

IMPACT OF RECENT TILTING PAD THRUST BEARING TESTS ON STEAM TURBINE DESIGN AND PERFORMANCE

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programs at General Electric and Kingsbury. Details are presented of a large thrust bearing test facility at the General Electric Mechanical Drive Turbine Products Department capable of testing bearings at speeds up to 14,000 rpm and loadings up to 60,000 pounds. Results of tests on bearings ranging in size from 12 inches to 17 inches are summarized along with the important conclusions relative to power loss, load-carrying capability, and oil flow optimization. Examples are presented to demonstrate the application of the new design data and its effect on the design of mechanical drive steam turbines.

INTRODUCTION

For variable speed operation, tilting pad thrust bearings are advantageous compared to conventional tapered land bearings. The pads are free to pivot to form a proper angle for lubrication over a wide speed range. The self leveling feature equalizes individual pad loadings and reduces the sensitivity to shaft misalignments which may occur during service. Recognizing these advantages, the authors' company in 1967 adopted the use of tilting pad thrust bearings for large mechanical drive steam turbines.

The growing demand for reliable rotating machinery capable of higher horsepower and higher speeds has significantly stretched the application of thrust bearings. Today's turbine designer faces the problem of trying to extrapolate data from a bearing vendor's catalog, which is usually intended for general use and is in most cases based on a laminar flow analysis, and account separately for the influence of turbulence in the design of bearings. If the designer used only catalog data, he runs the risk of underestimating the horsepower losses and lube oil flow rates and overestimating the load capacities under actual operating conditions. Complete understanding of these factors is fundamental to reliable turbine design. In addition to bearing reliability there are effects on turbine efficiency and accessory equipment.

Although there exist numerous references in the literature pertaining to laminar flow operation of tilting pad thrust bearings (1-10), there is a scarcity of reported experimental data on their superlaminar performance (11-13). The effects of turbulence and the need for reliable test data have long been recognized within the authors' company. In 1968 an extensive full-scale analytical (14-15) and experimental (16-17) program was initiated to investigate the effects of turbulence on the performance characteristics of fluid film bearings. A special tilting pad thrust bearing test machine was designed in conjunction with Worcester Polytechnic Institute.

ABSTRACT

Recent test programs on Kingsbury type tilting pad thrust bearings have altered application data and required the use of larger thrust bearing sizes with correspondingly larger oil flows and increased power loss. These results have had a significant impact on the design and performance of mechanical drive steam turbines, especially for high horsepower applications where thrust bearings operate in the superlaminar or turbulent flow regime. The authors cite data obtained from separate test

