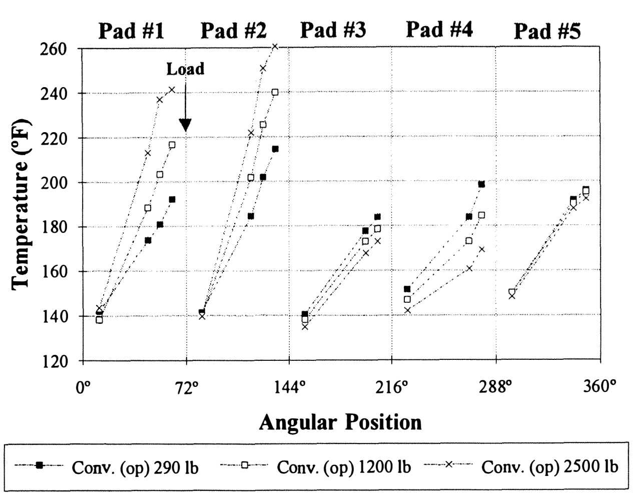


Figure 14. Pad Temperature Profiles; Conventional Offset Pivot Bearing 0.25 Preload, LBP, 100 Percent Nominal Flowrate 16,500 RPM. 290, 1,200, and 2,500 LB Loads.

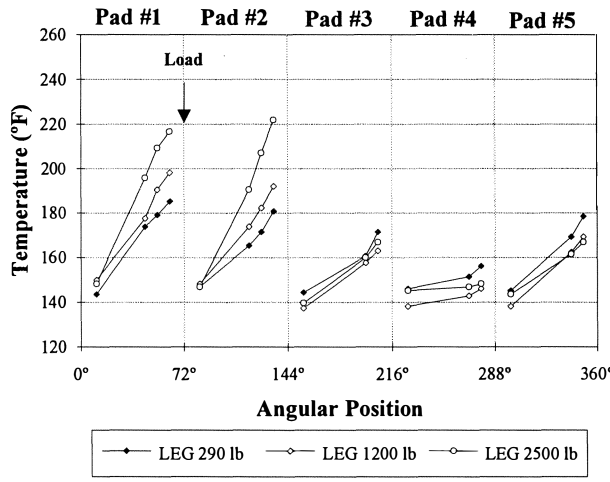


Figure 15. Pad Temperature Profiles; Leading Edge Groove Offset Pivot Bearing 0.25 Preload, LBP, 100 Percent Nominal Flowrate 16,500 RPM. 290, 1,200, and 2,500 LB Loads.

groove bearing. At the time of publication, data for the conventional offset pivot pad were not available. However, since this bearing was known to exhibit the highest operating temperatures during the group 1 bearing tests, excluding it from this section of the study was considered to be justifiable. Results of bearing temperature for LOP and LBP are summarized in Figures 16, 17, 18, and 19. Eighty percent location and maximum temperatures are plotted against shaft rotational speed for loads of 290 lb, 1,200 lb, and 2,500 lb. As with the group 1 bearing tests, the maximum temperature of the leading edge groove pad occurs at the trailing edge, and at the 80 percent location in the case of the conventional pad. In comparing the temperatures of the group 2 bearing with those of the group 1 bearing, it can be seen that increasing the bearing clearance generally results in a reduction in the maximum temperature of both the leading edge groove and conventional bearings. In some cases, this reduction is of the order of 18°F or more.

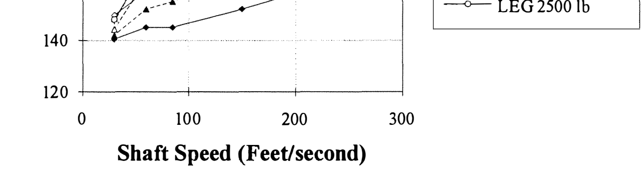


Figure 16. Variation of 80 Percent Pad Location Temperature with Shaft Speed Zero Preload Conventional and Leading Edge Groove Bearings LOP, 100 Percent Nominal Flowrate.

The results obtained from the zero preload bearings are similar to those obtained from the 0.25 preload bearings, in that the leading edge groove bearing runs noticeably cooler than the conventional bearing designs. This is particularly true of the

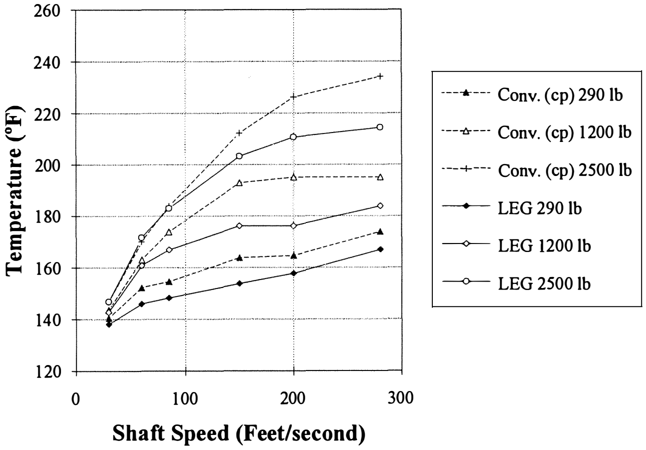


Figure 17. Variation of 80 Percent Pad Location Temperature with Shaft Speed Zero Preload Conventional and Leading Edge Groove Bearings LBP, 100 Percent Nominal Flowrate.

