

A STUDY OF PARAMETERS THAT AFFECT PIVOTED SHOE JOURNAL BEARING PERFORMANCE IN HIGH-SPEED TURBOMACHINERY

by

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ABSTRACT

Journal bearing pad temperatures, oil flow requirements, and power losses can impose limitations on the design and operation of high-speed turbomachinery. Over the past few years, the authors have conducted extensive tests and studies of parameters that affect pivoted shoe journal bearing performance. A special, high-speed test rig is described, which was designed and built for the purpose of measuring bearing steady-state performance characteristics under light to moderately heavy loads, and to very high operating speeds that are being approached in new turbine and compressor designs. Instrumentation includes a detailed array of pad temperature detectors and the direct measurement of frictional torque.

Data are presented that compare the effects of pivot offset, oil flow, load orientation, method of lubrication, and oil discharge configuration on 6 inch diameter pivoted shoe journal bearing performance. The parameters are shown to significantly influence bearing pad temperature and power loss, particularly at high loads and speeds. Pad temperature profiles, isotherms, torque, and oil outlet temperatures are compared and evaluated. Discussions address the prediction and application of these parameters, and how they may be used to improve the capacity and performance of high-speed turbomachinery. The data and discussions are intended to provide useful information to engineers, programmers, and personnel involved with the study or operation of pivoted shoe journal bearings.

INTRODUCTION

Turbomachinery operating speeds and loads have increased over time resulting in increased bearing temperatures and power losses. Gardner and Ulschmid (1973) addressed the concern of journal bearing limitations at turbulent operation, documenting a dramatic increase in pad temperature and power loss of a 17 inch pivoted shoe bearing at 3600 rpm (267 fps surface speed). The pad was a center pivot design, which has the advantage of being able to operate in either direction of rotation. There are many technical papers that study center pivot journal bearings. Offset pivots improve journal bearing pad temperature limitations, but are not as well documented in literature.

Large steam and gas turbine designers are presently considering 22 inch and larger diameter journal bearings for power generation where surface speeds at 3600 rpm exceed 330 fps. Designers of steam turbines and compressors have intentions for operating speeds approaching 400 fps. Such conditions are well into the turbulent regime where conventional bearing losses and

temperatures become so high that options must be considered to address limitations. Direct lubrication is one solution that has been successfully applied in thrust bearings for many years, and also in special journal bearing applications dating back to the mid-sixties. It is only recently that direct lube journal bearings have been seriously considered for general turbomachinery applications because high surface speeds now warrant such a consideration.

As in the case of offset pivots, technical papers on direct lube journal bearings are sparse. Data are published by Tanaka (1991) and Tanaka and Mishima (1989) comparing 100 mm (3.94 inch) diameter designs to 137 fps; Harangozo, et al. (1991), tested 5 inch diameter bearings to 152 fps; and Fillon, et al. (1993), report on 100 mm (3.94 inch) diameter data to 70 fps. The authors' (Brockwell, et al., 1992, 1994; Dmochowski, et al., 1993) tested 3.88 inch diameter, leading-edge-groove (LEG) designs to 270 fps; DeCamillo and Clayton (1997) provide data on an 18 inch LEG generator bearing at 283 fps; and Edney, et al. (1996), report on 5 inch diameter LEG steam turbine bearings running to 312 fps.

In assessing bearing limitations, there are many parameters that affect results including geometry, operating conditions, and even instrument location. For example, DeChoudhury and Barth (1981) show that the drop in oil outlet temperature between the bearing and drain line can lead to significant differences in thermal balance calculation of power loss. Pinkus (1990) describes peculiar behavior between laminar, transitional, and high-speed turbulent regimes of bearing operation. Pettinato and DeChoudhury (1999) note high edge temperatures from misalignment in ball-in-socket pivot geometry. Wygant, et al. (1999), show differences in steady-state and dynamic performance attributed to restriction of pad motion by friction in sliding contact pivots. Conventional flooded journal bearings were used in these references. The flooded journal bearing design has been studied for many years, and information is available for a wide range of operating loads and speeds.

In contrast, the few published papers on direct lube journal bearings cited earlier mostly report on low speed operation, in the laminar to transitional range of operation. There is some disagreement regarding the magnitude of the benefits of direct lubrication. This may be due to laminar/transitional influences, but there are also differences in the method of direct lubrication, as well as the type of pivot, instrument location, etc. In general, all references agree that direct lubrication provides pad temperature benefits that appear to improve with load and speed. There is also agreement that direct lube power loss is lower, although most authors report that thermal balance methods are of insufficient precision to allow an accurate assessment. Benefits are typically attributed to eliminating seal losses and reducing churning losses, but these are hypothetical because there are little data available for confirmation.

In order to address such issues, a special test rig was designed and built to measure steady-state performance under light to moderately high loads, and to very high operating speeds that are being approached in new turbine and compressor designs. An important feature of the rig is the direct measurement of frictional torque, which provides a more precise measurement of power loss than thermal balance techniques. Over the past few years, the authors have conducted extensive tests and studies of parameters that affect the performance of pivoted shoe journal bearings. The purpose of this work is to improve the capacity and performance of high-speed turbomachinery by extending bearing speed and/or load limitations and improving efficiency.

TEST RIG

Tests were performed on a new rig designed to investigate steady-state performance under high operating speeds and light to moderately heavy unit loads. The test apparatus is illustrated in Figure 1 and consists of a test bearing, the shaft and drive system, the loading system, and the rig supporting structure.

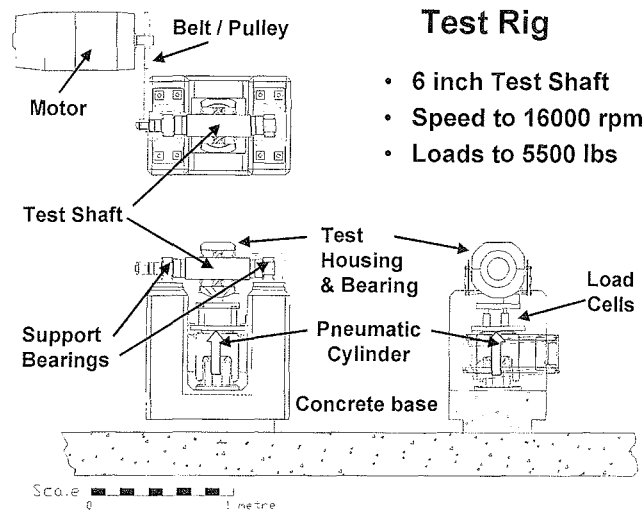


Figure 1. Test Rig Schematic.

The test shaft is driven by a 150 hp variable speed DC electric motor. A belt and pulley system connects the motor to the test shaft and provides a 4.5:1 speed step-up giving a maximum shaft speed of just over 16,000 rpm. The DC motor is linked to an electronic controller that ensures speed control to within ± 1 percent accuracy. The test shaft is supported by two 3.5 inch diameter pivoted shoe journal bearings spaced approximately 28 inches apart. The test shaft journal is 6 inches in diameter with circularity within 0.0005 inch. Shaft speed is measured using a slotted optical switch in conjunction with a shaft-mounted disk containing a number of drilled holes. This was found to be a robust system capable of reliable operation even at the top speed of the test facility.

An important feature of the rig is the direct measurement of frictional torque. This is accomplished by the design of the test housing (Figure 2). The two-piece housing is positioned on the test shaft, midway between the support bearings. The bottom face has been accurately machined and sits in a spherical hydrostatic bearing that eliminates friction between the housing and the loading device. The housing is held against rotation by a 100 lb capacity load cell mounted to a torque arm. Power loss is determined from the measured frictional force, and radii of the torque arm and bearing bore. The spherical hydrostatic bearing also provides good alignment between the test bearing and shaft. A second plane hydrostatic bearing supports the spherical hydrostatic bearing. This provides lateral freedom and allows the load line to be adjusted in relation to the test bearing. Proximity probes mounted on both ends of the test bearing housing measure the horizontal and vertical displacements of the test bearing with respect to the position of the shaft. These probes also provide an indication of the level of alignment between the test bearing and the shaft.

A pneumatic cylinder located between the test housing and base of the rig (Figure 1) generates the static load. This has a capacity of 5500 lb. Three load cells located between the top of the load applicator and the bottom surface of the plane hydrostatic bearing measure the vertical load applied to the test bearing. The load cells are arranged in such a way that they each share an equal proportion (one-third) of the load applied to the bearing. In calculating the net load on the test bearing, the combined weight of the test bearing, housing, and the hydrostatic bearing system is accounted for.

A pump with a capacity of 20 gpm and a maximum supply pressure of 300 psig delivers oil to the test bearing from a 150 gallon capacity tank. The flowrate is measured by a turbine type flowmeter with a linear flow range of 2.5 to 29 gpm. Feed oil temperature is measured before it enters the test bearing, and is controlled by a water-oil heat exchanger to $\pm 1^\circ\text{C}$ (33.8°F). An industrial type pressure transducer with a pressure range of zero to

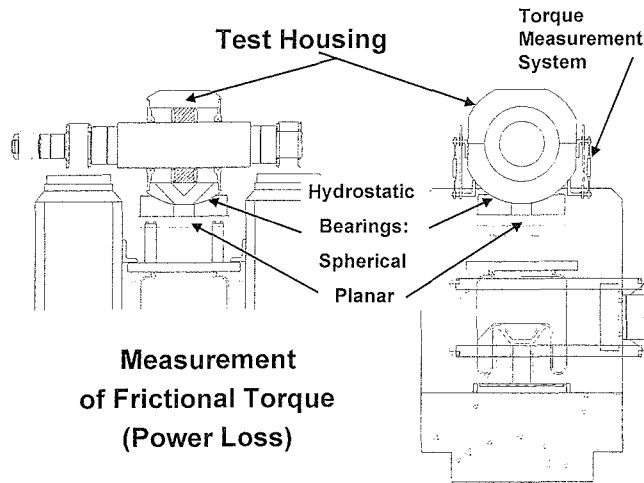


Figure 2. Test Housing and Torque Measurement System.

60 psig measures oil supply pressure to the test bearing. Thermocouples measure the oil inlet temperature, and the oil outlet temperature from each side of the test bearing. Flowrate and inlet and outlet oil temperatures are used in thermal heat balance calculations of power loss for comparison with values obtained from measured torque.

A special data acquisition system was developed for recording the data. In this system, process signals from the test facility are recorded using a Windows® graphical users interface program. This program provides an interface between the analog/digital hardware and the operator, permitting real-time display of process conditions, the acquisition of test data, and signal processing such as data trending, frequency analysis, and digital filtering. Signals processed and displayed by the data acquisition system include test and support bearing temperatures; oil supply and outlet temperature, supply pressure, tank temperature, and flow; shaft speed, bearing load, test bearing friction force, and X and Y displacements. The data acquisition hardware cards are rated at 1 MHz and the display screen is updated every second. Measurement uncertainties of the equipment are listed in Table 1.

Table 1. Test Rig Measurement Uncertainties.

