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Further Test Results of the Leading-Edge-Groove (LEG) Tilting Pad Thrust Bearing

This paper compares the LEG and pressurized controlled flow lubricant supply methods and evaluates their influence on the babbitt temperature and bearing power loss performance of a tilting pad, equalizing thrust bearing. The paper also presents new experimental temperature data from bidirectional testing of a unidirectional LEG bearing. The experimental data presented is from a 267 mm (10.5 in.) O.D. bearing, operating at shaft speeds up to 13000 rpm with applied loads that produced mean unit pressures of up to 3.45 MPa (500 psi). Conclusions are drawn based upon these test data.

Introduction

The leading-edge-groove (LEG) tilting pad thrust bearing is a low frictional loss hydrodynamic thrust bearing that utilizes a managed oil flow lubrication concept. The bearing is so named because the leading edge of each pad or shoe is extended to accommodate an oil distribution groove. Cool, undiluted lubricant is introduced from this groove directly into the fluid film of each shoe. This method of supplying oil into the hydrodynamic wedge has been found to significantly reduce bearing frictional power losses and babbitt temperatures [1, 2].

In this third paper of leading edge groove thrust bearing test results, the oil supply method was isolated and evaluated to determine its influence on bearing performance. The two previous papers [1, 2] compared offset pivot (60 percent) LEG and central pivot (50 percent) conventional thrust bearings. Elwell and Leopard in reference [1], and Martin and Gardner in reference [2] questioned whether the LEG temperature advantage was a result of the lubrication supply method or pivot location. This paper presents test data that addresses that question. The two primary indicators of bearing performance—frictional power loss and babbitt temperature—are used to contrast leading edge groove and pressurized supply (controlled flow) bearing results. Each bearing was tested under identical conditions of applied load, oil supply flow rate, shaft speed, oil supply temperature, pivot offset, and oil viscosity. Details of the bearing test rig can be found in reference [3].

Bidirectional operation test data for the LEG bearing is also presented. Babbitt temperature comparisons are made to contrast proper and reverse shaft rotation. Each bearing shaft rotation direction was tested under identical conditions of applied load, shaft speed, oil supply temperature and oil viscosity. Temperature differences can be attributed to shaft rotation direction and, therefore, pivot and oil supply groove location.

The bearings were evaluated using a light turbine (ISO VG 32) oil with a viscosity of 0.027 Pa•s @ 37.8°C and 0.006 Pa•s @ 98.9°C (150 SSU @ 100°F and 43 SSU @ 210°F) supplied at 46°C (115°F), for applied loads that produced mean unit pressures ranging from 0–3.45 MPa (0–500 PSI) and shaft speeds ranging from 4000–13000 rpm. The oil supplied to each bearing was controlled by a throttling valve and measured with a turbine flowmeter.

The performance data presented isolates and identifies the individual contributions of shoe pivot location and oil supply method. These issues were raised in the discussion of the two previous papers [1, 2], and based on this test data, should now be resolved.

Test Bearing Descriptions

The test configuration was a 267 mm (10.5 in.) O.D. equalizing tilting pad double element thrust bearing. Double thrust bearings are, as their name suggests, two single bearing elements such as shown in Fig. 1, one of which carries thrust load (loaded bearing), while the other (slack side bearing) positions the shaft and carries any transient reverse thrust loads. Details of this type of bearing arrangement can be found in reference [3].

The test bearings consisted of eight steel-backed and babbitt-faced pads or shoes on each side of a rotating collar for an (8 × 8) double thrust bearing configuration. Each shoe had a babbitt O.D. of 267 mm (10.5 in.) and a bore of 133 mm (5.25 in.) and, except for the leading-edge-groove (LEG) bearing, had a total bearing area of 356 cm^2 (55.1 in.^{2}) and 39 degrees of arc. The larger LEG shoes subtend 43-1/2 degrees of arc for a total effective bearing area of 349 cm^2 (54.1 in.^{2}). The LEG shoe is larger to accommodate the groove at the leading edge. Common to each bearing configuration was a radial pivot that was located 60% of the arc length from the leading edge of the shoe.

