Vol. 44 (2001), 4, 543-550
TRIBOLOGY TRANSACTIONS

Measured Temperature Characteristics of 152 mm Diameter Pivoted Shoe Journal Bearings with Flooded Lubrication©

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The steady-state performance characteristics of journal bearings sometimes impose limitations on the operation of high-speed rotating machinery. Such limitations in bearing performance might be the result of one or more of the following: inadequate load carrying capacity, unacceptably high operating temperatures, and inefficient component performance. Lowering operating temperatures helps to boost the bearing's load and/or speed capability, and reducing bearing power loss and/or oil flow requirements improves machine efficiency.

In response to the need to improve bearing performance, the authors have conducted extensive testing of the pivoted shoe journal (PSJ) bearing. This paper describes work from the first phase of the study, in which the effects of independent design and operating variables on the metal temperatures of flooded lubricated, PSJ bearings are examined experimentally. These variables include pivot location, load orientation, shaft speed, and bearing load. In conclusion, it is shown that some of the independent variables have a significant influence on bearing performance, after comparing pad temperature profiles, isotherms, and maximum temperatures.

The study was performed on a rig that measures steady-state performance under light to moderately heavy unit loads, and comparatively high operating speeds. These conditions are representative of modern-day rotating machinery, particularly new turbine and compressor designs.

Presented at the 56th Annual Meeting
Orlando, Florida
May 20-24, 2001
Final manuscript approved June 27, 2001
Review led by Alan Palazzolo

KEY WORDS

Bearings; Hydrodynamic; Fluid Film Bearings; Tilting-Pad Bearings

INTRODUCTION

As operating speeds have increased to improve the performance of rotating machinery, bearing designs have been pushed to their absolute limit. For example, steam and gas turbine designers are now considering the use of pivoted shoe journal (PSJ) bearings for applications operating in excess of 100 m/s. And compressor designers have intentions of operating PSJ bearings at speeds approaching 120 m/s. The dramatic increase in temperature and power loss of a 0.43 m diameter PSJ journal bearing operating in the turbulent regime has been well documented *(1)*. This is one of a number of studies of the steady-state performance of the center pivot PSJ bearing with flooded lubrication *(2)-(7)*. Unfortunately, other studies of offset pivot PSJ bearing designs with flooded lubrication are not so well documented *(8), (9)*.

Tests conducted on a center pivot PSJ bearing with flooded lubrication have shown that pivot design is influential in affecting bearing steady-state performance *(7)*. It was found that the spherical seat bearing has higher operating temperatures and power losses, and also consistently ran with a higher shaft eccentricity, than the key seat bearing. There have been other studies of the spherical seat PSJ bearing *(2), (3), (5), (10), (11)*, and a similar number of studies of the line contact pivot PSJ bearing *(4), (6),*

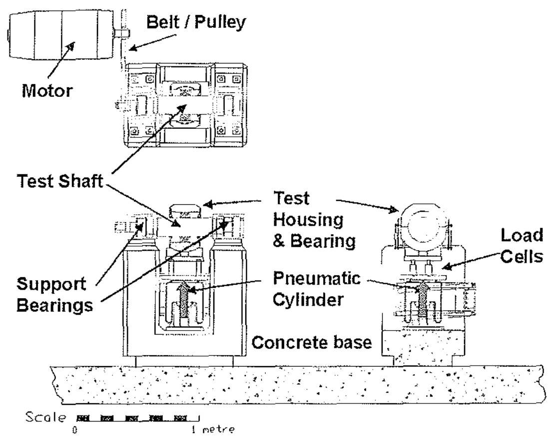

Fig. 1—Schematic of bearing test rig.

(9), (12), (13).

This is the first of a series of papers describing a new experimental and theoretical study that is intended to extend the speed and/or load capabilities, and to improve the efficiency, of PSJ bearings. The work was performed on 0.152 m (6 in) diameter PSJ bearings with rolling contact pivoted pads. This pivot is different from other types of pivot, in that the back of each pad is machined to a radius that is slightly smaller than the radius of the supporting ring. This allows each pad to roll in the supporting ring, so that their angle of inclination can change to accommodate changes in bearing operating conditions. Gardner and Ulschmid *(1)* tested a bearing with this type of pivot, and noted that the shift of the contact point is relatively small, in comparison to the pad circumferential length. Apart from this study, there seems to have been only one other published work on the rolling contact pivot bearing *(8)*.

This paper describes work from the first phase of the study, in which the effects of a number of independent design and operating variables on the operating temperatures of flooded lubricated PSJ bearings are examined experimentally. These variables include pivot location, load orientation, shaft speed and bearing load. The study was performed on a test rig that simulates the operating characteristics of bearings now used in modern-day rotating machinery.

TEST RIG

The test apparatus is illustrated in Fig. 1. It consists of a test bearing, the shaft and drive system, the loading system and the rig supporting structure.

