L. Costa
Universidade do Minho,
Departamento de Engenharia Mecânica,
4810 Guimarães, Portugal

M. Fillon
Université de Poitiers,
Laboratoire de Mécanique des Solides,
U.R.M. C.N.R.S 6610, SP2MI, BP 179,
86960 Futuroscope, France

A. S. Miranda

J. C. P. Claro
Universidade do Minho,
Departamento de Engenharia Mecânica,
4810 Guimarães, Portugal

An Experimental Investigation of the Effect of Groove Location and Supply Pressure on the THD Performance of a Steadily Loaded Journal Bearing

This paper aims to present the results of parametric experiments carried out in order to study the influence of groove location and supply pressure on the performance of a steadily loaded journal bearing with a single-axial groove. Hydrodynamic pressure and temperature distributions on the bush surface, shaft temperature, flow rate and bush torque were measured at variable supply pressure, using bushes with a single groove located at three different positions. A series of tests were carried out for variable applied load and rotational speed. The experimental evidence shows that some bearing characteristics are significantly sensitive to changes in groove location and supply pressure. One groove located at 30 degrees in relation to the load line, in the direction of shaft rotation, can conduct to reductions in maximum temperature, maximum hydrodynamic pressure and bush torque, with a moderate increase in oil flow rate.

1 Introduction

Hydrodynamic bearings are used in several applications and are considered a good choice to support rotating shafts due to their constructive simplicity, reliability, efficiency, and low cost. In modern applications, bearings are designed according a tendency toward "limit design" which implies high rotational speed and also high specific load.

In radial hydrodynamic bearings working in a steady-state regime, oil is usually supplied to the bearing through axial grooves. The supplying conditions (supply pressure, supply temperature, groove dimensions, and location) dictate the flow rate, affecting the oil temperature inside the bearing. As a consequence, the viscosity of the oil and the bearing load capacity will be affected. Thermal effects in hydrodynamic lubrication play, therefore, a crucial role on the prediction of bearing performance.

Very often in practical applications, the bush is mounted with the lubricant supply groove on the load line, allowing for shaft rotation in both directions. Slight changes in load direction may eventually occur. This will affect the clearance at the inlet, thus affecting the oil flow supplied to the bearing and, therefore, the bearing performance.

Experimental investigations on the thermohydrodynamic performance of journal bearings have been carried out over many decades (Swanson and Kirk, 1997), but there is little experimental information concerning the influence of the supply conditions on the thermal behavior of journal bearings with one axial groove. The work reported by Majumdar and Saha (1974) comprises an analysis of the effect of supply pressure on bush temperature distribution for a journal bearing with a single groove on the position of maximum film thickness. For a journal bearing with one groove on the load line, Dowson et al. (1966) have analyzed the effect of feed temperature on the bearing temperature distribution. Mitsui et al. (1983) have investigated the effect of feed

temperature and lubricant termoviscosity on maximum bearing temperature. In another experimental work, Mitsui et al. (1986), have investigated the cooling effect of the supply oil on bearing performance. Recently, Syverud and Tanaka (1997) carried out experiments to study the effect of feed temperature with an additional system for heating or cooling the shaft.

Pinkus (1990) has pointed out the need for experimental work in the field of thermal analysis in bearings in order to enable the improvement of theoretical models. In the present decade many experimental thermohydrodynamic investigations on the performance of journal bearings with two-axial grooves have been published. The influence of groove location has been investigated by Gethin and El-Deihi (1990, 1992), while Ma and Taylor (1995) have studied in detail the effect of feed temperature and supply pressure on bearing performance. According to their results, the supply conditions affect the bearing performance (maximum temperature, flow rate, power loss, friction torque) in different ways, the need to consider all supply parameters to make a more precise performance analysis is apparent.

In this work, the main results of an extensive experimental investigation of the thermohydrodynamic behavior of a single-groove, steadily loaded, journal bearing are presented. The influence of groove location and supply pressure on some bearing performance characteristics is discussed.

2 Test Apparatus, Measurements and Bearing Specifications

The test apparatus was designed and manufactured at the University of Poitiers and has been used before in a number of experimental studies, like those carried out by Ferron et al. (1983) and more recently, Monmousseau et al. (1997) and Kucinski and Fillon (1999). A typical arrangement of the test apparatus is illustrated in Fig. 1. The test set up consists mainly of the test bearing, the shaft driving system and the loading device. Ferron (1982), and more recently Kucinski and Fillon (1999), have given full details of the test apparatus. Only a brief description is given here.