Measurement	Type of sensor	Limit of error of sensor
Temperature	Type T thermocouple	1 °C or 0.75% (whichever is greater)
Shaft speed	Optical switch	± 5 rpm
Bearing load	0-2000 lb. load cells (× 3)	± 5 lb. at full scale
Friction force	0-100 lb. load cell	± 0.1 lb. at full scale
Displacements	Eddy current proximity probes	± 0.00004 in
Oil flowrate	Turbine flow meter	± 0.5% of reading
Oil supply pressure	Pressure transducer	± 0.6 lb/in ²
Leading edge groove oil pressure	Miniature pressure transducer	± 0.75 lb/in ²
Drive motor power consumption	Power cell	± 10 hp

TEST BEARINGS AND OPERATING CONDITIONS

The test bearings are 6 inches in diameter, have a 0.437 length-to-diameter ratio, a .0045 inch assembled radial clearance, and a .25 preload. The test bearing pads are of the same geometry, consisting of five pivoted pads with 60 degree pad angles. The pivot is a rolling contact design with curvature in both the axial and circumferential directions. This allows for axial alignment capability, and freedom of movement under changes in operating conditions.

The test bearing pads are instrumented with an array of 45 type T thermocouples with the tip of each thermocouple located .020 inches below the babbitt surface (Figure 3). The loaded shoes are more heavily instrumented along the centerline, and have additional thermocouples along the edges of the pads (Figure 4).

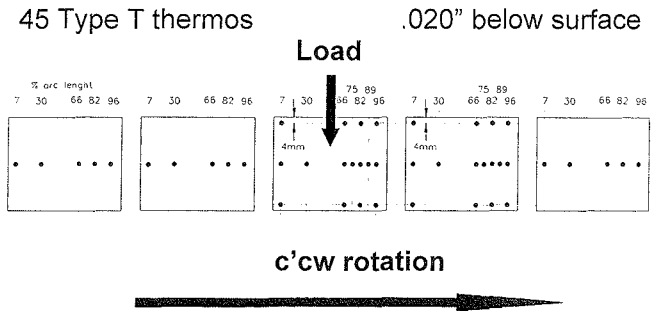


Figure 3. Babbitt Thermocouple Locations.

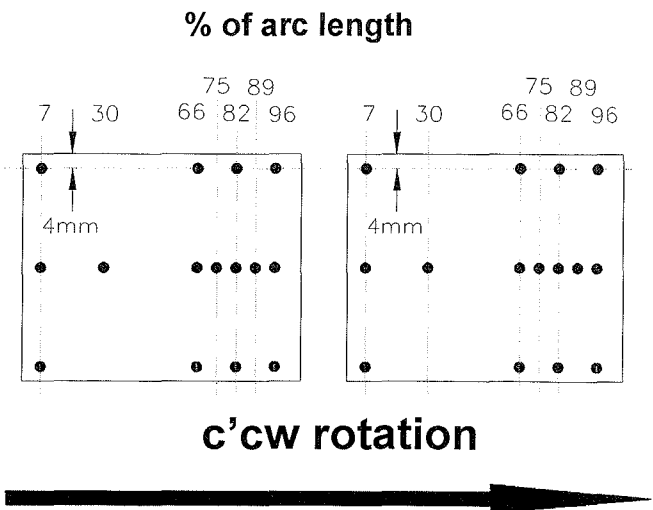


Figure 4. Loaded Shoe Instrumentation Locations.

Tests were performed over a range of loads and speeds to compare the effects of pivot offset, oil flow, load orientation, method of lubrication, and oil discharge configuration. A description of the specific parameter being tested is detailed in the associated section of this paper. The lubricant used in this series of tests was an ISO VG 32 turbine oil, with a measured viscosity of 32.76 centistokes at 40°C (104°F), and 5.41 centistokes at 100°C (212°F). All tests were run with an oil supply temperature of 49°C (120.2°F).

CONVENTIONAL FLOODED BEARING TESTS AND DISCUSSIONS

In hydrodynamic lubrication, the journal surface drags oil into the space between the journal and bearing pads. The terms “conventional” and “flooded” are terms used to describe the method of lubrication that provides oil to the journal surface. The conventional method is to restrict the hot discharge oil by sealing the sides of the bearing around the shaft such that the bearing cavity is flooded. This provides oil to the journal surface by filling the spaces between the pads. There are a couple of variations. Floating seal rings are sometimes used with tight shaft-to-seal clearances. Oil discharge is then restricted by oil outlet holes, sized to provide a slight back pressure to flood the housing. Another discharge configuration uses labyrinth seals where oil discharge is controlled by the gap between the seals and shaft. Labyrinth seals were used for the flooded bearing tests in this report. Referring to Figure 5, oil is supplied to an outer annulus around the bearing, through radial oil supply holes between each pad, and is restricted by seals on each side of the bearing.

Center Pivot Versus Offset Pivot Tests

This series of tests was performed with conventional, flooded pivoted shoe journal bearings, to compare the effect of pad pivot-location on bearing performance. The pivot locations tested are

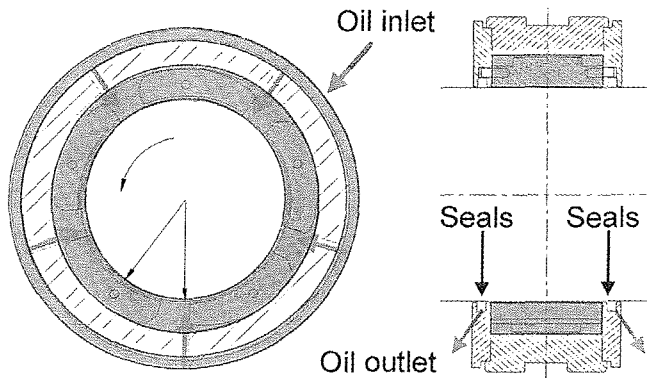


Figure 5. Conventional (Flooded) Bearing Lubrication.

shown schematically in Figure 6. The center pivot design has the pivot located in the center of the pad, 50 percent of the pad arc length from the leading edge in the direction of rotation. The offset pivot design has the pivot located at 60 percent of the pad arc length from the leading edge. Other than the location of the pivot, the bearing geometry for the two sets of pads is identical.

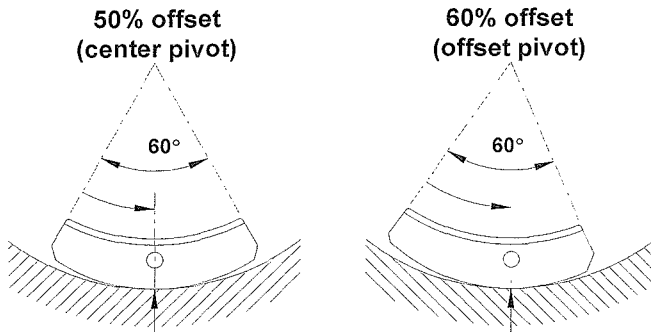


Figure 6. Center and Offset Pivot Geometry.

Pad Temperature Profiles

Pad temperature profiles are obtained by plotting the centerline thermocouple temperatures against the relative angular position of the detector in the bearings as depicted in Figure 7. This figure is for a load-on-pad (LOP) orientation where the load is directed toward a single pad. The loaded pad data are in the center of the figure. As speed and load are increased (Figures 8 and 9), the loaded pad temperature increases and will eventually limit the application. Figure 10 compares the loaded pad temperature profiles plotted according to position along the pad arc length.

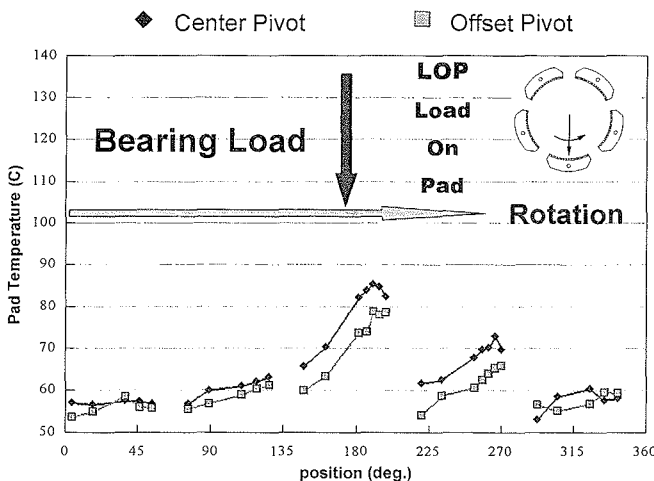


Figure 7. Pad Temperature Profiles, LOP, 200 PSI, 3600 RPM.

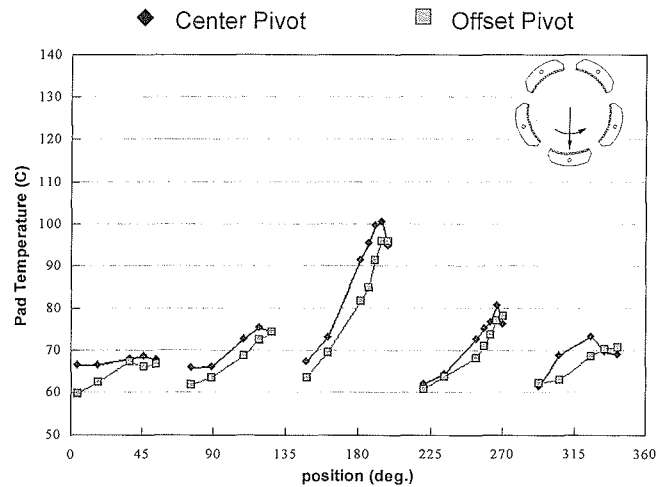


Figure 8. Pad Temperature Profiles, LOP, 200 PSI, 9000 RPM.

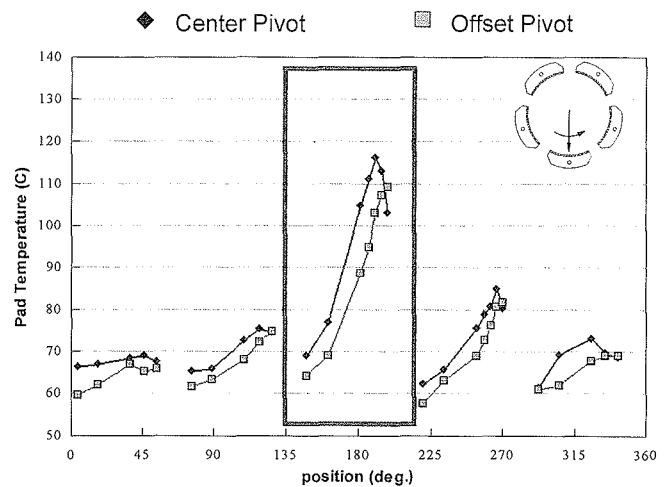


Figure 9. Pad Temperature Profiles, LOP, 320 PSI, 9000 RPM.