The test shaft is driven by a 112 kW variable speed DC electric motor. A belt and pulley system provides a 4.5:1 speed step-up to give a maximum shaft speed of just over 16,000 rpm. An electronic controller linked to the motor ensures speed control within ± 1%. The test shaft is supported by two 0.089 m diameter PSJ bearings, spaced approximately 0.71 m apart. The test shaft has a journal diameter of 0.152 m (6 in) and a circularity of 12.7 μm. Shaft speed is measured by a slotted optical switch in conjunction with a shaft-mounted disk with a number of drilled holes.

An important feature of the rig is the direct measurement of friction torque. This rig feature provides a more precise determination of bearing power loss than the thermal balance method *(5)-(7), (9), (10)*. The set-up comprises a two-piece bearing housing, which is located midway between the two support bearings. This housing has an accurately machined bottom face, which sits in a spherical hydrostatic bearing. In operation, the hydrostatic bearing eliminates friction between the housing and the loading device, and also serves to align the test bearing to the shaft. The housing is held against rotation by a 0.45 kN capacity load cell mounted on a torque arm. By measuring the force at this load cell, the bearing's power loss may be determined.

A second plane hydrostatic bearing supports the spherical hydrostatic bearing. This provides lateral freedom to the test bearing, and alignment between the test bearing and the applied load. Proximity probes mounted on both ends of the bearing housing measure the horizontal and vertical displacements of the test bearing, with respect to the shaft's location. They also provide a check of the level of alignment between the test bearing and the shaft.

A pneumatic cylinder located between the test bearing housing and the base of the rig generates a static load (to a maximum of 25 kN) on the test bearing. Three load cells, positioned between the top of the pneumatic cylinder and the bottom surface of the plane hydrostatic bearing, measure the applied static load. These load cells are arranged in such a way that each shares an equal proportion of the applied load. In calculating the net load on the test bearing, the combined weight of the test bearing, housing and the hydrostatic bearing system is always accounted for.

A pump with a capacity of $1.26\text{x}10^{-3}$ m^3/s and a maximum supply pressure of 2.1 MPa delivers oil to the test bearing from a 0.6 m^3 capacity tank. The flow rate is measured by a turbine type flow meter with a linear flow range of $0.16\text{x}10^{-3}$ m^3/s to $1.83\text{x}10^{-3}$ m^3/s. Feed oil temperature is measured as it enters the test bearing, and is controlled by a water-oil heat exchanger to within ± 1°C. An industrial type pressure transducer, with a 0 to 0.41 MPa operating range, measures the oil supply pressure to the test bearing. Thermocouples monitor the oil inlet temperature, as well as the oil outlet temperature from each side of the test bearing.

Measurement uncertainties are listed in Table 1

TEST BEARINGS

The test bearings had a nominal diameter of 0.152 m (6 in) and a pad axial length of 0.067 m (2.62 in). The assembled bearing diametral clearance was 229 μm, and the nominal pad preload was 0.25. The pads had a 60 degree included angle, with a nominal thickness at the pivot of 0.022 m. Pivot offsets were 50% in the case of the centre pivot bearing and 60% in the case of the offset pivot bearing, as shown in Fig. 2. Pads and bearing aligning rings were manufactured from steel.

Pad pivots were of the rolling contact design, with radii of curvature in the circumferential and axial directions of 0.086 m and 0.76 m, respectively. It should be noted that the circumferential curvature permits each pad to change its tilt to accommodate changing operating conditions, and that the axial curvature allows each pad to align itself with the shaft.

Lubricant was fed to the bearing through five radial holes. These holes directed oil from an annulus on the outside of the bearing carrier ring to the spaces between the pads. Drainage of

TABLE 1—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 -10 kN load cells (x3)	± 25 N at full scale
Oil flowrate	Turbine flow meter	± 0.5% of reading
Oil supply pressure	Pressure transducer	± 0.02 MPa

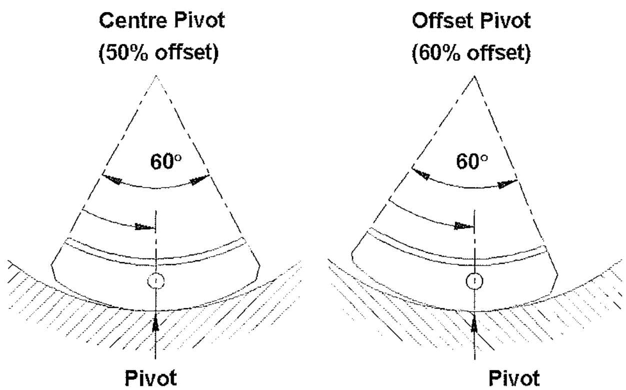

Fig. 2—Pivot geometries.

oil from the bearing was by way of a small annular clearance between the bearing end plates and the shaft. This set-up ensures that the bearing cavity always runs full of oil, and at a slight positive pressure. Details are shown in Fig. 3.

