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# A Comparison of Tilting Pad Thrust Bearing Lubricant Supply Methods

*This paper compares three different lubricant supply methods—pressurized supply (flooded), spray feed, and leading edge distribution groove—and analyzes their influence on the performance of tilting pad, equalizing thrust bearings. The paper presents experimental data on 267 mm (10-1/2 in.) o.d. bearings, operating at shaft speeds up to 13,000 rpm with loads ranging up to 3.45 MPa (500 psi). The data presented demonstrate the effect each lubricant supply method has on bearing power loss and temperature. Conclusions are drawn, based upon the effectiveness of each design, to guide the potential user.*

## Introduction

There are a wide variety of thrust bearing types available to machinery designers. In addition to initial cost, each type of thrust bearing has its own unique set of performance characteristics which serve as the evaluation criteria for identical operating conditions. The two primary indicators of bearing performance are power loss and babbitt temperature of the pads, or "shoes." Rising energy costs have made bearing power loss a very critical yardstick in the evaluation of relative performance. The maximum babbitt temperature effectively gages the degree of bearing operating risk, and can even limit the bearing's suitability for a specific application. The approach of this paper will be limited to a thorough discussion of power loss values and bearing operating temperatures attained in similar bearings employing three different lubricant supply methods.

In order to evaluate the three different lubricant supply methods—pressurized supply, spray feed, and leading edge distribution groove—each method 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 [1]. By reporting the effect of each lubricant supply method on bearing power loss and babbitt temperature, it is hoped that this paper will provide the necessary information for the designer to make a well-reasoned thrust bearing selection, based upon actual performance data.

All three bearings were evaluated in comprehensive tests using a light turbine 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 ranging from 0-3.45 MPa (0-500 psi) and shaft speeds ranging from 4000-13,000 rpm.

Contributed by the Lubrication Division of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS and presented at the ASLE-ASME International Lubrication Conference, New Orleans, La., October 4-7, 1981. Manuscript received by the Lubrication Conference February 16, 1981. Paper No. 81-Lub-8.

## Test Bearing Descriptions

This entire discussion is based upon the data obtained during an extensive series of tests on 267 mm (10.5 in.) tilting pad, equalizing double thrust bearings. A double thrust bearing consists of two elements such as that shown in Figs. 1 and 2, one of which normally carries thrust load and is termed the "loaded" or active bearing, while the other element, on the opposite side of the thrust collar, is called the "slack" side or inactive bearing because it merely serves to position the shaft. Exact details of the arrangement of the two elements in the bearing housing can be found in reference [1].

The primary test bearing has six babbitted, heavily instrumented pads or shoes on each side of the collar for a $(6 \times 6)$ double thrust bearing configuration. Additional, corroborative data are furnished in this paper for a similar bearing with eight babbitted pads on each side of the collar for an $(8 \times 8)$ double thrust bearing arrangement. In either instance, the shoes have 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 distribution groove bearings, have a total bearing area of 356 cm$^2$ (55.1 in.$^2$) with 51 deg of arc for the 6-shoe design, and 38 deg of arc for the 8-shoe design. The shoes used for the leading edge distribution groove bearings are different in surface area to accommodate the distribution groove as shown in Fig. 2. The shoes of the 6-shoe leading edge distribution groove design subtend 57 deg of arc for a total effective bearing area of 356 cm$^2$ (55.2 in.$^2$), while the 8-shoe design subtend 43-1/2 deg of arc for a total effective bearing area of 349 cm$^2$ (54.1 in.$^2$).

**Pressurized Supply (Flooded) Bearing.** This conventional thrust bearing style has been fully described in references [1, 3 and 4] and other literature. For this discussion of lubricant supply methods, it should be sufficient to say that this type of bearing is fed a specified volume of oil to an annular channel surrounding the bearing at the back end of the assembly. The oil then flows radially inward and axially between the bearing and rotating shaft until it reaches the working babbitt shoe faces. The oil is propelled more by the pumping action of the collar than by the pressurized supply, so only modest 0.03-

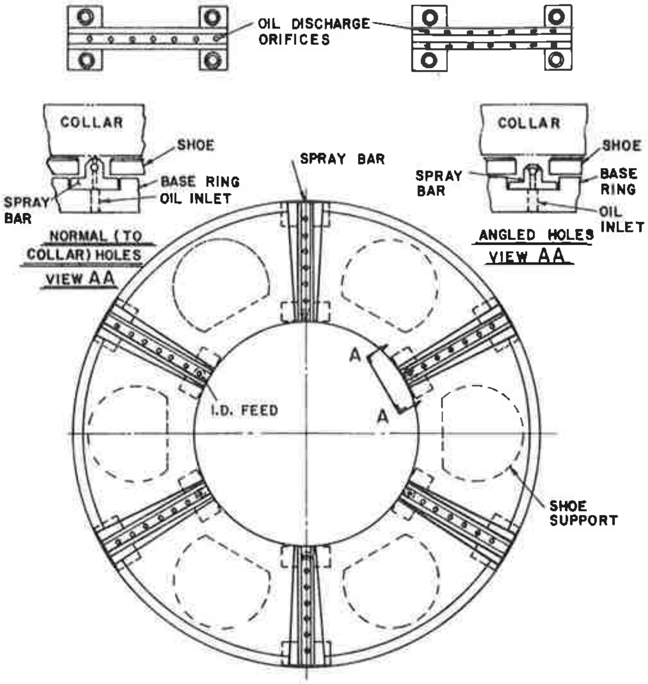


Fig. 1 Spray feed bearing showing the location of the spray bars and the different oil discharge orifice arrangements