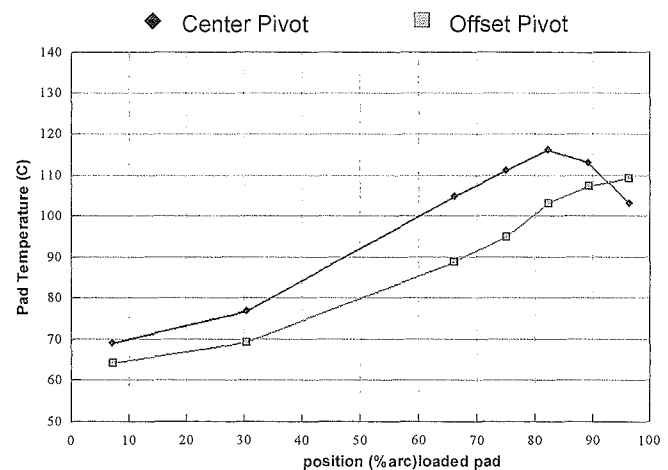


Figure 10. Loaded Pad Temperature Profiles, LOP, 320 PSI, 9000 RPM.

Pad Surface Isotherms

Before any comparisons are made, it is useful to look at the entire pad surface profile. This is generated by curve fitting data from the array of thermocouples in the loaded pad surface. Figure 11 is an isometric view of the loaded, center pivot pad. Figure 12 is the same data looking directly at the babbitt face. Figures 13 and

14 are similar plots for the loaded, offset pivot pad under identical conditions of load, speed, and oil flow. The surface temperature profiles are important in that they show if the pad is aligned with the shaft or if there is edge or skew loading that can distort comparisons and lead to erroneous conclusions. The symmetric patterns of Figures 11 through 14 indicate good axial alignment between the loaded pad and shaft, which is attributed to the alignment capability of the rolling contact pivot design. Another observation is that there is a fairly substantial axial temperature gradient noticeable in the figures. Temperature drops 15 to 20°C (59 to 68°F) from the center hot spot to the sides of the pad for the operating conditions shown.

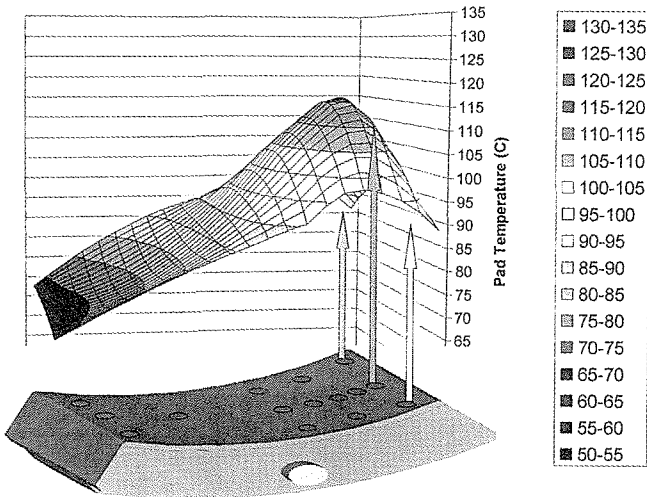


Figure 11. Center Pivot Pad Surface Isotherm, Isometric, LOP, 320 PSI, 9000 RPM.

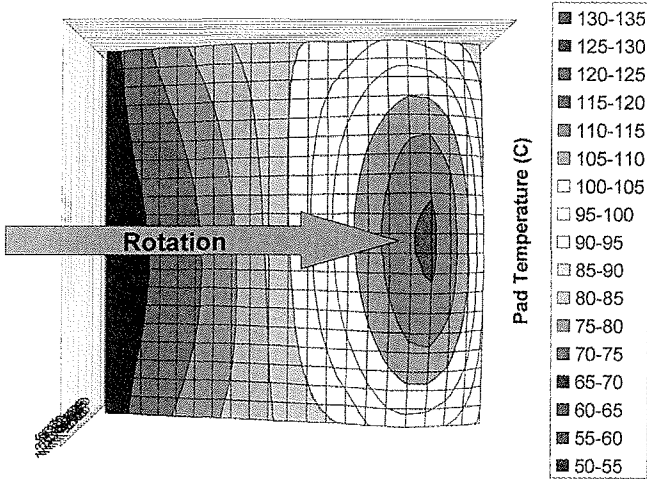


Figure 12. Center Pivot Pad Surface Isotherm, Plan View, LOP, 320 PSI, 9000 RPM.

Pad Temperature Comparisons, Center Versus Offset Pivot

A study of pad surface isotherms indicated that the pads were aligned with the shaft, and that peak temperatures were located along the circumferential centerline for the conditions tested. With this assurance, centerline pad temperature profiles can be used to provide an accurate indication of the highest pad temperatures.

Figures 15 through 17 are pad temperature profiles for a load-between-pad (LBP) orientation where the test load is applied directly between the two bottom pads. LBP pad temperature response is similar to the LOP data of Figures 7 through 9 in that as speed and load are increased, the loaded pad temperatures also

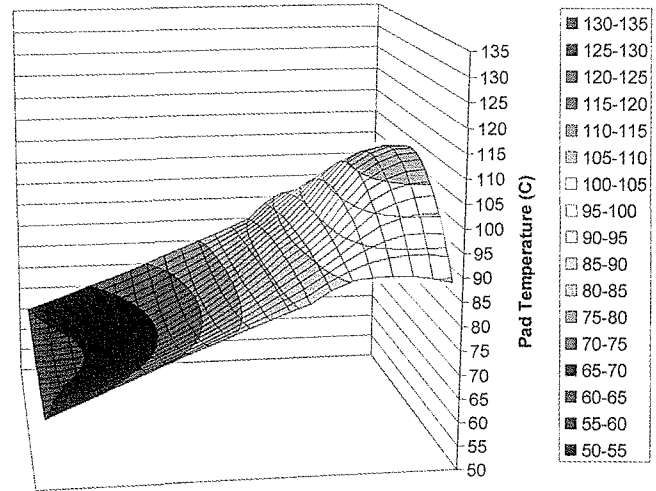


Figure 13. Offset Pivot Pad Surface Isotherm, Isometric, LOP, 320 PSI, 9000 RPM.

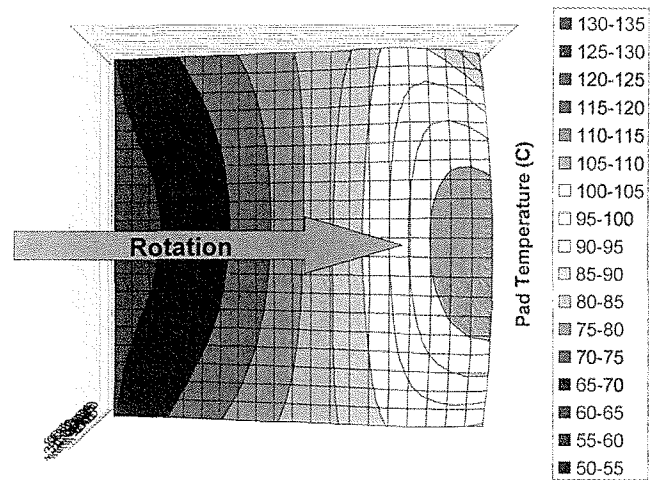


Figure 14. Offset Pivot Pad Surface Isotherm, Plan View, LOP, 320 PSI, 9000 RPM.

increase. For LBP orientation, it is noted that the second loaded pad in direction of rotation runs hotter than the first loaded pad. The effect is most noticeable at the higher load and speed. This is attributed to hot oil carryover. That is, a pad's temperature is influenced by oil from the preceding pad. The unloaded pad preceding the first loaded pad has relatively cool temperatures. The second loaded pad is influenced by much hotter oil from the first loaded pad. And so, for LBP orientation, the second loaded pad temperature becomes the limiting factor. Figure 18 compares second loaded pad temperature profiles for the center and offset pivot designs.

Studying LOP Figures 7 through 9 and LBP Figures 15 through 17, similar trends can be detected for the center and offset pivot designs and for LOP and LBP orientation. Except at the lowest speed and load where temperature profiles are fairly level, temperatures for all pads are lowest at the leading edge and increase toward the trailing edge. The temperature levels of all pads, and the temperature gradient from leading to trailing edge of all pads increase with speed. The temperature levels and gradient of the loaded pads increase with load. Other trends observed in this study are difficult to show in the few, reduced figures. A minor reduction in unloaded pad temperatures was noticed with increased load. The hottest temperature location of the loaded pads moved toward the trailing edge as speed was increased. This trend is reversed for increased load.

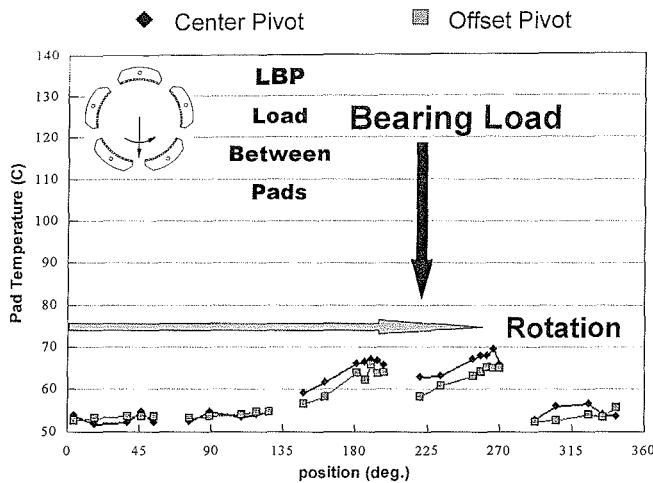


Figure 15. Pad Temperature Profiles, LBP, 200 PSI, 1800 RPM.

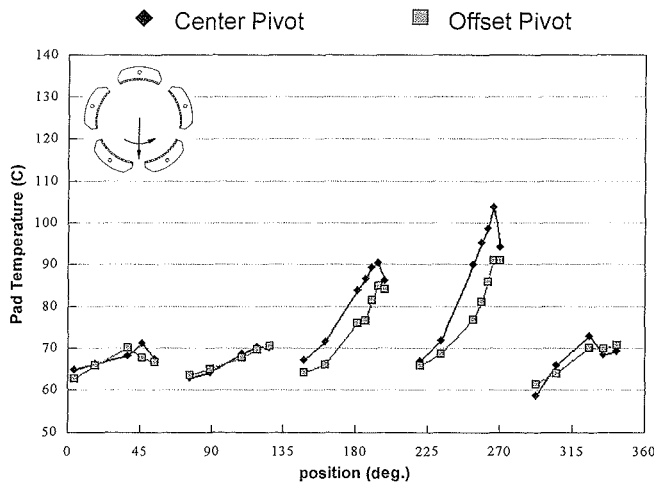


Figure 16. Pad Temperature Profiles, LBP, 200 PSI, 9000 RPM.

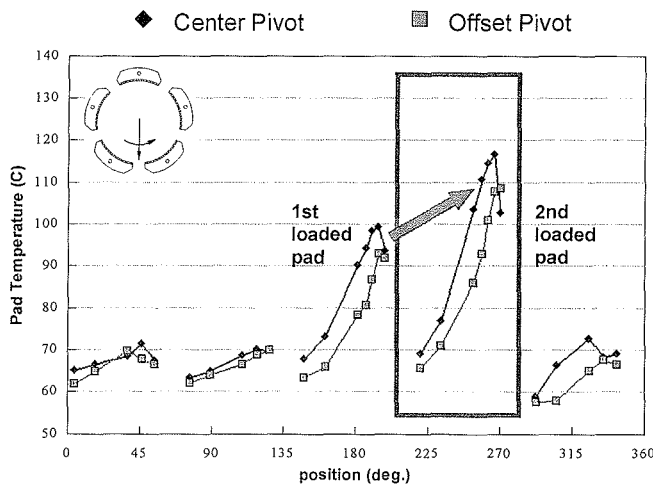


Figure 17. Pad Temperature Profiles, LBP, 320 PSI, 9000 RPM.