The test bearings were instrumented with an array of 45 type T thermocouples, with the tip of each thermocouple located 500 μm below the babbitt's surface. The loaded pads were more heavily instrumented along the centerline, and included additional thermocouples sited along the edges of the pads. Details are shown in Fig. 4.

TEST CONDITIONS

Tests on the bearings were performed over a range of speeds and loads. Shaft speed was limited to 10,650 rpm, as a result of the flowrate capacity of the test bearing lubrication system. Full details of the recommended flowrates are listed in Table 2. Measured oil feed pressures varied between 0.014 MPa and 0.345 MPa, depending on the amount of oil delivered to the bearing.

Applied bearing loads (with the respective unit loads in parenthesis) were 3.51 kN, 14 kN and 22.24 kN (0.35 MPa, 1.38 MPa, and 2.19 MPa). Tests were performed with both "load between pads" (LBP), and with "load on pad" (LOP).

An ISO VG32 turbine oil, with a supply temperature of 49°C, was used. This lubricant has a measured viscosity of 32.76×10^{-6} m^2/s (32.76 cSt) at 40°C and 5.41×10^{-6} m^2/s (5.41 cSt) at 100°C, and a manufacturer's published density of 0.87 g/ml at 15°C.

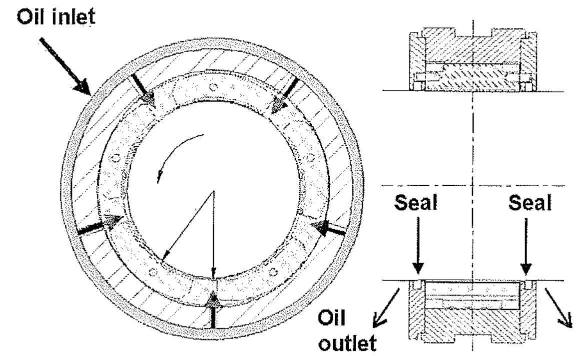

Fig. 3—Flooded lubricated PSJ bearing.

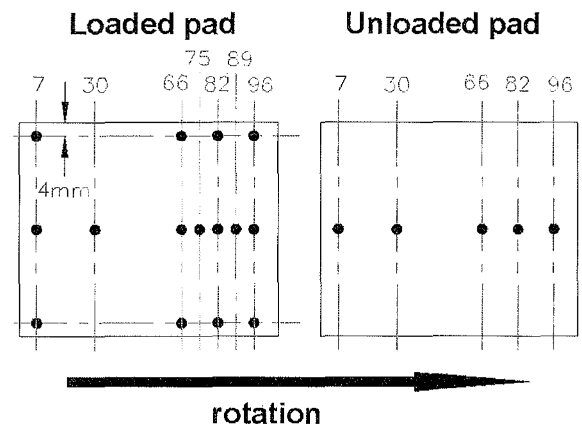

Fig. 4—Thermocouple locations-test bearing.

TEST RESULTS

Bearing Temperature Profiles

Measured temperature profiles from the offset and center pivot bearings, for LBP and LOP, are shown in Figs. 5 and 6, respectively. These plots relate to a load of 22.24 kN, and a shaft speed of 9040 rpm, and show measured center line temperatures plotted against detector angular location. In comparing these plots, the following is observed:

- Pad temperatures throughout the center pivot bearing are generally higher than those of the offset pivot bearing. This applies to both LOP and LBP. For example, Fig. 5 shows that, at some second loaded pad locations, the center pivot bearing was about 20°C hotter than the offset pivot bearing. At the unloaded pads, differences were smaller, but nonetheless still of some significance.
- With LOP, the maximum bearing temperature occurs at the single loaded pad (as shown in Fig. 6). Hence, this becomes the limiting feature of the bearing with regard to high performance machine applications.
- With LBP, the second loaded pad in the direction of rotation runs hotter than the first loaded pad. This is attributed to the effect of hot oil carry-over from one pad to the next *(14)*. The

TABLE 2—RECOMMENDED OIL FLOWRATES

LOAD, KN (lb.)	SPEED, RPM	FLOWRATE, m^3/s (usGPM)
3.51 (788)	1800	$5.7x10^{-5}$ (0.9)
	3600	$1.4x10^{-4}$ (2.2)
	5810	$2.9x10^{-4}$ (4.6)
	7750	$4.8x10^{-4}$ (7.5)
	9040	$6.5x10^{-4}$ (10.2)
	106506	$9.1x10^{-4}$ (14.4)
14 (3150)	1800	$7.3x10^{-5}$ (1.15)
	3600	$1.7x10^{-4}$ (2.7)
	5810	$3.4x10^{-4}$ (5.4)
	7750	$4.9x10^{-4}$ (8.5)
	9040	$7.2x10^{-4}$ (11.3)
	10650	$9.9x10^{-4}$ (15.6)
22.24 (5000)	1800	$8.2x10^{-5}$ (1.3)
	3600	$1.9x10^{-4}$ (3)
	5810	$3.7x10^{-4}$ (5.8)
	7750	$5.7x10^{-4}$ (9)
	9040	$7.5x10^{-4}$ (11.8)
	10650	$1x10^{-3}$ (16.2)

