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The Leading-Edge-Groove Tilting-Pad Thrust Bearing: Recent Developments

A. M. Mikula
Research Engineer,
Kingsbury, Inc.,
Philadelphia, PA 19154
Mem. ASME

This paper compares the leading edge groove and pressurized supply (flooded) lubricant supply methods, and analyzes their influence on the performance of equalizing tilting pad thrust bearings. This paper presents new experimental data on 6-shoe, 267 mm (10 1/2 in.) O.D. bearings, operating at shaft speeds up to 14000 rpm, with loads ranging up to 3.45 MP_a (500 psi) for two different lubricants. The data presented details the power loss and babbitt temperature performance of two versions of the leading-edge-groove bearing design and contrasts the results with a pressurized supply bearing design.

Introduction

The leading edge groove tilting pad thrust bearing is a hydrodynamic bearing that introduces the lubricant directly into the fluid film at the leading edge of the thrust shoe. This method of supplying cool, undiluted lubricant into the hydrodynamic wedge has been found to significantly reduce bearing power loss and babbitt temperatures [1].

This paper presents the most recent results of the extensive and ongoing testing performed on the leading edge groove bearing, and serves as a supplement to the test data published in reference [1]. These additional performance figures are the result of refinements that have been made to the leading-edge-groove bearing's design since the original results were first published. The net result of these refinements has been to reduce internal leakage and, therefore, maximize lubricant flow into the groove. Additional new test data has also been included for a more viscous lubricant, and a further reduction of oil flow rates.

The two primary indicators of bearing performance, power loss and babbitt temperature, will be used to evaluate the leading-edge-groove and pressurized supply (flooded) bearing designs. Each bearing was tested under identical conditions of applied load, shaft speed, inlet oil temperature, and oil viscosity. A detailed description of the test rig can be found in reference [2].

Each bearing was evaluated using both a light and heavy oil that was supplied at 46°C (115°F). Applied loads ranged from 0-3.45 MP_a (0-500 psi) and shaft speeds ranged from 2000-14000 rpm.

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VISCOSITY DATA

Oil Type	ISO VG	Pa·s @ 37.8°C	Pa·s @ 98.9°C	SSU @ 100°F	SSU @ 210°F
Light turbine oil	32	0.027	0.006	150	43
Heavy oil	None	0.067	0.0074	375	53

Test Bearing Descriptions

267 mm (10.5 in.) equalizing tilting pad double thrust bearings were tested. A double thrust bearing is, as the name suggests, two bearing elements such as that shown in Fig. 2, one of which carries thrust loads and is called the loaded bearing, while the other bearing is called the slack side bearing because its purpose is to position the shaft and carry any transient reverse thrust loads. Details of the arrangement of the two bearing elements in the housing can be found in reference [2].

The test bearings consisted of six babbitted and heavily instrumented shoes on each side of a rotating collar for a (6 × 6) double thrust bearing configuration. Each shoe had a babbitt O.D. of 267 mm (10.5 in.), a bore of 133 mm (5.25 in.) and, except for the leading-edge-groove distribution (LEG) bearing, had a total bearing area of 356 cm^2 (55.1 $in.^2$) with 51 deg of arc. The LEG shoes were of larger arcs (57°) to accommodate the distribution grooves, but had the same total effective bearing area of 356 cm^2 (55.2 $in.^2$). Figure 1 contrasts the LEG and conventional shoe designs.

All tests were conducted with a discharge restriction. The bearing collar was shrouded with an oil control ring that was bored with a 3.97 mm (5/32 in.) radial clearance over the collar diameter, and fitted with a 25.4 mm (1.0 in.) tangential discharge port. The geometry is shown in Fig. 12, reference [6].

