OPERATING TEMPERATURES AND POWER LOSS OF THE LEADING-EDGE-GROOVE TILTING PAD JOURNAL BEARING

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ABSTRACT

Because of it's superior anti-vibration characteristics, the tilting pad journal bearing is a popular choice for supporting the shafts of rotating machinery. However, recent increases in shaft speed are causing concern about the high levels of bearing temperatures and frictional losses. Accordingly, a new design of bearing has been developed in which cool oil is supplied directly to the leading edge of each pad.

This paper presents results from the testing of a 0.098 m diameter, five pad bearing operating with a load of 11.1 kN and speeds up to 275 Hz. These experimental data are compared with results obtained from a three-dimensional thermohydrodynamic model of the tilting pad journal bearing. This model includes oil mixing at the leading edge of each pad. Good correlation between the experimental and calculated results has confirmed the hypothesis that hot-oil-carry-over may be significantly reduced by supplying cool oil directly to the leading-edge-groove. Also, a comparison of results from the conventional and LEG bearings shows that the latter has substantially lower operating temperatures. Furthermore, calculations indicate that the power loss of the LEG bearing may be reduced significantly.

INTRODUCTION

The tilting pad bearing, in either the thrust or journal configuration, is a popular choice of bearing for supporting the shafts of rotating machinery. However, machine operating speeds have now increased to the point where power loss and bearing metal temperature may be unacceptably high. A number of studies have, therefore, been aimed at improving bearing steady state performance [1,2]. Basically, apart from reducing the churning losses of the tilting pad thrust bearings, effort was focused on directing "fresh" oil to the leading edge of the pads. Bielec and Leopard [3] demonstrated the beneficial effect of supplying oil to nozzles located between the pads of a thrust bearing and a similar method was used for a tilting pad journal bearing [4]. The authors of both papers claimed that by supplying oil under pressure to the oil film gap, the effect of hot-oil-carry-over was reduced.

The leading-edge-groove journal bearing offers a different approach. In this design, each pad is extended to accommodate an axial oil distribution groove, from which cool lubricant is introduced directly into the oil film. It has been shown that there are reductions in temperature and friction losses of both thrust [2, 5] and journal [6, 7] bearings when this method of oil supply is used.

Harangozo et al. [8] concluded that these oil supply methods had a greater effect on the performance of the thrust bearing and were less important in the case of the journal bearing. However, this conclusion was based on test results where the oil flow was fixed, regardless of bearing operating conditions. Recently, Fillon et al. [9] concluded that directed lubrication is effective at higher operating speeds (above 30 m/s).

In most cases, results from investigations into oil supply methods for tilting pad bearings have not been supported by theoretical considerations of the flow and heat exchange in the space between two adjacent pads. Some authors [4, 9] attributed the benefits of directed lubrication to a reduction in hot-oil-carry-over, but in their computer models the mixing coefficients were chosen without consideration of the bearing operating conditions.

Previous attempts to both explain and model the effect of the leading-edge-groove have been presented by the authors [6, 7]. In this paper, additional test results and theoretical data are presented to support the finding that the decrease in the operating temperature of the LEG bearing is through hot-oil-carry-over reduction. Furthermore, it will be shown that this effect is dependent on the oil flow rate. The results from the LEG bearing are compared with data from a conventional tilting pad journal bearing, and also with theoretical data from quasi three-dimensional thermohydrodynamic computer models of the bearings. In these models, oil film turbulence and pad thermal and elastic distortions are accounted for. Full details are given in reference [6].

EXPERIMENTAL APPARATUS

The test rig is the classical arrangement of a free test bearing and a fixed shaft. The shaft, which has a 0.098 m diameter journal, is supported on two pairs of high precision, preloaded, angular contact ball bearings. A 37 kW variable speed electric motor, driving through a belt and pulley system, provides shaft speeds that range between 30 and 275 Hz.

Eddy current probes mounted on each end of the bearing housing measure the horizontal and vertical displacements of the housing with respect to the shaft. Tensioned wires attached to the sides of the bearing housing minimize non-parallel movement of the bearing housing that may arise from misalignment of the bearing loading system. These wires are flexible in a direction perpendicular to the bearing axis so that they do not influence the movement of the bearing housing. Full details are provided in reference [6].

The five pad test bearing had a length/diameter ratio of 0.387, a bearing radial clearance of 0.076 mm and a pad radial clearance of 0.102 mm. The resulting pad preload factor was 0.25. Thermocouples were imbedded in the babbitt lining to within 0.5 mm of the bearing surface.