All tests were conducted with an oil control ring to control

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the discharge of oil from the bearing (Fig. 2). The oil control ring shrouded the bearing collar and was bored with a 3.97 mm (5/32 in.) radial clearance over the collar diameter and was fitted with a 25.4 mm (1.0 in.) tangential discharge port. The design of the oil discharge port is critical to the performance of any low frictional loss bearing. The discharge must be designed so that only a very slight positive bearing cavity pressure (0.007 to 0.014 MPa/1 to 2 psi) develops during operation.

Detailed descriptions of both the conventional pressurized and LEG lubricant supply methods can be found in references [1, 2, and 3].

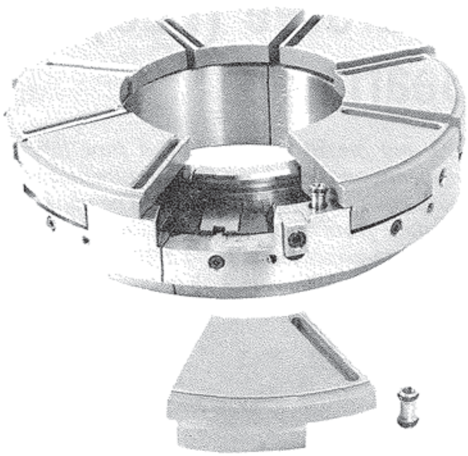

Fig. 1 Single element LEG thrust bearing

Bearing Operating Temperatures

Bearing operating temperatures were monitored by thermocouples puddled in the babbitt itself, approximately 0.8 mm (1/32 in.) below the actual babbitt surface. The location of the thermocouples across the babbitt face of both the LEG and conventional shoes is shown in Fig. 3. Thermocouples were also used to measure the oil supply and drain temperatures.

Bearing operating babbitt temperatures provide a convenient means of assessing thrust bearing performance [4]. Where possible, temperature data presented in this paper will be from the thermocouples located at the 75/75 percent position (Fig. 3). The 75/75 percent position was selected because it is at this location that the oil-film pressure-temperature combination provides the most accurate picture of bearing operating risk [4, 5]. Comparisons of operating babbitt temperatures are made between the maximum measured 75/75 percent temperature of each bearing (the hottest of the eight 75/75 percent temperatures for each bearing).

The influence of lubricant supply method on operating babbitt temperature at the 75/75 percent location is illustrated in Figs. 4 and 5. The corresponding oil flow rate for each of these tests can be found in Figs. 6 and 7.

Figure 4 compares the hottest 75/75 percent location temperatures of both the LEG and conventional pressurized supply bearing designs when operating at an applied load of 2.07 MPa (300 psi). Each bearing had a 60 percent pivot offset and was supplied with identical oil flow rates (Fig. 6). The LEG design consistently operated with maximum 75/75 percent temperatures that ranged from 2.8 to 17.8°C (5 to 32°F) lower than those of the conventional pressurized supply design. The lower temperatures are attributed to the introduc-