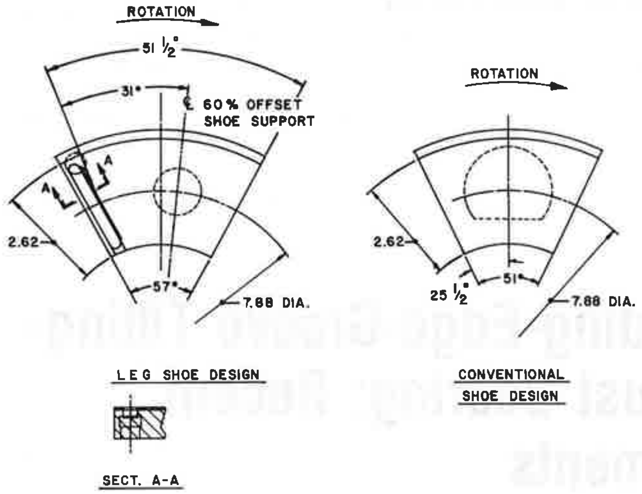

Fig. 1 Comparison of the conventional and LEG shoe designs

Fig. 2 Leading-edge-groove bearing showing the oil feed tube

Pressurized Supply (Flooded) Bearing

This style of conventional thrust bearing has been fully described in reference [2] and other literature, but a brief description of its lubricant supply method should prove beneficial. The bearing is supplied with a specific volume of oil that enters an annulus in the base ring. From there, the oil flows radially inward through slots in the back face of the base ring and then axially through the clearance between the base ring bore and shaft. When the oil reaches the rotating thrust collar, it is pumped radially outward between the shoes. Rotation of the collar carries some of it into the oil film on each shoe. Due to the pumping action of the collar, supply pressures of only 0.03–0.14 MP_a gage (5–20 psi) are all that is required. As the oil travels through the bearing, it is warmed considerably, so that oil supplied at 46°C (115°F) is 54–60°C (130–140°F) by the time it enters the oil film wedges.

Only a small portion of the total oil flow supplied to the bearing (around 10 percent) actually finds its way into the hydrodynamic wedges [3]. Most of the oil is intended to be used for cooling of the bearing components. In the process, however, it increases the churning losses around the collar. A reduction in the volume of oil supplied will therefore reduce the collar churning losses, but it will also reduce the beneficial cooling available, which will be reflected by an increase in shoe babbitt temperatures.

Leading-Edge-Groove (LEG) Bearing

The original design of the LEG bearing is described in reference [1]. Since that time, further design refinements have been introduced. The LEG design introduces cool, undiluted oil directly into the hydrodynamic wedge. This initially cool layer in intimate contact with the shoe babbitt results in dramatic temperature reductions.

The leading edge of the shoe is extended to accommodate the distribution groove by the addition of nonload-carrying areas as shown in Fig. 2. A chamfer on the trailing edge side of the groove is used to facilitate oil flow even when the shoe contacts the collar, such as an "at rest" condition. The pressurized oil supply is directly connected at the O.D. with the distribution groove by means of an oil feed tube that is designed to provide a positive "O" ring seal while, at the same time, allowing unencumbered shoe movement. The groove is relieved to the I.D. to facilitate oil flow down the full length of the groove. The oil path through the bearing was redesigned to reduce internal leakage and, at the same time, pressure drop. These improvements have resulted in not only a reduction of the oil supply pressure necessary to supply the designated amount of oil to the distribution grooves, but also any oil leakage from the oil supply system before the oil enters the distribution groove.

Reverse Rotation

Originally, it was stated that the LEG design precludes operation in two directions of rotation because of the distribution groove [1]. Recent tests, with the LEG bearing purposely installed with the distribution at the trailing edge, prove that the bearing can support loads up to 3.45 MP_a (500 psi) with some loss of efficiency. These tests will be detailed in a subsequent paper.

Oil Supply Flow Rates

The influence of oil supply flow rates on thrust bearing performance has been found to be significant [1, 2]. If 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 and later refined in order to effectively manage reduced oil flows in such a way as to eliminate this usual tradeoff.

The most recent development work on the LEG design, described in this paper, has centered on two subjects: 1) a further reduction in total oil flow rates; 2) maximizing the oil flow through the distribution groove and into the hydrodynamic wedge.

Overall, oil flow rates to the LEG bearings during the most recent testing were reduced by cutting back the slack bearing flows as shown in Fig. 3. The rationale behind this was that, because the distribution groove had proven so effective at placing the oil into the loaded bearing's hydrodynamic wedge, it should be just as effective in the slack side bearing. The slack bearing oil flows were reduced on a sliding scale, the maximum reduction occurring at no-load and the minimum reduction at maximum load. Figures 4 and 5 show the loaded and slack bearing oil flow rates of ISO VG 32 light turbine oil at loads of 2.07 and 3.45 MP_a (300 and 500 psi). Tests of the LEG bearing using heavy oil were only conducted with reduced slack oil flows as shown in Figs. 6 and 7.

Design modifications to the original LEG bearing were

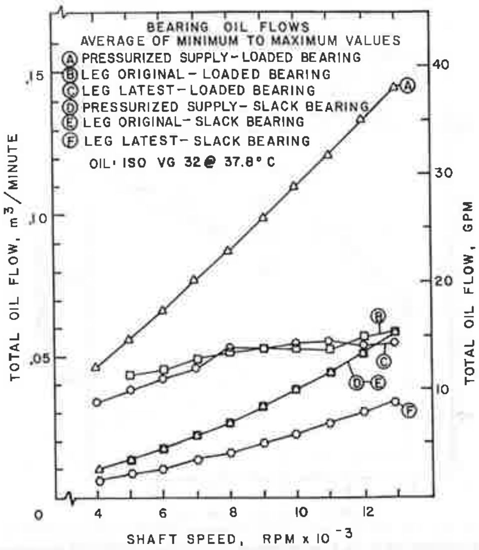

Fig. 3 A comparison of the average oil flow rate supplied to each bearing. This average was computed as follows:

