Reprinted from the January 1976
Journal of Lubrication Technology

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Effects of High-Operating Speeds on Tilting Pad Thrust Bearing Performance

A comparison of the high-speed performance characteristics of tilting-pad, self-equalizing type thrust bearings through two independent full-scale programs is reported. This paper presents experimental data on centrally pivoted, 6-pad, 267-mm (10½-in.) and 304-mm (12-in.) O.D. bearings operating at shaft speeds up to 14000 rpm and bearing loads ranging up to 2.76 MPa (400 psi). Data presented demonstrate the effects of speed and loading on bearing power loss and metal temperatures. Included is a discussion of optimum oil supply flow rate based upon considerations of bearing pad temperatures and power loss values.

Introduction

This paper presents the experimentally measured values of bearing power loss and pad metal temperatures under variable load, speed, and oil supply flow rate. Information of this nature, especially in turbulent operation, has been very limited [1, 2, 3].[1] The data presented have been obtained from two completely independent test programs, which were developed on two dissimilar test machines. When compared, the overall bearing performance is consistent. This paper is intended to present new test data as a contribution to the better understanding of the phenomenon of turbulence in fluid film bearings.

The bearings under study were the conventional 6-pad, steel-backed, centrally pivoted, self-equalizing, 267-mm (10½-in.) and 304-mm (12-in.) O.D. tilting pad type illustrated in Fig. 1. The experimental data reported on the 267-mm (10½-in.) and the 304-mm (12-in.) bearings were collected on two independent test facilities.

The investigations studied the effects of external parameters such as applied load, shaft speed, and oil flow rate on the performance of tilting pad thrust bearings operating under simulated field conditions. Table 1 indicates the bearing sizes, areas, shaft speeds and applied loads. Bearing performance was evaluated by an analysis of pad metal temperatures and bearing power loss as influenced by variation of external parameters. A petroleum-based, light turbine oil with a viscosity of $32 \mu m^2/S$ at 38° C (150 SUS at 100° F) was used as a lubricant in both test programs.

The data herein represent only a portion of testing so far undertaken in the two research programs. A previous paper [2] has reported preliminary results on the 267-mm (10½-in.) bearing, and an additional paper [4] has been submitted relating to 432-mm (17-in.) and 381-mm (15-in.) bearings.

Test Apparatus

Bearing Description. The bearings involved in both test programs are geometrically similar (see Fig. 1). The bearings are 6-pad, self-equalizing tilting pad thrust bearings with sector shaped pads each subtending an arc of 51 deg. The steel pads are faced with a 1.6-mm (1/16-in.) thick layer of high tin babbitt (military specification QQ-T-390 Grade 2, ASTM-B23 Grade 2). The babbitted pads are positioned by a series of interlocking leveling links, which distribute the total applied thrust load equally among the pads. In order to achieve self-alignment during operation, each pad assembly incorporates a hardened steel pad support button on the back surface with a spherical radius at the point of contact with the leveling link. The pad inner babbitt diameter was 50 percent of the pad outer babbitt diameter. Additional data are described in Table 1.

For the portion of the experimental work relating to the 267-mm (10½-in.) bearing (Kingsbury, Inc.), the complete bearing assembly consists of active and inactive bearing elements, together with their individual oil seal rings plus a central oil discharge ring

[1] Numbers in brackets designate References at end of paper.

Contributed by the Lubrication Division and presented at the Joint Fluids Engineering and Lubrication Conference, Minneapolis, Minn., May 5–7, 1975, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Manuscript received at ASME Headquarters, February 19, 1975. Paper No. 75–LubS–6.

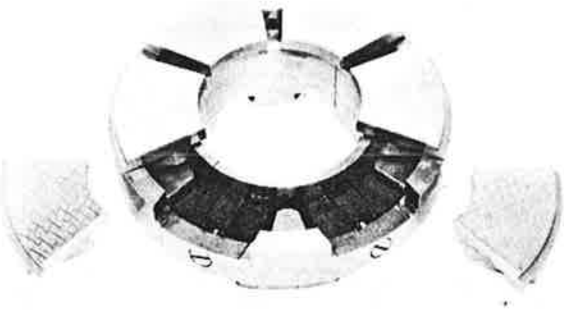

Fig. 1 Single element thrust bearing

Table 1 Bearing test parameters

BEARING OUTSIDE DIAMETER	BEARING SURFACE AREA	SHAFT SPEED RANGE	MAXIMUM MEAN VELOCITY	MAXIMUM APPLIED BEARING LOAD
mm / inch	cm^2 / $Inch^2$	RPM	M/Sec / Ft/Min	MPa / PSI
267 / 10½	358 / 55.5	4000-14000	146 / 28810	2.8 / 400
304 / 12.	465 / 72.	4000-13000	156 / 30631	2.3 / 400

(4-mm ($^5\!/_{32}$-in.) radial clearance) encircling the integral shaft collar. This bearing assembly is typical of many field installations. The reported power loss values apply to the complete bearing assembly (two bearings). The individual active and inactive bearing elements (as shown in Fig. 1) are identical in dimension and design.

All tests conducted on the 304-mm (12-in.) thrust bearing (MDTPD, General Electric Company) were of a single element bearing type. This was done in order to develop performance data on each individual size of bearing over a range of operating conditions. Since it is normal practice to use two opposing thrust bearings to axially locate the shaft in rotating machinery, the available performance data will allow the maximum flexibility in optimizing the selection of each bearing independently. It is important to note that oil control rings were not employed during the 304-mm (12-in.) test series.

Mechanical Arrangement. Since the test data were collected from two independent programs, the test machines used were somewhat different in design.