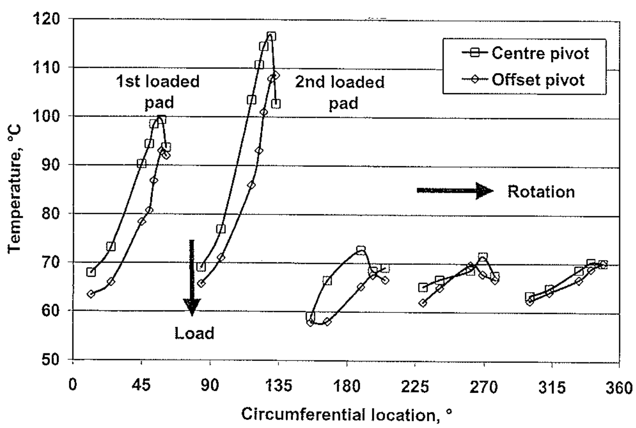

Fig. 5—PSJ bearing temperature profiles LBP, 22.24 kN load, 9040 rpm.

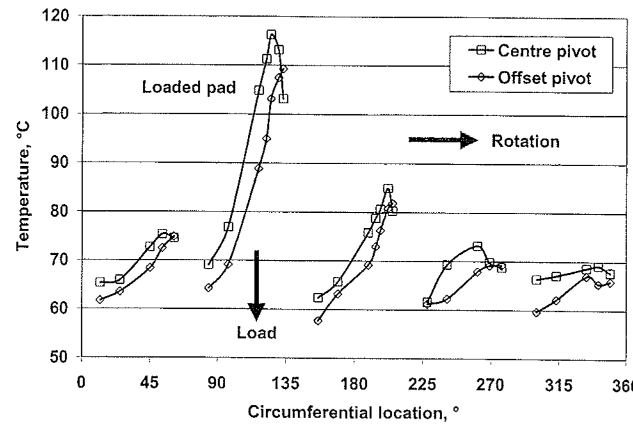

Fig. 6—PSJ bearing temperature profiles LOP, 22.24 kN load, 9040 rpm.

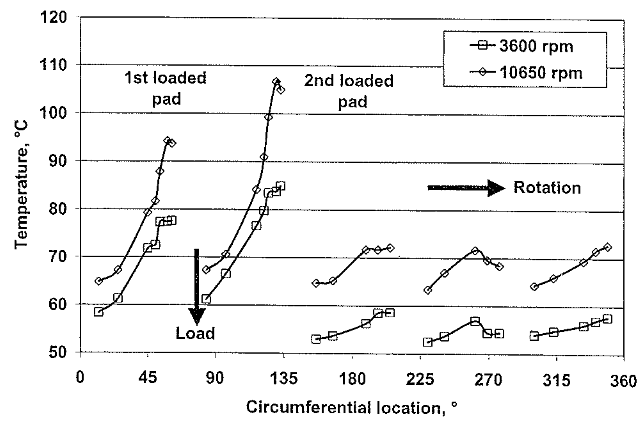

Fig. 7—PSJ bearing temperature profiles - offset pivot LBP, 22.24 kN load, 3600 rpm and 10,650 rpm.

difference between first and second loaded pad maximum temperatures also increases as shaft speed increases, as shown in Fig. 7. Thus, the maximum temperature of the second loaded pad is the limiting feature for LBP with regard to high performance machine applications.

An increase in shaft speed also resulted in a general rise in temperature at both the loaded and unloaded pads. An example is shown in Fig. 7, where bearing temperature profiles from the offset pivot bearing, for a load of 22.24 kN, and shaft speeds of 3600 rpm and 10,650 rpm, are displayed. An increase in bearing load (with constant shaft speed) also caused temperatures at the loaded pads to increase, sometimes in quite a dramatic fashion. This increase in load also caused temperatures in the leading half of the unloaded pads either to remain unchanged, or to increase slightly. However, in the trailing half of the unloaded pads, temperatures sometimes declined. An example of such thermal behavior, taken from the offset pivot bearing, is shown in Fig. 8. Although not shown in this paper, these trends in temperature also applied to the center pivot bearing.

Pad Temperature Profiles

Measured second loaded pad temperature profiles from the offset and center pivot bearings, for LBP, are displayed in Figs. 9 and 10. Figure 9 shows profiles for a constant load of 14 kN, and shaft speeds of 3600 rpm and 10,650 rpm. Profiles for a constant shaft speed of 10,650 rpm, and loads of 14 kN and 22.24 kN, are shown in Fig. 10. In comparing these plots, the following is observed:

- Figure 9 confirms that, with LBP, the second loaded pad of the center pivot bearing runs hotter than the second loaded pad of the offset pivot bearing. In this example, differences in maximum pad temperature are of the order of 8°C at 3600 rpm, and 10°C at 10,650 rpm.
- In the case of the centre pivot bearing, the temperature rises to a maximum in the vicinity of the 85% to 90% pad location, and then falls to a lower value at the pad's trailing edge.