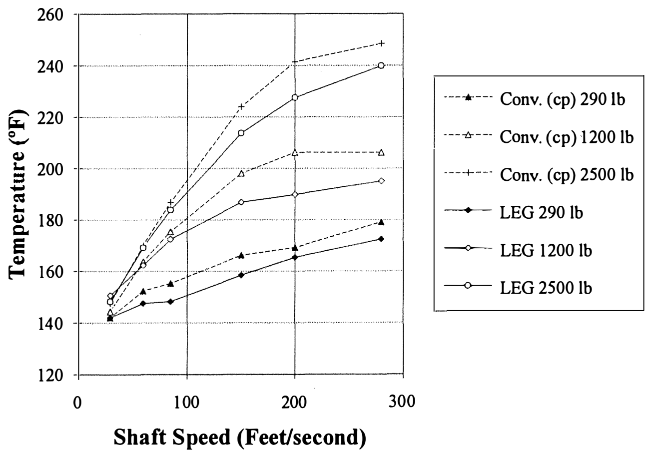


Figure 18. Variation of Maximum Pad Temperature with Shaft Speed Zero Preload Conventional and Leading Edge Groove Bearings LOP, 100 Percent Nominal Flowrate.

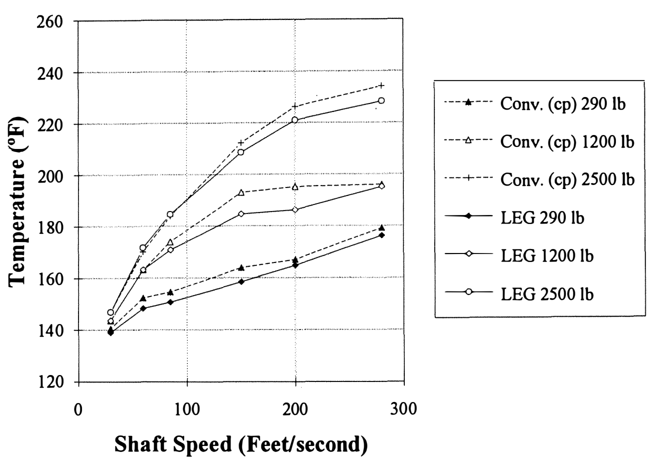


Figure 19. Variation of Maximum Pad Temperature with Shaft Speed Zero Preload Conventional and Leading Edge Groove Bearings LBP, 100 Percent Nominal Flowrate.

LOP tests, where this difference in temperature is evident to a much lower speed.

Oil Flow Reduction

The oil flowrate to the tilting pad journal bearing is normally adjusted for each combination of load and speed, so as to give a temperature rise between supply and drain of approximately 25°F to 35°F. A temperature rise of this order is typical of industrial usage, and it is upon this basis that the flowrates given in Table 2 have been derived. It is of interest to note that the flow varies more with speed than load, since shaft speed has a much larger influence on the power loss of the bearing. However, as long as the bearing is not actually starved of oil, usually it is possible to operate with oil flowrates that are lower than those recommended for normal industrial usage. The three main effects of reducing the oil flowrate are: (1) an increase in the operating temperature of the bearing; (2) an increase in the temperature of the oil leaving the bearing casing, and; (3) a reduction in the power loss of the bearing.

In a previous study [3] of the conventional and leading edge groove bearings, it was shown that flow reductions of up to 50 percent could be made without increasing excessively the operating temperature of the bearing. In this latest study, the absolute minimum oil flow requirements of the leading edge groove and conventional bearings, for a wide range of speeds and loads, were investigated. These tests were carried out by gradually reducing the oil flow to the bearing until one or more of the following four operating conditions were observed:

- The maximum pad temperature exceeded 266°F (limit for babbitt type bearings)
- Pad temperatures did not stabilize. Usually this is a sign that either the oil flowrate is insufficient to remove the heat generated in the bearing, or that there is local contact between the shaft and bearing surfaces.
- Pad temperatures fluctuated rapidly. Usually an indication of local contact between the shaft and bearing surfaces.
- Severe nonsynchronous vibration of the test bearing, with fluctuating oil inlet pressure. This effect was observed at low speeds and loads, and mainly on the conventional bearing. The effect of oil flowrate on the vibration levels of the conventional bearing, operating at 9,000 rpm with a load of 290 lb, is shown in Figure 20. Note how vibration levels increase as a result of reducing the oil flowrate from 4.0 gpm to 1.0 gpm.

The effect of reduced oil flowrates on the 80 percent pad location temperature of the zero preload bearings, for LOP and LBP, is shown in Figures 21 and 22, respectively. The operating speed is 16,500 rpm and the loads are 290 lb, 1,200 lb, and 2,500 lb. As expected, a reduction in oil flowrate raises the operating temperature of the bearings, although it seems clear that quite significant reductions in flow (in some cases greater than 50 percent) are possible. For LBP, the reduction in oil flowrate does seem to have more effect on the conventional bearing than on the leading edge groove bearing. For example, in the case of the 1200 lb results, the temperature of the conventional bearing begins to rise quite rapidly when the flow is reduced below 50 percent of the nominal recommended value. On the other hand, this same rapid rise in temperature does not occur on the leading edge groove bearing until the flowrate is below 40 percent of nominal.

Pad temperature distributions from the zero preload conventional and leading edge groove bearings with different flowrates, for both LOP and LBP, are shown in Figures 23 and 24, respectively. The load is 1,200 lb and the shaft speed is 16,500 rpm. The LOP results indicate that the increase in pad temperature of both the leading edge groove and conventional bearings is of the order of 15°F as a result of reducing the flow to about 40 percent of the recommended nominal value. For LBP, the

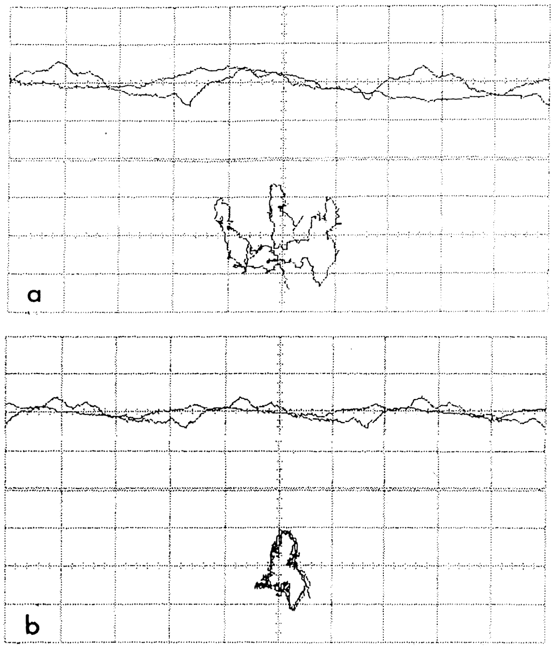


Figure 20. Effect of Volume Oil Flowrate on Bearing Whirl Orbit. a) 1 US gpm; b) 4 USgpm.