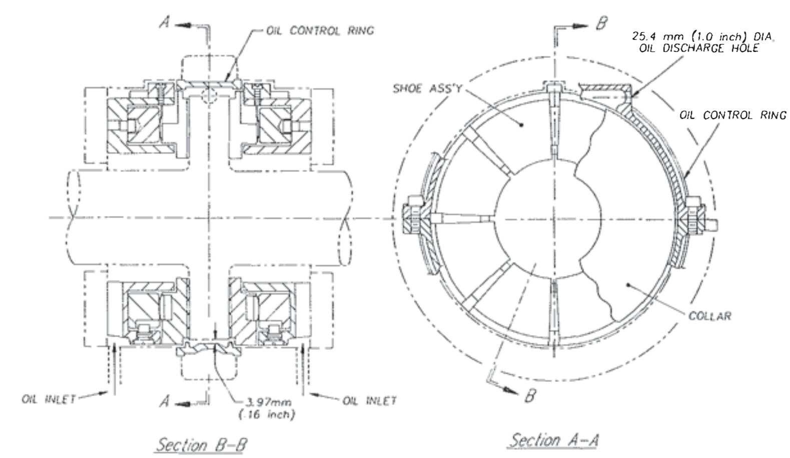

Fig. 2 Oil control ring configuration

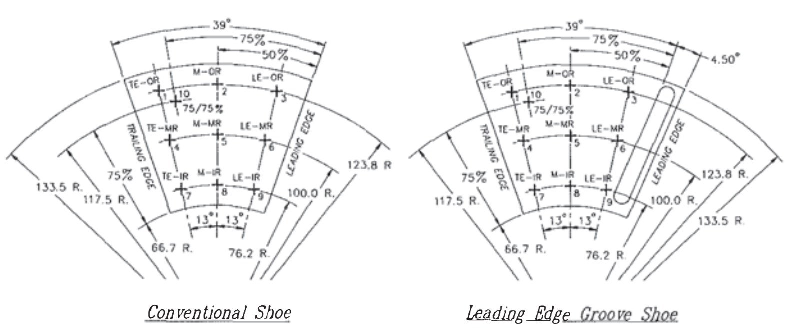

Fig. 3 Thrust shoe thermocouple locations

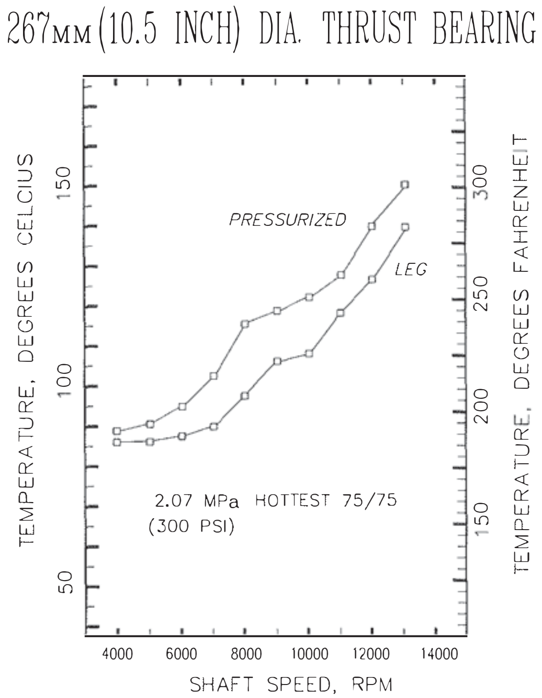

Fig. 4 A comparison of the hottest measured 75/75 percent babbitt temperature locations for conventional pressurized and LEG thrust bearings when loaded to 2.07 MPa at shaft speeds of 4000 to 13,000 rpm

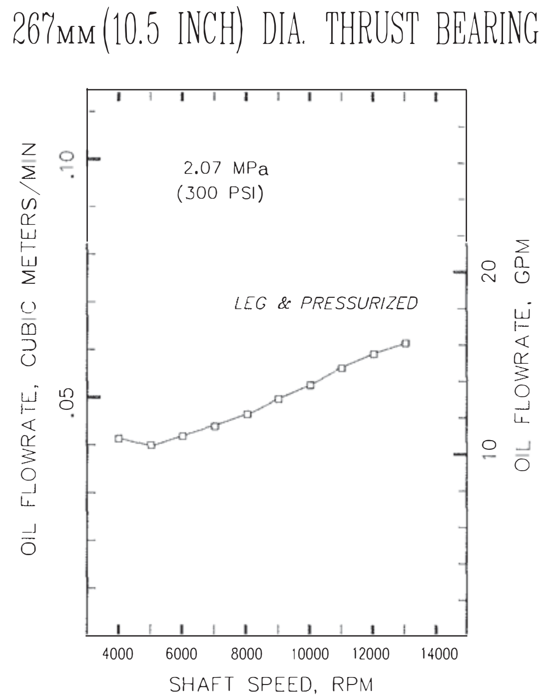

Fig. 6 Oil flowrate supplied to conventional pressurized and LEG thrust bearings when loaded to 2.07 MPa at shaft speeds of 4000 to 13,000 rpm