Let

A = average oil flow rate
i = oil flow rate at tested load
N = number of loads tested for each shaft speed

Then

$$A = \frac{\sum_{i=0}^{3.45\ \mathrm{MPa}} i}{N}$$

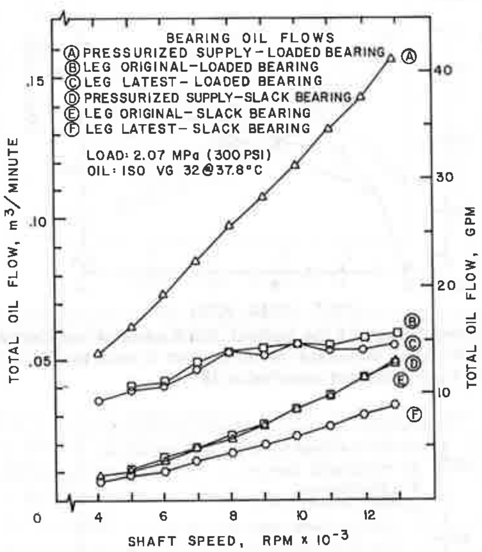

Fig. 4 A comparison of oil flows to the individual thrust bearings when loaded to 2.07 MP$_a$ for an ISO VG 32 lubricant supplied at 46°C

made to reduce internal leakage and pressure drop through the bearing. These design changes have resulted in a reduction in the supply pressure from 0.10–0.14 MP$_a$ gage (15–20 psig) to 0.048–0.069 MP$_a$ gage (7–10 psig).

Bearing Operating Temperatures

Thermocouples puddled in the babbitt itself, approximately 0.8 mm (1/32 in.) below the actual shoe surface, were used to measure operating temperatures. Specific details of thermocouple placement can be found in reference [2]. Shoe surface temperatures are subject to the influence of location and load equalization. Across the shoe surface, there are many operating temperatures, each a function of location as shown in the typical result, Fig. 8. Small variations in tem-

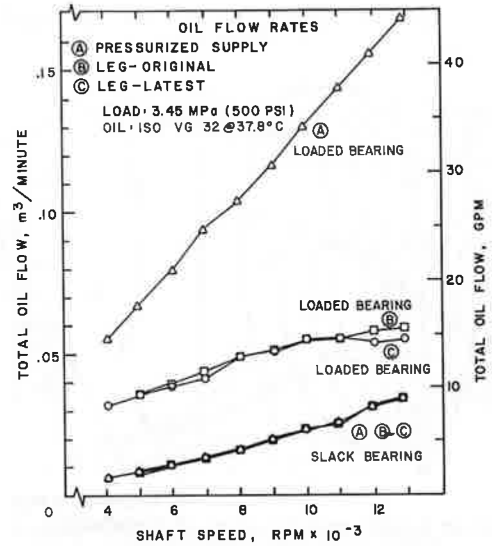

Fig. 5 A comparison of oil flows to the individual thrust bearings when loaded to 3.45 MP$_a$ for an ISO VG 32 lubricant supplied at 46°C

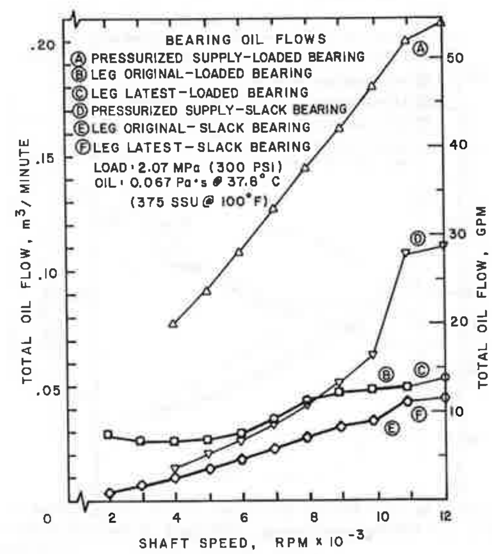

Fig. 6 A comparison of oil flows to the individual thrust bearings when loaded to 2.07 MP$_a$ for a 0.067 P$_a$·s lubricant supplied at 46°C

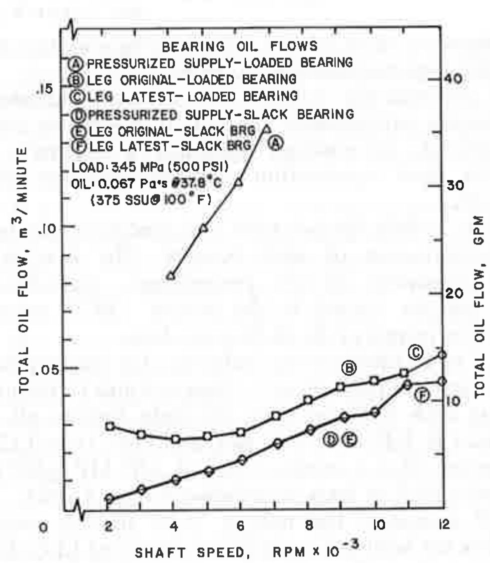

Fig. 7 A comparison of oil flows to the individual thrust bearings when loaded to 3.45 MP$_a$ for a 0.067 P$_a$·s lubricant supplied to 46°C