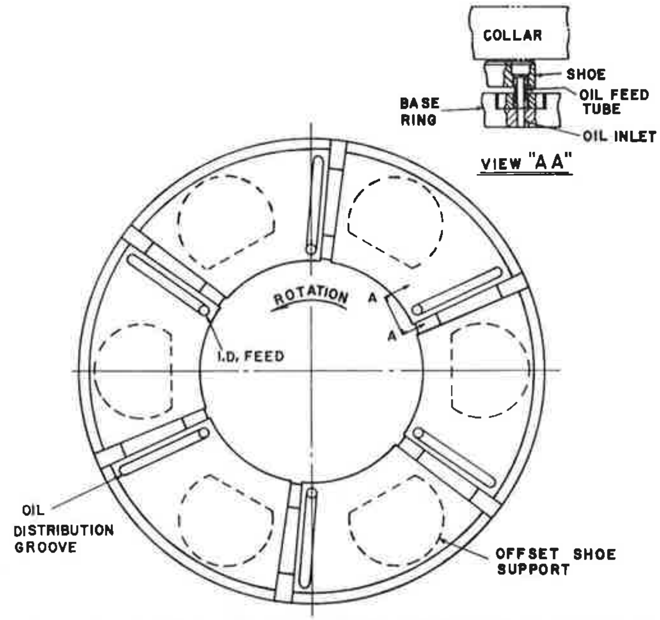


Fig. 2 Leading edge distribution groove bearing showing the location and details of the oil supply to the oil distribution groove

0.14 MPa gage (5-20 psig) supply pressures are required. The oil is warmed considerably in its journey, so that oil supplied at 46°C (115°F) has a bulk oil temperature of 54-60°C (130-140°F) at the time of introduction to the oil film wedge. Only a small portion of the total oil supply volume (estimated at 5-10 percent) actually finds its way into the hydrodynamic film wedge. The larger portion of excess oil is used for cooling of the bearing components and merely adds to the churning losses around the collar until it is expelled. Reducing the amount of supply oil (as long as the film wedge requirements are satisfied) will reduce the amount of beneficial cooling available, and result in a slight increase in operating temperature, measured in the shoe babbitt.

**Spray-Feed Bearing.** This specialized bearing design attemptes to introduce cool oil closer to the point of entry into the film wedge. The distribution of the spray jets, and hence the shape of the spray element, is a controversial subject, but it would seem reasonable that a greater number of spray orifices parallel to the leading edge of the shoe would be more effective at introducing cool oil into the film wedge than a design that featured fewer spray orifices. Accordingly, the test bearing discussed in this paper incorporates a spray bar that fits between adjacent shoes and is fastened to the bearing base ring at the o.d. and i.d., as shown in Fig. 1. Lubrication oil is supplied to the spray bars under pressure from an external source. The spray bars then serve to direct the oil at the rotating collar. Because of the improved efficiency of this distribution method, the amount of lubricant supplied can be reduced. The reduced oil volume, it will be shown later, accounts for lower power loss values.

Another potential benefit of the spray-fed bearing would be to utilize the pressurized spray jets to scour off the hot oil carryover that adheres to the rotating collar. Obviously, the number, size, pressure intensity, orientation, and gap between collar and spray bar would all influence the impingement velocity of the oil on the collar and, therefore, the effectiveness of this approach. During the course of the extensive testing reported here, all these parameters were varied, with only limited success. Two versions of the spray bars tested are also depicted in Fig. 1.

The drawbacks of this particular design would appear to be the impracticality of supplying sufficiently high pressures (as high as 0.55 MPa gage for these tests) to get true, effective scouring of the hot oil carryover, the recapture of any hot oil carryover disturbed by the jets, the tendency of the small jet holes to clog with foreign material, and the dilution of the cool inlet oil before it is swept into the film wedge. On the other hand, the reduction in the volume of supply oil and its delivery close to the point of usage are both steps in the direction of more efficient operation. Although this bearing can be used with the housing cavity flooded or evacuated, the best power loss advantage is found with dry sump operation.

**Leading Edge Distribution Groove Bearing.** This specialized bearing design introduces cool, undiluted oil directly into the hydrodynamic wedge in a laminar layer that forms between the hot oil carryover and the stationary shoe. The result is an initially cool layer in intimate contact with the shoe babbitt, resulting in dramatic temperature reductions. The leading edge of the shoe is extended to incorporate the distribution groove by the addition of noneffective (nonload-carrying) area, as shown in Fig. 2. A chamfer or edge bevel on the groove facilitates oil flow even when the shoe contacts the collar, such as an "at rest" condition.

The pressurized oil supply is directly connected (either at bearing o.d. or i.d.) with the distribution groove to maximize the oil flow into the groove and, therefore, the oil film. It should be noted that any oil leakage from the oil supply system before the oil enters the distribution groove will reduce the amount of oil the groove can supply to the oil film. The supply pressure can be varied over a large range, but the bearing never truly becomes a hydrostatic or hybrid type [6]. Isothermal temperature maps show no indication of oil starvation at the trailing edge, i.d. location of the shoes, due to the centrifugal (inertial) effects of collar rotation on the oil with this method of oil supply.