Differences between center and offset pivot temperatures are also noticeable in Figures 7 through 9 and 15 through 17. In all cases, the offset pivot runs with cooler overall pad temperatures, particularly noticeable on the loaded pads. Figure 10 for LOP and Figure 18 for LBP focus on the hotter, loaded pad. For the center pivot, the pad temperature is noticed to increase to a maximum at a position of 82 and 89 percent of the arc length for LOP and LBP, respectively, after which it falls to a lower level. For the offset pivot

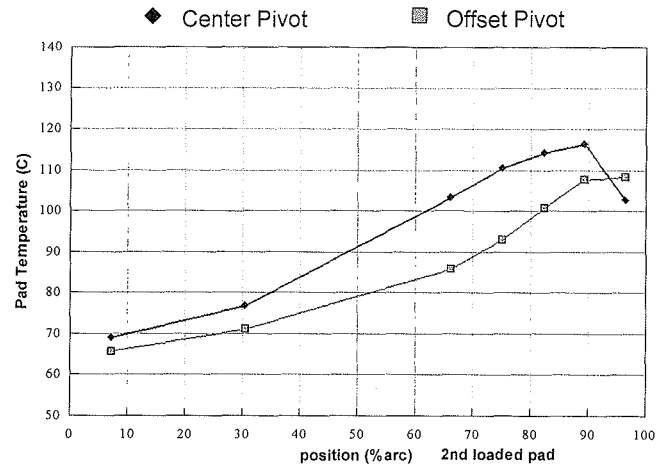


Figure 18. Loaded Pad Temperature Profiles, LBP, 320 PSI, 9000 RPM.

design, the pad temperatures reach a maximum close to the trailing edge of the pad. These observations are consistent with 3.88 inch diameter results in earlier work by the authors (Brockwell, et al., 1994). For the range of loads and speeds tested, the center pivot maximum temperature varied between the 65 percent and 90 percent location. The offset pivot maximum temperature varied between the 82 percent location and trailing edge. The specific location of the maximum temperature varied with speed and load.

To further reduce the data for generalized comparison, it is helpful to view a single location. The 75 percent position is a typical location used in industry (Nicholas, 1994) and is recommended in many specs such as American Petroleum Institute (API). With the entire profile measured in detail, and data showing different maximum temperature locations for the center and offset pivot designs, some authors choose to compare maximum measured temperatures (Simmons and Lawrence, 1996). Figures 19 through 22 compare 75 percent location and maximum temperatures for LOP and LBP orientation at 100 percent of recommended flow. The speed range for these comparisons was limited because the lubrication system could not provide the recommended flow for higher speeds. However, tests were also performed with 50 percent flow, shown in Figures 23 and 24.

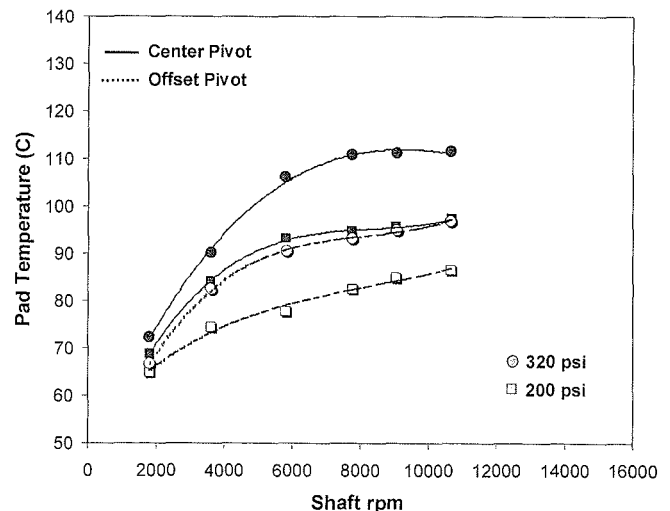


Figure 19. 75 Percent Location Pad Temperatures, LOP.

The change in slope and inflection noticeable in Figures 23 and 24 indicate that the speed range covers laminar, transitional, and turbulent operating conditions. Figures 19 through 24 show that the offset pivot design has lower 75 percent location and lower maximum temperatures than the center pivot design for all conditions

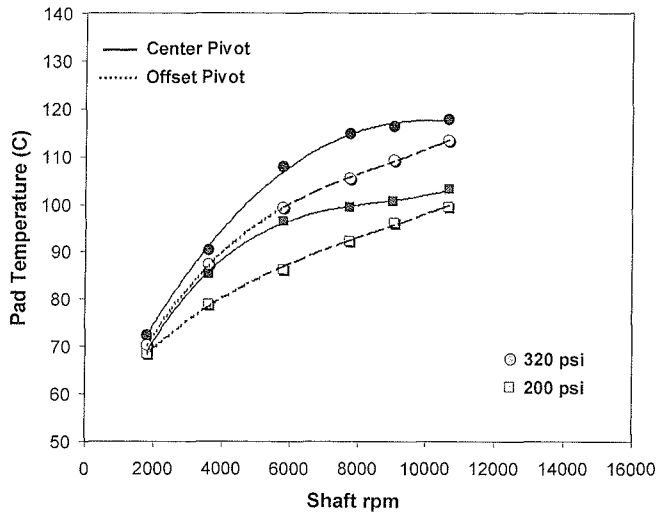


Figure 20. Maximum Pad Temperatures, LOP.

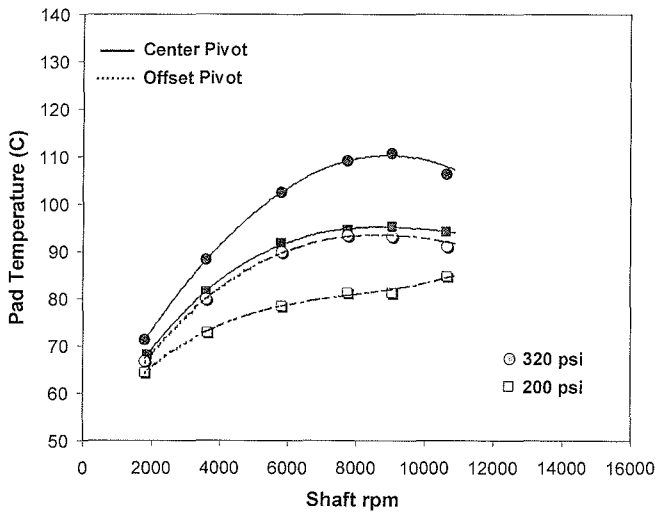


Figure 21. 75 Percent Location Pad Temperatures, LBP.

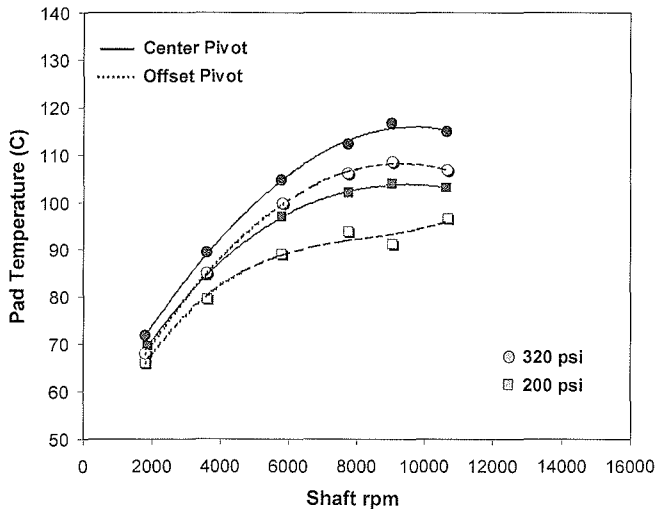


Figure 22. Maximum Pad Temperatures, LBP.

tested. The 75 percent location temperatures are as much as 20°C (68°F) cooler, and maximum temperatures are up to 10°C (50°F) cooler than the center pivot pad temperatures. The benefit is noticed to improve with speed and load, and extends through the laminar, transitional, and turbulent regimes of operation.

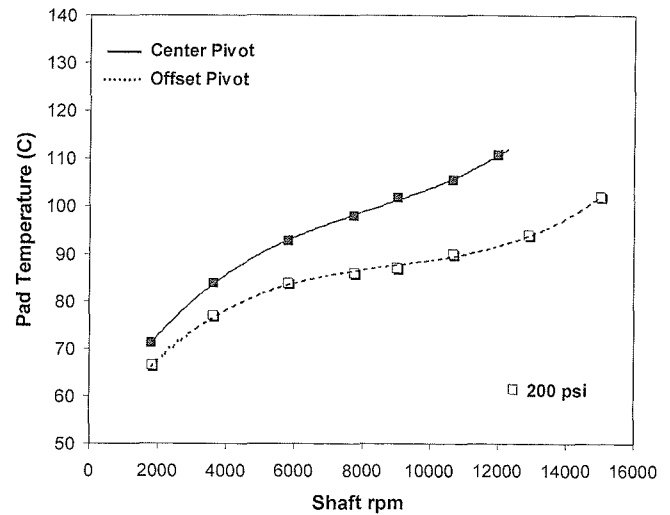


Figure 23. 75 Percent Location Pad Temperatures, LOP, 50 Percent Flow.

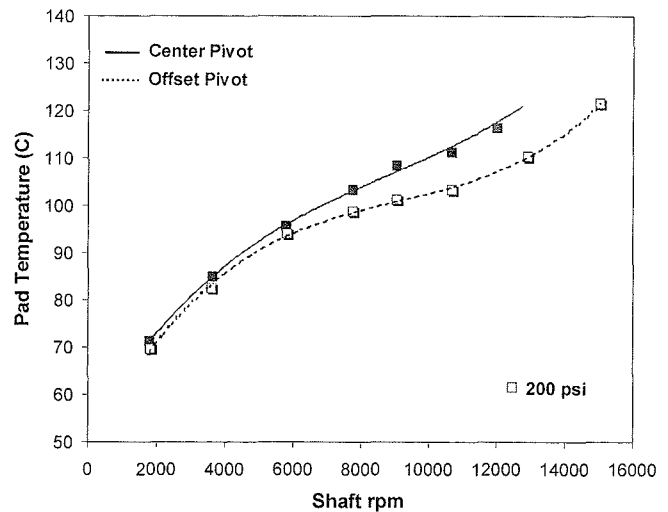


Figure 24. Maximum Pad Temperatures, LOP, 50 Percent Flow.