The 267-mm (10½-in.) bearing test facility is fully described in reference [2]. The test rig used to obtain the 267-mm (10½-in.) thrust bearing data consists of two identical thrust bearing assemblies installed in separate housings, but sharing a common shaft. One housing is fixed to the foundation while the other is free to slide axially. By applying load to the free end of the sliding housing, it is possible to load both bearings in a symmetric fashion. The amount of applied load is measured hydraulically and electronically by external and internal strain-gage load cells.

The 304-mm (12-in.) bearing test facility is fully described in reference [4]. The major components of the test machine were two journal bearing housings, two thrust bearing housings, supporting structure, test shaft, and a loading mechanism. The input force was applied to the "facility" thrust bearing through an external hydraulic system. The facility bearing is free to slide axially, thus loading the shaft collar and in turn the test bearing. As a result of a single applied load, both thrust bearings experienced the same load simultaneously. This duplication of bearing loading not only permitted the testing of two identical thrust bearings as a check on test repeatability, but also allowed the testing of two dissimilar size bearings under identical operating conditions. Calibrated mechanical extensometers at the load yoke end of the piston rods served as a check on the hydraulic system to insure that the full load was being applied, and to insure that each piston carried its share of the load in a totally symmetrical fashion.

Instrumentation. The instrumentation incorporated into the two independent test programs was somewhat different. The 267-mm (10½-in.) test bearings were instrumented with a variety of transducers including strain-gage load cells and proximity probes. One pad of the test bearing was equipped with an array of nine thermocouples embedded in the babbitt metal and arranged as shown in Fig. 2. This pattern of thermocouples was used to detect temperature gradients in both the radial and circumferential directions.

The instrumentation incorporated into the 304-mm (12-in.) test bearing consisted of thermocouples installed at radial locations along both the leading and trailing edge of the pads as well as embedded into the babbitt surface at various radial and circumferential coordinates. Each pad of the thrust bearings tested was equipped with a 75-percent radial–75-percent circumferential ($T_{R75-C75}$) thermocouple location. Fig. 3 illustrates the instrumentation locations on the 304-mm (12-in.) bearing.

Certain types of instrumentation were common to both test facilities. Calibrated flowmeters were installed in the oil supply piping to measure the amount of lubricant supplied to each test bearing. In each case, the bearings were supplied with varying amounts of light turbine oil at a constant inlet temperature. For the 267-mm (10½-in.) and 304-mm (12-in.) bearings the oil inlet temperatures were held constant at 46° ± 0.5° C (115° ± 1° F) and 49° ± 0.5° C (120° ± 1° F), respectively. This paper will discuss the data obtained from iron-constantan thermocouples used to monitor the oil supply and discharge temperatures as well as pad babbitt metal temperatures.

Test Procedure. The area of investigation reported in this paper centered on three independent variables for each test point: applied bearing load, shaft speed, and oil flow rate. Prior to the test, the required lubricant supply rate was determined for each load and speed in accordance with conventional design practice. These values constituted the "nominal" flow rates.

Since the 267-mm (10½-in.) testing was conducted on a double bearing assembly, a distinction must be made between the oil supply flow rate to the active and inactive elements. Typical values of nominal flow rate and the distribution of oil flow between active and inactive elements, for the 267-mm (10½-in.) test bearings, are shown in Fig. 4. Fig. 5 illustrates the nominal flow rate used for the 304-mm (12-in.) test bearing. Only one value of flow rate is required since only single element bearings were tested.

The normal test procedure called for additional test points at oil supply rates 50 percent above and 50 percent below the nominal value. Thus, each load and speed point included oil flow rates of 50, 100, and 150 percent of the nominal value. (At some of the higher speeds conducted on the 267-mm (10½-in.) test bearing it was not possible to attain a true 150-percent flow rate because of limited pump capacity. In fact, at 13,000 and 14,000 rpm, it was necessary to set oil flows at 50, 75, and 100 percent of the nominal to obtain three distinct test points.) Data were collected at each test point only after the bearing load, shaft speed, and oil flow rate were determined in accordance with the preestablished test schedule.

A unique value of bearing power loss was experimentally determined for each flow rate condition tested. The bearing power loss was computed for the thrust bearing by an energy balance method based upon the measured oil flow, measured temperature rise, and lubricant specific heat. The total power loss associated with an operating thrust bearing of this type can be attributed to shear in the oil film between the shaft collar and the pads, churning in the region between the shaft collar and oil discharge ring (if present), seal ring frictional loss, and other lesser parasitic losses. The total loss manifests itself as an increase in bulk oil temperature as the lubricant flows through the bearing, by virtue of the work performed by the bearing and its accessories. Thus, a measurement of oil temperature at the inlet and drain and a known oil flow rate are sufficient to compute the increase in energy level for the lubricant and equate it with the bearing power loss. Radiation from housing surfaces, conduction by the shaft and other minor energy losses are ignored in this analysis since they are small and constant values.

Test Results

In the discussion of test results, the presentation of the observed general overall performance characteristics of the subject tilting pad thrust bearings will be emphasized. Although the bearings dif-

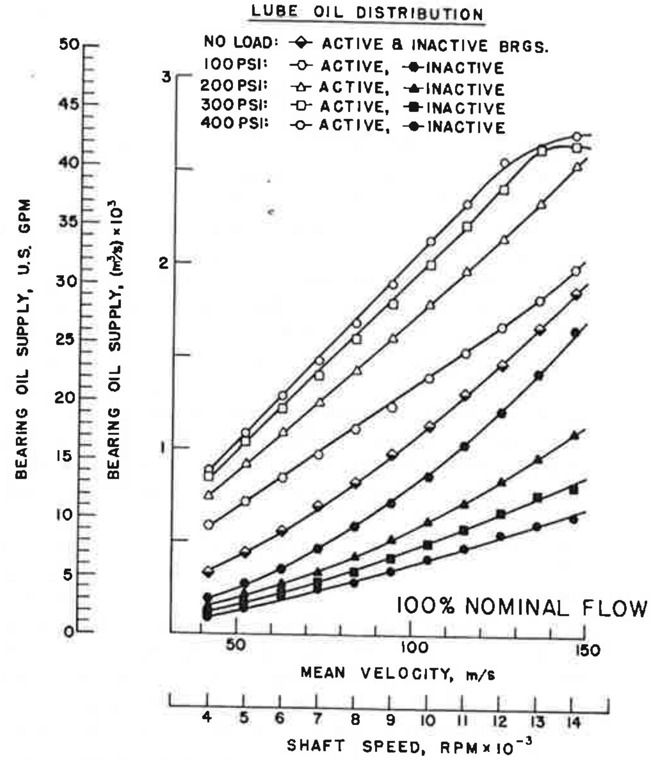

Fig. 4 Lube oil distribution to active and inactive elements of 267-mm (10½-in.) bearing