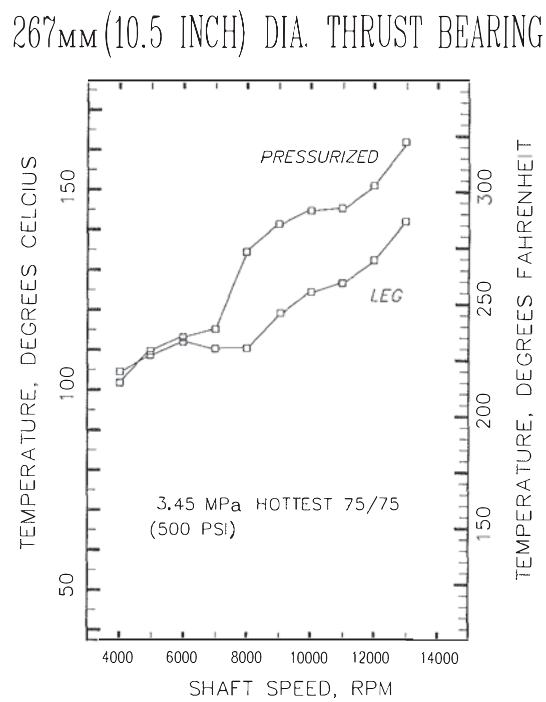

Fig. 5 A comparison of the hottest measured 75/75 percent babbitt temperature locations for conventional pressurized and LEG thrust bearings when loaded to 3.45 MPa at shaft speeds of 4000 to 13,000 rpm

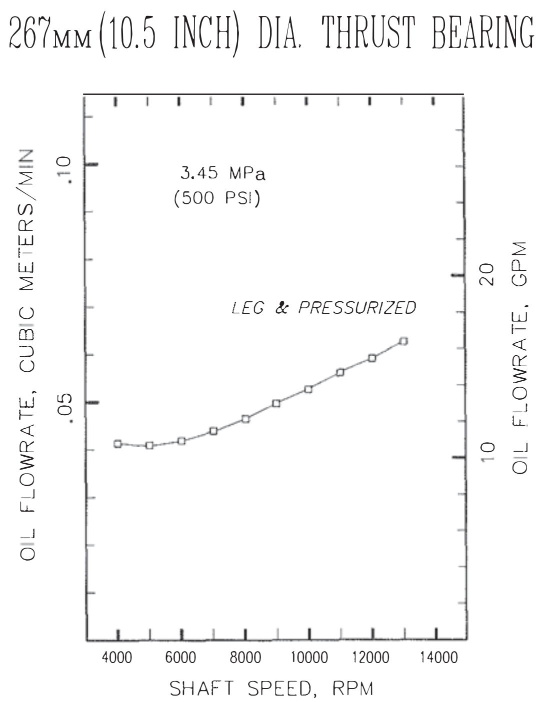

Fig. 7 Oil flowrate supplied to conventional pressurized and LEG thrust bearings when loaded to 3.45 MPa at shaft speeds of 4000 to 13,000 rpm

tion of cool, undiluted supply oil directly into the oil film wedge from the leading edge groove. This cool oil seems to insulate the shoe surface from the hot oil carryover adhering to the rotating collar.

Figure 5 is similar to Fig. 4 except this time the comparison is made for an applied load of 3.45 MPa (500 psi). The increased loading on the bearings has caused the temperature excursions to become larger (a maximum of 23°C/42°F @ 8K rpm) for all shaft speeds above 7000 rpm and smaller for shaft speeds of 7000 rpm and lower. In fact, the conventional design actually runs 2.2°C (4°F) cooler at a shaft speed of 4000 rpm. Once again, the oil flow rates were identical for each bearing (Fig. 7).

Reverse Rotation

The design of a tilting pad thrust bearing can be configured

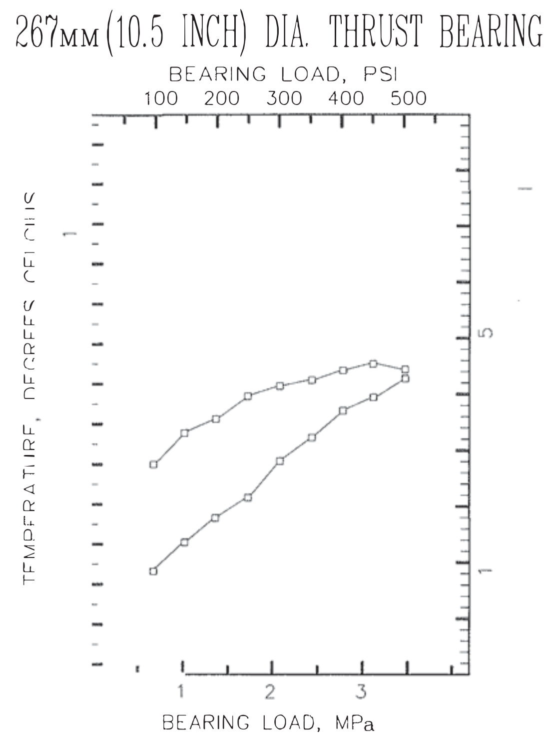

Fig. 8 A comparison of the hottest measured TE-MR babbitt temperature locations for an LEG bearing with proper and reverse shaft rotation at 4000 rpm and loaded from .69 to 3.45 MPa