Power Loss Comparisons, Center Versus Offset Pivot

Power loss using frictional torque measurements are compared in Figures 25 and 26 for select cases of load orientation and flow. These figures are representative of the cases studied. For 100 percent flow (Figure 25), there was a slight but noticeably higher power loss for the offset pivot design, which is consistent with theory. Cooler temperatures have higher oil viscosities and, therefore, more frictional drag. In the case of 50 percent flow (Figure 26), there was no measurable difference in power loss between the offset and center pivot designs. It may be that the higher pad temperatures negate any measurable difference. Considering all test cases, a fair conclusion is that offset pivot power losses were the same to slightly higher than the center pivot design.

Reduced Oil Flow

Figure 27 compares the effects of reducing oil flow by 50 percent for the center pivot LOP configuration. One would expect pad temperatures to increase with reduced flow across the range of conditions. However, Figure 27 shows that only higher speed temperatures are affected, a phenomenon also noted by DeChoudhury and Masters (1983). The authors have similar experience in independent thrust bearing tests. Specifically, some variation in pad temperature with flow at low speed, an intermittent range where flow has little effect, and a significant variation of pad temperature

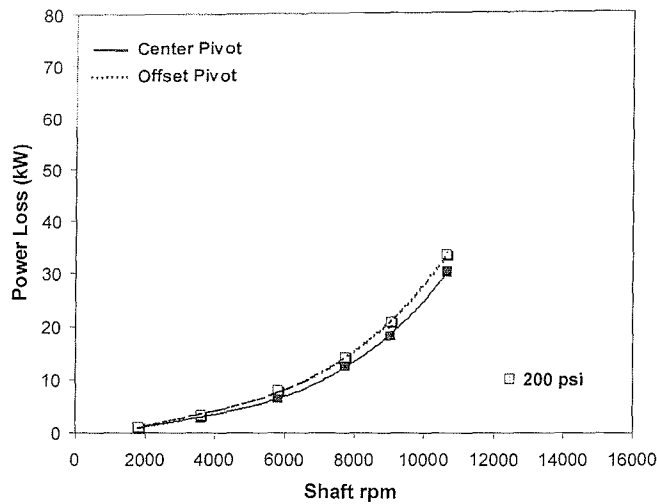


Figure 25. Center Versus Offset Pivot Power Loss, LOP, 100 Percent Flow.

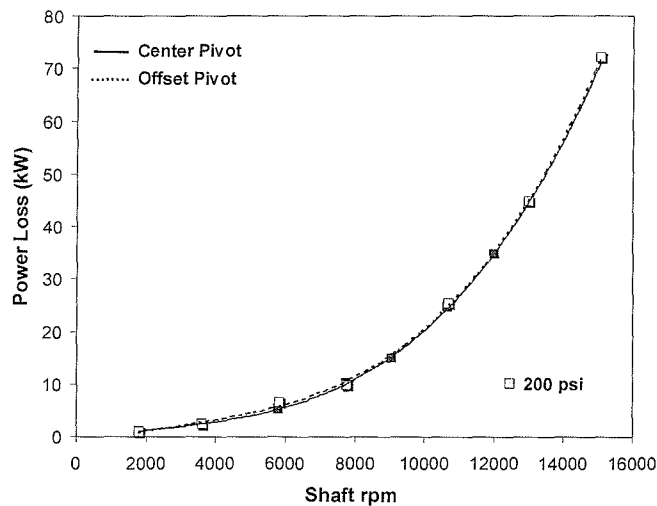


Figure 26. Center Versus Offset Pivot Power Loss, LBP, 50 Percent Flow.

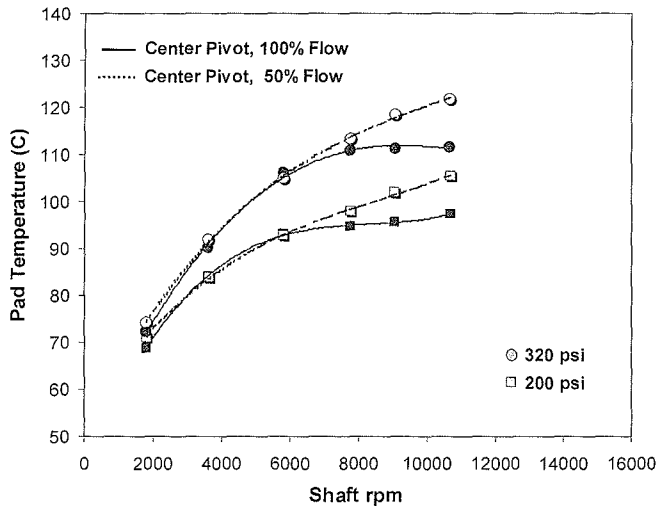


Figure 27. 75 Percent Location Pad Temperatures, LOP, 50 Versus 100 Percent Flow.

with flow at high speeds. These appear to coincide with the laminar, transitional, and turbulent film regimes. And so it would seem that the observed phenomenon is associated with film turbulence.

Be that as it may, the purpose of the work is to improve bearing limitations. Flow reduction for the center pivot design is inappropriate because pad temperatures (Figure 27) are already high. However, the lower temperatures of the offset pivot design may allow for an improvement in efficiency with reduced oil flow.

With center pivot pad temperatures as a baseline, Figure 28 compares the effects of decreasing the offset pivot bearing oil flow for a LOP orientation at 320 psi projected load and 10,660 rpm. For a 50 percent reduction in flow, the offset pivot loaded pad temperatures increase fairly evenly from leading to trailing edge, such that the entire pad is running approximately 10°C (50°F) hotter. With 50 percent flow, the offset pivot 75 percent location is still cooler than the center pivot baseline, but the maximum pad temperature is hotter. The question as to which comparison should be used to judge the two designs becomes more relevant.

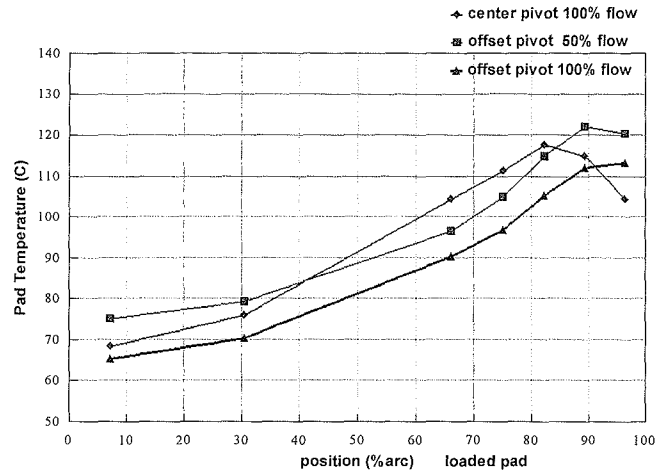


Figure 28. Effects of Flow on Loaded Pad Temperature Profiles, LOP, 320 PSI, 10,660 RPM.

In the authors' experience, both are important. Maximum pad temperature is of concern in regard to the oil, where additives and base stock can break down or deposit on the pad surfaces causing bearing temperatures to increase over time. In regard to the bearing, the 75 percent location is a general area known from experience to suffer distress under adverse conditions. This is typified by Figure 29, which shows mechanical fatigue of the babbitt near the 75 percent location of a center pivot pad. The 75 percent location is only a rule-of-thumb. The actual location changes with load and speed and is also influenced by other parameters. A technically valid comparison requires an analysis of pressure and temperature related to mechanical criteria for the babbitt. One method used by the authors is as follows:

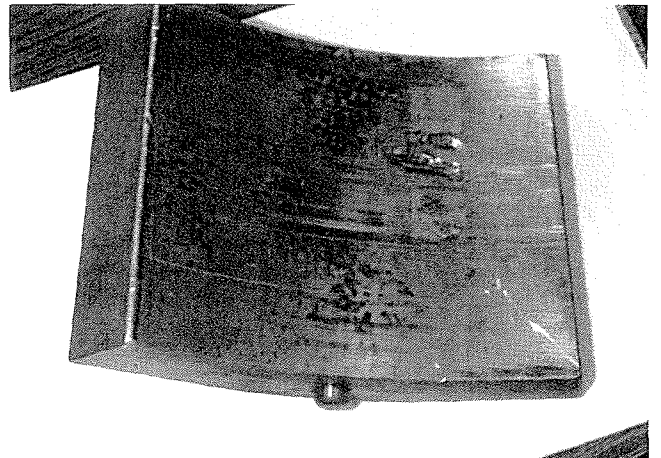


Figure 29. Mechanical Fatigue on a Center Pivot Pad.

Figure 30 is a plot of yield point versus temperature based on data from ASTM-B23 for grade 2 babbitt. In the normal range of bearing operating temperatures, the yield point is approximately 4000 psi at 70°C (158°F), and reduces to 2000 psi at 130°C (266°F). The purpose of the figure is to establish a material reference point to judge the two designs.

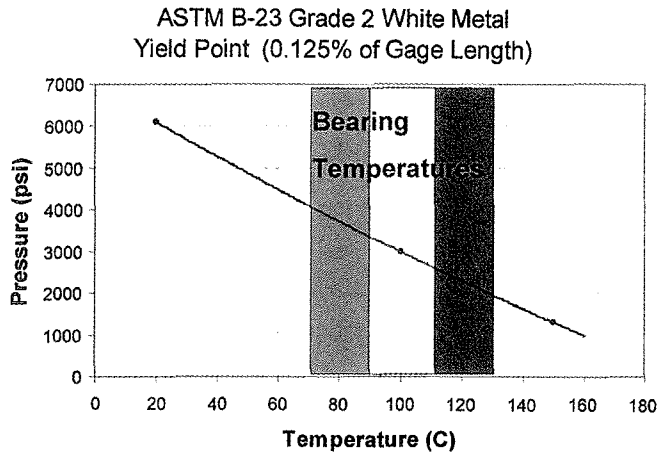


Figure 30. Babbitt .125 Percent Offset Yield Point.

Figure 31 plots the calculated pressure profile and curve-fitted measured pad temperatures for the center pivot pad of Figure 28. Notice that the peaks occur at different locations such that at peak temperature, the pressure on the babbitt is very low and at peak pressure, the temperature is low. Figure 32 is a similar plot for the offset bearing with 50 percent flow. By comparing the pressure and temperature to the babbitt yield point, the condition along the pad arc can be evaluated. For example, from Figure 31 at the 52 percent location, the film pressure and temperature are 1400 psi and 91°C (195.8°F), respectively. From Figure 30, the yield point at 91°C (195.8°F) is 3200 psi. The ratio of the film pressure to yield point is $1400/3200 \text{ psi} = .46$ for this location. By repeating this procedure for each position along the pad, a map of the pressure to yield ratio can be generated as shown in Figure 33. The closest approach to the yield point is found at the 62 percent location for the center pivot design, and at the 82 percent location for the offset pivot design. The highest ratio for each is approximately .47 or, in other words, both have a worst case pressure location that is 47 percent of the babbitt yield point.