The drawback of this particular design appears to be its incapability of operation in two directions of rotation because the loaded shoe pressure distribution precludes the addition of a groove at the trailing edge of the shoe. The favorable performance of the design far outweighs this consideration for most high speed, unidirectional equipment.

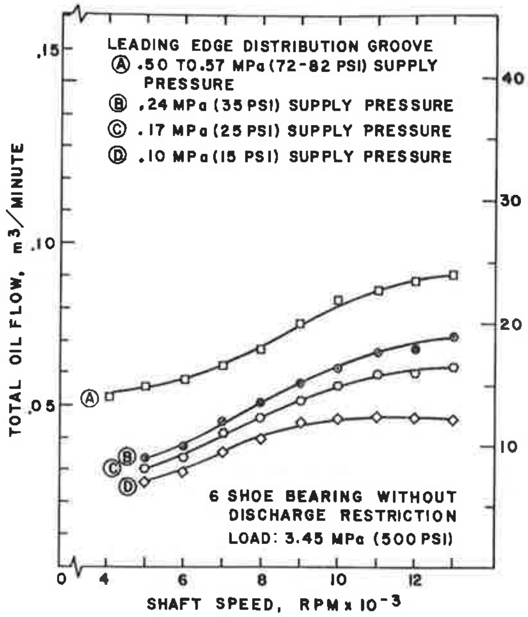


Fig. 3 A comparison of total oil flow for a 6-shoe leading edge distribution groove bearing operating without discharge restriction, with supply pressures ranging from 0.10 to 0.57 MPa

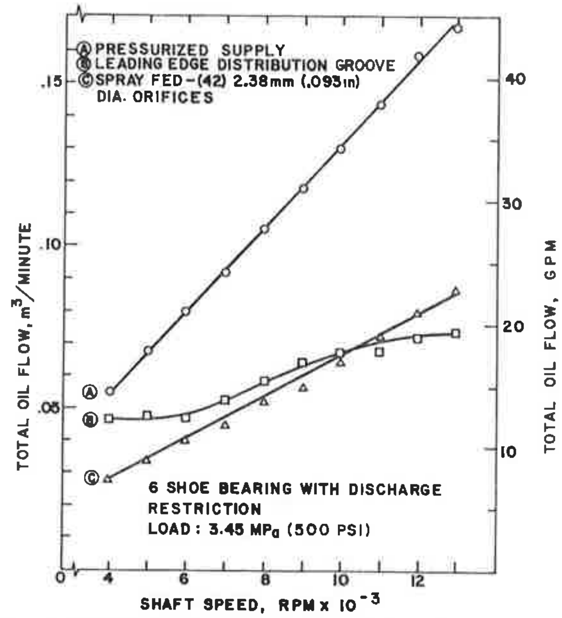


Fig. 5 A comparison of total oil flow for 6-shoe bearings operating with a discharge restriction and with these oil supply pressures: pressurized supply @ 0.03 to 0.14 MPa; leading edge distribution groove @ 0.50 to 0.57 MPa; and spray feed @ 0.04-0.35 MPa

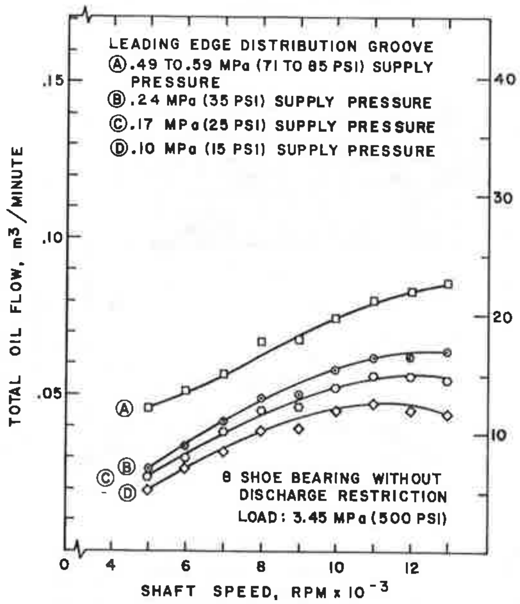


Fig. 4 A comparison of total oil flow for an 8-shoe leading edge distribution groove bearing operating without discharge restriction, with supply pressures ranging from 0.10 to 0.59 MPa

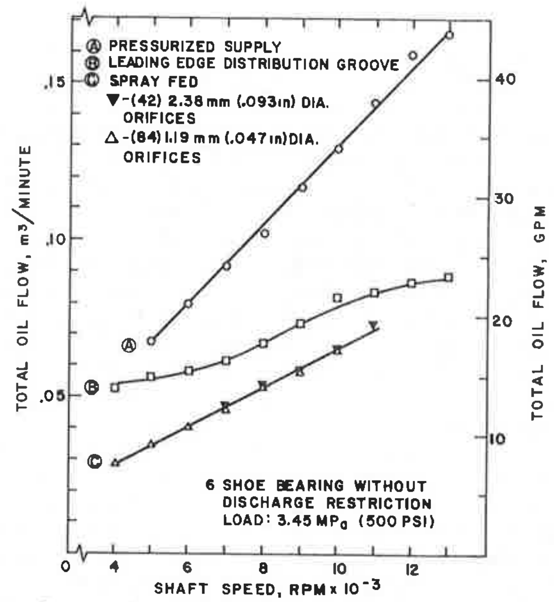


Fig. 6 A comparison of total oil flow for 6-shoe bearings operating without a discharge restriction and with these oil supply pressures: pressurized supply @ 0.03 to 0.14 MPa; leading edge distribution groove @ 0.50 to 0.57 MPa; spray feed with (42) 2.38 mm dia orifices @ 0.03 to 0.27 MPa; and spray feed with (84) 1.19 mm dia angled orifices @ 0.09 to 0.38 MPa