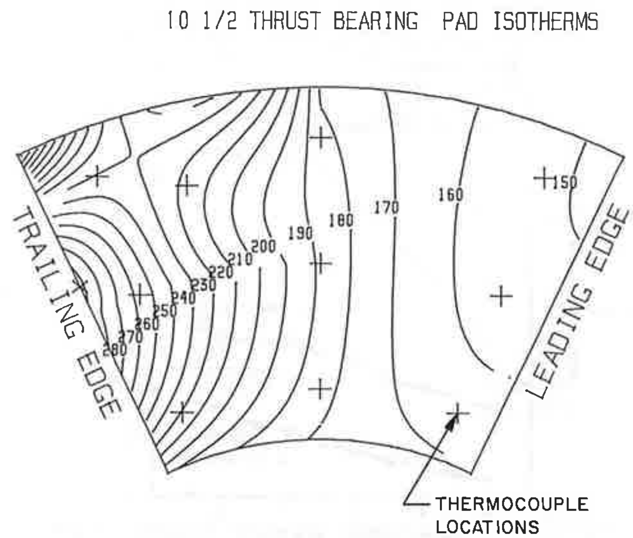

Fig. 8 An isotherm of a 10 1/2 in. thrust bearing shoe loaded to 3.45 MP_a at a shaft speed of 7000 rpm, showing the temperature gradient across the shoe in degrees Fahrenheit when using the light turbine oil

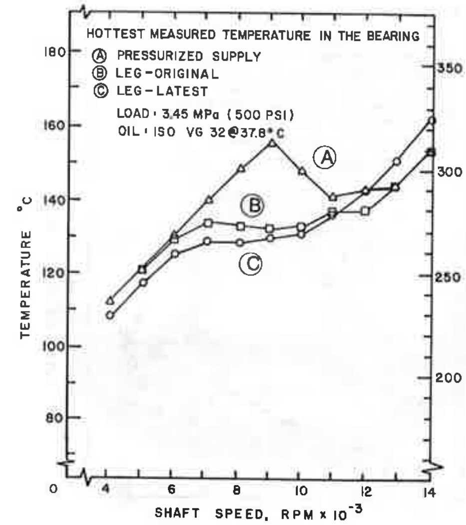

Fig. 10 A comparison of the hottest measured babbitt temperatures of each individual bearing when loaded to 3.45 MP_a and using an ISO VG 32 lubricant supplied at 46°C

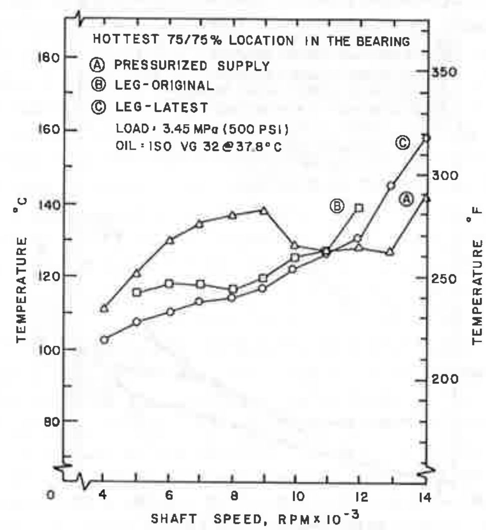

Fig. 9 A comparison of the hottest 75/75 percent location babbitt temperature for each individual bearing when loaded to 3.45 MP_a and using an ISO VG 32 lubricant supplied at 46°C

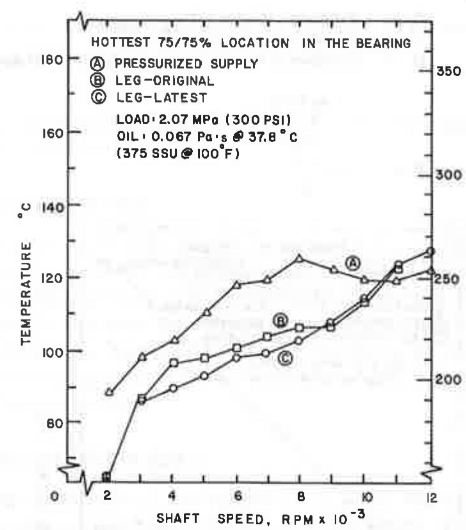

Fig. 11 A comparison of the hottest 75/75 percent location babbitt temperature for each individual bearing when loaded to 2.07 MP_a and using a 0.067 P_a·s lubricant supplied at 46°C

perature between shoes of the same bearing can also result from less than optimal load equalization.