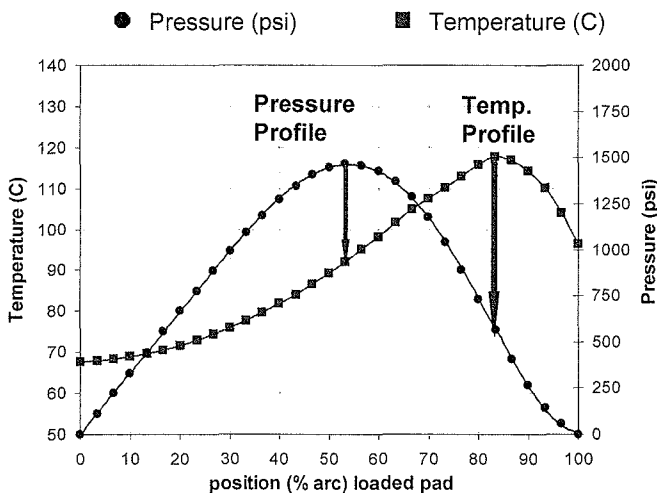


Figure 31. Center Pivot Loaded Pad Pressure and Temperature Profiles, 100 Percent Flow.

The comparison shows that the offset pivot bearing running with 50 percent flow and the center pivot bearing running with 100

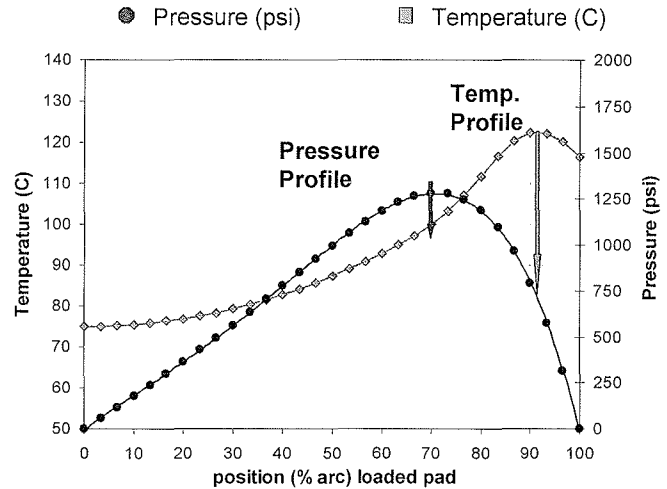


Figure 32. Offset Pivot Loaded Pad Pressure and Temperature Profiles, 50 Percent Flow.

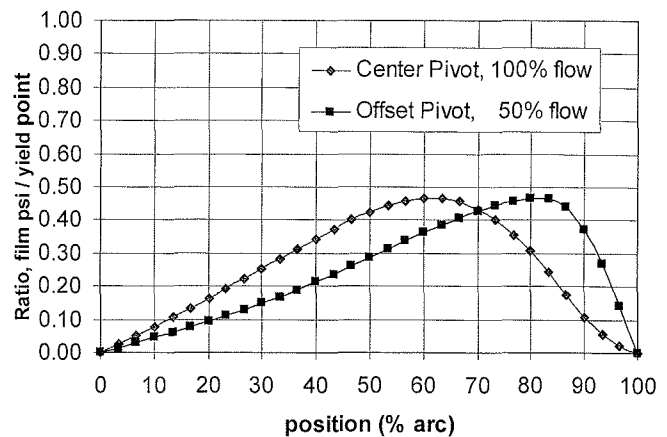


Figure 33. Ratio of Film Pressure to Babbitt Yield Point, LOP, 320 PSI, 10,660 RPM.

percent flow are equivalent in regard to the integrity of the babbitt for the operating condition shown. This exercise also brings out two important points. First, the closest approach to the yield point occurs at a location before the peak temperature. Second, this location is quite different between the offset and center pivot designs, and is not obvious from the temperature profiles of Figure 28 alone.

To complete the study on improving efficiency, Figure 34 compares power loss for the offset pivot pad with 50 percent flow to the center pivot pad with 100 percent flow. There is a reduction in power loss on the order of 10 to 20 percent, which provides a small increase in machine efficiency. With reduced flow, the offset pivot bearing's oil outlet temperatures are 5°C (41°F) higher (Figure 35). The higher outlet temperatures would need to be checked to assure they are still within acceptable specification limits. (Please note that this section compares only one test condition, and is not intended as recommendation to reduce flow by 50 percent in all applications.)

LOP Versus LBP Orientation

When a LOP bearing application is running hot, it is straightforward to assume that reorienting the bearing for LBP will significantly reduce pad temperature, because the load would then be shared by two shoes. In a five-pad bearing, the unit pad load is reduced 38 percent going from LOP to LBP orientation. However, earlier figures indicate that the second loaded pad ran up to 12°C (53.6°F) hotter than the first loaded pad in LBP tests.

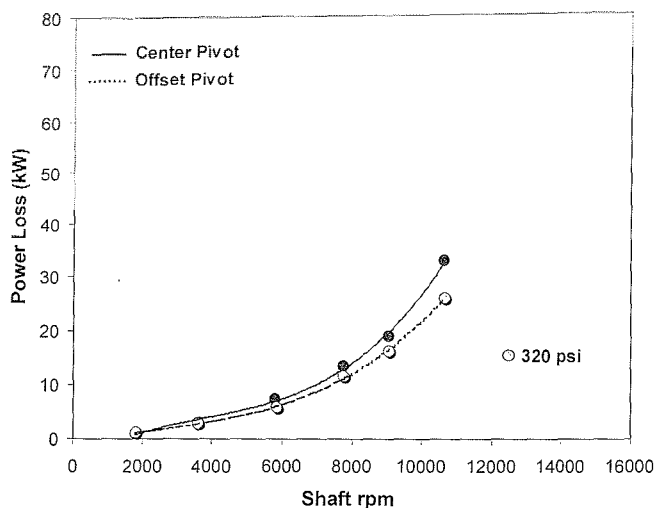


Figure 34. Power Loss—Center Pivot 100 Percent Flow Versus Offset Pivot 50 Percent Flow.

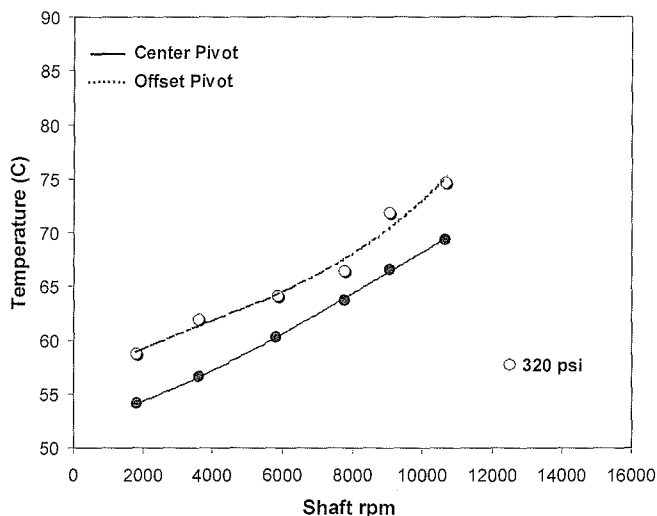


Figure 35. Drain Temperature—Center Pivot 100 Percent Flow Versus Offset Pivot 50 Percent Flow.

Figure 36 compares temperature profiles for LOP and LBP orientations for the center pivot bearing at identical operating conditions. It is noticed that the hottest pad temperature for each orientation reaches similar levels, just above 100°C (212°F). In other words, there is not a significant reduction in pad temperature going from LOP to LBP orientation for the case shown. Figure 37 compares pad temperature profiles of the LOP loaded pad with the LBP second loaded pad. There is little difference in pad temperature profile between the two orientations.

Figure 38 plots center pivot 75 percent location pad temperatures, and Figure 39 plots offset pivot power loss measurements comparing LOP and LBP performance. These figures were chosen to represent general observations of a study of all operating conditions tested that include center and offset pivots at various flows and over a range of operating loads and speeds. LBP orientation in most cases ran with lower pad temperatures than the LOP configuration, but not as much as would be expected. This is attributed to the effects of hot oil carryover. In general, the temperature benefits of LBP orientation improve with load and speed. In regard to power loss, there was no measurable difference between LOP and LBP orientation (Figure 39).

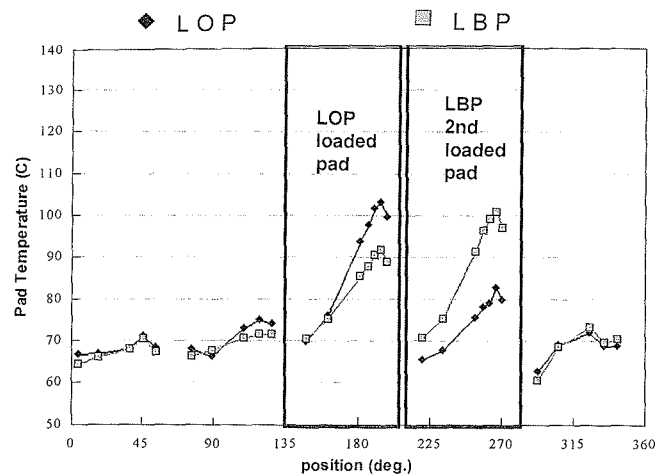


Figure 36. LOP Versus LBP Pad Temperature Profiles, 200 PSI, 7760 RPM, 50 Percent Flow.

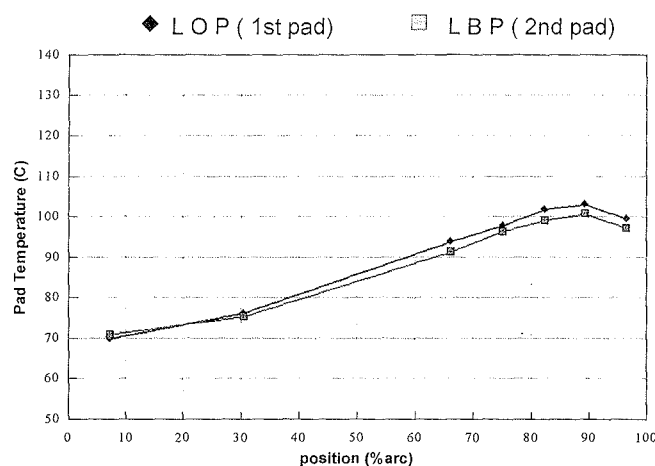


Figure 37. LOP Versus LBP Loaded Pad Temperature Profiles, 200 PSI, 7760 RPM, 50 Percent Flow.

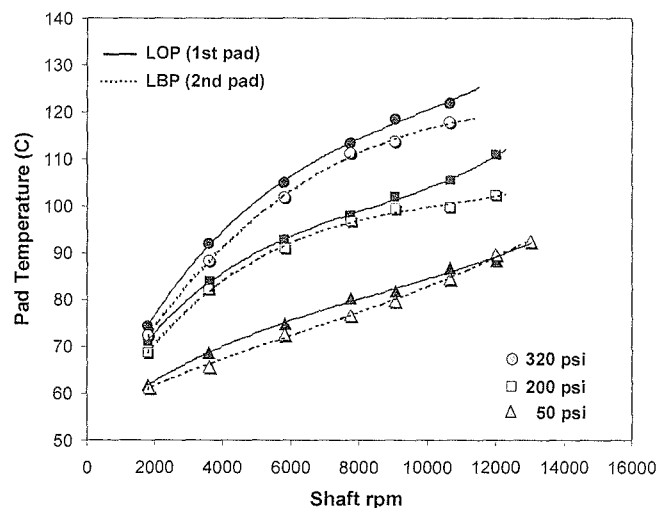


Figure 38. 75 Percent Location Loaded Pad Temperatures, Center Pivot, 50 Percent Flow.