The shaft is driven by a variable speed DC motor (21 kW) by means of a belt multiplying transmission being supported on three

Contributed by the Tribology Division of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS and presented at the STLE/ASME Tribology Conference, Orlando, FL, October 10-13, 1999. Manuscript received by the Tribology Division February 1, 1999; revised manuscript received May 11, 1999. Paper No. 99-Trib-8. Associate Technical Editor: L. San Andrés.

Fig. 1 Schematic representation of test apparatus

preloaded high precision ball bearings, whose mounting arrangement provides proper stiffness as well as high rotational accuracy. The DC motor is linked to an electronic controller, which ensures speed control to an accuracy of ± 10 rpm. The circular imprecision of the shaft is about $5 \mu\text{m}$. An electronic comparator was used to measure the inner diameter of the bush, which was taken as the average of several measurements. This average value was less than $\pm 5 \mu\text{m}$ away from each individual measurements. Thus, the error on the value of radial clearance quoted (Table 2) is in the range $\pm 5 \mu\text{m}$. The bearing geometry is shown in Fig. 2. The test bush was made in bronze having one single groove. Using always the same bush it was possible to change the angular location of the groove from $\phi = -30$ degrees to $\phi = +30$ degrees in relation to the load line.

The loading system (Fig. 3) is a pneumatic cylinder where high pressure air is applied, being the load intensity calculated by knowing the internal air pressure, the cylinder cross section and the weight of the bush. The load direction is vertical and according to Kucinski and Fillon (1999), the error on load evaluation is about ± 0.05 kN. There are two hydrostatic bearings, one of spherical shape to ensure correct alignment between journal and test bush, and the other of plane shape for a correct centring of the load line in relation to the bearing. The friction torque in the bush was measured using a torque-measuring device mounted on a two-armed assembly connected to the bush. Apart from the torque-meter, the only connections from the test bush to the supporting system were the flexible thermocouple leads, the oil supply tube and the pressure measurement tubes, which were assumed to introduce a negligible torsional constraint. The lower part of the bush is machined to form the mating surface of a spherical hydrostatic bearing. This arrangement automatically provides vertical and horizontal alignment between shaft and test bush, allowing for good precision in bush torque measurements.

Oil flow rate was measured using a flowmeter, with an accuracy of ± 0.05 l/min. Supply pressure and feed temperature were controlled. Oil can be supplied to the bearing at a pressure equal or greater than atmospheric. A Bourdon gauge, manometer at entry indicates the value of supply pressure. Oil supply temperature was controlled by a water-oil heat exchanger and was measured just before entering the supply groove.

Nomenclature

a = length of supply groove (m)
 C_r = radial bearing clearance at 20°C (m)
 d = bearing diameter (m)
 D = Bush outer diameter (m)
 L = bearing length (m)
 N = shaft speed (rpm)

P_s = oil supply pressure (kPa)
 T_a = ambient temperature ($^\circ\text{C}$)
 T_s = oil supply temperature ($^\circ\text{C}$)
 W = load (kN)
 w = width of supply groove (m)
 Ψ = attitude angle (degrees)

α = angle measured from the middle of the groove (degrees)
 ω = shaft angular velocity (rad/s)
 ϕ = groove location in relation to the load line (degrees)
 μ_{35} = oil viscosity at 35°C (Pa.s)
 μ_{40} = oil viscosity at 40°C (Pa.s)
 μ_{70} = oil viscosity at 70°C (Pa.s)

Fig. 2 Bearing geometry

For temperature measurements, a set of 34 type K thermocouples 1 mm diameter were mounted in the bush to bring out bush temperatures, environment temperature, and oil supply temperature. The active part of most of them is flushed with the internal surface of the bush; 16 are located at the mid-plane of the bush, being their circumferential locations shown in Table 1 and depicted in Fig. 4. The film/shaft interface temperature was also measured using three thermocouples as described elsewhere (Feron, 1982). Each thermocouple was calibrated by comparing its reading of the water temperature in a thermostatic vessel with that of a high precision analogic thermometer. The discrepancies observed were within the range $\pm 0.5^\circ\text{C}$, this being, therefore, assumed as the accuracy of the thermocouples.