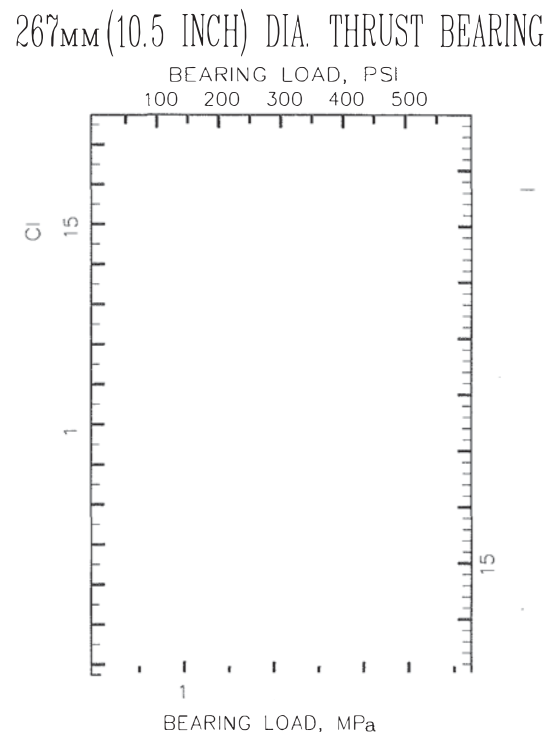

Fig. 10 A comparison of the hottest measured TE-MR babbitt temperature locations for an LEG bearing with proper and reverse shaft rotation at 10000 rpm and loaded from .69 to 3.45 MPa

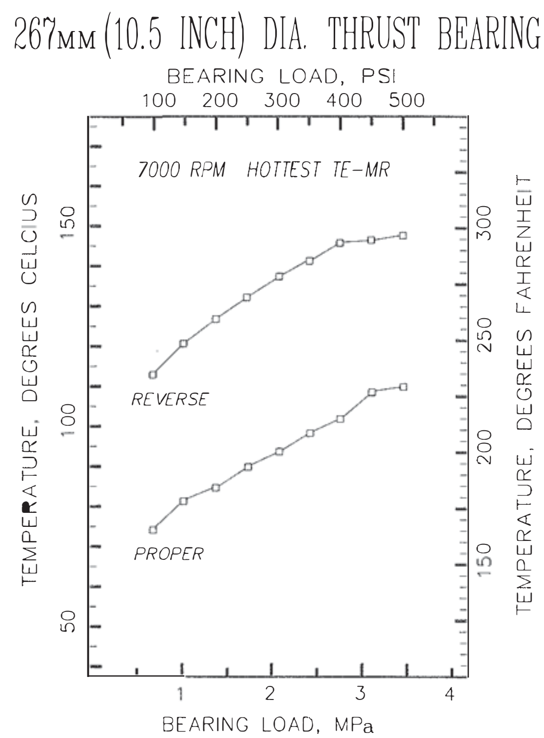

Fig. 9 A comparison of the hottest measured TE-MR babbitt temperature locations for an LEG bearing with proper and reverse shaft rotation at 7000 rpm and loaded from .69 to 3.45 MPa