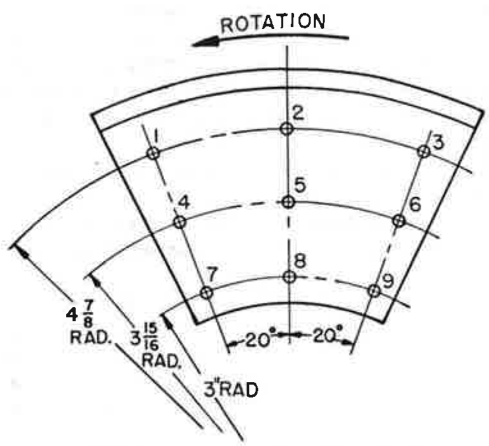

Fig. 2 Thermocouple locations on the 267-mm (10½-in.) bearing

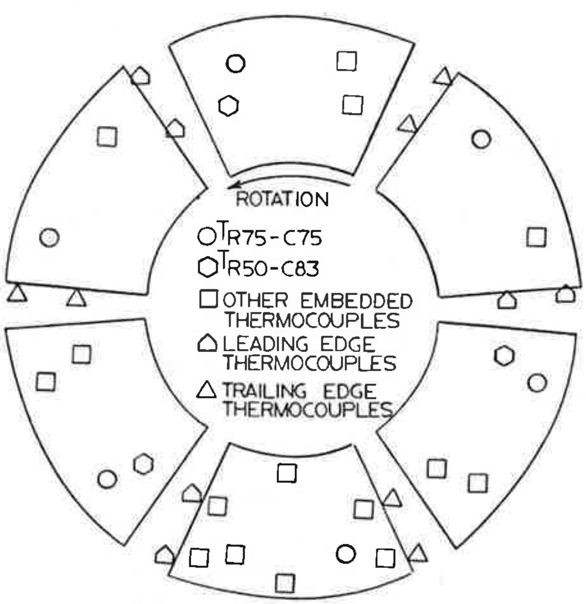

Fig. 3 Bearing instrumentation on the 304-mm (12-in.) bearing

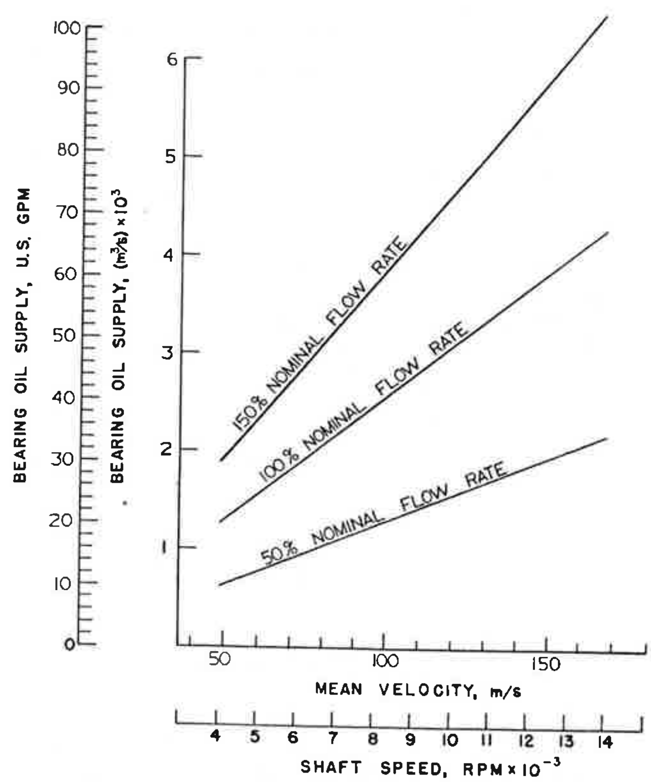

Fig. 5 Lube oil supplied to the 304-mm (12-in.) bearing

fer slightly in design details and the operating performance characteristics were obtained on different test machines, the results are consistent.

Bearing Temperatures. During the 267-mm (10½-in.) bearing test program, the maximum babbitt temperature location was found to occur at the trailing edge of the pad along the middle radius (location 4 of Fig. 2—$T_{R50\text{–}C89}$) in almost all instances. Typical temperature behavior for the nine thermocouples of Fig. 2 is shown in Fig. 6 for 9000 rpm and the entire load range. Although the 75 percent–75 percent thermocouple location ($T_{R75\text{–}C75}$) was generally observed to be the maximum metal temperature during the 304-mm (12-in.) bearing test program, comparative data were obtained for other locations. Fig. 7 represents a comparison of the 75 percent–75 percent location ($T_{R75\text{–}C75}$) and a secondary location ($T_{R50\text{–}C83}$) near the trailing edge at the pitch line of the pad. Since there is little temperature difference between these two thermocouple locations over a wide speed and load range, the $T_{R50\text{–}C83}$ location will be used for direct comparison to the thermocouple readings reported for the 267-mm (10½-in.) test bearing discussed in the foregoing.