## Supply Oil Flow Rates

Probably no single factor influences pressurized supply or "flooded" thrust bearing performance (babbitt surface temperature, power loss, load capacity, film thickness, etc.) as significantly as the rate of oil supply to the bearing [1]. Fortunately, control of the oil flow rate is usually very easily accomplished by external adjustment without disturbing the bearings.

Depending upon the hydraulic pressure drop inherent in each of the three bearing designs, the lubricant supply pressures can be adjusted to obtain the desired oil flow rate. In the case of the spray-feed bearings with small jet holes, the high internal pressure drops require higher supply pressures (in the range of 0.51-0.55 MPa gage or 75-80 psig) to maintain oil flows even approaching those of the flooded bearings. Similarly, the oil flow rates through the leading edge distribution groove bearing are established by the oil film thickness, and shoe side leakage and bleed grooves (if any), since all the film oil enters, bypasses, or exits the hydrodynamic oil wedge through these gaps. Figures 3 and 4 show the range of oil flow rates obtained by adjusting the supply pressures of the distribution groove bearing.

It is also true that higher supply pressures necessitate improved sealing between bearing and housing, or between split bearing halves, in order to minimize the leakage of cool inlet oil and eliminate the tendency of bypassing supply oil to the discharge area. However, it is a redeeming feature of the spray-fed and distribution groove bearings that their better efficiency at providing oil at the point of need reduces the

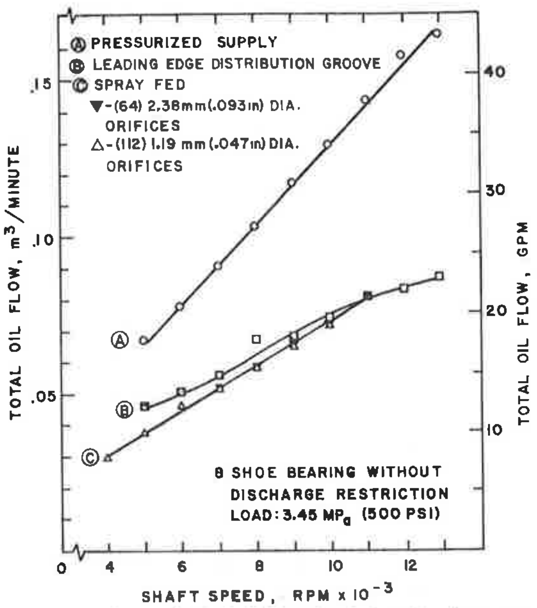


Fig. 7 A comparison of total oil flow for 8-shoe bearings operating without a discharge restriction and with these oil supply pressures: pressurized supply @ 0.03 to 0.14 MPa; leading edge distribution groove @ 0.49 to 0.59 MPa; spray feed with (64) 2.38 mm dia orifices @ 0.10 to 0.24 MPa; and spray feed with (112) 1.19 mm dia angled orifices @ 0.03 to 0.17 MPa

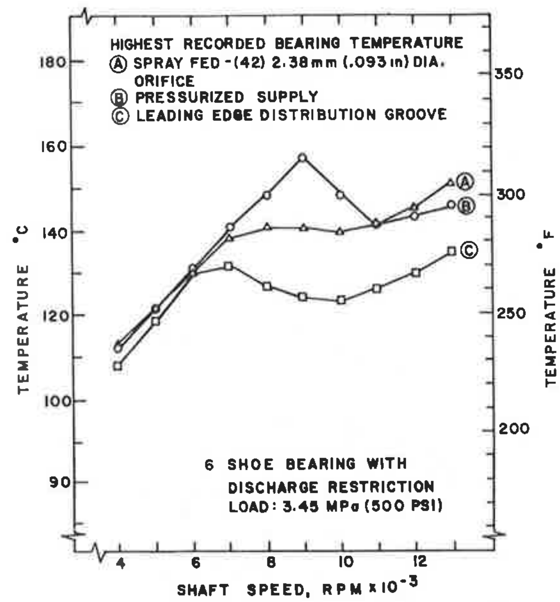


Fig. 8 Effect of various lubricant supply methods on the highest shoe babbitt temperature recorded for a 6-shoe bearing operating with discharge restriction and with these oil supply pressures: spray feed with (42) 2.38 mm dia orifices @ 0.04 to 0.37 MPa; pressurized supply @ 0.03 to 0.14 MPa; and leading edge distribution groove @ 0.22 to 0.45 MPa

requirement for excess oil. An adequate lubricant supply can be provided at a practical 0.10-0.14 MPa gage (15-20 psig) supply pressure. Therefore, variations in the flow rates between the three bearing designs compared in this paper are to be expected because they are determined by the choices of hardware and supply pressures, and they are clearly shown in Figs. 3-7 for the three different test configurations.

