ANALYSIS AND TESTING OF
THE LEG TILTING PAD JOURNAL BEARING—
A NEW DESIGN FOR INCREASING LOAD CAPACITY,
REDUCING OPERATING TEMPERATURES AND CONSERVING ENERGY

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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 exper-

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, preload-
ed, 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 spacing. 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.

<table>
<thead>
<tr>
<th>Group #</th>
<th>Bearing clearance, in</th>
<th>Pad clearance, in</th>
<th>Preload, in</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.003</td>
<td>0.004</td>
<td>0.25</td>
</tr>
<tr>
<td>2</td>
<td>0.004</td>
<td>0.004</td>
<td>0.0</td>
</tr>
</tbody>
</table>

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 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).

<table>
<thead>
<tr>
<th>Load, lb (lb/in³)</th>
<th>Shaft speed, rpm (ft/sec)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1800</td>
</tr>
<tr>
<td>290 (50)</td>
<td>0.23</td>
</tr>
<tr>
<td>1200 (207)</td>
<td>0.34</td>
</tr>
<tr>
<td>2500 (430)</td>
<td>0.46</td>
</tr>
</tbody>
</table>

*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.

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

**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-
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:

\[ t_1 = \frac{q_1 t_1 + (q_1 - q) t_2}{q_1} \]  

(1)

where:
- \( t_1 \) = oil temperature at the leading edge
- \( t_2 \) = oil temperature at the trailing edge of the preceding pad
- \( t_1 \) = oil inlet temperature to the bearing
- \( q_1 \) = oil flow at the leading edge
- \( q \) = oil flow at the trailing edge of the preceding pad
- \( q \) = 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:

\[ t_1 = \frac{(q_1 - q_2) t_1 + q_1 t_2}{q_1} \]  

(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 location 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 10 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 drive, 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 flow rate. 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.

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 backled, 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 0.15 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

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

<table>
<thead>
<tr>
<th>Flooded</th>
<th>Thrust Bearing calculated</th>
<th>Journal Bearings field</th>
<th>Total Thrust &amp; Journals</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil Outlet (F)</td>
<td>150</td>
<td>141</td>
<td></td>
</tr>
<tr>
<td>75 Pad Temp. (F)</td>
<td>249</td>
<td>218</td>
<td></td>
</tr>
<tr>
<td>Oil Flow (GPM)</td>
<td>11.5</td>
<td>32.0</td>
<td>43.5</td>
</tr>
<tr>
<td>Power Loss (hp)</td>
<td>28</td>
<td>56</td>
<td>84</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Leg</th>
<th>Thrust Bearing field</th>
<th>Journal Bearings field</th>
<th>Total Thrust &amp; Journals</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oil Outlet (F)</td>
<td>159</td>
<td>155</td>
<td></td>
</tr>
<tr>
<td>75 Pad Temp. (F)</td>
<td>222</td>
<td>183</td>
<td></td>
</tr>
<tr>
<td>Oil Flow (GPM)</td>
<td>6.0</td>
<td>14.2</td>
<td>20.2</td>
</tr>
<tr>
<td>Power Loss (hp)</td>
<td>19</td>
<td>41</td>
<td>60</td>
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</table>

Comparison Leg vs. Flooded

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

<table>
<thead>
<tr>
<th></th>
<th>Thrust Bearing rec.</th>
<th>Journal Bearings field</th>
<th>Total Thrust &amp; Journals</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Oil Outlet (F)</strong></td>
<td>153</td>
<td>164</td>
<td></td>
</tr>
<tr>
<td><strong>75 Pad Temp. (F)</strong></td>
<td>205</td>
<td>226</td>
<td></td>
</tr>
<tr>
<td><strong>Oil Flow (GPM)</strong></td>
<td>456</td>
<td>360</td>
<td>816</td>
</tr>
<tr>
<td><strong>Power Loss (KW)</strong></td>
<td>646</td>
<td>734</td>
<td>1401</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th></th>
<th>Thrust Bearing field</th>
<th>Journal Bearings predicted</th>
<th>Total Thrust &amp; Journals</th>
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<tbody>
<tr>
<td><strong>Oil Outlet (F)</strong></td>
<td>162</td>
<td>178</td>
<td></td>
</tr>
<tr>
<td><strong>75 Pad Temp. (F)</strong></td>
<td>186</td>
<td>205</td>
<td></td>
</tr>
<tr>
<td><strong>Oil Flow (GPM)</strong></td>
<td>254</td>
<td>180</td>
<td>434</td>
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<tr>
<td><strong>Power Loss (KW)</strong></td>
<td>500</td>
<td>532</td>
<td>1033</td>
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<table>
<thead>
<tr>
<th></th>
<th>Thrust Bearing</th>
<th>Journal Bearings</th>
<th>Thrust &amp; Journals</th>
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</thead>
<tbody>
<tr>
<td><strong>Reduced Pad T (F)</strong></td>
<td>19</td>
<td>21</td>
<td></td>
</tr>
<tr>
<td><strong>Reduced Flow (GPM)</strong></td>
<td>202</td>
<td>180</td>
<td>382</td>
</tr>
<tr>
<td><strong>% Reduction, (GPM)</strong></td>
<td>44</td>
<td>50</td>
<td>47</td>
</tr>
<tr>
<td><strong>Reduced Loss (KW)</strong></td>
<td>146</td>
<td>222</td>
<td>368</td>
</tr>
<tr>
<td><strong>% of 200 MW</strong></td>
<td>0.07</td>
<td>0.11</td>
<td>0.18</td>
</tr>
<tr>
<td><strong>% kW Reduction 23</strong></td>
<td>29</td>
<td>26</td>
<td></td>
</tr>
</tbody>
</table>

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.

REFERENCES


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