Figure 1 shows the pad arrangement of the conventional and LEG bearings. The effective pad angle was 56.1°, although the overall angle of the LEG pad was larger because of the inclusion of the leading edge groove. The dimensions of this groove were as follows: circumferential width=4.8 mm; axial length=33.5 mm; depth=4.6 mm. The lubricant was a light turbine oil (ISO VG32), supplied at a temperature that was controlled between 48°C and 50°C. Bearing test conditions were as follows:

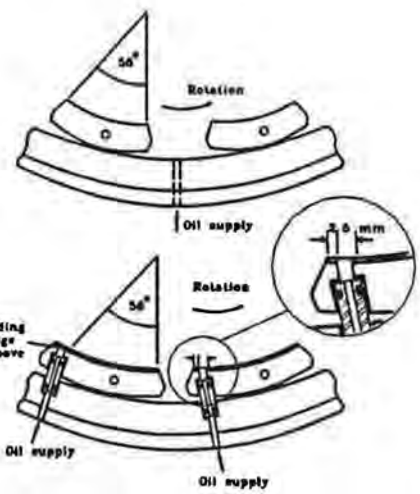

Figure 1 **Pad arrangements:**

a) conventional **b) LEG**

- bearing load: 11.1 kN
- load direction: load between pads (LBP)
- nominal oil flow rate: dependent on speed: 0.21×10^{-4} m^3/s @ 30 Hz; 1.85×10^{-4} m^3/s @ 150 Hz; 4.27×10^{-4} m^3/s @ 275 Hz.

In addition, the bearings were tested at 50%, 150%, and 250% of the nominal flow rates.

RESULTS AND DISCUSSION

To illustrate the effect of the oil supply method on the temperature of the bearing, the measured circumferential pad temperature profiles of the conventional and the LEG bearings, at 275 Hz shaft speed with a load of 11.1 kN, are shown in Figure 2. It is of interest to note that the maximum temperature of the LEG was approximately 20°C lower than the conventional bearing, although leading edge temperatures were similar. This is consistent with the observations of Ettles and Cameron [10] and Heshmat and Pinkus [11], who concluded that two separate layers of oil enter the oil film gap. The first is a layer of hot oil, carried over from the preceding pad, that adheres to the surface of the shaft. The second cooler layer of oil enters the oil film directly adjacent to the pad surface.

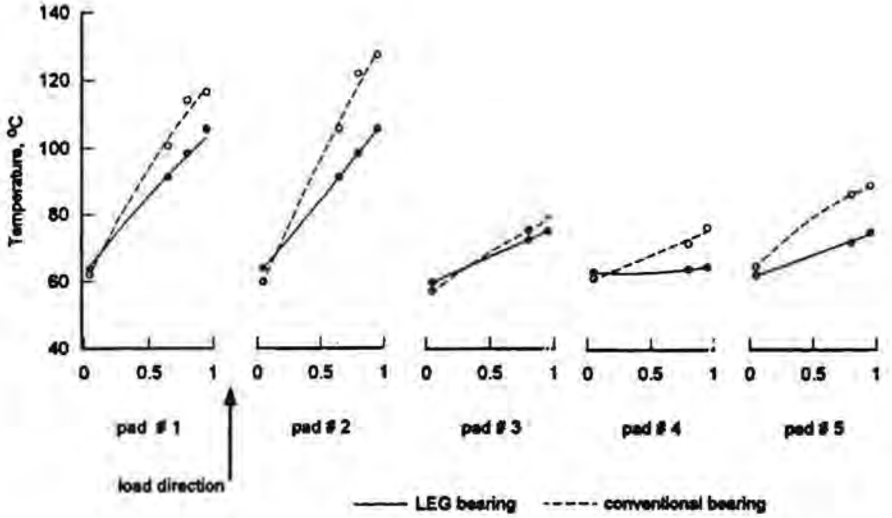

Figure 2 Pad temperature profiles, load 11.1 kN, shaft speed 275 Hz.