Figs. 6 and 7 represent the response of pad metal temperature as a function of load for several shaft speeds and locations. Figs. 8 and 9 depict the variation response of the pad metal temperature with shaft speed, for several loads, for the 267-mm (10½-in.) and 304-mm (12-in.) test bearings, respectively. Fig. 9 also includes the response of the oil discharge temperature as a function of speed, for several loads, for the 304-mm (12-in.) test bearings.

In comparing the characteristics of the pad metal temperature and oil discharge temperature (Fig. 9), it is possible to note the corresponding increase in temperature levels as a function of speed. It is understandable from this observation, how the erroneous practice of using the oil discharge temperature as a relative indication of the pad metal temperature could be adopted. Figs. 10 and 11 illustrate these temperatures as a function of load at constant speeds for the 267-mm (10½-in.) and 304-mm (12-in.) test bearings, respectively. The constant speed condition is generally accepted to be the mode of operation for most rotating machinery design work.

Contrast the relative insensitivity of the oil discharge temperatures with the response of the babbitt thermocouples to changing load conditions. This shows that the oil discharge temperature is not a suitable criterion for bearing monitoring. The effect that lubricant flow rate has on the pad metal temperature is shown in Figs. 11 and 12 for the 267 mm (10½-in.) and 304 mm (12-m.) test bearings, respectively. In the past, attempts have been made to specify oil flow rates based upon an expected or desired temperature rise in the bulk oil flow rate or in a desired bearing power loss. Fig. 13 illustrates the small relative change in metal temperature in increasing the oil flow rate 50% compared to the dramatic increase in metal temperature in decreasing the oil flow rate 50%. This illustrates that the quantity of oil used should be optimized to achieve the lowest maximum pad metal temperature with respect to the amount of oil supplied. Figs. 11, 12, and 13 all demonstrate the optimization of the nominal oil flow rate supplied to the test bearings in both laminar and turbulent flow regimes.

Bearing Power Loss. Typical values of measured power loss for the 267 mm (10½-in.) and 304 mm (12-in.) test bearings are depicted in Figs. 14 and 15. These figures represent data for the entire range of bearing loads and shaft speeds and for nominal flow rates. It is important to note that the power loss levels reported for the 267 mm (10½-in.) test bearing represents the loss of a complete assembly of an active and an inactive bearing element, together with their individual oil seal rings plus a central oil discharge ring 4 mm (5/32-in.) radial clearance encircling the integral shaft collar.

In contrast, the power loss levels reported for the 304-mm (12-in.) test bearing are for a single element bearing. The bearing power loss appears to be relatively constant as the bearing load is increased. Variation with shaft speed is much more evident (Figs. 14 and 15).

Figs. 16 and 17 illustrate the effect of varying the oil flow rate on bearing power loss at 2.07 MPa (300 psi) load for the 267-mm (10½-in.) and 304-mm (12-in.) test bearings, respectively. Substantial economies in bearing power loss can be achieved if the bearing

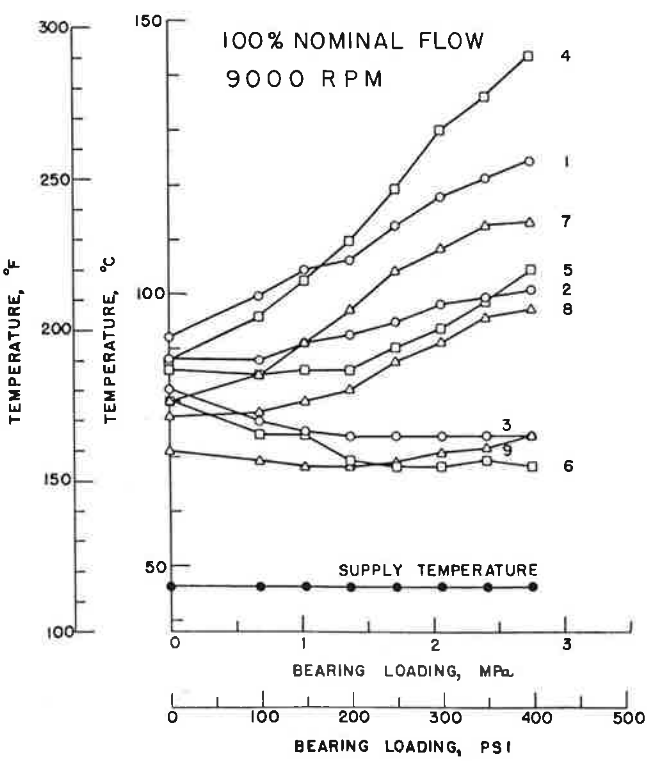

Fig. 6 Temperature distribution across babbitt of 267-mm (10½-in.) pad at 9000 rpm

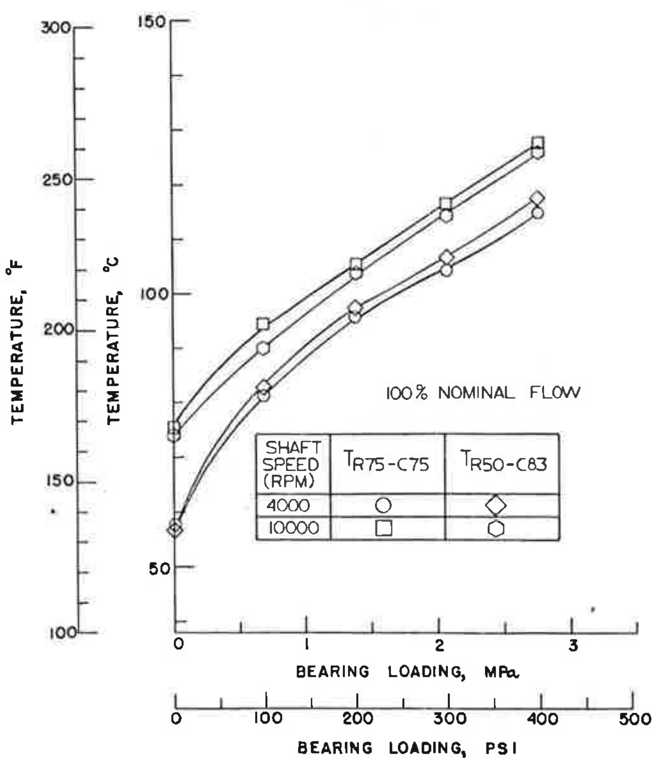

Fig. 7 Effect of load and shaft speed on pad babbitt temperatures—304-mm (12-in.) bearing