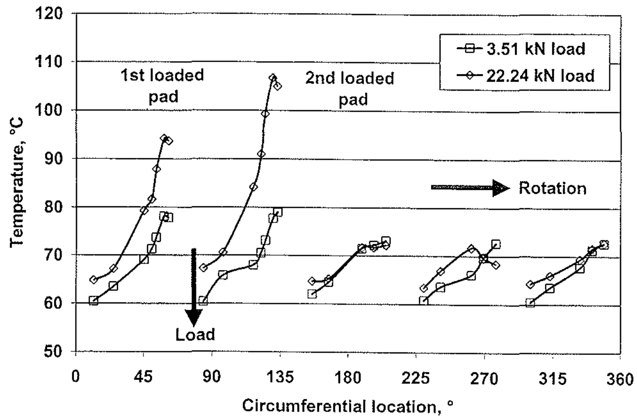

Fig. 8—PSJ bearing temperature profiles - offset pivot LBP, 10,650 rpm, 3.51 kN and 22.24 kN loads.

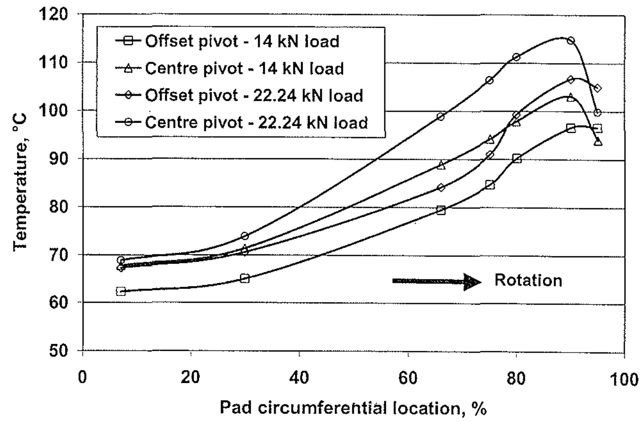

Fig. 10—Second loaded pad temperature profiles - center and offset pivots LBP, 10,650 rpm, 14 kN and 22.24 kN loads.

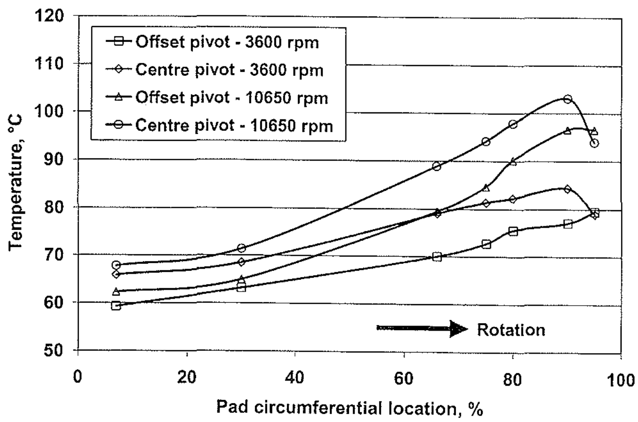

Fig. 9—Second loaded pad temperature profiles - center and offset pivots LBP, 14 kN load, 3600 rpm and 10,650 rpm.

Other profiles (not shown in this paper) have shown that, for some low speed - high load test conditions, the maximum temperature was sited even further away from the pad's trailing edge, at the 65% location. In the case of offset pivot design, the location of the maximum pad temperature always remained close to the pad's trailing edge. This observation is consistent with that reported by Brockwell, et al. *(14)*, following tests on 0.098 m diameter PSJ bearings.

- In regard to second loaded pad temperatures, the leading edge temperature of the centre pivot bearing is generally higher than the leading edge temperature of the offset pivot bearing. At the trailing edge, however, differences in temperature between the two pad designs are smaller. Indeed, for some test cases not shown in this paper, the difference was virtually negligible. Typical pad temperature profiles are shown in Fig. 9.
- In Fig. 10, trends in second loaded pad temperatures are similar to those shown in Fig. 9, with one significant difference. Maximum pad temperatures are higher, as a result of the greater load on the bearing. In the case of the center pivot PSJ bearing, the tests have confirmed that, as a result of an increase in load, the location of the maximum temperature moves away from the pad's trailing edge.

Loaded pad temperature profiles for LOP are displayed in Figs. 11 and 12. Trends in temperature are similar to those associated with LBP, but with one significant difference. In the case of the center pivot bearing with LOP, a change in operating conditions caused a more pronounced movement in the location of the maximum temperature. For example, Fig. 11 clearly shows that the location of the maximum temperature moves towards the trailing edge as the speed is increased. On the other hand, an increase in load (Fig. 12) reverses this trend, and the location of the maximum temperature moves away from the trailing edge.