For hydrodynamic pressure measurements a set of 15 holes are drilled in the bush. Each hole is connected via a manifold valve to a high precision manometer. The manometers used are specified as 1 percent accurate over the total measuring range. Pressures measured were generally greater than 30 percent of the total range of each manometer, thus resulting in a maximum error of 3 percent of the pressure measured. Table 1 and Fig. 4 show the locations of pressure pick up points.

All measurements were performed under steady-state conditions. Table 2 lists the essential bearing dimensions at 20°C , oil viscosity, and test conditions.

Repeatability of results was checked by running selected tests three times under the same test conditions. The largest differences observed were less than $\pm 1^\circ\text{C}$ for maximum bearing temperature and for average shaft temperature, and less than ± 18 kPa for maximum hydrodynamic pressure. The latter is about one percent of the maximum pressure observed. All measured values of flow rate were within 4.4 percent of the average value.

3 Results and Discussion

The experiments yielded a large volume of data. Part of them will be presented and analyzed. Generally, the oil supply temper-

Fig. 3 Test bush mounting and loading system

ature was fixed at 35°C. In one case this value was increased to 40°C. Hydrodynamic pressures and bush temperatures presented refer to the mid-plane of the bearing.

3.1 Temperature and Pressure Profiles. The circumferential temperature variation was briefly examined for some operating conditions. The results obtained at different groove locations, with a constant load of 8 kN and a supply pressure equal to 300 kPa, are shown in Fig. 5. It can be seen that for $\phi = -30$ degrees there is a general decrease in the bush internal surface temperature. This occurs, because the groove is placed in the divergent film zone where the viscous dissipation is undoubtedly lower. It can be observed that the maximum temperature is higher in the case of a groove located on the load line (crown bearing configuration, $\phi = 0$ degrees), being 3°C above the maximum temperature measured for $\phi = -30$ degrees. For the groove location corresponding to $\phi = 30$ degrees, the angular position of the maximum temperature is 50 degrees further away from the downstream groove section, comparatively to the other two configurations.

In Fig. 6 is shown the influence of supply pressure and oil feed temperature on the circumferential bush temperature for the crown bearing configuration. There is a generalized increase in bush temperature with increasing oil feed temperature. When the oil feed temperature changed from 35°C to 40°C, the maximum bearing temperature increased only by 3°C. The decrease in viscosity due to an increase in oil temperature resulted in a reduction on viscous dissipation, this explaining the moderate increase observed in the maximum bearing temperature. Furthermore, when the bearing was moderately loaded the circumferential temperature variation and maximum temperature were only slightly affected by oil supply pressure. On the contrary, for the lightly loaded bearing, the influence of supply pressure on temperature is well marked.

Figures 7 and 8 show the variation of hydrodynamic pressure for variable groove location and oil supply pressure. At $\phi = -30$ degrees the maximum hydrodynamic pressure measured was 8% and 12% higher than that measured for $\phi = 30$ degrees and $\phi = 0$ degrees, respectively. Additionally, for $\phi = -30$ degrees, the measured hydrodynamic pressure only started raising at 90 degrees

Fig. 4 Location of thermocouple and pressure holes

Table 2 Essential bearing dimensions and test conditions

Bearing diameter	d	m	0.10
Bearing length	L	m	0.08
Bush outer diameter	D	m	0.20
Radial clearance at 20°C	C_r	10^{-3} m	0.0935
Length of supply groove	a	10^{-3} m	70
Width of supply groove	w	10^{-3} m	16
Groove location	ϕ	degrees	-30, 0.0, +30
Speed	N	rpm	2000 ... 4000
Load	W	kN	2 ... 8
Oil supply pressure	P_s	kPa	50 ... 300
Oil supply temperature	T_s	°C	35, 40
Lubricant type	—	—	ISO VG 32
Oil viscosity at 35°C	μ_{35}	Pa.s	0.0358
Oil viscosity at 40°C	μ_{40}	Pa.s	0.0293
Oil viscosity at 70°C	μ_{70}	Pa.s	0.0111
Ambient temperature	T_a	°C	18 ... 24