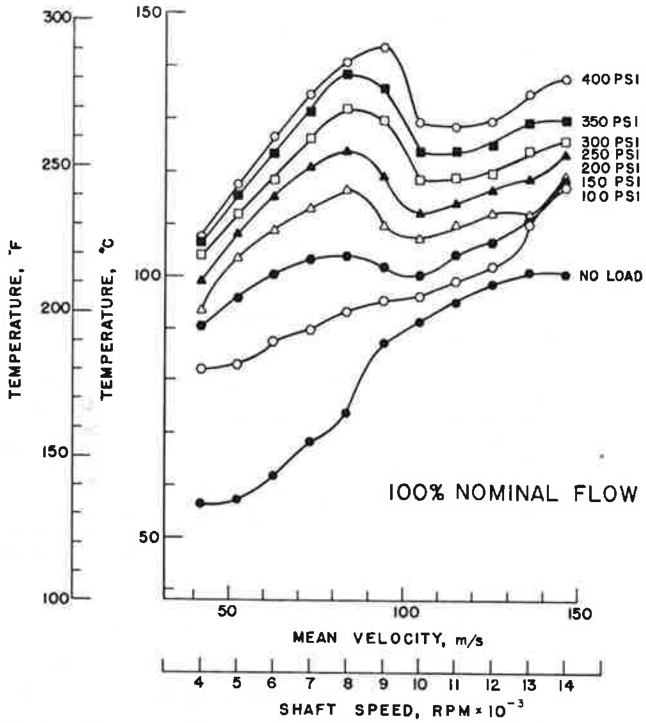

Fig. 8 267-mm thrust bearing maximum babbitt temperatures at pad location 4 ($T_{R50\text{-}C89}$)

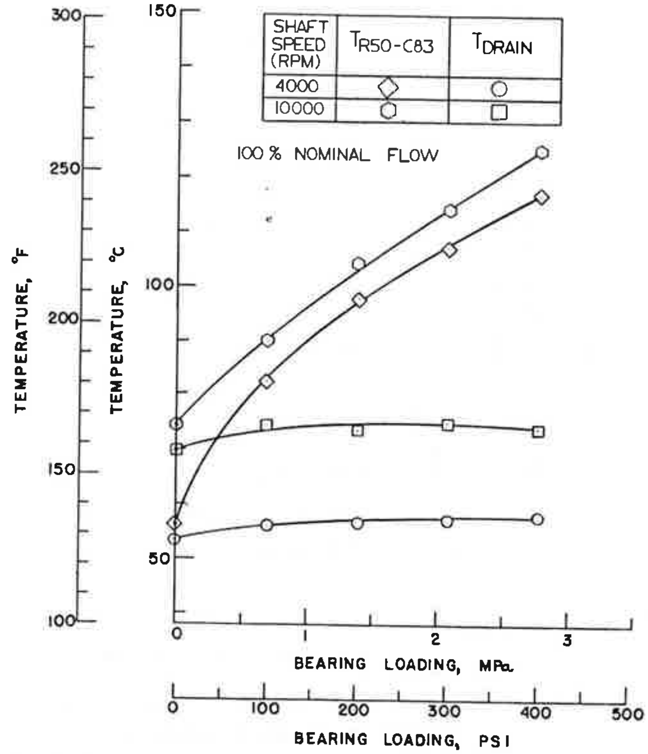

Fig. 10 Effect of load and shaft speed on pad babbitt temperatures—304-mm (12-in.) bearing

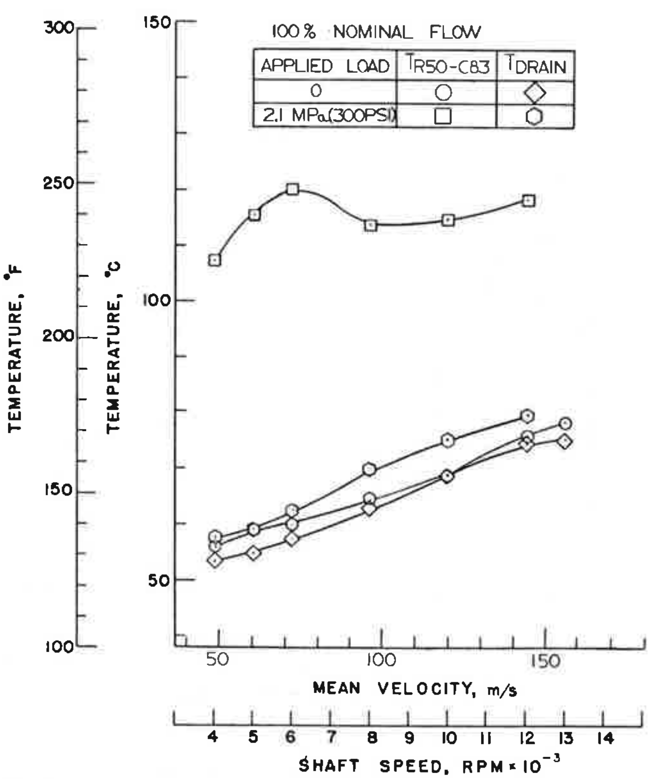

Fig. 9 Effect of shaft speed and load on pad babbitt temperatures—304-mm (12-in.) bearing