Tests of each lubricant supply method were conducted, both with and without restrictions on the discharge, for both the 8 and 6-shoe thrust bearings. For tests conducted with discharge restriction, the bearing collar was shrouded with an "oil control ring" bored with 3.97 mm (5/32 in.) radial clearance over the collar diameter, and fitted with a 25.4 mm (1.0 in.) diameter tangential discharge port. For tests without discharge restrictions, the oil control ring was removed, and the discharge oil drained through holes in the base of the housing. Although the use of an oil control ring added to the total measured power loss (compare curves of Figs. 12 and 13) at high operating speeds, it provided the necessary discharge backpressure at low shaft speeds to prevent thrust shoe instability and flutter in the unloaded bearing (see references [1, 3, and 4]). A cooler operating temperature is also evident for the flooded (pressurized supply) design with the use of the oil control ring, as can be seen in the comparison of curve B of Fig. 8 and curve A of Fig. 9.

Comparison of Figs. 5 and 6 will reveal that the flow rates with a discharge restriction were found to be identical for both the pressurized and spray-fed lubricant designs. However, differences in flow rates were evident for the leading edge distribution groove design. The use of a discharge restriction resulted in a 16-20 percent reduction in the lubricant supply flow rate for the 6-shoe configuration of the leading edge distribution groove method, as shown in Figs. 5 and 6. Figure 7 shows the lubricant supply flow rates for the 8-shoe configuration without discharge restriction under the various supply methods tested.

## Bearing Operating Temperatures

The operating temperature values were reported by thermocouples puddled in the babbitt itself, approximately

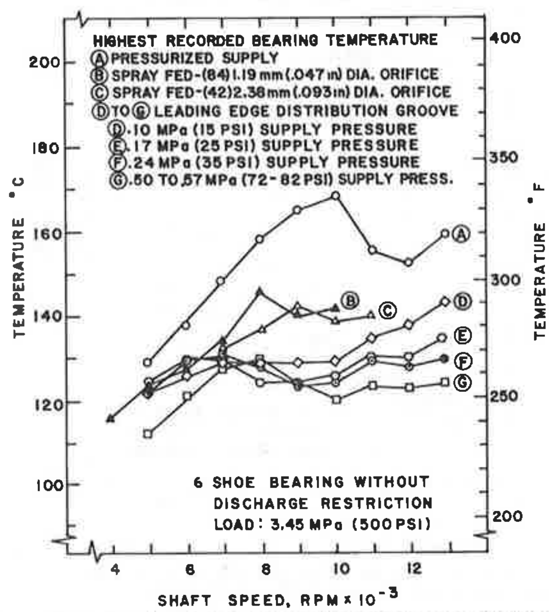


Fig. 9 Effect of various lubricant supply methods on the highest shoe babbitt temperature recorded for 6-shoe bearing operating without a discharge restriction and with these oil supply pressures: pressurized supply @ 0.03 to 0.14 MPa; spray feed with (84) 1.19 mm dia orifices @ 0.09 to 0.38 MPa; spray feed with (42) 2.38 mm dia orifices @ 0.03 to 0.27 MPa; and leading edge distribution @ various supply pressures as noted

0.8 mm (1/32 in.) below the actual shoe surface. Additional information on thermocouple placement can be found in references [1, 3, and 4]. Variation in shoe surface temperatures occurs as a function of position on the shoe surface – the coolest temperatures reported at the leading edge of the shoe, while the hottest temperatures are reported at the trailing edge. Also, due to less than optimal load equalization caused by differences in housing deflections, there are small variations in temperature between shoes of the same bearing. Generally, a bearing that is operating within its design limits

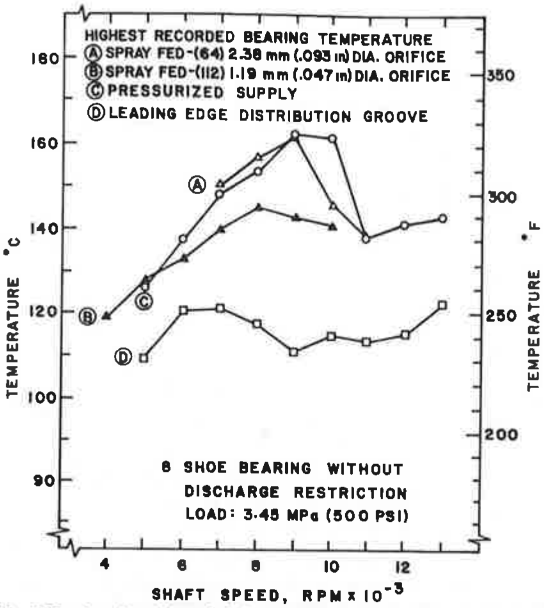


Fig. 10 Effects of various lubricant supply methods on the highest shoe babbitt temperature recorded for an 8-shoe bearing operating without a discharge restriction and with these oil supply pressures: spray feed with (64) 2.38 mm dia orifices @ 0.10 to 0.24 MPa; spray feed with (112) 1.19 mm dia orifices @ 0.03 to 0.17 MPa; pressurized supply @ 0.03 to 0.14 MPa; and leading edge distribution groove @ 0.49 to 0.59 MPa