FLOODED VERSUS DIRECT LUBE TEST DATA AND DISCUSSIONS

This section compares the effects of direct lubrication and discharge configuration. As mentioned earlier, the term "flooded"

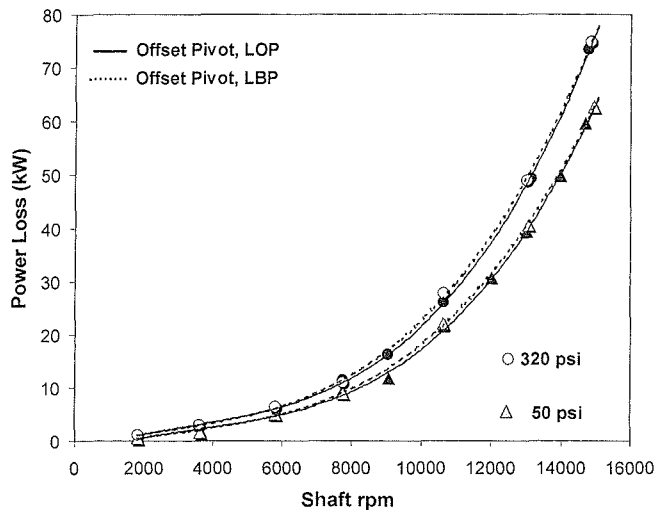


Figure 39. Power Loss—Offset Pivot, LOP and LBP, 50 Percent Flow.

refers to a method of lubrication. For the flooded design tested, oil is provided to the bearing between the pads and the discharge is restricted by labyrinth seals, which provides oil to the journal surface by flooding the spaces between pads (Figure 40). Leading-edge-groove is a method of lubrication that provides oil directly to the journal surface at the entrance to the oil film (Figure 41). In both of these methods, the principles of hydrodynamic operation are the same. Specifically, the journal drags oil into the space between the journal and bearing pads.

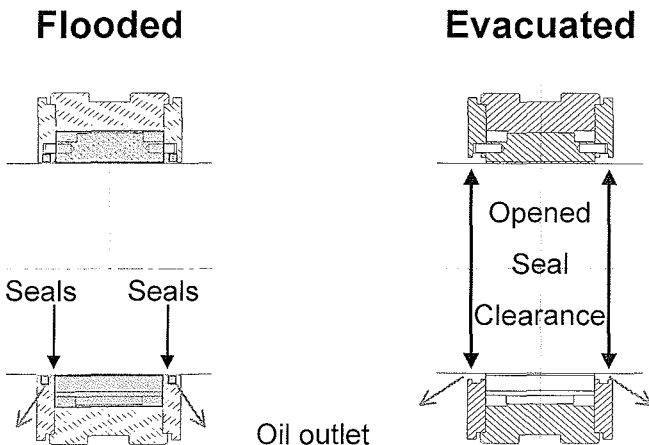


Figure 40. Discharge Configuration, Flooded Versus Evacuated.

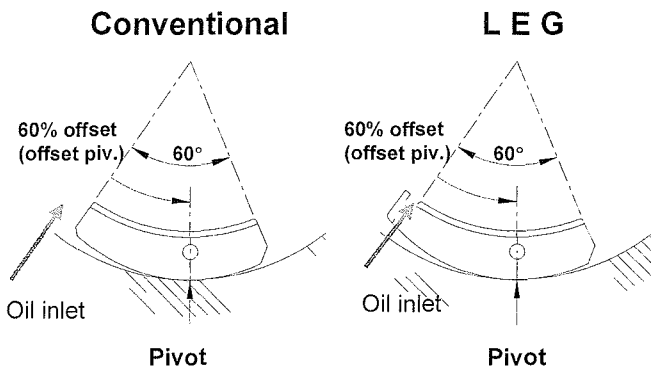


Figure 41. Method of Lubrication, Conventional Versus Direct Lube.

In the case of direct lubrication, it is desirable to allow the hot oil to exit freely from the bearing to reduce parasitic losses and efficiently remove heat. There are a few different methods for evacuating the bearing cavity. For test data reported in this paper, the labyrinth seal clearances were enlarged (Figure 40). This is a most efficient method because the side leakage (exiting oil) is discharged between the end seals and shaft without being captured and recirculated within the bearing. Note that in these comparisons, two parameters are changed—the method of lubrication and the discharge configuration. Otherwise, the flooded and LEG bearings are the same. The pads have 60 percent offset pivots (Figure 41), and data are compared under identical operating conditions and oil flow.

Conventional (Flooded) Versus LEG (Evacuated)

Figure 42 compares conventional, flooded bearing pad temperature profiles to direct lube, evacuated pad temperature profiles at conditions of 320 psi and 10,660 rpm. The direct lube design has noticeably lower pad temperature levels, especially at the leading edges of the pads. This is attributed to providing cool oil directly to the leading edge, which affects the entire loaded pad (Figure 43). Comparing 75 percent location temperatures for 200 psi in Figure 44, and 320 psi in Figure 45, this benefit is noted to improve with load and speed, up to 16°C (60.8°F) cooler at extreme test conditions. It is also noticed that the differences begin around the turbulent transition, evident by the inflection in the curves. This seems to tie in with the behavior of pad temperature and flow with turbulence discussed in the reduced oil flow section of this paper.

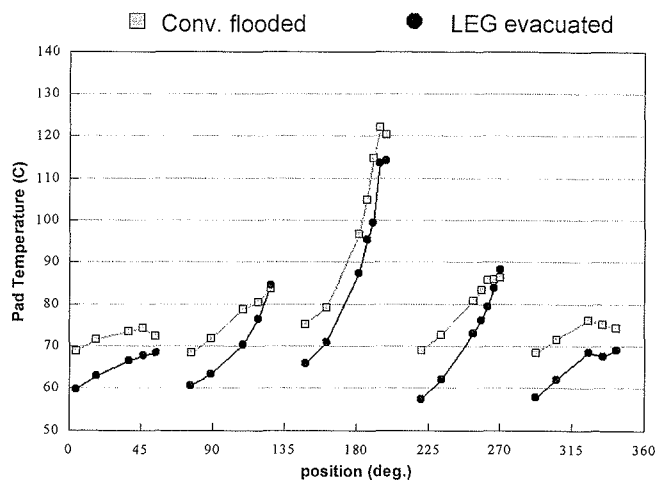


Figure 42. Pad Temperature Profiles, LOP, 320 PSI, 10,660 RPM, 50 Percent Flow.

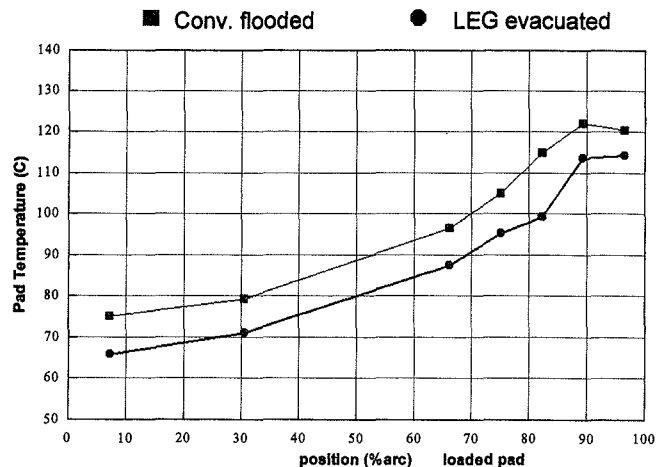


Figure 43. Loaded Pad Temperature Profiles, LOP, 320 PSI, 10,660 RPM, 50 Percent Flow.

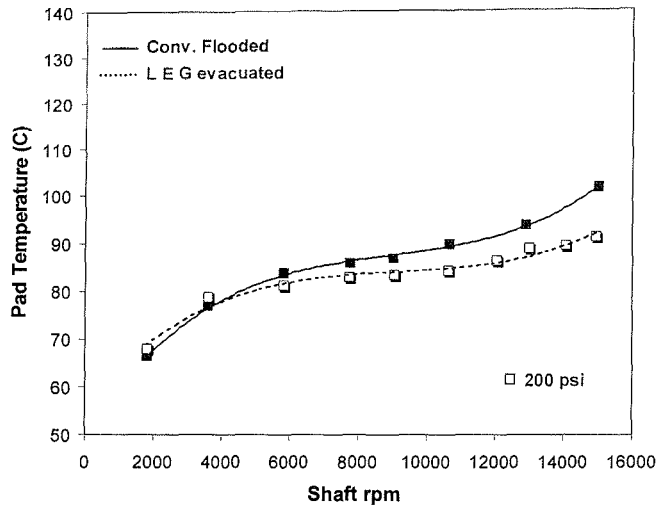


Figure 44. 75 Percent Location Pad Temperatures, LOP, 50 Percent Flow, 200 PSI.

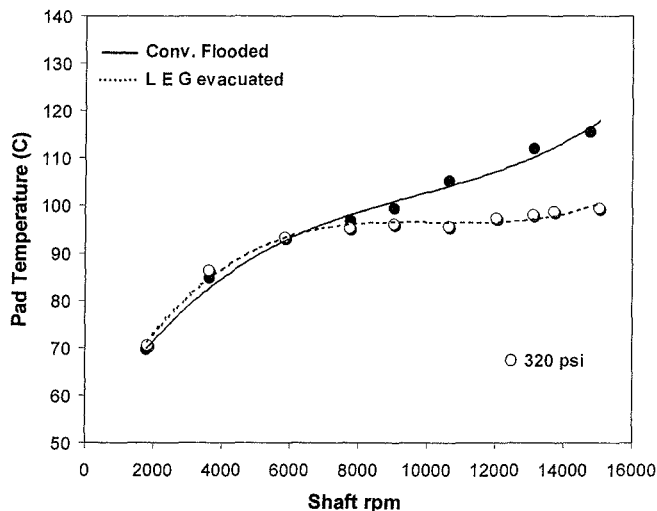


Figure 45. 75 Percent Location Pad Temperatures, LOP, 50 Percent Flow, 320 PSI.

Figure 46 compares power losses for the two designs. Power loss in the evacuated, direct lube bearing measured significantly lower than the flooded bearing design, on the order of 45 percent at extreme operating conditions. Resulting oil drain temperatures are plotted in Figure 47. It is noticed that the flooded design oil outlet temperatures are very high, exceeding typical specification limits at the higher speeds. The evacuated, direct lube design oil outlet temperatures ran up to 15°C (59°F) cooler, and support the reduction in power loss from measured torque. As with pad temperatures, a study of power loss and oil outlet temperatures indicates that the benefits improve with increased speed and load.