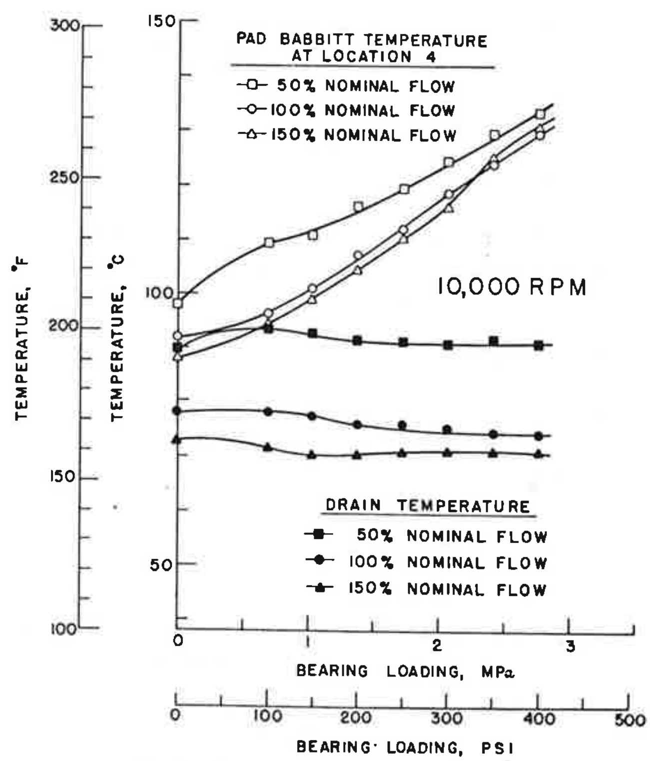

Fig. 11 Effect of lube oil flow rate on 267-mm (10½-in.) bearing temperatures at 10000 rpm

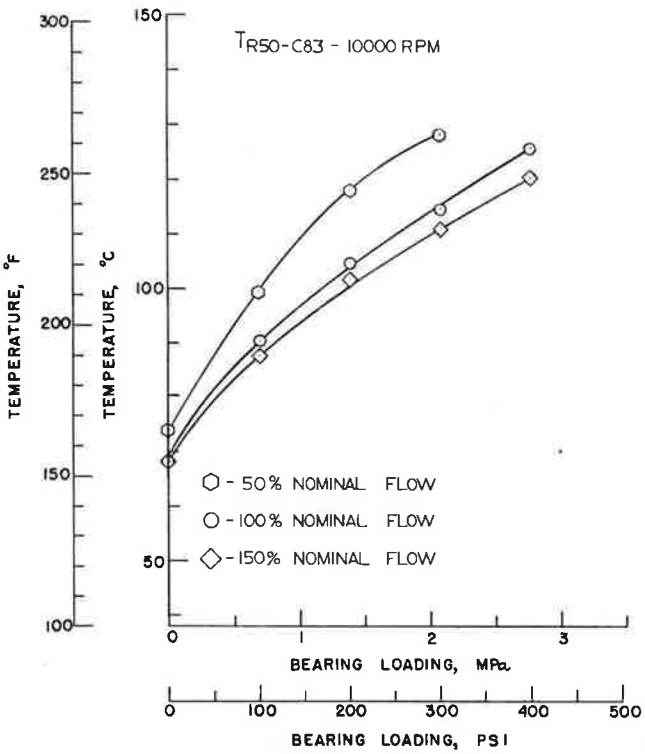

Fig. 12 Effect of load and lube oil flow rate on pad babbitt temperatures—304-mm (12-in.) bearing

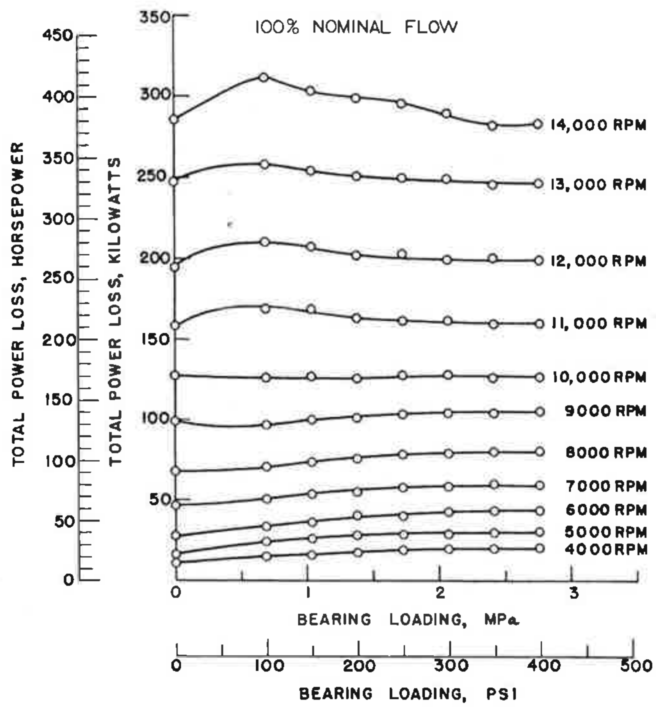

Fig. 14 Effect of load and shaft speed on 267-mm (10½-in.) bearing total power loss