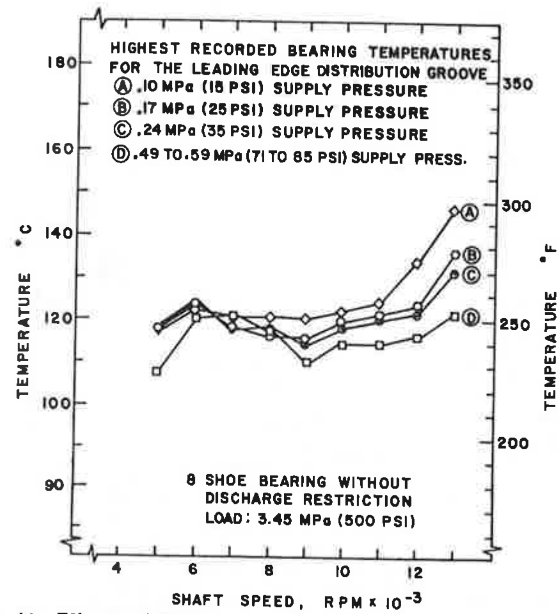


Fig. 11 Effects of various gage supply pressures on the highest recorded shoe babbitt temperature for an 8-shoe, leading edge distribution bearing operating without discharge restriction

will develop shoe surface temperatures that fall within a safe specified range, while temperatures that exceed this normal range of operation are indicative of a bearing in trouble.

The maximum operating temperature of a bearing is of the most interest to the user because of the limitations that temperature places on bearing operation. The high tin content babbitt used on the working surface of the bearing shoe loses its tensile and compressive strength, and is subject to creep at elevated temperatures. Also, high tin babbitt, being a non-cubic crystal, is subject to grain boundary dislocations when cyclically heated, which can cause the development of a nonflat bearing surface, with ripples of the order of oil film thickness [2].

Because the maximum bearing temperature is such a critical indicator of bearing performance, the maximum value of the more than 36 thermocouples installed in each bearing was selected to typify the temperature results reported here.

Figures 8 and 9 illustrate the influence that the various lubricant supply methods have on temperature performance of the 6-shoe thrust bearing, while Figs. 10 and 11 report similar temperature data for the 8-shoe thrust bearing. These temperature values are shown for constant load cases of 3.45 MPa (500 psi), at the flow rates shown in Figs. 3-7.

Figure 8 compares the three competitive bearing designs operating with the restricted discharge offered by the oil control ring. While for some test points the spray-feed bearing operates cooler than the flooded bearing, especially at the turbulent transition point of 9000 rpm, the distribution groove bearing shows a substantial improvement over the other two designs, an improvement that is evident across more of the speed range. A temperature advantage of 14°C (25°F) or more is possible. Figure 9 presents a similar comparison for operation without the oil control ring. Again, the distribution groove bearing (curves D-G) operates at a much lower temperature than the other two bearing designs (curve A for pressurized supply and curves B and C for spray feed). This figure also demonstrates the effect that reducing the supply pressure and flowrate will have on the distribution groove bearing. It is evident that the higher supply pressures and flows will offer improved temperature performance, even though a modest 0.10-0.14 MPa gage (15-20 psig) supply is perfectly adequate for a cool operating bearing.

Figure 10 demonstrates comparable results for the 8-shoe thrust bearing opearing without an oil control ring. While the spray-fed bearing with holes normal to the collar face operates at about the same temperature as the flooded bearing, the spray-fed bearing with angled holes offers a modest temperature reduction. However, the distribution groove bearing shows a marked temperature reduction across the entire speed range. Figure 11 demonstrates the deterioration in temperature performance that occurs as the supply pressure and flow is reduced incrementally for the distribution groove bearing. The external supply pressure was varied from a low of 0.10 MPa gage (15 psig) for curve A through 0.17 MPa gage (25 psig) for curve B, 0.24 MPa gage (35 psig) for curve C, and higher pressures (0.49-0.59 MPa gage or 71-85 psig) required for high flow conditions of curve D.

## Bearing Power Loss

Power loss values are computed by the familiar energy balance technique, whereby the loss is computed as a direct function of measured oil temperature rise (inlet to discharge), measured oil flowrate, and lubricant specific heat. Radiation losses from the housings and conduction losses via shafting and foundation are considered small and constant for this entire series of tests, and so are omitted from this analysis.

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. It has been stated that one of the aims of spray-feed or directed lubrication is to "eliminate the parasitic churning losses by only providing lubricant at the places it was required" [5]. The reduced flows (see Figs. 5, 6, and 7) associated with the spray-fed bearing are a result of the metering effect of the spray orifices, and the inevitable cause of the power loss reduction. A similar reduction in bearing power loss could be achieved for a conventional flooded thrust bearing by reducing the supply flow rate and accepting the higher operating temperatures that would naturally result

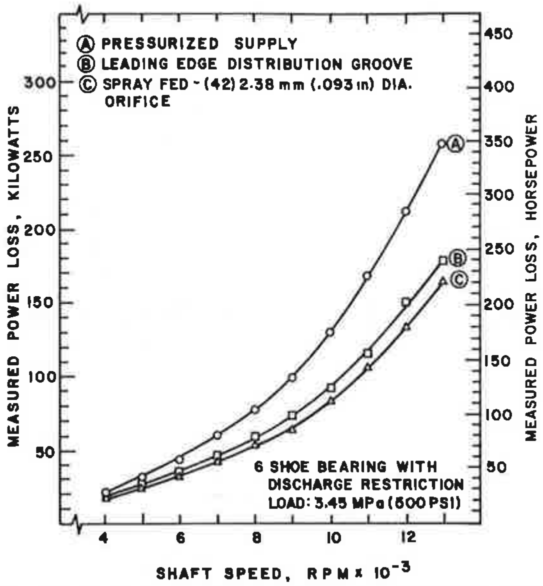


**Fig. 12 Effects of various lubricant supply methods on measured power loss for a 6-shoe bearing operating with a discharge restriction and with these oil supply pressures: pressurized supply @ 0.03 to 0.14 MPa; leading edge distribution groove @ 0.50 to 0.57 MPa; and spray feed @ 0.04 to 0.35 MPa**