Babbitt temperatures serve as a convenient indicator of overall bearing performance. Babbitt temperatures can be used to provide information regarding a bearing's load capacity [4], load equalization, and the condition of the babbitt itself.

Two shoe surface temperatures are used to evaluate the relative performance of each bearing. The first is the maximum measured babbitt temperature, regardless of location; and the second is the hottest "75-75 percent" location found on any of the six bearing shoes.

Figures 9 to 12 illustrate the influence that bearing design has on temperature performance. Figures 9 and 10 report the temperature data collected using the light turbine oil at a constant load of 3.45 MP_a (500 psi), and Figs. 11 and 12 are for the heavier oil at a constant load of 2.07 MP_a (300 psi). The respective oil flow rates were shown in Figs. 5 and 6.

Figure 9 compares the hottest 75-75 percent location temperatures for both the original and improved LEG design and the conventional flooded design. Both LEG designs have a temperature advantage over the flooded bearing until 11,000

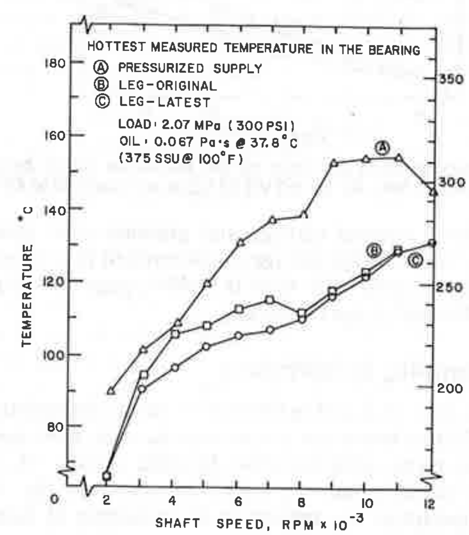

Fig. 12 A comparison of the hottest measured babbitt temperatures of each individual bearing when loaded to 2.07 MP_a and using an ISO VG 32 lubricant supplied at 46°C

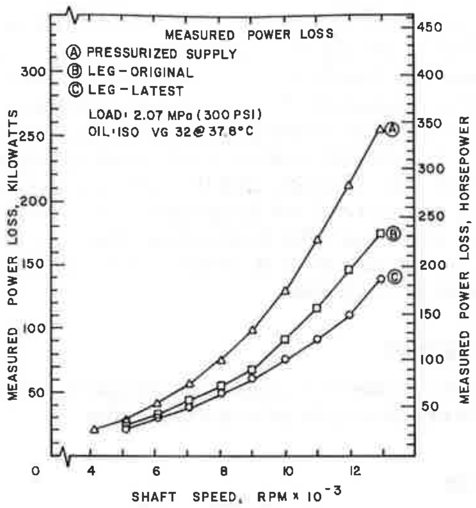

Fig. 13 A comparison of the measured power loss of the individual bearings when loaded to 2.07 MP$_a$ and using an ISO VG 32 lubricant supplied at 46°C

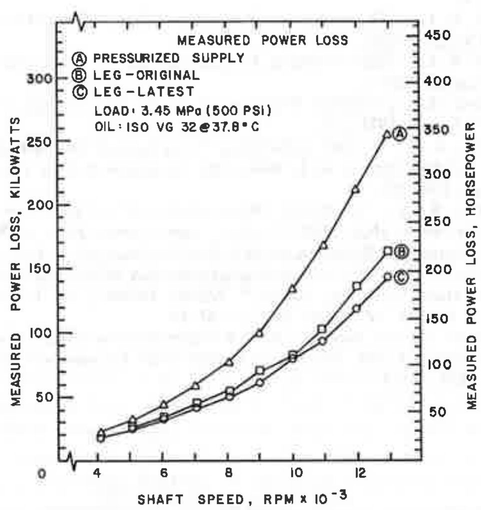

Fig. 14 A comparison of the measured power loss of the individual bearings when loaded to 3.45 MP$_a$ and using an ISO VG 32 lubricant supplied at 46°C

rpm. While it is true that after 11,000 rpm the flooded design is running cooler, it should be remembered that the flooded design is being supplied with over 2 1/2 times the oil flow of the LEG designs. This temperature disadvantage can be changed by increasing the oil flow rates to the LEG designs [1]. Figure 10 makes a similar comparison, but this time for the hottest babbitt temperature regardless of location. The results are also similar except that the LEG designs maintain their advantage until after 12,000 rpm. Comparisons between the two LEG designs show that, overall, the latest design runs generally cooler than the original design.