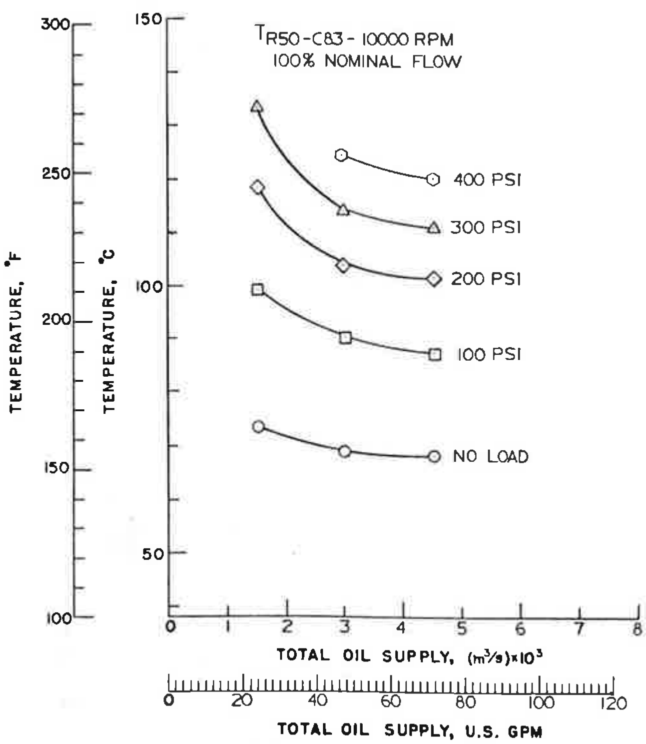

Fig. 13 Effect of lube oil flow rate and load on pad babbitt temperatures—304-mm (12-in.) bearings

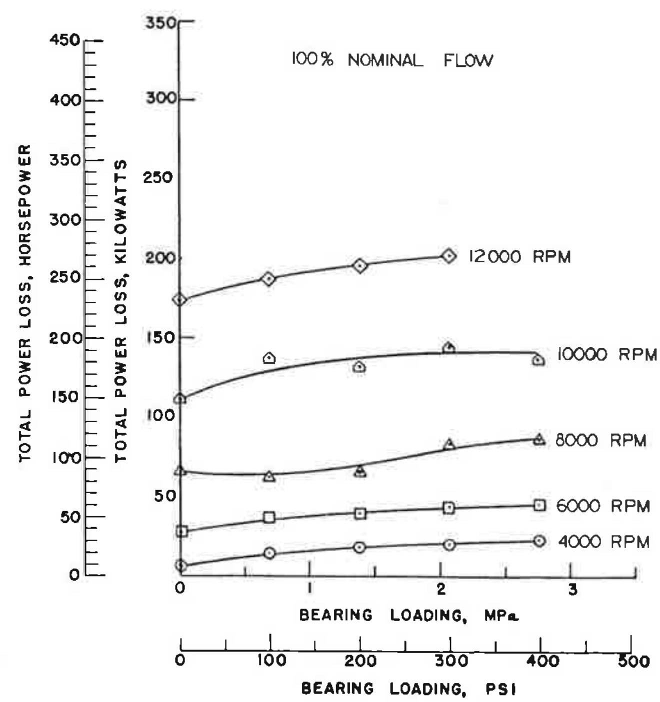

Fig. 15 Effect of load and shaft speed on 304-mm (12-in.) bearing total power loss

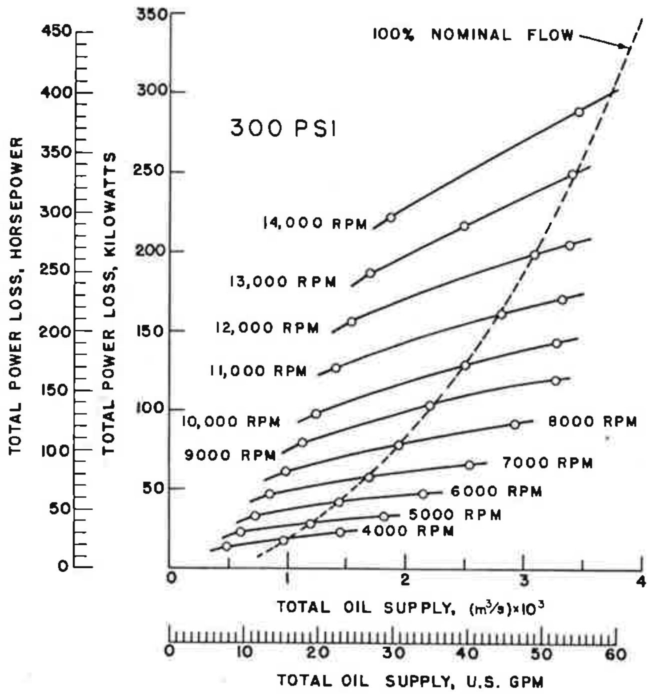

Fig. 16 Effect of lube oil flow rate on 267-mm (10½-in.) bearing total power loss at 2.07 MPa (300 psi)

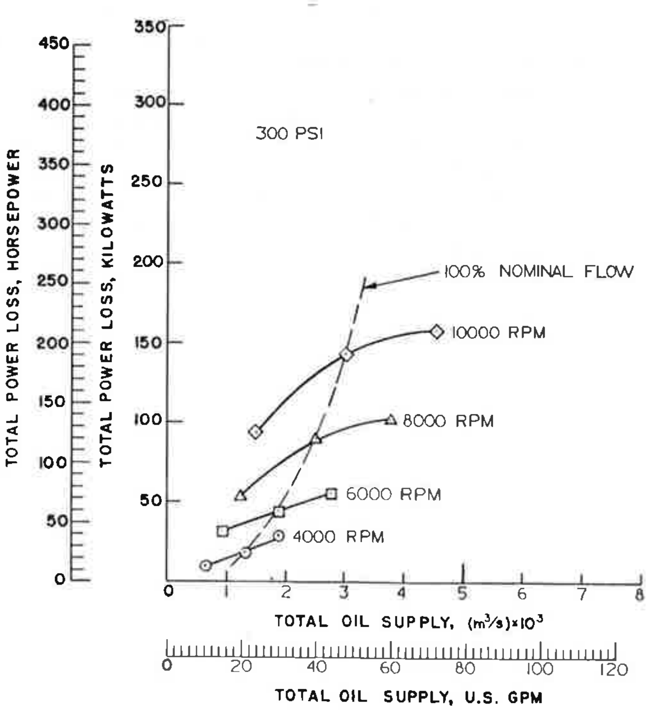

Fig. 17 Effect of lube oil flow rate on 304-mm (12-in.) bearing total power loss at 2.07 MPa (300 psi)

metal temperature will permit a reduction in oil flow rate.