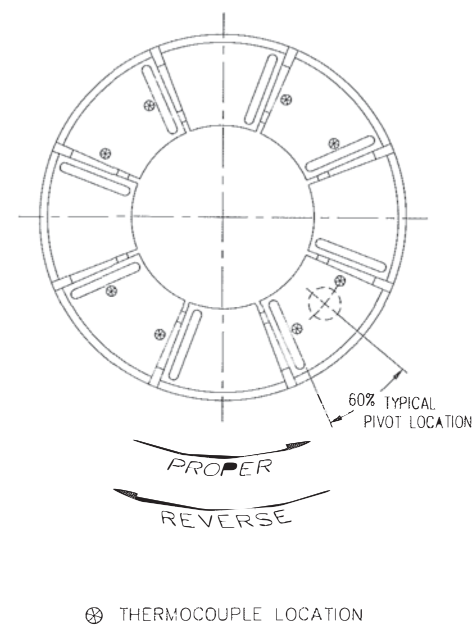

Fig. 11 Thermocouple location in the reverse rotation LEG thrust bearing

in such a way as to preclude bidirectional operation. Design features intended to improve performance sometimes do so at the expense of operational flexibility. Any design that would presuppose the direction of shaft rotation places that bearing in the unidirectional category. Unfortunately, this has prevented the use of certain bearing designs in applications which have the potential for bidirectional operation.

The leading-edge-groove (LEG) tilting pad thrust bearing is considered unidirectional because of its offset pivot and oil distribution groove. Figures 8, 9, and 10 present the results of testing conducted on the LEG bearing when the shaft is rotating in the reverse direction. Bidirectional LEG operation will be evaluated on the basis of operating shoe babbitt temperatures.

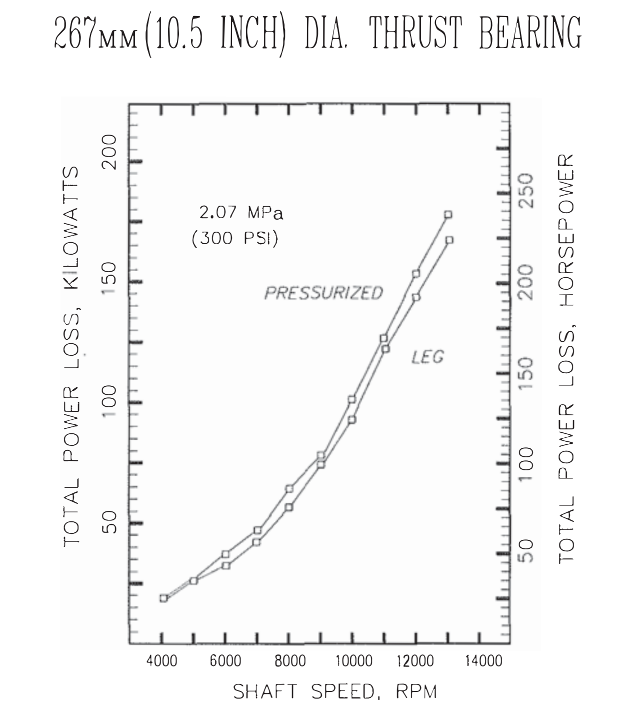

Fig. 12 A comparison of the measured power loss for conventional pressurized and LEG bearings supplied with identical oil flowrates when loaded to 2.07 MPa for shaft speeds of 4000 to 13000 rpm

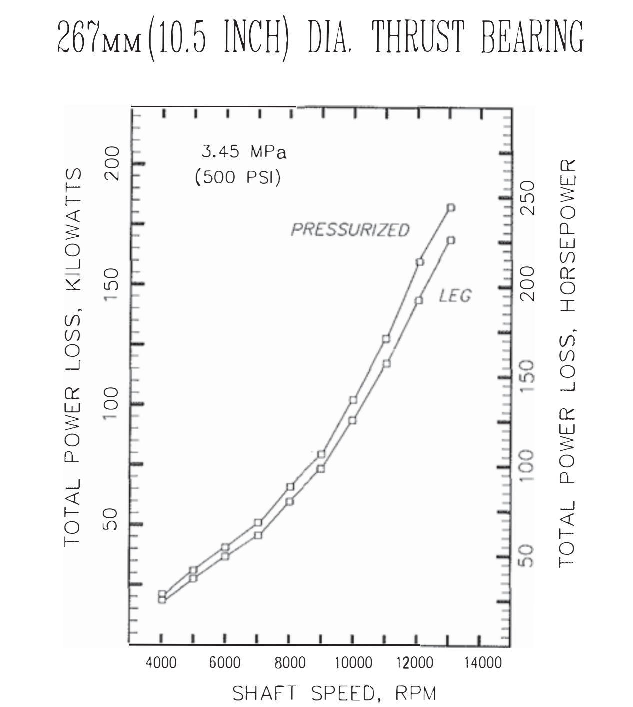

Fig. 13 A comparison of the measured power loss for conventional pressurized and LEG bearings supplied with identical oil flowrates when loaded to 3.45 MPa for shaft speeds of 4000 to 13000 rpm

The tests were conducted under identical operating conditions, and bearing operating temperatures were measured by thermocouples. Unfortunately, during these particular tests, limited temperature data were collected because not every shoe was instrumented. This resulted because to achieve the reverse shaft rotation, the normally heavily instrumented loaded and lightly instrumented unloaded bearings were transposed in the housings (Fig. 2 in reference [7]). This transposing of the bearings oriented the pivot 40 percent of the arc length from the leading edge of the shoe, and the oil supply groove at the trailing edge.