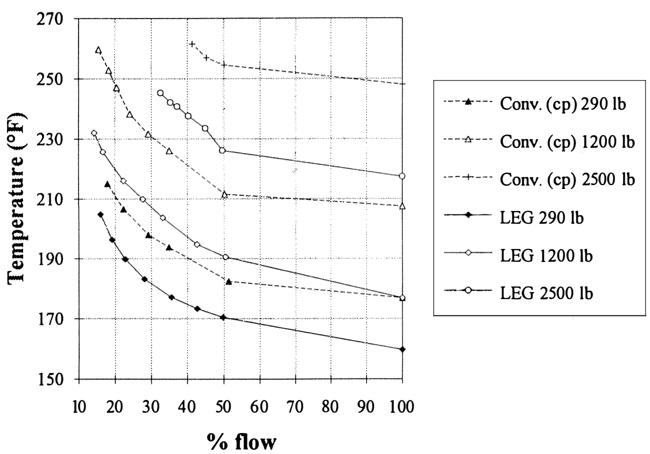


Figure 21. Variation of 80 Percent Pad Location Temperature with Percent Oil Flowrate Zero Preload Conventional and Leading Edge Groove Bearings LOP, 16,500 RPM.

same decrease in flow increases the temperature of the conventional bearing by 20°F, but only 5°F in the case of the leading edge groove bearing. Also, it should be noted that the 60 and 80 percent location temperatures of the leading edge groove loaded pads, with this reduced flowrate, are approximately 10°F lower than those of the conventional bearing with the 100 percent nominal flowrate. Furthermore, in the case of the conventional bearing with LBP, the reduction in oil flow raises the temperature of pad number 1 by only 5°F. However, much larger increases in temperature are evident on pad number 2.

Oil Drain Temperature

Information on the variation of the temperature rise of the oil (between supply and drain) with oil flowrate, for loads of 1,200 lb and 2,500 lb, is given in Figures 25 and 26. Plots are given for the conventional and leading edge groove bearings, with both

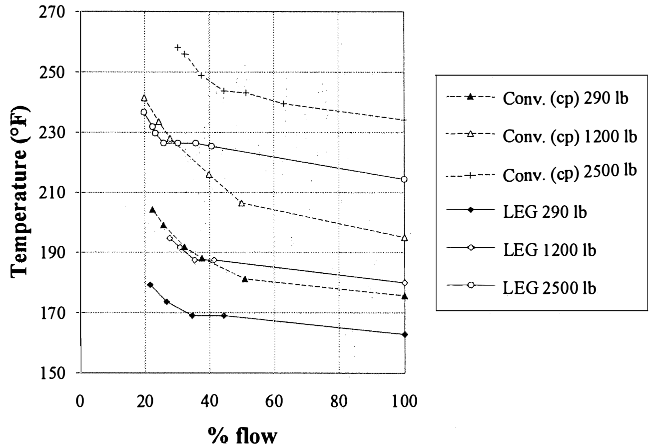


Figure 22. Variation of 80 Percent Pad Location Temperature with Percent Oil Flowrate Zero Preload Conventional and Leading Edge Groove Bearings LBP, 16,500 RPM.

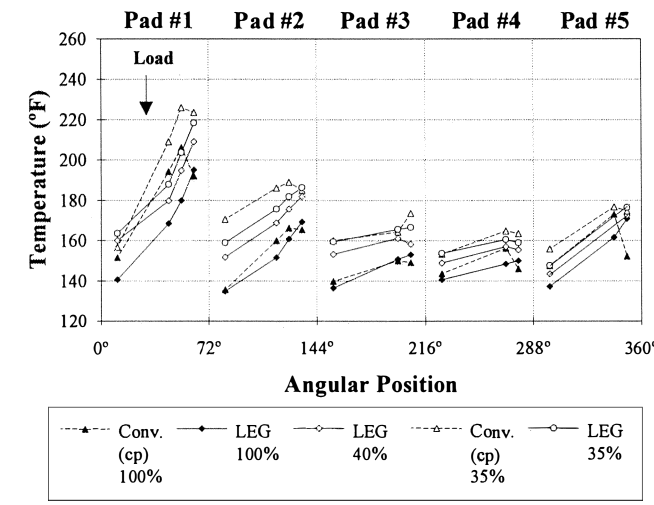


Figure 23. Pad Temperature Profiles; Zero Preload Conventional and Leading Edge Groove Bearings LOP,16,500 RPM, 1,200 LB 100, 40 and 35 Percent Nominal Flowrates.

LOP and LBP. The shaft rotational speed is 16,500 rpm. There are a number of features associated with these plots. First, it is confirmed that the 100 percent nominal flowrate results in a temperature rise that is generally of the order of 30°F, and is therefore in accordance with what is considered to be normal industrial practice. Second, the temperature rise increases as the flow is reduced, which also increases the drain temperature of the oil. At 50 percent of the nominal flowrate, the temperature rise has climbed to approximately 40°F to 45°F, and with extremely low flowrates of 20 to 30 percent of nominal, the temperature rise has, in some cases, further increased to upwards of 50°F. Third, for a particular flowrate, the temperature rise associated with the conventional bearing is slightly larger than that of the leading edge groove bearing.