Figure 11 also compares the hottest 75-75 percent location temperatures of the various designs, but this time for the heavier oil. As was the case with the lighter oil, there is a decided temperature advantage for the LEG designs until 11,000 rpm. The hottest recorded babbitt temperatures are compared in Fig. 12. As was the case with the light oil, both LEG designs ran significantly cooler than the flooded design, and the latest LEG design ran cooler than the original design. Excessive operating temperatures measured in the conventional flooded bearing limited the maximum load that could be run to 2.07 MP$_a$ (300 psi) across the entire speed range.

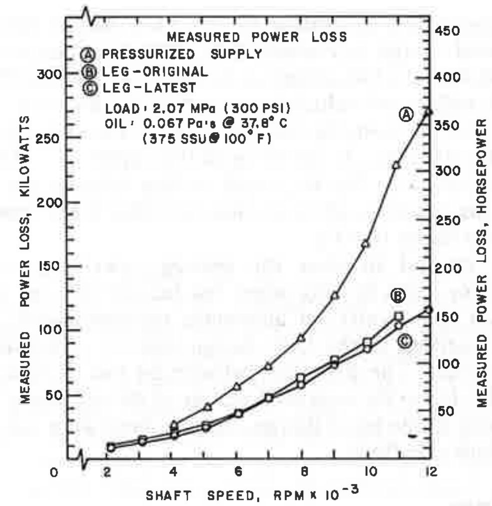

Fig. 15 A comparison of the measured power loss of the individual bearings when loaded to 2.07 MP$_a$ and using a 0.067 P$_a$·s lubricant supplied at 46°C

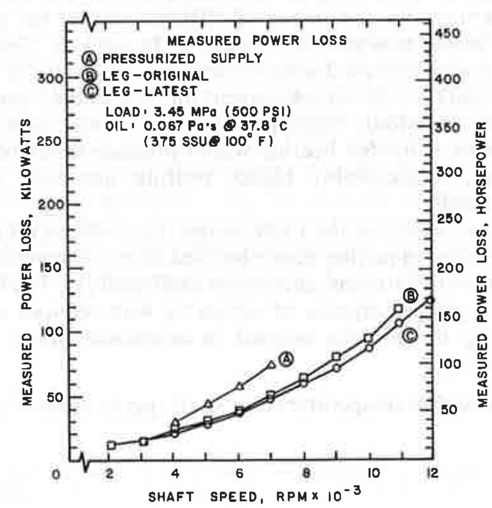

Fig. 16 A comparison of the measured power loss of the individual bearings when loaded to 3.45 MP$_a$ and using a 0.067 P$_a$·s lubricant supplied at 46°C

Bearing Power Loss

Bearing power loss values were calculated using an energy balance technique that computed loss from the measured rise in oil temperature (supply to drain), oil flow rate, and lubricant specific heat. Radiation and conduction losses are considered small and constant, and are therefore omitted from this analysis.

"One of the most critical factors influencing bearing power loss is the oil supply flow rate" [1]. As a result, much work has gone into the development of bearing designs that can run at flow rates, and therefore power losses, that are substantially less than conventional flooded bearings. The reduced flow rates (see Figs. 3 to 6) associated with the LEG design are possible because the oil is supplied directly into the hydrodynamic wedge by the distribution groove. This results in the meager requirements of the oil films [3], dictating the greatly reduced oil flow rates that are the inevitable cause of the substantial power loss reductions. Reducing the flow rate to a conventional flooded bearing would also reduce power loss, but unlike the LEG design, operating babbitt temperatures would suffer.

Figures 13 and 14 show the bearing power loss values calculated for each bearing with light turbine oil. The substantial power loss advantage of the LEG design over the conventional design is evident. The difference between the original and latest LEG designs is also evident. The benefits of improved sealing and reduced slack bearing flows account for the difference between the LEG designs in Fig. 13, while in Fig. 14, the difference is due solely to the improved sealing of the oil supply path. The improved sealing reduced parasitic churning losses that result from lubricant that is not supplied directly into the oil film [5].

Figures 15 and 16 show the bearing power loss values measured for each bearing when the heavier oil was used. Once again, the significant advantage (approximately a 50 percent reduction) of the LEG design over the conventional design is evident. The difference between the two LEG designs can only be due to the improved sealing of the oil supply path incorporated in the latest design, because both were run with reduced slack side flows.

Conclusions

1 The leading-edge-groove (LEG) bearing demonstrated an ability to operate at oil flow rates that were up to 75 percent less than those supplied to the conventional flooded thrust bearing with the heavy oil. These reduced oil flows produced power loss savings as high as 56 percent. Similar, although somewhat less dramatic, results were produced with the lighter ISO VG 32 oil (66 percent oil flow and 45 percent power loss reduction). While it is true that similar flow rate reductions to a flooded bearing would produce lower power loss values, unacceptably higher bearing operating temperatures result.