Effects of Discharge Configuration on Direct Lubrication

To distinguish between the effects of direct lubrication and evacuated discharge, direct lube tests were also performed in a flooded configuration with restrictive labyrinth seals. Pad temperature results are over plotted in Figure 48. In the flooded condition, the direct lube loaded pad temperatures increased to a level between the LEG evacuated and conventional flooded bearing designs. Based on this and comparisons of other test conditions, part of the direct lubrication pad temperature reduction is attributed to the method of lubrication, and part is attributed to the discharge configuration.

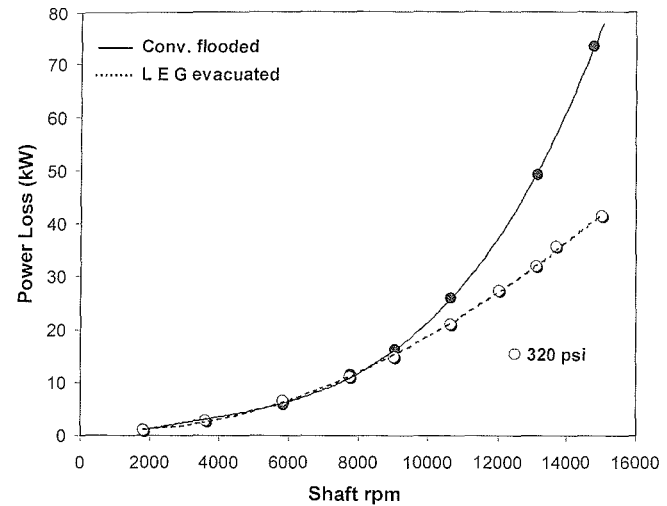


Figure 46. Power Loss—Conventional (Flooded) Versus Direct Lube (Evacuated), 50 Percent Flow.

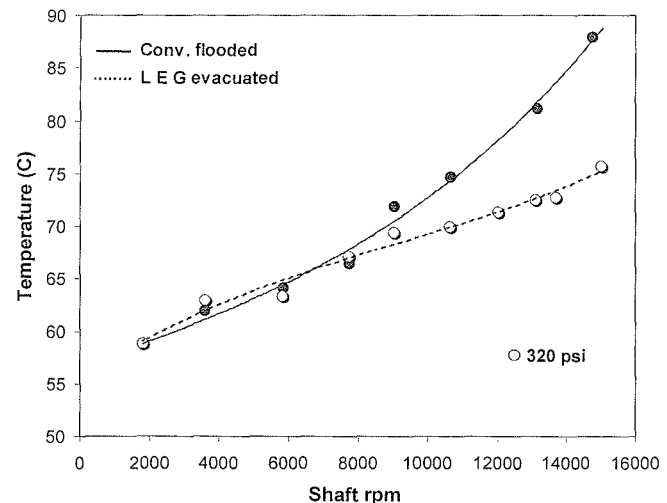


Figure 47. Oil Outlet Temperatures—Conventional (Flooded) Versus Direct Lube (Evacuated), 50 Percent Flow.

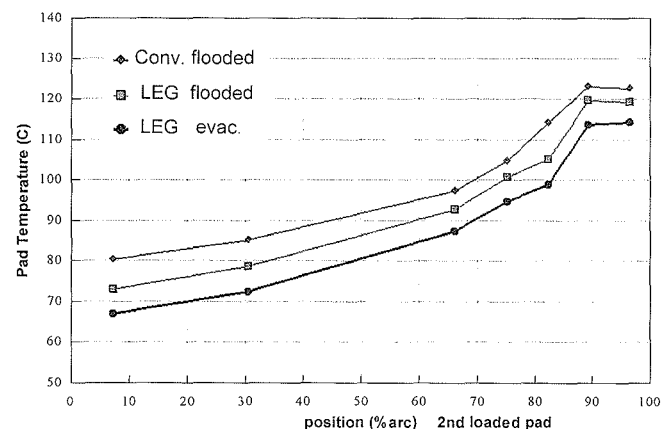


Figure 48. Affects of Oil Discharge Restriction on Direct Lube Pad Temperature.

Resulting power losses are compared in Figure 49, which also shows an increase to a level between the LEG evacuated and conventional flooded bearing losses. The data indicate that a significant portion of the power loss reduction is due to the discharge configuration, and a smaller portion to the method of lubrication.

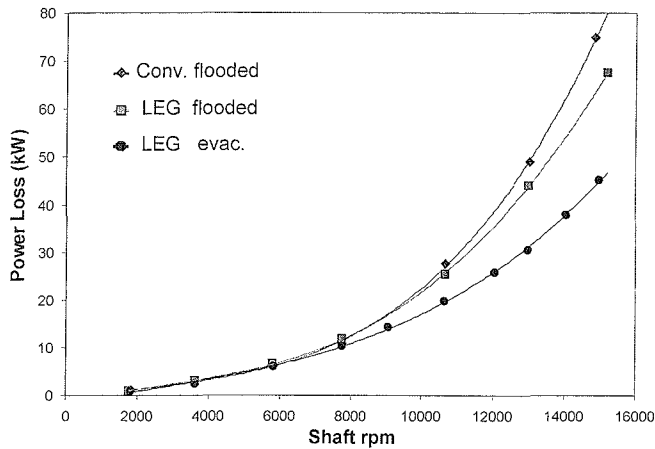


Figure 49. Affects of Oil Discharge Restriction on Direct Lube Power Loss.

SUMMARY

Returning to the original purpose of the work, the tests were undertaken to improve the capacity and performance of high-speed turbomachinery by extending bearing speed and/or load limitations and improving efficiency.

Extension of speed and load capability is accomplished by parameters that reduce bearing temperatures. In this respect, the offset pivot reduces temperatures. Referring to Figure 19 for a comparative example (say a 100°C (212°F) temperature limit and 320 psi load) the offset pivot can extend the application's speed to 11,000 rpm compared to the center pivot, which reaches 100°C (212°F) at approximately 5000 rpm. For a constant speed, say 11,000 rpm, the offset allows loads on the order of 320 psi where the center pivot design is at 200 psi.

In regard to improving machine efficiency, this can be attained by reducing oil flow and power loss, explained earlier with Figure 34. Because of the lower pad temperatures, it is possible to reduce oil flow in the offset design without jeopardizing the integrity of the babbit. This is accompanied by a small increase in machine efficiency, on the order of 10 to 20 percent. With reduced flow, the oil outlet temperatures are higher and need to be checked to assure that the levels do not exceed acceptable limits.

Temperatures are further reduced in the direct lube design (Figures 44 and 45). Continuing the example, the LEG evacuated design allows for 320 psi operation to 15,000 rpm compared to the conventional flooded design which reaches 100°C (212°F) at 9000 rpm (both with 50 percent flow). For efficiency, direct lubrication with an evacuated discharge configuration can attain most significant reductions in power loss, on the order of 45 percent (Figure 46), with lower pad temperatures as well as a significant reduction in oil outlet temperature (Figure 47). The reductions in power loss are significant enough to allow an additional benefit considering design for a smaller lubrication system.

CONCLUSIONS

Conventional (Flooded) Bearings

This series of tests was performed with conventional, flooded pivoted shoe journal bearings to compare the effect of pad pivot location on bearing performance. The pivot locations tested were center (50 percent) and offset (60 percent). Other than the location of the pivot, bearing geometry for the two sets of pads was identical. The bearings were tested in LOP and LBP configurations. Conclusions arising from these tests are as follows:

Center Pivot Versus Offset Pivot

- Center pivot maximum temperatures occurred between the 65 percent and 90 percent location.

- Offset pivot maximum temperatures occurred between the 82 percent location and trailing edge.
- As speed increases, the hottest temperature location of the loaded pads moves toward the trailing edge. The trend is reversed for increasing load.
- The offset pivot design ran cooler in all cases tested, as much as 20°C (68°F) lower than the center pivot design. The benefit improved with speed and load.
- Offset pivot power losses were the same to slightly higher than the center pivot pad.

Effects of Reduced Flow

- Offset pivot design oil flow can be reduced without jeopardizing the integrity of the babbit compared to a center pivot design at similar operating conditions.
- Reducing the flowrate to the offset pivot bearing by 50 percent reduced power losses 10 to 20 percent. This provides a small increase in machine efficiency.
- For conditions tested, oil outlet temperatures ran 5°C (41°F) higher with a 50 percent flow reduction.

LOP Versus LBP Orientation

- For LBP orientation, the second loaded pad ran up to 12°C (53.6°F) hotter than the first loaded pad. This is attributed to hot oil carryover.
- The LBP orientation in most cases ran cooler than LOP, but not as cool as expected, which is also attributed to hot oil carryover.
- The benefits of LBP orientation improve with load and speed.
- There was no measurable difference in power loss between LOP and LBP orientation.

Flooded Versus Direct Lubrication

This section compares the effects of direct lubrication and discharge configuration. The conventional, flooded bearing has oil discharge restricted by labyrinth seals. The direct lube LEG design is tested with two discharge configurations described below. Other than this, the two bearings are of the same geometry and compared at identical operating conditions and oil flow.

Conventional Flooded Versus Direct Lube (Evacuated)

In these comparisons, the direct lube bearing has an evacuated discharge configuration via large clearance end seals.

- Direct lubrication with an evacuated discharge design has significantly lower pad temperature, power loss, and oil outlet temperature. At extreme conditions, pad leading edge temperatures were significantly cooler than the conventional, flooded design.
- Direct lube, 75 percent location pad temperatures ran up to 16°C (60.8°F) cooler than the flooded bearing at extreme operating conditions. Maximum pad temperatures were as much as 12°C (53.6°F) cooler.
- The direct lube design with an evacuated discharge reduces power loss on the order of 45 percent compared to the conventional, flooded design. Oil outlet temperatures were up to 15°C (59°F) cooler.
- The pad temperature and power loss benefits improved with increased speed and load.

Direct Lube (Flooded) Versus Direct Lube (Evacuated)

In this section, the effects of oil discharge configuration on direct lube performance were compared. The LEG flooded discharge configuration had the oil discharge restricted by

labyrinth seals. The LEG evacuated discharge configuration had large clearance end seals. The conclusions are:

- Part of the reduction in LEG pad temperatures is due to the direct lube design, and part is due to the evacuated discharge configuration.
- A significant portion of the power loss reduction is attributable to evacuation of the housing, and a smaller portion to the method of lubrication.

ADDITIONAL COMMENTS

Data contained in this report show the necessity to consider other parasitic losses (such as churning and seal losses) in computer models. Parasitic influences are discussed by many authors (e.g., Booser, 1990), and are required to predict the higher losses in flooded designs, and the lower losses in evacuated designs. Other phenomena mentioned throughout the paper need to be better understood to develop accurate computer models. A study of pad temperature profiles can give valuable insight into hydrodynamic behavior, and is necessary for programming models to accurately predict performance. The surface isotherms showing axial temperature gradients are also useful data for program development. It is hoped that the information and high-speed data contained in this paper will enhance the study and prediction of bearing performance in turbomachinery.

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ACKNOWLEDGEMENT

The authors sincerely thank all personnel at Kingsbury, Inc., and the National Research Council of Canada, who contributed to the work documented in this paper. The authors are grateful to Kingsbury, Inc., and the National Research Council of Canada, for permission to publish this information.