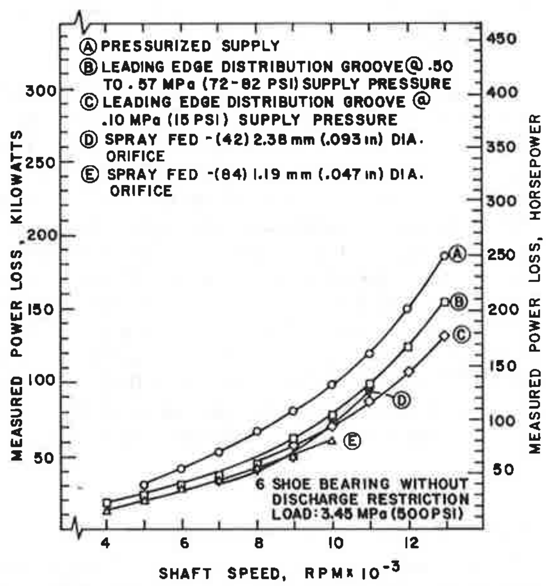


**Fig. 13 Effects of various lubricant supply methods on measured power loss for a 6-shoe bearing operating without a discharge restriction and with these oil supply pressures: pressurized supply @ 0.03 to 0.14 MPa; leading edge distribution groove @ 0.10 and 0.50 to 0.57 MPa; spray feed with (42) 2.38 mm dia orifices @ 0.03 to 0.27 MPa; and spray feed with (84) 1.19 mm dia angled orifices @ 0.09 to 0.38 MPa**

from that action. That, of course, would be an unacceptable trade-off because of the high temperature levels involved. Therefore, the ideal bearing design should first provide for satisfactory temperature performance, while at the same time, the oil flow rate consistent with this primary objective should be as low as possible to minimize power loss.

The leading edge distribution groove bearing flow rates, as illustrated by Figs. 3-7, are also lower than those associated with the pressurized flooded lubrication supply method. The reduced oil flow is possible because the meager requirements of the bearing's oil film, to a large extent control the amount of oil the bearing will accept.

Figure 12 illustrates the 6-shoe thrust bearing power loss values measured for each of the three designs, operating with

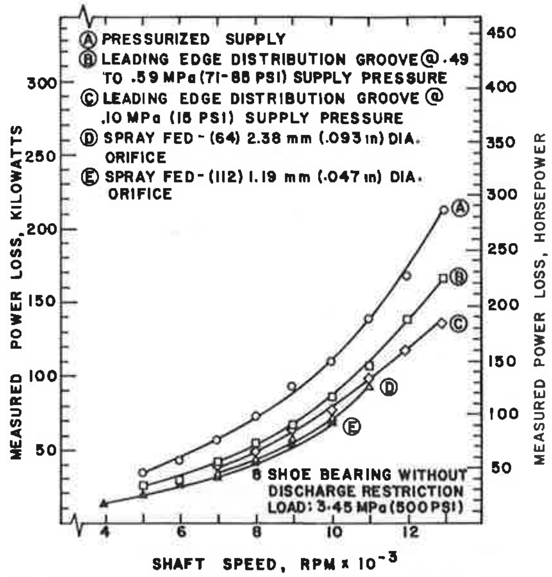


**Fig. 14 Effects of various lubricant supply methods on measured power loss for an 8-shoe bearing operating without a discharge restriction and with these oil supply pressures: pressurized supply @ 0.03 to 0.14 MPa; leading edge distribution groove @ 0.10 and 0.49 to 0.59 MPa; spray feed with (64) 2.38 mm dia orifices @ 0.10 to 0.24 MPa; and spray feed with (112) 1.19 mm dia angled orifices @ 0.03 to 0.17 MPa**

the restricted discharge (oil control ring). Reductions of 25-35 percent can be achieved for the reduced oil flow distribution groove and spray-fed bearings, compared to the conventional, full flow, flooded bearing. Figure 13 shows similar results for the same bearings operating without the oil control rings. That the power loss reduction parallels the reduction in oil flowrate is also demonstrated by the two curves for the distribution groove bearing, each for a different oil flowrate. Two curves have also been plotted for the spray-fed bearings to demonstrate the typical differences encountered for the different spray configurations discussed earlier.

Figure 14 shows the same type of results for the 8-shoe bearing design operating without an oil control ring. These curves corroborate the previous figures in that a substantial power loss reduction can be achieved by operating at reduced oil flowrates, with a reduction in operating temperatures for the two special bearing designs.

## Conclusions

The performance of these test bearings equipped with different lubricant supply methods has been evaluated on the basis of measured power loss and maximum measured shoe operating temperatures. While the optimum bearing should exhibit both the lowest babbitt temperatures and the minimum power loss values at the same time, none of the test bearing designs matched that ideal goal. However, substantial progress toward achieving that goal was evidenced in these tests.