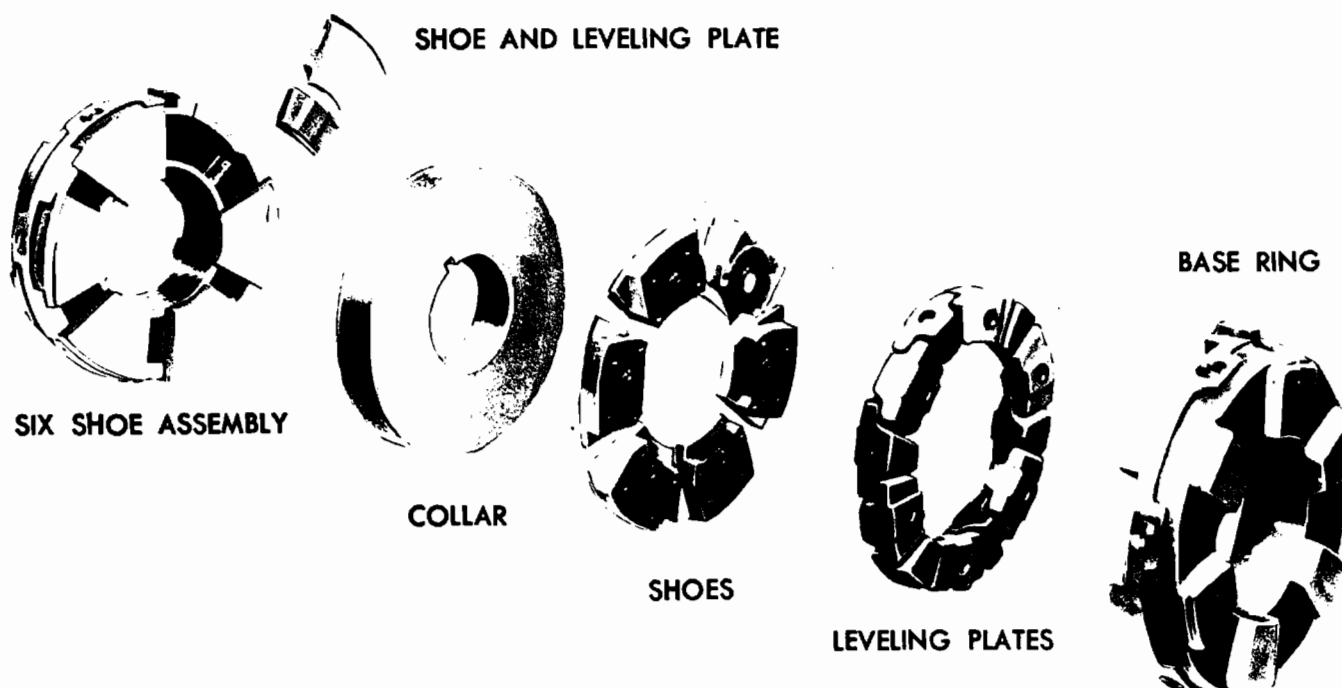


Figure 1. Typical Double Element Thrust Bearing.

A series of full-scale, superlaminar flow, tilting pad thrust bearing tests were conducted in this test facility. Test programs on the Kingsbury type, Fig. 1, tilting pad thrust bearings have altered application data and led to the use of larger thrust bearing sizes with larger oil flows and increased power loss. Table 1 indicates the bearing sizes, areas, shaft speeds, and applied loads.

TEST APPARATUS

Bearing Description

The tests reported in this paper were all conducted on geometrically similar tilting pad thrust bearings. Full-scale, 6-pad, steel-backed, centrally pivoted, self-equalizing bearings were investigated. The pad inner babbitt diameter was 50 percent of the pad outer babbitt diameter. A petroleum based, light turbine oil with a viscosity of 150 SSU at 100° F was used as a lubricant for the tests presented. Additional information is given in Table 1 and the typical thrust bearing construction tested is illustrated in Fig. 1.

Mechanical Arrangement

A schematic overview of the high speed thrust bearing test facility is shown in Fig. 2. The test facility is fully described in reference (16). The major components are: a single-stage steam turbine drive, a speed step-up gear box, and the thrust bearing test rig.

Instrumentation

The instrumentation installed in the test bearings consisted of thermocouples installed at radial locations along both the leading and trailing edges of the pads as well as embedded into the babbitt surface at various radial and circumferential coordinates. Although each bearing was instrumented in a somewhat different manner, all were instrumented with the same general philosophy in mind. Each pad was equipped with the 75 percent radial-75 percent circumferential (TR75-C75). In addition a variety of secondary locations, in close proximity to the

TABLE 1

Bearing Test Parameters

Bearing Outside Diameter inch	Bearing Surface Area inch ²	Shaft Speed Range RPM	Maximum Mean Velocity Ft/min	Applied Load Range PSI
17	144.5	1,000-9,000	30,041	0-400
15	112.5	2,000-10,000	29,452	0-500
12	72	4,000-13,000	30,631	0-500

