stantial 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 carryover 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.

Acknowledgment

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

References


DISCUSSION

F. A. Martin¹

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 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 opinion 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?

W. Gardner²

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 pivot 0.6 as compared to center pivots 0.5 for the conventional bearing.

¹Principal Engineer, Department of Applications Engineering, The Glacier Metal Co., Ltd., Middlesex HA1 1HD England.

²Vice President – Engineering, Waukesha Bearings Corp., Waukesha, Wisc. 53187
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 author 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 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
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).