downstream of the groove, thus suggesting that oil film reformation occurred at about the same location. At $\phi = 30$ degrees, increasing supply pressure from 70 kPa to 300 kPa caused an increase in maximum hydrodynamic pressure of 19%. At low supply pressure (70 kPa) the effect of groove location is similar to that observed at higher supply pressures (300 kPa). The highest values of maximum hydrodynamic pressure observed for $\phi = -30$ degrees, for both values of supply pressure tested, is mainly due to the reduction of the angular extent of the pressure field, suggesting that elastic effects may eventually be more meaningful for that configuration. This is also in agreement with reductions observed for both oil flow rate and shaft temperature.

3.2 Bearing Performance Characteristics. Two values of applied load were considered, 2 kN and 8 kN, corresponding to specific loads of 0.25 MPa and 1.0 MPa, respectively. The bearing performance characteristics were measured for supply pressures ranging from 50 kPa to 300 kPa. The experimental results of maximum temperature, shaft temperature, flow rate, and bush torque are shown in Fig. 9. Figure 9(a) shows that at moderate load (8 kN), for all groove locations, the variation observed on maximum temperature did not exceed 1°C. At low applied load (2 kN) the maximum bearing temperature decreased with increasing supply pressure. Measured values of maximum temperature tended to get closer at higher supply pressures, for all groove locations. For both low and moderate loads, the lowest maximum temperature was observed for the groove location corresponding to $\phi = -30$ degrees. Shaft temperatures measured are depicted in Fig. 9(b). A low applied load (2 kN) originated the highest shaft temperature and a significant decrease in the value of this parameter with

Table 1 Circumferential location of thermocouples and holes for pressure measurements in the mid-plane of the bush (angle origin at the middle of the groove)

Thermocouples location (degrees)	12.5	45	75	105	135	150	165	180
Pressure holes location (degrees)	195	210	225	240	260	285	315	347.5
	30	60	90	120	142.5	157.5	172.5	187.5
	202.5	217.5	232.5	247.5	270	300	330

Fig. 5 Temperature variation in the mid-plane of the bush surface for different groove locations

increasing supply pressures. At a moderate load (8 kN) the variation of shaft temperature did not exceed 2°C. For low loads shaft eccentricity is small, being low the flow of oil supplied to the bearing, thus resulting in a lesser cooling effect. Also the construc-

Fig. 6 Temperature variation in the mid-plane of the bush surface as a function of oil supply pressure and supply temperature

Fig. 7 Hydrodynamic pressure variation in the mid-plane of the bush surface for different groove locations

Fig. 8 Hydrodynamic pressure variation in the mid-plane of the bush surface as a function of supply pressure and groove location

tive features associated to the shaft assembly may have affected the availability to remove heat in the axial direction, contributing to a higher shaft operating temperature.

The effect of supply pressure on the measured flow rate is shown in Fig. 9(c). For all values of applied load and all groove locations, oil flow rate increased significantly with increasing supply pressure. The minimum values of flow rate were observed for the groove location corresponding to $\phi = -30$ degrees. This was due to the fact that the clearance at inlet is the smallest of all three groove locations. These low values of flow rate can justify the lowest temperatures observed in Fig. 9(a) and (b). For a given

Fig. 9 Bearing performance characteristics as a function of supply pressure, applied load and groove location ($N = 3000$ rpm)

Fig. 10 Bearing performance characteristics as a function of shaft speed, oil supply pressure and groove location ($W = 8$ kN)

supply pressure, at a moderate load (8 kN), the highest variation of the measured flow rate occurred.

The influence of supply pressure on bush torque is depicted in Fig. 9(d). Torque variations are only significant for a groove location of $\phi = -30$ degrees, occurring an increase in bush torque for increasing oil supply pressure. This grooving arrangement is associated with the lowest temperature (therefore the highest viscosity) in the bearing (Figs. 9(a) and (b)), originating high viscous dissipation. Also, at low supply pressure, the lowest torque values were observed. This may be due to the low flow rate present under such conditions, which did not fill completely the clearance space for a considerable angular extent, as shown in Fig. 8. At low supply pressures the lowest oil flow rate occurred for all groove locations (Fig. 9(c)). Therefore, the cooling effect of the supply oil was reduced thus originating high bearing temperatures (Fig. 9(a), (b)) and low bush torque (Fig. 9(d)).