Energy Reduction

In the course of testing the group 2 bearings to observe the effect of reduced oil flowrate on bearing temperatures, the authors noted that this same reduction in flowrate resulted in considerable reductions in energy consumption. These readings

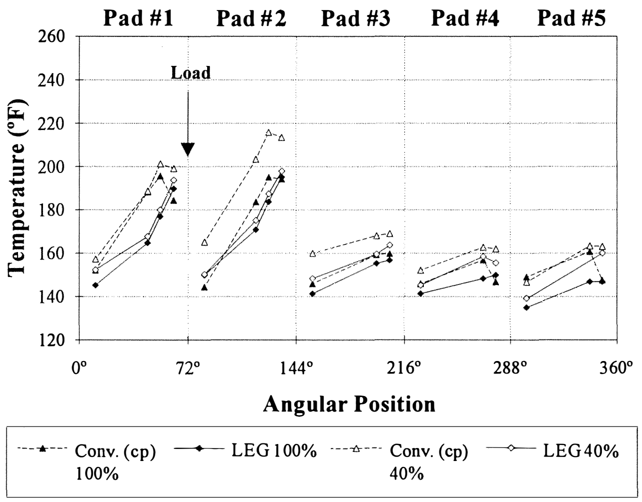


Figure 24. Pad Temperature Profiles; Zero Preload Conventional and Leading Edge Groove Bearings LBP, 16,500 RPM, 1,200 LB 100 and 40 Percent Nominal Flowrates.

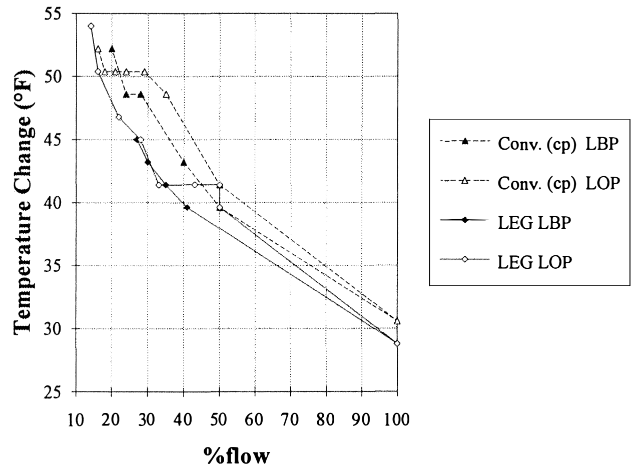


Figure 25. Variation of Lubricant Temperature Rise with Oil Flowrate Zero Preload Conventional and Leading Edge Groove Bearings 16,500 RPM, 1,200 LB.

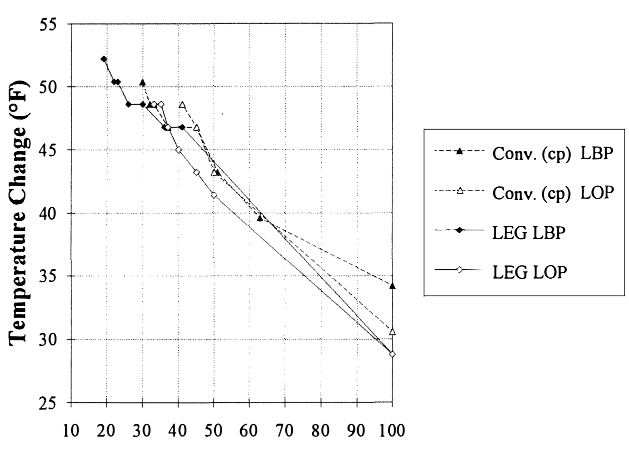


Figure 26. Variation of Lubricant Temperature Rise with Oil Flowrate Zero Preload Conventional and Leading Edge Groove Bearings 16,500 RPM, 2,500 LB.

of energy consumption were obtained from the power cell fitted to the control system of the electric motor which, in effect, was measuring the total power consumed by the test rig. Thus, losses from the support bearings, and other miscellaneous sources of power consumption, were included in these readings. However, the authors concluded that if changes were made only to those variables that would alter the power consumption of the test bearing, and not to any of the other losses associated with the test rig, then this power cell could be used to monitor changes in the energy consumption of the test bearing. Such a variable is oil flowrate.

The variation in energy savings of the conventional and leading edge groove bearings for different oil flowrates are presented in Figures 27, 28, 29, and 30. The operating speed is 16,500 rpm and the loads are 1,200 lb and 2,500 lb. In these plots, the power saving associated with the test rig is presented as a function of the percentage of the nominal oil flowrate through the bearing. To calculate this power saving, the power loss of the test rig with the flowrate at 100 percent of nominal is first determined. The results show that power loss savings associated with the conventional and leading edge groove bearings are similar and reach upwards of 30 percent when the flow is reduced to 30 percent of the recommended nominal oil flowrate. It should be noted that, at the conclusion of the tests, the bearings were in good condition and showed no signs of distress as a result of operating with these reduced flowrates.

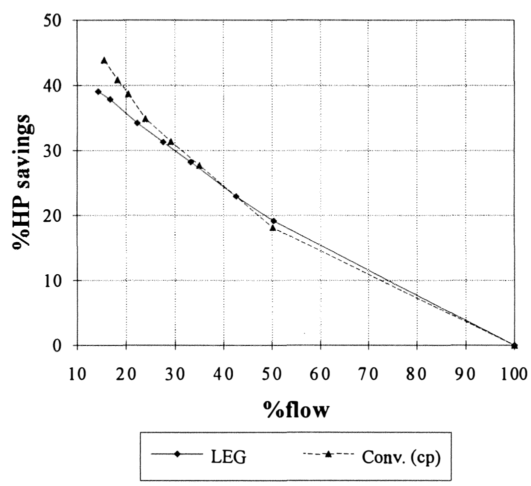


Figure 27. Percentage Power Savings Vs *Percent Oil Flowrate LOP, 16,500 RPM, 1,200 LB.*