Both the spray-fed and the leading edge distribution groove bearing demonstrated an ability to operate at power loss levels substantially lower than the conventional flooded thrust bearing. This is attributed to the elimination of churning and pumping losses, brought about by the reduction in oil supply flow rate. Such a flow reduction in a conventional bearing leads to higher operating temperatures, but the two specialized designs were not adversely affected.

The leading edge distribution groove bearing proved capable of operating over a very broad range of supply pressures and flows. This makes it possible to "fine tune" this bearing to achieve either minimal power loss or minimal operating temperature. The spray-fed bearing proved less adaptable, requiring a change in the number, diameter, or orientation of the jet holes to achieve flow or pressure variations that would result in power loss or temperature changes.

All three bearing designs were prone to shoe instabilities and flutter in the unloaded bearings when operating without a restriction to provide suitable backpressure at the discharge. This was especially true at shaft speeds below 7000 rpm. The instability manifests itself as an audible, metallic clicking emanating from the housings. Fatigue failure of instrumentation wires indicates actual movement of the unloaded shoes. This problem was eliminated by the use of the oil control ring with tangential discharge opening. Bearing operating temperatures were reduced as a result of using the oil control ring. However, the overall power loss was adversely affected by this discharge restriction.

The maximum measured shoe operating temperatures were found to be very responsive to the choice of lubricant supply method. The leading edge distribution groove proved superior at reducing maximum shoe temperatures. This is attributed to the positive introduction of cool supply oil directly into the oil film wedge between the hot oil carryover adhering to the high speed rotating collar and the shoe working surface. Test work is in progress to further refine and optimize the reduction of both operating temperature and power loss offered by the leading edge distribution groove bearing.

The conventional pressurized supply or flooded bearing makes no attempt to minimize the effects of hot oil carryover from the previous shoes, and thus demonstrates high operating temperatures. The spray-fed bearing could not possibly scour away a thin, tenacious oil film moving at high speed with the moderate supply pressures normally encountered in field installations. The slight reduction in operating temperatures obtained for the spray-fed bearing is more likely due to the rapid introduction of spray-cooled oil at the leading edge of the shoe.

It is hoped that the information herein presented will prove useful to the designers of rotating machinery. Thrust bearing selection should be based upon empirical evidence of superior performance, with an appreciation of the reasons behind performance variations.


## Acknowledgments

The facilities and personnel of Kingsbury, Inc., were utilized to perform these bearing tests, collect and process the data included in this paper, and prepare this manuscript for presentation. The gratitude of the authors is expressed to Kingsbury, Inc., for the opportunity to publish these results.


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## DISCUSSION

### R. C. Elwell[1]

The results in this paper are a testimonial to the quality of these authors' test work. However, I question their explanation for the reduced babbitt temperatures obtained with the leading edge distribution groove. This bearing geometry (Fig. 2) employs offset pivots, in contrast to the center-pivoted reference-bearings (Fig. 1). Our experience has been that offsetting the pivots reduces babbitt temperatures. Furthermore, pressurizing a feed groove at the leading edge of a pivoted pad generates a force tending to tilt the pad away from the thrust collar in a beneficial way.

There is another possible reason for improved babbitt temperatures that is not so easy to visualize. Nearly 50 years ago F. Ribary[2] performed experiments in which he took photographs of oil feed streamlines through a transparent thrust runner operating against a tilting-pad bearing. He found that admission of oil directly into the film, as in the case of the authors' leading edge groove, was the only way to get the benefit of fresh lubricant. He shows in an excellent set of photographs that spray bars, flood lubrication and scrapers are not effective, just as these present day Authors do.

### C. M. McC. Ettles[3]

The authors are to be congratulated on publishing these results which contain a definite improvement in the state of the art of thrust bearings.

The result of most interest is that the leading edge distribution groove gives a substantial decréase in shoe maximum temperature compared to the conventional fully flooded design. The leading edge distribution groove also shows an improvement over the spray feed method.

The authors refer to hot oil carry-over and conclude that a spray between the shoes "cannot possibly scour away a thin, tenacious oil film." The discusser has carried out tests in a fully flooded bearing with a spray bar fitted between two of the shoes. The jets were arranged to impinge directly on the runner surface. Jet velocities of over twice the sliding velocity failed to give away measurable improvement in the downstream shoe temperature. This result tends to agree with the authors' that the removal of the hot layer is most difficult.

The theory of thermal boundary layers can give some explanation for this. The temperature profile of the oil drawn across the groove is strongly dominated by the runner surface temperature. Even if the hot oil is actually removed (say by some scraping device), the specific heat and density of oil and steel are such that any cold oil applied over a small area has a negligible cooling effect. The very thin layer of oil due to enter the following shoe almost immediately heats to the runner surface temperature.

The introduction of cold oil directly into the film is clearly much more effective as the authors' resuls show. It is interesting that the beneficial effect increases with the flow rate of cold oil introduced. At the highest flow rate a large proportion of the cold oil may be flowing against the direction of sliding and deflecting the hot layer from entering the film.

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[2] Ribary, F., "Some Results of Tests Made With Segmental Thrust Bearings," *The Brown Boveri Review*, Vol. 20, July/August 1933, pp.119-122.

[3] Rensselaer Polytechnic Institute, Troy, N.Y. 12181.