2 Improvements to the LEG design that reduced oil path leakage resulted in further power loss reductions in comparing the present results to those previously published [1]. The LEG design also proved capable of operating with reduced slack side bearing flows. This resulted in additional power loss reductions.

3 Shoe babbitt temperatures for shaft speeds under 12,000 rpm were found to be up to 20 percent lower with the LEG design. This is attributed to the introduction of cool, undiluted supply oil directly into the oil film wedge. This cool oil is believed to insulate the shoe surface from the hot oil carry-over adhering to the rotating collar. At sliding velocities above 11,000 rpm, the combined effects of the significant LEG oil flow rate reductions and the turbulent oil film flow [7, 8, 9] in the conventional design produced mixed results. Reductions to the slack side bearing's oil flow rate generally produced no increase in shoe surface temperatures because this assembly was unloaded.

Acknowlegment

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

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DISCUSSION

F. A. Martin[1]

The leading-edge groove bearing would appear to be most suitable for direct lubrication in the pads with a freely drained casing. It was first envisaged that this was the topic of the authors paper since the abstract refers to "performance of the *leading edge groove bearing* design and *constrasts* the results with a *pressurized supply bearing* design". On reading the paper further it is understood that the casing in these particular tests is subjected to supply pressures from 0.10 to 0.14 MPa for the original LEG bearing tests and 0.048 to 0.069 MPa for the latest LEG results and that *all tests* were *conducted with* a *discharge restriction*. The discusser is now of the optinion that the authors present results all relate to pressurized casing (flooded) bearings. If this is so, it would be useful to know the general global temperature rise through the assembly compared with other bearings.

As commented on in previous papers, the tests results for the pressurized supply bearing (without leading edge groove) used by the author, relates to a centrally pivoted bearing. As the LEG bearings have offset pivots, it would appear more appropriate to compare them with other offset pivot bearings rather than with the special case of centrally pivoted bearings which must have different performance characteristics (the latter relying on the crowning of the pad – by machining, thermal and elastic distortion – for successful operation). The authors experimental results are always a welcome input to the literature. However, does the pivot position "cloud" the comparison of results and are these particular LEG tests for a nominally pressurized casing?

[1]Principal Engineer, Department of Applications Engineering, The Glacier Metal Co., Ltd., Middlesex HAO 1HD England.

W. Gardner[2]

As the author indicates, the present paper is a supplement to an earlier paper [1], both giving test results of the leading-edge-groove bearing as compared to a "conventional" design. In the present paper, as in the discussion to the first paper, it is noted that the LEG bearing uses circumferentially offset pivots (0.6) as compared to center pivots (0.5) for the conventional bearing.

[2]Vice President – Engineering, Waukesha Bearings Corp., Waukesha, Wisc. 53187

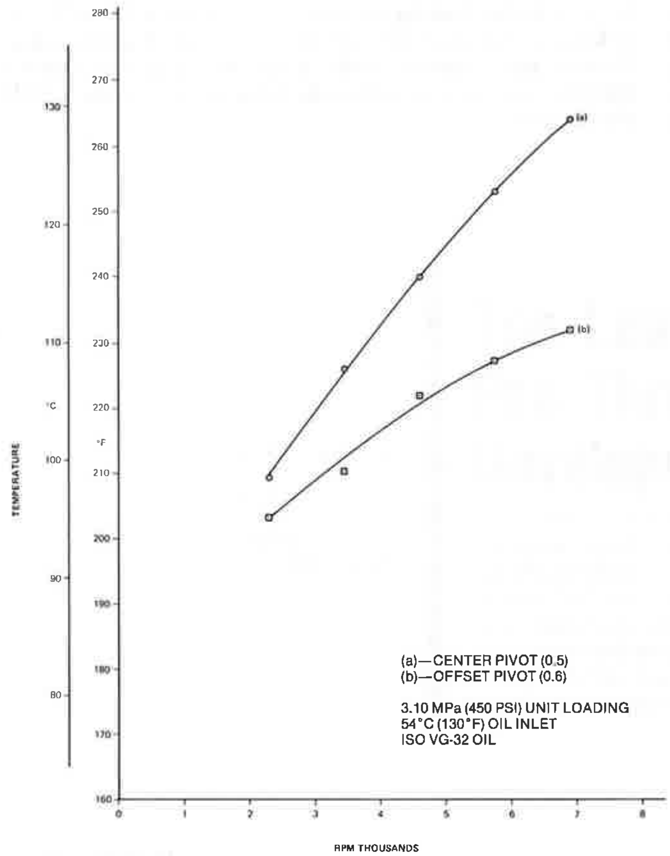

Fig. A1 A comparison of the average 75/75 location babbitt temperatures for center and offset pivot pads

The reduced operating temperatures found with the LEG bearing have been attributed (in both papers) to the change in the lubrication supply method. The discussions of the first paper suggested that the use of offset pivots in the LEG bearing was contributing to reduced pad temperatures. The authors responded to this in their closure with test data from their conventional bearing that showed no temperature advantage for the offset pivot construction, as compared to center pivot, below about 11,000 RPM.