Maximum *vs.* 75% Pad Location Temperatures

To further reduce the data for generalized comparison, it may be desirable to view a single data point. Some choose to compare maximum measured temperature *(9)*, while industry specifications, such as those published by the API, often recommend the 75% location *(15)*. Accordingly, measured maximum and 75% pad location temperatures from the offset and center pivot bearings, for LBP and LOP, are shown in Figs. 13 and 14, respectively. In comparing these plots, the following is observed:

- Maximum pad temperatures are greater than 75% pad location temperatures. This applied to both the center and offset pivot bearings, for both LOP and LBP.
- In the case of the offset pivot bearing, there can be quite large differences between the maximum and 75% pad location temperatures. This is not really surprising, given the fact that the location of the maximum temperature is usually in close proximity to the pad's trailing edge. However, in the case of the center pivot bearing, the location of the maximum temperature is inboard of the pad's trailing edge, and in closer proximity to the 75% pad location.
- In comparing the maximum and 75% location temperatures of the offset and center pivot bearings, it was found that, at

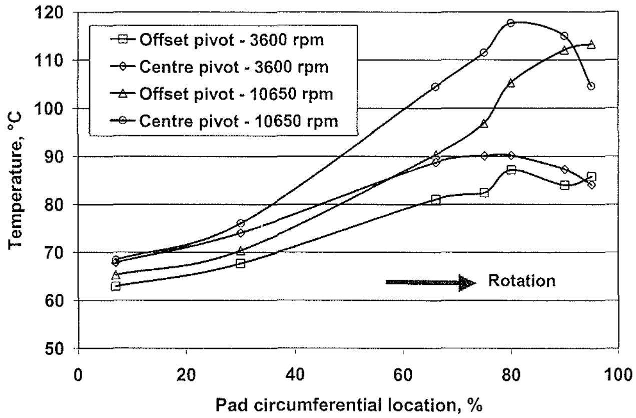

Fig. 11—Loaded pad temperature profiles - center and offset pivots LOP, 14 kN load, 3600 rpm and 10,650 rpm.

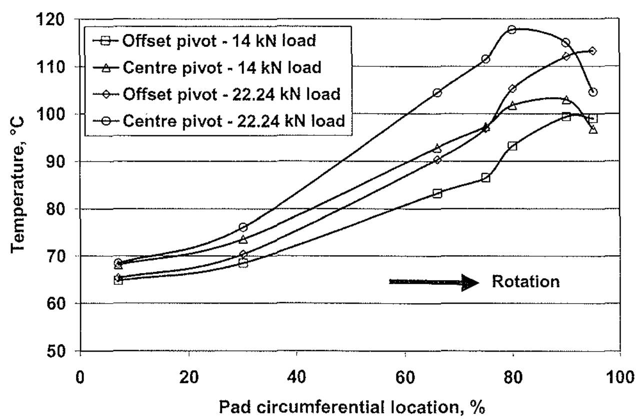

Fig. 12—Loaded pad temperature profiles - center and offset pivots LOP, 10,650 rpm, 14 kN and 22.24 kN loads.

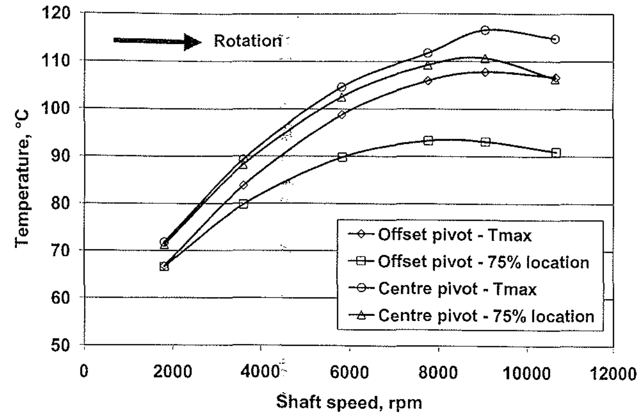

Fig. 13—Maximum and 75% pad location temperatures - center and offset pivots LBP, 22.24 kN load.

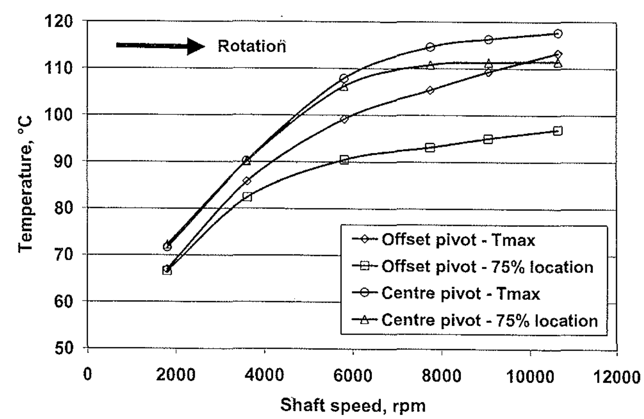

Fig. 14—Maximum and 75% pad location temperatures - center and offset pivots LOP, 22.24 kN load.

both locations, the offset pivot bearing always ran with lower operating temperatures. This was true for all of the test conditions considered in this study. For example, at 10,650 rpm with LOP (Fig. 14), the difference between the offset and center pivot bearing 75% location temperatures is of the order of 15°C. For this same test condition, the difference in maximum pad temperature is smaller, at about 5°C. Generally, it was found that differences between offset and center pivot bearing temperatures (both maximum and 75% pad locations) increased with speed and load.