The eight thermocouple locations shown in Fig. 11 include four trailing edge-middle radius (TE-MR) and four leading edge-middle radius (LE-MR) locations. The hottest of the four TE-MR locations was selected to evaluate bearing risk. While it is true that this may not be the hottest TE-MR babbitt location on the bearing, the fact that the bearing is self-equalizing [6] minimizes any possible excursions.

The influence of shaft rotation direction on TE-MR babbitt temperatures for applied loads that ranged from 0–3.45 MPa (0–500 psi) and shaft speeds of 4000, 7000, and 10000 rpm can be found in Figs. 8 through 10.

Figure 8 reports the TE-MR babbitt temperatures for the various applied loads at a shaft speed of 4000 rpm. The response of the TE-MR babbitt temperature to changes in applied load is very different for each direction of shaft rotation. The "proper" rotation bearing (60 percent offset) responded to the constant rate of change in applied load with two separate rates of change in the TE-MR temperature. The initial temperature response occurred from 0.69 to 2.76 MPa (100 to 400 psi), and the less responsive second occurred above 2.76 MPa (400 psi). The "reverse" rotation bearing (40 percent offset) also demonstrated two temperature response trends. The first from 0.69 to 2.07 MPa (100 to 300 psi), and the second above 2.07 MPa (300 psi). Although the reverse rotation bearing initially runs 26.6°C (48°F) hotter, this difference decreased with load until at 3.45 MPa (500 psi), there is only a 1.6°C (3°F) difference.

The comparison of temperature performance at 7000 rpm is shown in Fig. 9. The increase in sliding velocity has significantly changed the response of the reverse rotation bearing at the higher shaft speed. There is no converging of the two temperature curves, but a consistent 39 to 44°C (70 to 80°F) spread in favor of the proper rotation bearing.

The temperature response at a shaft speed of 10,000 rpm (Fig. 10) is similar to that at 7000 rpm—a consistent 39 to 47°C (70 to 85°F) spread in favor of the proper rotation bearing. Unfortunately, due to high TE-MR babbitt temperatures (154°C/310°F), the test had to be concluded at 2.42 MPa (350 psi). No indication of bearing distress could be found during the visual inspection of the reverse rotation bearing at the conclusion of the testing.

The tests conducted were intended to establish if a unidirectional bearing, such as the LEG, could operate as a bidirectional bearing. Both the measured babbitt temperatures and visual inspections of the bearings demonstrated that the LEG design can operate with either direction of shaft rotation. Conventional hydrodynamic theory provides no explanation for the performance of the reverse rotation (40 percent offset) bearing. Theoretically, unless the pivot offset is greater than 50 percent, the collar can never lift off and allow the shoes to form a tapered wedge. Perhaps one possible explanation for the performance of the reverse rotation bearing might be the collar's ability to supply oil to the bearing. This scenario requires: oil adhesion, a rounded leading edge on the shoe, and shaft rotation to provide the required volume of oil to the shoe for collar lift-off. Thermal and elastic deformations provide the tapered wedge. Kettleborough [8], and Horner [9] have also shown that offset-pivoted pads do have a substantial reverse rotation load-carrying capacity.

Bearing Power Loss

Bearing frictional power losses were calculated by the familiar energy balance technique whereby the frictional loss is computed as a function of the measured oil temperature rise (supply to discharge), measured oil flow rate, and the specific heat of the oil. The analysis, however, does not include radiation losses from the housing or conduction losses via the shafting and foundation because they are small and consistent.