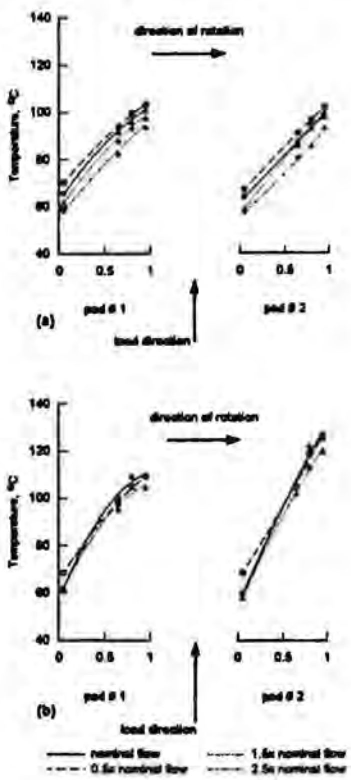

Figure 3 Loaded pads temperature profiles, load 11.1 kN, shaft speed 200 Hz.
a) LEG bearing b) conventional bearing

The loaded pad temperature profiles of the LEG and conventional bearings, for different flow rates, are shown in Figures 3a and 3b respectively. Analyses have shown that, in the case of the conventional bearing, most hot oil leaving the trailing edge of one pad enters the leading edge of the next adjacent pad. The balance comes from the cool "supply" oil. Consequently, even with a reduced flow rate of 50% of nominal, there is still adequate oil within the cavity of the bearing for "full film" conditions to be maintained at the pads, and bearing temperatures do not change substantially. The situation is somewhat different in the case of the LEG bearing. Because all cool oil supplied to the leading edge groove enters the oil film gap, any reduction in this flow will naturally result in a need for more hot "carried-over" oil to maintain the balance. Consequently, bearing temperatures rise as shown in Figure 3a, but do not exceed those of the conventional bearing.

The benefit, in terms of maximum pad

temperature, of using the LEG bearing becomes increasingly evident for shaft speeds above 150 Hz, as shown in Figure 4. At lower speeds, the rise in temperature of the oil over the pad length is not sufficient to affect the oil temperature at the leading edge of the next pad. Consequently, even though the LEG bearing runs with a reduced amount of hot oil at the pad leading edge (see Figure 5), the pad operating temperatures of both bearings are similar. At higher speeds, the oil leaving the trailing edge of the pad is hotter and is reflected in the operating temperature of the next pad. In the case of the conventional bearing this effect is stronger.

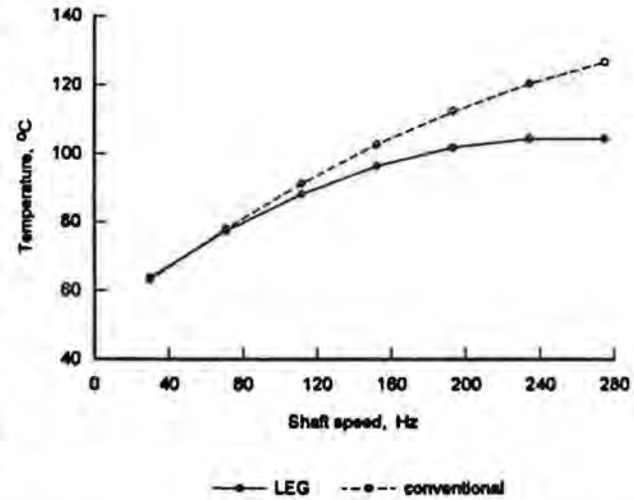

Figure 4 Maximum pad temperature, load 11.1 kN.

Advantage can be taken of the lower operating temperature of the LEG bearing to reduce it's power loss. This may be done in one of two ways (or possibly in combination) - by increasing the oil inlet temperature or by using an oil with a lower viscosity grade. In this paper, where the former option is considered, it will be shown that there can be substantial reductions in power loss without compromising bearing temperature. As an example, the computer models have been used to calculate the power losses of both the conventional and LEG bearings; the former with an inlet temperature of 49°C and the latter with an inlet temperature of 74°C. This raised inlet temperature results in a maximum bearing temperature that is similar to that of the conventional bearing with an oil supply temperature of 49oC . The results of this exercise are shown in Figure 6 and indicate that the friction torque of the LEG bearing can be reduced by as much as 40%. Table I lists the pad losses of both bearing designs and indicates that there are savings at all pads. The same table also shows that, by increasing the oil supply temperature to 60oC, the power loss of the LEG bearing is still reduced by over 20%. Furthermore, maximum pad temperature is still 13oC lower than the conventional bearing.