Pad Isotherms

Isotherms from the loaded pad of the center and offset pivot bearings are shown in Figs. 15 and 16, respectively. They apply to the LOP condition, with a load of 22.24 kN and a shaft speed of 9040 rpm. These 3-dimensional curves were generated by curve fitting data taken from thermocouples fitted to the loaded pad of each test bearing. Profiles such as these are an important tool, since they are able to show that the test bearing is properly aligned with the shaft, and not subjected to edge or skew loading. These latter conditions are undesirable, can distort comparisons and may lead to erroneous conclusions. The symmetric patterns shown in Figs. 15 and 16 are confirmation of the good axial alignment of the test bearings.

SUMMARY OF RESULTS

The long-term goal of this study is to improve PSJ bearing performance, by extending speed and load capability, and by increasing efficiency. In this paper, it is shown that using offset pivot pads can increase the operational envelope of the flooded lubricated PSJ bearing. This is the result of the bearing's lower operating temperatures. For example, Fig. 13 shows that the center pivot bearing reached a maximum temperature of 100°C at about 5000 rpm, with LBP and a load of 22.24 kN. However, for the same maximum temperature and load configuration, the offset pivot extends the bearing's speed to just under 6500 rpm. Based on 75% location temperatures, the offset pivot bearing extends the bearing's speed capability even further, to about 11,000 rpm. This rep-

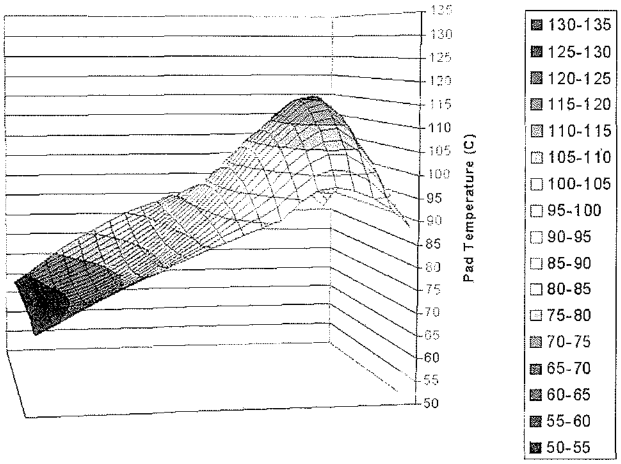

Fig. 15—Loaded pad isotherms - center pivot LOP, 22.24 kN load, 9040 rpm.

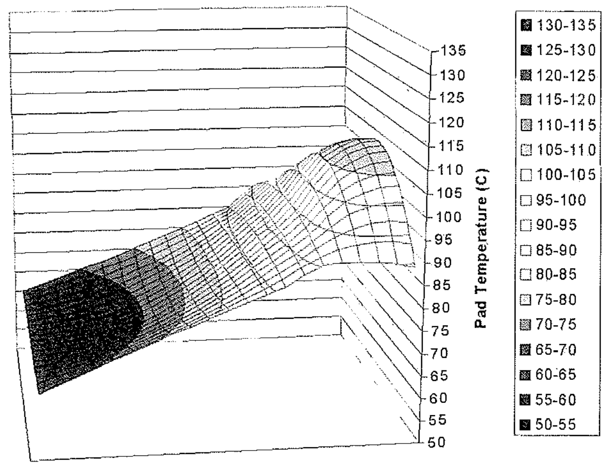

Fig. 16—Loaded pad isotherms - offset pivot LOP, 22.24 kN load, 9040 rpm.

resents a substantial increase in the operational capability of the bearing. Similarly, Fig. 10 shows that, for a shaft speed of 10,650 rpm, the offset pivot bearing has a load capacity which is about 50% greater than that of the center pivot bearing (based on a similar maximum pad operating temperature).

In comparing the temperature data presented in Figs. 13 and 14, it is concluded that, in reorienting the bearing from LOP to LBP, pad operating temperatures in both the center and offset pivot bearings are reduced. However, as Figs. 13 and 14 also show, such reductions in pad temperature only become of real advantage at higher operating speeds, and not at the lower operating speeds.

Other work to further enhance the performance of PSJ bearings will be described in later papers. This will include comparisons of experimental and theoretical data and measurements of friction torque and bulk oil temperatures. Also, the adaptation of directed lubrication technology to the PSJ bearing, to further reduce operating temperatures, power loss and oil flowrate requirements, will be discussed.