The influence of oil supply flow rate is a critical factor in determining bearing power loss. Oil supplied to the bearing which is not utilized in the formation of the oil film enacts a frictional power loss penalty attributable to the pumping and churning losses associated with the excess oil. Low frictional power loss fluid film bearings are possible because the managed oil flows permit a significant reduction in oil supply flow rates.

Figures 12 and 13 compare the frictional power losses of a conventional and LEG tilting pad fluid film thrust bearing for applied loads of 2.07 and 3.45 MPa (300 and 500 psi). The LEG bearing design, all things being equal, demonstrated frictional power losses that ranged from 0 to 13 percent lower than the conventional design bearing. The lower LEG power losses result because the oil is introduced directly into the fluid film of the shoe minimizing parasitic churning losses.

Conclusions

1. The LEG design, all things being equal (applied load, shaft speed, oil viscosity, supply temperature, flow rate and pivot offset) operates with lower babbitt temperatures and frictional power losses than a conventional design thrust bearing.

2. The design features (leading edge distribution groove and offset pivot) that categorize the LEG thrust bearing as unidirectional do not preclude occasional successful bidirectional operation.

Acknowledgment

The author wishes to express his gratitude to Kingsbury, Inc. for the opportunity to publish these results.

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DISCUSSION

K. Brockwell[1]

This is the latest of a series of informative papers from Mr. Mikula, concerning the operating characteristics of the tilting pad thrust bearing. In this age of the computer where the trend is toward a theoretical approach, this experimental work is especially welcome.

The type of bearing described by Mr. Mikula is one of a new breed of thrust bearings, developed by bearing vendors over the last two decades, principally for use in high speed machinery. They are designed to reduce bearing power requirements and to lower bearing operating temperatures.

The results presented in this paper demonstrate that useful reductions in bearing temperature can be obtained with the LEG bearing, when compared to the temperatures gathered from the pressurized bearing. Savings in power loss, however, do not appear to be so significant and I wonder if the author would care to comment on this. Could it be associated with the use of an oil control ring? In the discusser's experience, to obtain really large reductions in bearing churning losses, a completely open space around the thrust collar, with a large gravity drain at the bottom of the housing, is necessary.

I am pleased to see that the author has addressed the question of reverse rotation. Specifically, he has shown that the LEG design of thrust pad will perform satisfactorily, when running backwards under heavily loaded conditions. Horner et al.[2] successfully ran offset pivot pads backwards and compared temperatures with those obtained from centre pivot and forward running offset pivots. Of particular interest are the results which show only a small difference between the centre pivot pad and the backward running, offset pivot pad.

The question of reverse rotation of offset pivot pads is one that is raised frequently by both original equipment manufacturers and end users. Usually, it is the bearing vendor who must provide the right answers. Publication of test data as described in this paper will help to reduce concerns for classes of machine that require extended periods of reverse operation.

[1]National Research Council of Canada, Tribology and Mechanics Laboratory, Vancouver, B.C. V6S 2L2 Canada.

[2]Horner, D., Simmons, J. E. L., and S. D. Advani, "Measurements of Maximum Temperature in Tilting-Pad Thrust Bearings," Presented at the 41st ASLE Annual Meeting, Toronto, Ontario, Canada. May 12-15, 1986.

Author's Closure

The author would like to express his gratitude and appreciation to Mr. Brockwell for the interest and comments he expressed in this paper.

The lack of a significant power loss difference between the LEG and pressurized bearings was noted by Mr. Brockwell. The reason for the similar power loss results is the identical flow rates that were supplied to each bearing (Figs. 6 and 7). One of the most critical factors influencing bearing power loss is the oil supply flow rate. Oil supplied to the bearing, which is not utilized directly in the formation of the load supporting film, enacts a penalty in the form of increased power loss due to churning losses and pumping of the excess oil. If bearing power loss is to be minimized, then so must oil flow rates (all other things being equal). Unfortunately, the babbitt temperatures and safe load capacity of the bearing usually suffer as the flow rates are reduced. With this tradeoff in mind, the LEG was developed to effectively manage reduced oil flows in such a way as to eliminate this usual tradeoff.

The purpose of the oil control ring is to control the discharge of oil from the bearing so that only a very slight positive bearing cavity pressure (0.007 to 0.014 MPa/1 to 2 psi) develops during operation. This discharge arrangement permits the efficient removal of oil from the bearing cavity and prevents the possible cavitation damage due to a negative cavity pressure.