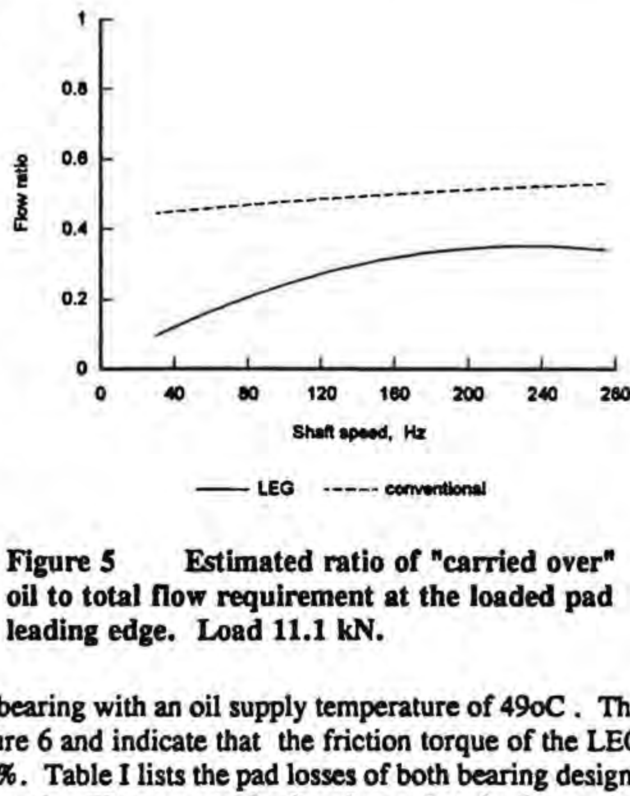

Figure 5 Estimated ratio of "carried over" oil to total flow requirement at the loaded pad leading edge. Load 11.1 kN.

Table I **Calculated power losses and maximum pad temperatures. Load=11.1 kN, shaft speed=275 Hz.**

Bearing	Power losses, kW						Max.pad temp., °C
	pad 1	pad 2	pad 3	pad 4	pad 5	total	
conventional	3.48	4.48	1.88	1.55	1.85	13.24	124
LEG	3.36	4.03	2.39	1.72	1.87	13.37	107
LEG (T_i=60°C)	2.90	3.33	1.61	1.16	1.39	10.39	111
LEG (T_i=74°C)	2.36	2.53	1.05	1.19	0.97	8.1	121

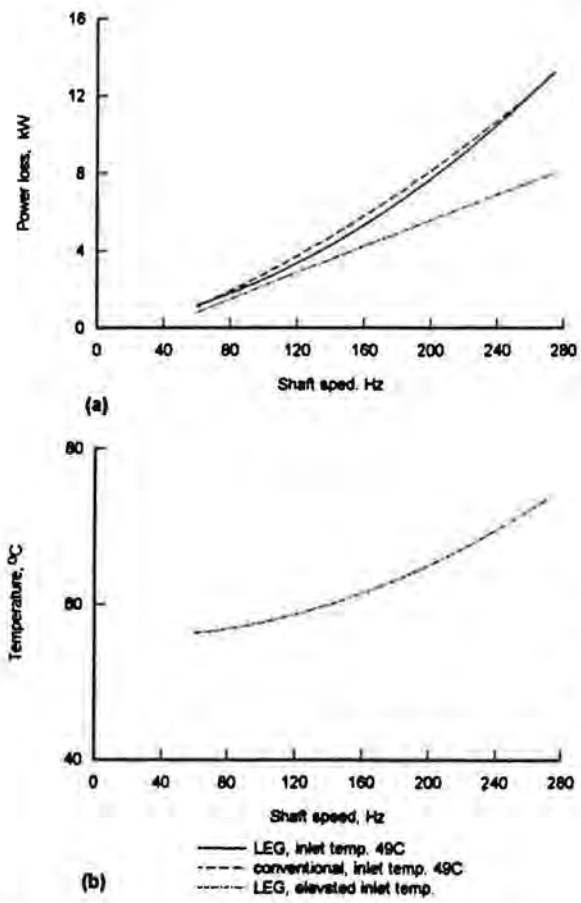

Figure 6 **Power loss analysis. Load 11.1 kN , a) calculated power losses, b) elevated oil supply temperatures - LEG bearing**

CONCLUDING REMARKS

1. By reducing the effect of hot-oil-carry-over, the LEG bearing operates with substantially lower temperatures to those of the conventional bearing.
2. The LEG bearing offers the potential to reduce significantly the frictional losses as a result of increasing the oil inlet temperature or lowering the viscosity of the oil.
3. Unlike the conventional bearing, the amount of oil supplied to the LEG bearing can have a profound effect on oil film temperatures.

ACKNOWLEDGEMENTS

The authors would like to thank the National Research Council Canada and Kingsbury, Inc. for permission to publish this paper.

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