DISCUSSION

Bearing Temperatures

The effect of the different pad designs on the operating temperature of the bearing can be assessed by examining Figures 4 through 19, inclusively. As indicated earlier, the leading edge groove bearing has a lower maximum temperature than the conventional bearings, particularly at higher shaft speeds. In the case of this bearing with offset pivot pads, the maximum temperature of the loaded pad(s) occurs at the trailing edge. The maximum temperature of the conventional offset pivot bearing is also at the trailing edge, but in the case of the center pivoted conventional bearing, the hot spot is in the vicinity of the 80 percent location. Interestingly, the shape of the temperature profiles of the conventional center pivot bearing are similar to those recorded by others [7, 8]. Depending on the bearing configuration and operating conditions, the maximum tempera-

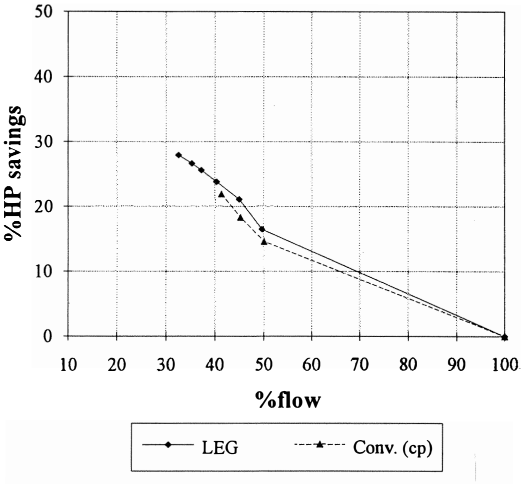


Figure 28. Percentage Power Savings Vs *Percent Oil Flowrate LOP, 16,500 RPM, 2,500 LB.*

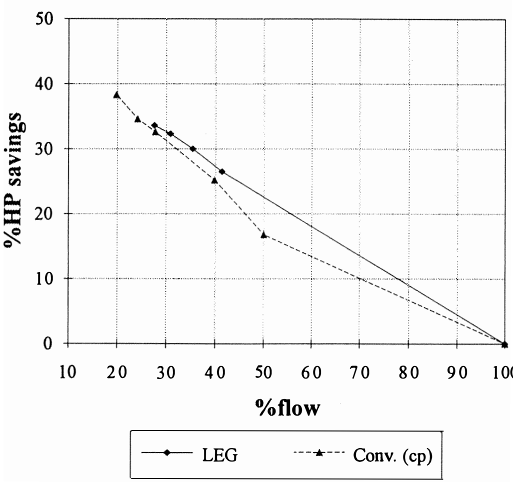


Figure 29. Percentage Power Savings Vs *Percent Oil Flowrate LBP, 16,500 RPM, 1,200 LB.*

ture of the leading edge groove bearing is as much as 44°F lower than the conventional bearing.

The reason for the lower operating temperature of the leading edge groove bearing has been discussed in detail [2]. Briefly, the amount of cool lubricant entering the oil film directly adjacent to the pad surface is increased by feeding cool oil directly to the leading edge groove. Additionally, this has the effect of reducing the amount of hot oil carried over from the previous pad (earlier studies [9, 10] have confirmed that this hot oil adheres to the surface of the shaft). Thus, the oil temperature at the leading edge of the leading edge groove pad may be represented by the following equation:

$$t_1 = \frac{q_i t_i + (q_1 - q_i) t_2}{q_1} \tag{1}$$

where:

t_1 = oil temperature at the leading edge
t_2 = oil temperature at the trailing edge of the preceding pad
t_i = oil inlet temperature to the bearing
q_1 = oil flow at the leading edge

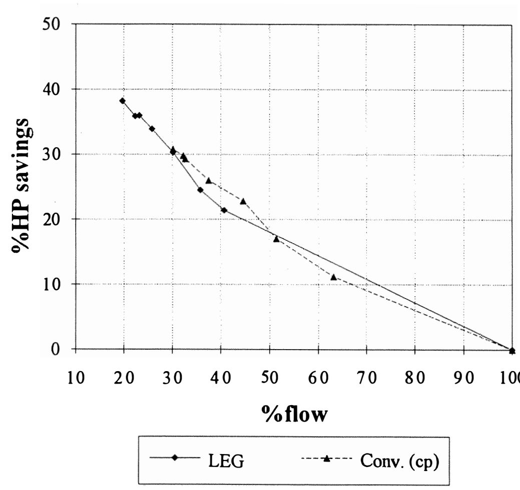


Figure 30. Percentage Power Savings Vs *Percent Oil Flowrate LBP, 16,500 RPM, 2,500 LB.*

q_2 = oil flow at the trailing edge of the preceding pad
q_i = oil flow to each pad (total flow/number of pads)

In the case of the conventional bearing, most hot oil leaving the trailing edge of one pad is carried over to the next pad, where it enters the oil film via the pad's leading edge. The balance of the flow comes directly from the cool oil fed to the bearing. Thus:

$$t_1 = \frac{(q_1 - q_2) t_i + q_2 t_2}{q_1} \tag{2}$$

Equations (1) and (2), when used in conjunction with the author's computer models of the conventional and leading edge groove bearings, gave calculated pad temperatures that were in good agreement with measured values [2].

In regard to bearing safety, there is the question on how to judge temperatures between different pad designs, because of the difference in location of the maximum temperature. To address this point, LOP and LBP pad temperature distributions for the three designs are shown in Figures 10, 11, 12, 13, 14, and 15 so as to compare features in respect to the placement of temperature detectors in accordance with API Standard 670 [6]. First, in regard to placement at the 75 percent location, it is shown that the maximum temperature of the loaded pad(s) shifts from the trailing edge towards the 80 percent location as the load is increased. This is most obvious for the conventional center pivot bearing. The shift in location of the maximum pad temperature of the conventional offset pivot bearing is only noticeable at the highest load, although the trailing edge temperature remains the hottest. This shift is also noticeable with the leading edge groove bearing with LOP, but less so with LBP as the pad operating temperatures are, by comparison, considerably lower. Second, in regard to LBP and placement in the second loaded pad in the direction of motion, it can be seen that, for the conventional center and offset pivot bearings, the second loaded pad in the direction of motion is significantly hotter. This is attributed to hot oil carryover, which increases the temperature of the second pad by as much as 30°F. In contrast, the temperature of the leading edge groove bearing does not increase dramatically from one pad to the next, which suggests that the second loaded pad in the direction of motion is sheltered from the effects of hot oil carryover.