The Laminar-Turbulent Region. The transition from laminar to super-laminar or turbulent operation was carefully studied as a point of major interest. In examining Figs. 8 and 9, the response of the pad metal temperature to load and speed variations is shown. With increasing speed at the higher load conditions, a temperature peak is attained, followed by a dramatic decrease in temperature. After a quiescent zone, the temperature begins to climb rapidly. This temperature behavior is associated with the transition from laminar to the turbulent flow regime in the oil film. In a deductive manner, this decrease in pad metal temperature confirms the common hypothesis [5, 6] that an increase in load capacity will accompany the onset of fluid film turbulence. This lower operating pad metal temperature allows operating the bearings at a higher load in this speed range if the design criteria is the maximum pad metal temperature.

This decrease occurs in the higher temperature range, 120–150° C (250–300° F) and will not significantly increase the practical application range of this type of tilting pad bearing.

A second phenomenon attributed to the onset of fluid film turbulence is a dramatic increase in bearing power loss at the higher speeds. This is apparent when Figs. 14 and 15 are examined.

Conclusions

The limited test data sample presented here is sufficient to show the interrelationship that exists between bearing temperature and power loss and the parameters of load, speed, and oil flow rate. The experimental data presented can be further used for comparison with present analytical data for the prediction of tilting pad thrust bearing performance.

Data have been presented which show that bearing power loss is not appreciably affected by load changes at constant speed. Power loss for a given bearing is largely a function of oil flow rate and most dramatically a function of operating shaft speeds.

There is strong evidence that a reduction in oil flow rate will result in a beneficial power loss decrease. However, this apparent gain is offset by the resulting higher pad metal temperature and the determining factor would be whether the penalty in increased pad metal temperature can be tolerated. Figs. 11, 12 and 13 are representative of this oil flow rate change.

In contrast, increasing the oil flow rates (as is often recommended for the cooling of hot running bearings) by 50 percent may not significantly improve the pad metal temperature. It therefore becomes apparent that there exists an optimum oil supply flow rate based upon the bearing power loss and pad metal temperature.

Acknowledgments

The facilities and personnel of Kingsbury, Inc., and the Mechanical Drive Turbine Products Department, General Electric Company, were utilized to perform these bearing tests and to gather the data presented in this paper. The authors wish to express their gratitude for permission to publish these results.

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D. F. Wilcock[2]

This paper provides a welcome addition to the available experimental data on tilting pad thrust bearings. Particularly useful is the author's demonstration of the lack of correlation between oil outlet temperature and bearing pad temperatures. The simplicity of a relationship between the two has tempted machine designers and users for many decades, and still does, despite analytical and experimental studies that have shown the contrary. Tilting pad thrust bearings are particularly subject to this lack of correlation since lubricant flow is not controlled by the bearing geometry but by inlet orifices or outlet seals.

The value of experiments of the type reported would be very significantly increased, from an engineering point of view, if some critical measurements were added. In particular, film thickness at several points on a pad, and circumferential force on a pad, would permit comparison with some of the analytical predictions which should be available to the authors. They could then begin to quantify the churning losses in a type of bearing which is designed to run immersed in lubricant, a loss which is becoming increasingly important as bearings are run at higher Reynolds' numbers in large power machinery.

W. W. Gardner[3]

As portions of this paper present tilting pad thrust bearing operating data from the same bearing test facility as that used for the tests reported in LubS-7 (a companion paper at this conference), several of my comments there apply here also, but will not be repeated. The data presented in this paper on two additional bearing sizes of similar geometry and construction adds to the field of knowledge in this specialized area and will aid in the design of machinery using such bearings.

The instrumentation of the 267 mm (10½ in.) bearing is noted to include proximity probes, presumably for film thickness measurements. The effects of turbulence on bearing temperature and power loss are presented. In this respect, would the authors comment on what corresponding effects were noted in film thickness measurements?

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The data presented in Figs. 6 and 7, comparing pad temperatures at various locations on the pad face, are helpful in establishing a location for one thermocouple in a pad (as often done in bearing applications) as a monitor on bearing operation. The discusser is familiar with test data on similar type bearings, but at lower speeds (laminar operation) where the temperature at location #5 of Fig. 2 increased rapidly and to a level higher than that at location #4 just prior to bearing failure. This does not appear to be the situation in Figure 6, although the extent to which the load could be increased without failure is not known. Can the authors expand on this point in view of their test results?

Authors' Closure

The comments of Dr. Wilcock and Mr. Gardner regarding the content of this paper are appreciated by the authors. Unfortunately, space limitations would not permit the inclusion of all the data collected during these bearing tests. Measured oil film thickness values were omitted because the proximity probes were used only for the 267 mm (10½ in.) bearing tests so that comparison of the two different bearing sizes was not possible. In general, the data available for the 267 mm bearing shows no detectable increase in film thickness due to the onset of turbulence, although the anticipated variations with shaft speed and applied load are readily apparent. No measurements of actual pad circumferential force, as such, were made in either test program.

The phenomenon of peak pad temperature occurring at pad center (location #5) just prior to failure as described by Mr. Gardner was not encountered during this series of tests. The measured peak pad temperature was always closer to pad trailing edge than pad center. However, it is apparent from the data presented in this paper that the actual location of the peak pad temperature is dependent on the load and speed parameters, and it will change as the operating conditions are varied. It is entirely possible that a heavily loaded pad will crown sufficiently to force the peak temperature location back to the pad center (location #5) just before failure.