CONCLUSIONS

As part of a long-term goal to improve the performance of PSJ bearings, tests have been carried out on conventional, flooded lubricated center and offset pivot PSJ bearings with rolling contact pivots. Conclusions arising from this study are as follows:

- The location of the maximum temperature of both the center and offset pads is dependent on the actual operating conditions of the bearing.
- In the case of the center pivot bearing, the maximum temperature occurs anywhere between the 65 percent and 90 percent pad locations. In the case of the offset pivot bearing, the maximum temperature is closer to the trailing edge, usually between the 90 percent and trailing edge pad locations.
- Differences between the 75% pad location and maximum temperatures can be quite large in the case of the offset pivot bearing, but smaller in the case of the center pivot bearing.
- In all test conditions, the offset pivot bearing ran cooler than the center pivot bearing. In some cases, this difference was as much as 20°C. This benefit improves with speed and load, and can be used as a means of extending the load and/or speed capability of the bearing.
- For LBP, the second loaded pad always ran hotter than the first loaded pad. This is attributed to the effect of hot oil carry over from one pad to the next.
- In terms of bearing operating temperatures, there does seem to be an advantage in switching from LOP to LBP. This advantage improves at higher shaft speeds.

ACKNOWLEDGMENTS

The authors are grateful to Kingsbury, Inc. and the National Research Council of Canada for permission to publish the results of this study. The authors particularly want to thank Brian Liko and Tim Breithaupt of the National Research Council of Canada, without whose efforts this work would not have been possible.

REFERENCES

(1) Gardner, W. and Ulschmid, J., "Turbulence Effects in Two Journal Bearing Applications," ASME Paper No. **73-LubS-7**, (1973).
(2) DeChoudhury, P. and Barth, E., "A Comparison of Film Temperatures and Oil Discharge Temperature for a Tilting-Pad Journal Bearing," *ASME Jour. of Lubr. Tech.*, **103**, pp 115-119, (1981).
(3) Tripp, H. and Murphy, B., "Eccentricity Measurements on a Tilting-Pad Bearing," *ASME Trans.*, **28**, 2, pp 217-224, (1984).
(4) Brockwell, K. and Kleinbub, D., "Measurements of the Steady-State Operating Characteristics of the Five Shoe Tilting Pad Journal Bearing," *Trib. Trans.*, **32**, 2, pp 267-275, (1989).
(5) Harangozo, A., Stolarski, T. and Gozdawa, R., "The Effect of Different Lubrication Methods on the Performance of a Tilting Pad Journal Bearing," *Trib. Trans.*, **34**, pp 529-536, (1991).
(6) Simmons, J. and Dixon, S., "Effect of Load Direction, Preload, Clearance Ratio, and Oil Flow on the Performance of a 200 mm Journal Pad Bearing," STLE Preprint No. **93-AM-2E-3**, (1993).
(7) Pettinato, B. and DeChoudhury, P., "Test Results of Key and Spherical Pivot Five-Shoe Tilt Pad Journal Bearings — Part I: Performance Measurements," *Trib. Trans.*, **42**, 3, pp 541-547, (1999).
(8) Dmochowski, W., Brockwell, K. and DeCamillo, S., "A Study of the Thermal Characteristics of the Leading Edge Groove and Conventional Tilting Pad Journal Bearings," *ASME Jour. of Trib.*, **115**, pp 219-226, (1993).

(9) Simmons, J. and Lawrence, C., "Performance Experiments with a 200 mm, Offset Pivot Journal Pad Bearing," STLE Preprint No. **96-AM-7B-4**, (1996).

(10) DeChoudhury, P. and Masters, D., "Performance Tests of Five-Shoe Tilting-Pad Journal Bearing," *ASLE Trans.*, **27**, 1, pp 61-66, (1983).

(11) DeChoudhury, P. and Masters, D., "Slow Roll Tests on a Five Shoe Tilting-Pad Journal Bearing," STLE Preprint No. **84-AM-2D-1**, (1984).

(12) Brockwell, K. and Dmochowski, W., "Thermal Effects in the Tilting Pad Journal Bearing," *Jour. Phys. D: Appl. Phys.*, **25**, pp 384-392, (1992).

(13) Makino, T. and Taniguchi, S., "Thermohydrodynamic Performance of High Speed Journal Bearings," in *Proc. Instn. Mech. Engrs.*, **210**, pp 179-187, (1996).

(14) Brockwell, K., Dmochowski, W. and DeCamillo, S., "Analysis and Testing of the Leading Edge Groove Tilting Pad Journal Bearing - A New Design for Increasing Load Capacity, Reducing Operating Temperatures and Conserving Energy," in *Proc. of the Texas A&M 23rd Turbomachinery Symp.*, (1994).

(15) Nicholas, J., "Tilting Pad Bearing Design," in *Proc. of the Texas A&M 23rd Turbomachinery Symp.*, (1994).