With all three pad designs, as indicated in Figure 10, 11, 12, 13, 14, and 15, the maximum temperature moves towards the 80 percent location under more severe operating conditions. It is worth noting that actual bearings experiencing duress in the form of wiping, polishing or babbitt fatigue usually exhibit this damage in the vicinity of the 75 percent location. Indeed, this is the critical location where the peak film pressure, minimum film thickness and maximum pad temperature usually occurs, which suggests that this is the best location in terms of judging the merits of one bearing design against another. In the case of this study, sufficient measurements were recorded such that comparisons can be supported by entire pad temperature profiles of the alternate bearing designs.

Oil Flow

Oil flowrate can be important in a number of different ways. Decreasing the flowrate will reduce the size of the lubrication system, amounting to lower initial costs, which can be of the order of hundreds of dollars per gpm. The flowrate also affects the size of the machine in regards to oil paths, and drain sizes, and from an environmental viewpoint, a reduction in flowrate helps to reduce potential leakage rates.

The tests have confirmed that considerable reductions in oil flowrate can be made. In practice, the leading edge groove design can allow smaller flowrates as a result of its lower operating temperature. Results of pad temperature versus percent oil flowrate for the zero preload bearings are shown in Figures 21 and 22.Interestingly, even with the heaviest load of 2,500 lb, the leading edge groove bearing continued to work satisfactorily with a temperature of 227°F at a flowrate of 30 percent of the recommended nominal value, whereas the maximum temperature of the conventional bearing rose to 260°F. It should be noted that the tests were conducted under laboratory conditions, and it is not the intention of the authors to recommend that such low flowrates be used in industrial applications. However, the results from the tests do show that some reduction is possible without excessively increasing the operating temperature of the bearing.

Oil Drain Temperature

Measurements of oil drain temperature have confirmed that when the recommended nominal oil flowrates are used, the temperature rise through the bearing housing is in accordance with the guidelines outlined for normal industrial usage, i.e., 25°F to 35°F. However, when the flow to the bearing is reduced below these nominal recommended values, the temperature rise between inlet and drain increases, and may be as high as 50°F to 60°F for flows of 20 to 30 percent of the nominal recommended value. The reduced oil flowrate and higher temperature rise combine to effectively reduce the amount of energy that must be removed by the cooler. This causes no problems with existing lubrication systems, and new systems can be designed with smaller, more efficient coolers. However, at extreme conditions, a temperature rise of the order of 55°F may give rise to a high bulk oil temperature, and whether or not this is acceptable will depend on the quality and type of lubricant being used. In any case, reducing the flowrate to 50 percent of nominal, which gives a temperature rise of the order of 35°F to 45°F, would seem to be acceptable for most applications.

Power Loss

It is clear from Figures 27, 28, 29 and 30 that there are considerable reductions in energy consumption as a result of reducing the oil flowrate to the bearing. Most of these savings are from reductions in shearing losses as a result of a higher operating temperatures, and from the oil surrounding the bearing in the form of reduced churning losses. According to this study, the energy savings associated with the bearing can be as high as 40 percent if a 60 to 70 percent reduction in flowrate is made. However, such large reductions in flowrate are probably unacceptable from the point of view of bearing and oil drain temperatures, and it is more likely that the flowrate should be kept to 40 to 50 percent of the nominal recommended value. Even so, this flowrate is expected to give energy reductions that are of the order of 20 to 25 percent. These savings are similar to those observed by Simmons, et al. [7].

Flow Reduction, Efficiency and Safety

So far, data presented herein have shown that similar reductions in power loss can be obtained from the conventional and leading edge groove designs. However, the advantage of the leading edge groove bearing is that it has significantly lower pad operating temperatures. A careful study of Figures 21 and 22 shows that the leading edge groove pad temperature at the 80 percent location, with a flowrate of 25 to 30 percent of nominal, just begins to approach the temperature of the conventional bearing when the flowrate is 100 percent of nominal. Pad temperature profiles are compared in Figures 23 and 24. For LOP (Figure 23), the maximum temperature of the conventional and leading edge groove bearings are 205°F and 195°F, respectively, with a 100 percent nominal flowrate. Reducing the flowrate by 60 percent increases the temperature of both bearings by about 15°F to 20°F. However, the leading edge groove bearing with this reduced flowrate is still approximately 10°F cooler than the conventional bearing (with 100 percent nominal flowrate) over most of the active pad angle, especially near the critical 80 percent location. Thus, it is possible to reduce the flow to the leading edge groove bearing by as much as 60 percent, and still maintain a lower operating temperature than the conventional bearing with the normal flowrate. In addition, the power loss of the leading edge groove bearing is at least 25 percent lower as a result of this reduction in flowrate. Similarly, for the LBP condition (Figure 24), the maximum temperature of the conventional and leading edge groove bearings are 196°F and 195°F, respectively. Shaft speed is 16,500 rpm and the load is 1,200 lb. Reducing the flow by 60 percent increases the pad temperature of the conventional bearing by 20°F, but only by 5°F in the case of the leading edge groove bearing. Again, it is noticed that the leading edge groove bearing, with the 40 percent flowrate, is cooler over most of the active part of the loaded pad when compared to the conventional bearing temperatures with the 100 percent flowrate. However, the reduction in power loss associated with the leading edge groove bearing as a result of the 60 percent reduction in flow is approximately 28 percent.

That the temperature of pad number 2 of the conventional bearing, with LBP, is affected significantly by the reduction in flowrate is shown in Figure 24. Studying these results further, it would seem that the conventional bearing, with the normal flowrate, receives an adequate amount of oil to counteract hot oil carryover effects. However, when reduced to 40 percent of nominal, the flowrate is inadequate and there is a significant increase in the temperature of pad number 2. In contrast, the leading edge groove bearing maintained low temperatures in both loaded pads, even when the flowrate was reduced to 40 percent of nominal.