The discusser accepts this but wants to note that it is not in agreement with his experience in test work on bearings similar to the author's "conventional" bearing. In this respect, Fig. A1 gives test data from a 267 mm (10.5 in.), 356 sq cm (55.1 sq in.), six pad thrust bearing in both center (0.5) and offset (0.6) pivot constructions. Unfortunately, the operating conditions are not identical to those of the author, but a comparison to Fig. 9 can still be made. The temperature values in the discusser's Fig. A1 are the average of the thermocouples in the 75-75 location rather than the hottest of these, as the author has used.

It is recognized that this (Fig. A1) is not necessarily a comparison of the hot spots on the pads. This is certainly one of the problems in such test work because, as the author notes, the hot spot location varies depending on the operating conditions and bearing design. However, Fig. A1 indicates that reduced temperatures (similar to those in Fig. 9) can result from offsetting the pivot. The question for the author from all this is whether any tests have been run on a center pivot LEG bearing? This would allow a direct comparison to the conventional bearing and isolate the influence of the leading edge groove from any influence of the offset pivot.

In the "Conclusions" to this paper, the author states that flow rate reductions in a flooded bearing similar to those in the LEG bearing result in unacceptably high bearing temperatures. Has this been confirmed by tests?

Author's Closure

The author would like to express his gratitude and appreciation to both Mr. Gardner and Mr. Martin for the comments and interest they have expressed in this paper.

The question of whether or not the exceptional temperature performance of the Leading Edge Groove (LEG) Bearing should be attributed to pivot location or lubricant supply method has been raised by both Mr. Gardner and Mr. Martin. This is the same question that was raised in the discussion of reference [1]. The reply to this question has not changed, but the fact that this question is raised again suggests that a more definitive answer is required. To resolve this question requires the construction and testing of a centrally pivoted LEG bearing. Unfortunately, test results are not currently available, but will be in the near future.

Mr. Gardner's question concerning test results for a flooded bearing operating at reduced oil flow rates is addressed in Fig. 17. As a rule, bearing operating babbitt temperatures in a high film pressure region, such as the 75/75 percent location, that exceed 130°C (266°F) are considered excessive for most applications. Clearly, for all shaft speeds except 4000 and 5000 rpm, the reduced oil flows produced unacceptable 75/75 percent babbitt temperatures.

Mr. Martin also questioned whether all the bearing tests were conducted with pressurized (flooded) casings. The

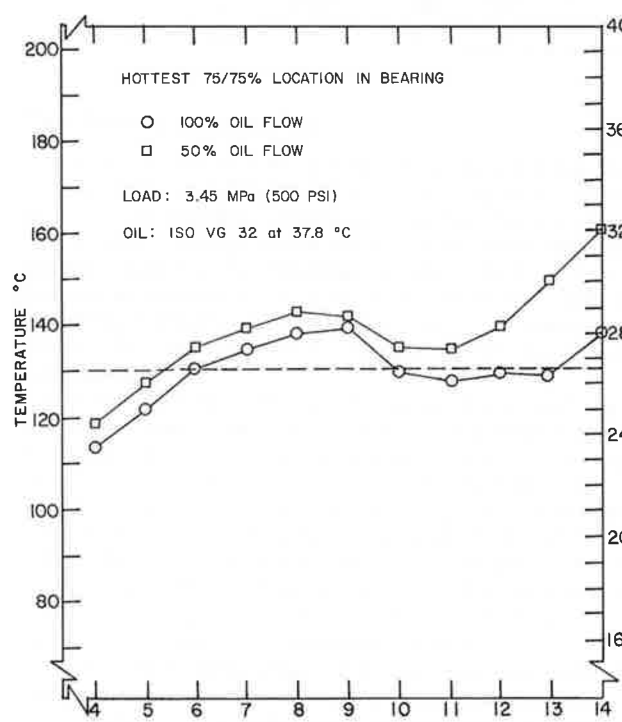

Fig. 17 A comparison of the hottest 75/75 percent location babbitt temperature for bearing supplied with 100 and 50 percent of recommended oil flow rates

answer to this is "no." The confusion probably results from the statement in the text that states "All tests were conducted with a discharge restriction." The restriction is not on casing drains which were open fully, but on the tangential discharge port of the oil ring (OCR). The purpose of the oil control ring is to minimize bearing pumping and churning losses by expeditiously removing the oil from around the bearing collar. Pressure measurements made during the LEG tests indicate that the static pressure within the OCR never exceeded 0.0138 MPa (2 psi).