The effect of shaft speed on the bearing performance characteristics, for a fixed applied load of 8 kN, was investigated. The experimental results of maximum bush temperature, oil flow rate, bush torque and maximum hydrodynamic pressure are shown in Fig. 10. Two groove locations ($\phi = 0$ and $\phi = +30$ degrees) and oil supply pressure equal to 70 kPa and 300 kPa were considered.

Figure 10(a) shows that increasing shaft speed resulted in increasing maximum bush temperature. The effect of groove location and supply pressure is similar to the one discussed earlier in relation to Fig. 9(a) for shaft speed of 3000 rpm. The effect of shaft speed on flow rate is shown in Fig. 10(b). Flow rate increased markedly as shaft speed increased due to increasing oil entrainment by the shaft. Also the increase in oil supply pressure for a given location ($\phi = 0$ degrees) resulted in a considerable increase in oil flow rate. The effect of groove location on flow rate was not significantly affected by shaft speed.

The effect of shaft speed on the measured bush torque is depicted in Fig. 10(c). Bush torque increased with increasing shaft speed, as a consequence of an increase of the rate of shear in the

oil. At the groove location of $\phi = +30$ degrees the measured bush torque was always lower than for $\phi = 0$ degrees, as already observed in Fig. 9(d). For $\phi = 0$ degrees, the influence of supply pressure on bush torque was not noticeable regardless shaft speed, because for this groove location the effect of oil supply pressure on the full film extent is not significant as for the case of $\phi = -30$ degrees (Fig. 9(d)). Figure 10(d) shows the influence of shaft speed on maximum hydrodynamic pressure. For all oil supply conditions tested an increase in shaft speed originated a decrease in maximum hydrodynamic pressure, as a result of the increasing carrying capacity associated to increasing speeds. For a supply pressure of 70 kPa the groove location of $\phi = +30$ degrees resulted in lower values of maximum pressures (for the same applied load) than for $\phi = 0$ degrees, except at low shaft speed. For the crown bearing configuration ($\phi = 0$ degrees), an increase in oil supply pressure from 70 kPa to 300 kPa originated a significant increase in maximum pressure for all values of shaft speed. This may be explained by the effect of the hydrostatic pressure over the inlet region, which generates a pressure force component added to the applied load.

As already mentioned (Table 2), the bush was made in bronze, 50 mm thick. Typical bearing bushes are usually much thinner and constituted of two distinct materials, a layer of a soft bearing alloy on a steel backing. A thinner bush wall should increase heat conduction, resulting in lower bearing temperatures. Regarding the influence of the bush material, Swanson and Kirk (1995) compared temperature and pressure profiles for a steel and a bronze bearing otherwise identical. Under the same operating conditions the bearings showed nearly identical temperatures. The pressure profiles, however, were different this being attributed to thermoelastic deformation effects.

4 Conclusions

Experimental measurements of bush temperatures and pressure distribution, shaft temperature, flow rate, and bush torque have been carried out on a hydrodynamic journal bearing operating under steady state conditions. Experimental results have been presented concerning the influence of groove location and supply pressure on the bearing performance characteristics. For the range of operating conditions tested, the following conclusions can be drawn:

- The role of the supply groove on the performance of the bearing cannot be ignored. Comparatively to the crown bearing configuration, the location of the groove at $\phi = -30$ degrees (in the divergent film zone) has promoted a reduction in maximum temperature and a large increase in maximum hydrodynamic pressure. Shaft temperature has been reduced slightly, bush friction torque has increased significantly and the reduction in oil flow rate has been also significant. The location of the groove at $\phi = +30$ degrees has originated a reduction in maximum temperature, a moderate increase in oil flow rate, a significant reduction in bush friction torque and a slight increase in maximum hydrodynamic pressure.
- Increasing oil supply pressure has increased markedly oil flow rate with a decrease in bearing operating temperatures, which has been more significant for low loads. Increasing oil supply pressure has also resulted in an increase in maximum hydrodynamic pressure in the bearing (for a fixed applied load), being the bush torque more significantly affected for the groove location of $\phi = -30$ degrees.
- The influence of shaft speed has also been investigated. Increasing shaft speed has resulted in increasing bush temperature, flow rate and bush torque, for low and high supply pressures, regardless the groove location. On the contrary, increasing shaft speed has caused a significant decrease in maximum hydrodynamic pressure.