COMPARISONS AND IMPLICATIONS

Over the past ten years, the benefits of leading edge groove thrust bearings have been demonstrated in many applications. Although the leading edge groove pivoted shoe journal bearing has only recently been introduced, advantages have already been obtained in steam turbine applications. At the time of publication, compressor, large motor and gas turbine applications were being evaluated.

In the following case studies, background information and data from steam turbine tests, where the leading edge groove and flooded journal bearing designs were compared, are given. The predicted benefits of applying a leading edge groove tilting pad journal bearing to the gas turbine of an existing 200 MW generator are also presented.

CASE 1: Mechanical Drive Steam Turbine

In a steam turbine test rig, leading edge groove tilting pad journal bearings were independently tested against conventional flooded bearings. The tests were meant to assess journal bearing designs for a high speed compressor driver, with up to a 6500 hp rating and operational speeds to 17,000 rpm. At these high speeds, there was concern about power loss, oil flow and pad temperature of the conventional bearing.

Previous applications at speeds above 10,000 rpm indicated that the pad temperatures of the existing flooded bearing designs were approaching uncomfortably high levels. The most important objective, therefore, was to reduce these temperatures. The test directive also included selection of a bearing which would give a significant reduction in the oil flow, thus reducing the lubrication console size and permitting a smaller pump, cooler and piping. This directive envisioned using the benefits of a smaller lubrication system (in terms of size and lower initial costs) for new applications.

Test results from the 4.0 in diameter × 3.0 in long, four shoe journal bearings at 16,000 rpm are shown in Table 3. The flooded design required a total of 32 gpm for the two journal bearings, just to keep pad temperatures below 220°F. The corresponding loss was 56 hp based on a thermal balance of the outlet temperature and oil flow. For the leading edge groove bearing, a total of 17.8 gpm for the two journal bearings represented a 56 percent reduction in oil flowrate. Even with reduced flow, pad temperatures were 35°F cooler than the flooded design representing a significant benefit in bearing operation (note the good agreement with test data presented herein). Furthermore, there was no noticeable difference in the dynamic characteristics of the rotor between the flooded and leading edge groove bearing designs. As a consequence of these successful tests, a 5.0 in diameter leading edge groove thrust bearing and 5.0 in diameter × 3.75 in long, five shoe, leading edge groove journal bearings were incorporated in the field application steam turbine. This turbine is currently operating at 15,600 rpm with similar results.

Table 3. Comparison of Leading Edge Groove and Flooded Bearing Designs in a Steam Turbine Application.

Turbine Bearings
5″ Diameter, (6×6) Shoe Thrust Bearing, Steel Pads.
Two 4.0″ Dia. × 3.00″ Long, 4-Shoe Journal Bearings. LBP
Conditions: ISO VG 32, 16000 RPM, 120 DEG F Oil Inlet Temp.

Flooded		Thrust Bearing calculated	Journal Bearings field	Total Thrust & Journals
Oil Outlet	(F)	150	141	
75 Pad Temp.	(F)	249	218	
Oil Flow	(GPM)	11.5	32.0	43.5
Power Loss	(hp)	28	56	84
Leg		**Thrust Bearing field**	**Journal Bearings field**	**Total Thrust & Journals**
Oil Outlet	(F)	159	155	
75 Pad Temp.	(F)	222	183	
Oil Flow	(GPM)	6.0	14.2	20.2
Power Loss	(hp)	19	41	60
Comparison Leg vs. Flooded		**Thrust Bearing**	**Journal Bearings**	**Thrust & Journals**
Reduced Pad T	(F)	27	35	
Reduced Flow	(GPM)	5.5	17.8	23.3
% Reduction,	GPM	48	56	54
Reduced Loss	(hp)	9	14	23
% of 2600 hp		0.34	0.56	0.90
% hp Reduction		31	26	28

So that a complete comparison may be made, calculated flooded thrust bearing data are listed for the first field application (rated at 2600 hp). The reduced losses shown in Table 3 represent a significant portion of this rated power; approximately 0.9 percent. If the compressor also included leading edge groove bearings, the reduction in power loss could approach or exceed two percent of the power of the machine (assuming bearings of approximately the same size). At 16,500 rpm, a typical turbine compressor train with flooded bearings might require of the order of 90 gpm of oil (including the compressor bearings). Allowing for leakage and lubrication of other components in the machine, a normal lubrication system for this application may be of the order of 150 gpm capacity, with a 1500 gal sump. If the leading edge groove design were applied to both the turbine and compressor bearings, a 33 percent reduction in the capacity of the lubrication system could be attained. This represents a saving of approximately 50 gpm.

The success of the leading edge groove pivoted shoe journal bearings has resulted in two more applications in similar turbines: one for a recycle gas compressor and the second for a coker gas compressor.

CASE 2: 200 MW Gas Turbine/Generator

In 1988, bearing design calculations for a large gas turbine were initiated, resulting in a 30 in (12 × 12) thrust bearing recommendation. Chrome-copper offset thrust shoes were required due to the high surface speeds, loads and temperatures. The analysis included a study of flooded bearing operation. The first field application was a 200 MW turbine/generator for a modular plant design, where the turbine, generator, and diffuser are each housed separately. The auxiliary equipment is mounted on skids, and is also housed separately for serviceability. As such, reducing the size of the lubrication system was important, resulting in leading edge groove lubrication being recommended for the thrust bearing. Actual data from full power field tests are listed in Table 4.

Two 21.5 in diameter four-shoe journal bearings were used in this first application. These are steel backed, offset pad, flood lubricated bearings. Data from field tests are summed for both journal bearings and tabulated in Table 4. The thermocouples for the thrust and journal bearings are mounted in the metal at a depth of 0.19 in, and so reflect a lower temperature than at the babbitt surface.

During design, careful attention was paid to the thrust bearing, by keeping it as small and as lightly loaded as possible. This particular application was chosen as an example for predicting the benefits of the leading edge groove journal bearing because, as shown in Table 4, the journal bearing losses are large in relation to the thrust losses, and also in relation to the power of the machine. To allow a complete comparison, the recommended flooded thrust bearing design conditions are listed, although field data for this design are not available. In any case, the calculated total bearing losses for the flooded design are 1,401 kW, which represents 0.7 percent of the rated power of the machine.

In the actual application, the leading edge groove thrust bearing has already attained reductions in power loss, pad

Table 4. Comparison of Predicted Leading Edge Groove Tilting Pad Journal Bearing Data with Flooded Bearings for a 200 MW Gas Turbine Generator.

Turbine Bearings
30″ Diameter, (12×12) Shoe Thrust, Offset, Chrome-Copper.
Two 21.5″ Diameter × 14.7″ Long, 4-Shoe, Off-Set, LBP Journal Bearings, Steel Pads.
Conditions: ISO VG 32, 3000 RPM, 130 DEG F Oil Inlet Temp.

Flooded		Thrust Bearing rec.	Journal Bearings field	Total Thrust & Journals
Oil Outlet	(F)	153	164	
75 Pad Temp.	(F)	205	226	
Oil Flow	(GPM)	456	360	816
Power Loss	(KW)	646	754	1401
Leg		**Thrust Bearing field**	**Journal Bearings predicted**	**Total Thrust & Journals**
Oil Outlet	(F)	162	178	
75 Pad Temp.	(F)	186	205	
Oil Flow	(GPM)	254	180	434
Power Loss	(KW)	500	532	1033
Comparison Leg vs. Flooded		**Thrust Bearing**	**Journal Bearings**	**Thrust & Journals**
Reduced Pad T	(F)	19	21	
Reduced Flow	(GPM)	202	180	382
% Reduction,	(GPM)	44	50	47
Reduced Loss	(KW)	146	222	368
% of 200 MW		0.07	0.11	0.18
% kW Reduction	23	29	26	

temperature, and oil flow. Journal bearing predictions based on test results presented in this paper indicate that flows can be reduced 50 percent with a reduction in pad temperature of the order of 20°F. This would lower the flow requirements by a further 180 gpm and reduce the power consumption by a further 222 kW (which represents 0.11 percent of the machine's power). By switching to the leading edge groove design, savings of approximately 0.2 percent are predicted for the turbine alone. Additional benefits may be obtained by considering the generator bearings as well.

Although this study is a prediction in regard to the journal bearings, the implications of a 0.2 percent reduction in losses are enormous in terms of the operating cost of rotating machinery. For example, taking a power generation utility with a capacity of 20,000 MW, and assuming that the turbogenerator sets run for an average of 6,500 hr/year and that the cost/kWh is $0.075, the savings will be of the order of $20M/year. This is in line with the findings of Jost [11], who discusses the energy losses in large turbogenerators. He indicates that the loss in each bearing can be of the order of 0.5 MW, and goes on to show that if the losses in each bearing are reduced by 15 to 20 percent, then the average savings to the United Kingdom are of the order of $50M/annum.

CONCLUSIONS

An extensive study that compares the performance of 3.875 in diameter conventional and leading edge groove tilting pad journal bearings has been made, and the following conclusions are drawn:

- Application of leading edge groove lubrication to an offset pivoted shoe journal bearing has resulted in substantial reductions in pad operating temperature. These reductions become more significant at higher surface speeds. Compared to the center and offset pivot conventional bearings, the difference in favor of the leading edge groove bearing may be as much as 40°F to 45°F at extreme operating conditions.
- Results from experimental and theoretical studies show that the temperature advantage of the leading edge groove bearing comes from the reduction of hot oil carryover. In "load between pads" tests on the conventional bearing, it is shown that the second pad in the direction of motion runs significantly hotter than the preceding pad. With the leading edge groove design, both loaded pads run cooler, and at about the same temperature.
- For most operating conditions, the temperature of the loaded pad of the conventional center pivot bearing reaches a maximum in the vicinity of the 80 percent location and then falls to a lower value at the trailing edge. In the case of the conventional offset pivot pad, the temperature at the 80 percent location is approximately the same as that of the center pivot pad, although the trailing edge temperature is higher. Thus, the loaded pad of the conventional offset pivot bearing has the highest maximum temperature.
- The maximum temperature of the offset leading edge groove bearing also occurs at the trailing edge, although temperature levels are much lower than those recorded on the conventional bearings. At extreme operating conditions, the maximum temperature of both the leading edge groove and conventional offset pivot bearings shows a tendency to move towards the 80 percent location. This would support the API directive that the 75 percent pad location should be used as the recommended position for critical pad temperature measurement.
- Reductions in flow to both the conventional and leading edge groove bearings can be made without excessively increasing pad operating temperatures. Reducing the flow by 50 percent raised pad temperature 5°F to 20°F, increased the temperature rise of the oil to approximately 40°F, and reduced bearing power loss by 20 percent or more.
- Compared to the conventional bearing, larger reductions in flowrate can be made with the leading edge groove bearing as a result of its lower operating temperature. The leading edge groove bearing with the flowrate reduced to 40 percent of nominal ran cooler than the conventional bearing with a flowrate of 100 percent of nominal.
- When the flowrate to the conventional bearing was reduced to 40 percent of nominal, a sharp rise in the temperature of the second loaded pad was observed. This indicates that insufficient oil was being supplied to the bearing to limit the effect of hot oil carryover. In the case of the leading edge groove bearing, this sharp rise in temperature did not take place until the flowrate had been reduced below 35 percent of nominal.
- Data presented have shown that the leading edge groove bearing can be used to significantly reduce both power loss and oil flowrate. This is achieved without exceeding the temperature of the conventional bearing with a normal oil flowrate. Consequently, machine efficiency is increased without compromising operational safety.

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