Acknowledgments

The first author was supported by the PRAXIS XXI Programme implemented by "Fundação para a Ciência e Tecnologia" of Portugal. Part of this experimental work was carried out with the support of the "Mission de Cooperation" and C.I.E.S., both of France. All institutions are gratefully acknowledged. Thanks are also extended to Mr. J. C. Bligoud and Mrs. I. Pierre for their technical support during tests.

References

- Dowson, D., Hudson, J., Hunter, B., and March, C., 1966-67, "An Experimental Investigation of the Thermal Equilibrium of Steadily Loaded Journal Bearings," *Proc. Instn. Mech. Engrs.*, Vol. 101, Part 3B, pp. 70-80.
- Ferron, J., 1982, "Contribution à l'étude des phénomènes thermiques dans les paliers hydrodynamiques," Thèse de Doctorat de 3^{ème} Cycle, Université de Poitiers.
- Ferron, J., Frêne J., and Boncompain, R., 1983, "A Study of the Thermohydrodynamic Performance of a Plain Journal Bearing. Comparison Between Theory and Experiments," *ASME JOURNAL OF LUBRICATION TECHNOLOGY*, Vol. 105, pp. 422-428.
- Gethin, D. T., and El-Deihi, M. K. I., 1990, "Thermal Behaviour of a Twin Axial Groove Bearing Under Varying Loading Direction," *Proc. Instn. Mech. Engrs.*, Vol. 204, Part C, pp. 77-90.
- Gethin, D. T., and El-Deihi, M. K. I., 1992, "A Thermohydrodynamic Analysis of Twin Axial Groove Bearing Under Different Loading Directions and Comparison With Experiment," *ASME JOURNAL OF TRIBOLOGY*, Vol. 114, pp. 304-310.
- Kucinski, B., and Fillon, M., 1999, "An Experimental Study of Transient Thermal Effects in a Plain Journal Bearing," *ASME JOURNAL OF TRIBOLOGY*, Vol. 121, pp. 327-332.
- Ma, M. T., and Taylor, C. M., 1995, "Effects of Oil Temperature on the Performance of an Elliptical Bearing," *Proc. of the 21st Leeds-Lyon Symposium on Tribology*, 1994, Elsevier, Tribology Series no. 30, pp. 143-151.
- Majumdar, C. B., and Saha, A. K., 1974, "Temperature Distribution in Oil Journal Bearings," *Wear*, Vol. 28, pp. 259-266.
- Mitsui, J., Hori, Y., and Tanaka, M., 1983, "Thermohydrodynamic Analysis of Cooling Effect of Supply Oil in Circular Journal Bearings," *ASME JOURNAL OF TRIBOLOGY*, Vol. 105, pp. 414-421.
- Mitsui, J., Hori, Y., and Tanaka, M., 1986, "An Experimental Investigation on the Temperature Distribution in Circular Journal Bearings," *ASME JOURNAL OF TRIBOLOGY*, Vol. 108, pp. 621-627.
- Monmousseau, P., Fillon, M., and Frêne, J., 1997, "Transient Thermoelastohydrodynamic Study of Tilting-Pad Journal Bearings-Comparison Between Experimental Data and Theoretical Results," *ASME JOURNAL OF TRIBOLOGY*, Vol. 119, pp. 401-407.
- Pinkus, O., 1990, *Thermal Aspects of Fluid Film Tribology*, ASME Press, New York.
- Swanson, E. E., and Kirk, R. G., 1995, "Experimental Temperature and Pressure Profiles for Two Steadily Loaded Journal Bearing," *STLE Tribology Transactions*, Vol. 38, pp. 601-606.
- Swanson, E. E., and Kirk, R. G., 1997, "Survey of Experimental Data for Fixed Geometry Hydrodynamic Journal Bearings," *ASME JOURNAL OF TRIBOLOGY*, Vol. 119, pp. 704-710.
- Syverud, T., and Tanaka, M., 1997, "Experimental Investigation of the Effect of Shaft Heating and Cooling on Single Bore Journal Bearing," *Wear*, Vol. 207, pp. 112-117.