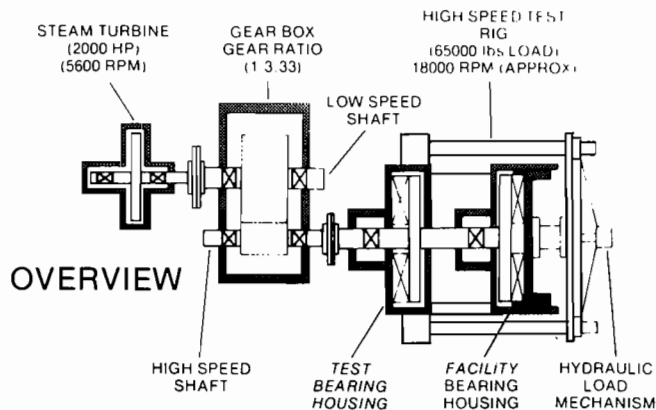


Figure 2. Schematic of Thrust Bearing Test Apparatus.

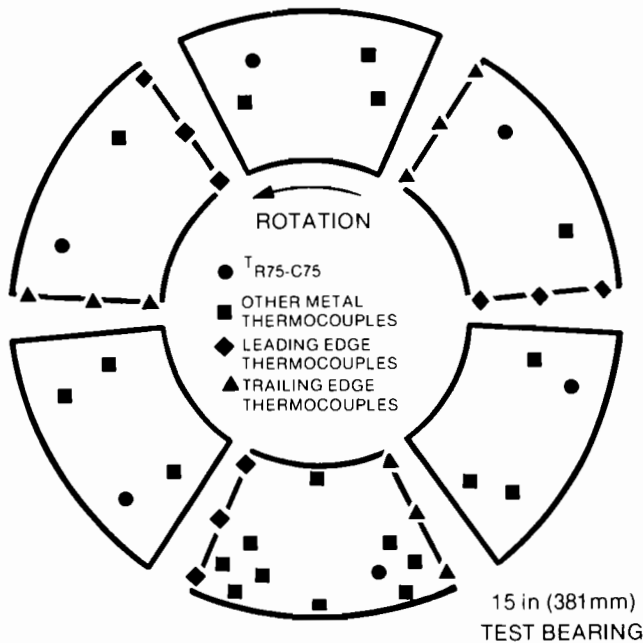


Figure 3. Typical Test Bearing Instrumentation.

TR75-C75 thermocouple were installed to insure that the maximum pad metal temperature would be measured. Fig. 3 illustrates the thermocouple locations indicative of this practice on the 15 inch test bearing.

Each of the oil supply and discharge lines was equipped with a thermocouple and a pressure gage. A rotometer type flow meter was also installed in the test bearing lube oil supply line. In addition each bottom oil discharge drain line was supplied with a throttle valve. It was this throttle valve that allowed the testing of the bearings in a flooded condition (bottom drain closed) or in an evacuated condition (bottom drain open).

Test Procedure

The area of investigation during the testing program centered on four basic variables: shaft speed, applied load, lube oil flow rate, and evacuated versus flooded bearing chamber. Table 1 indicates the minimum and maximum speeds and loads at which each size bearing tested. It was determined that the temperatures were repeatable to within $\pm 2 - 5^\circ\text{F}$ in the range from $150-300^\circ\text{F}$. The oil supply to the bearings was held constant at $120^\circ\text{F} \pm 1^\circ\text{F}$. Test results reported herein will be confined to the evacuated (bottom drain open) configuration only. The evacuated (bottom drain open) mode of operation results in lower bearing metal temperatures and power loss consumption.

TEST RESULTS

Bearing Power Loss

It is important for the designer of large mechanical drivers to be able to accurately assess bearing power loss. These losses must be accounted for in the prediction of turbine efficiency and in the design of the associated oil supply system. The increasing cost of energy coupled with a rapid growth in the power output of modern mechanical drive turbines have greatly increased emphasis on efficiency levels. Oil systems must be properly sized in terms of pumping and cooling capacity. The test data generated in the recent test programs (16, 17) supplied

TOTAL POWER LOSS

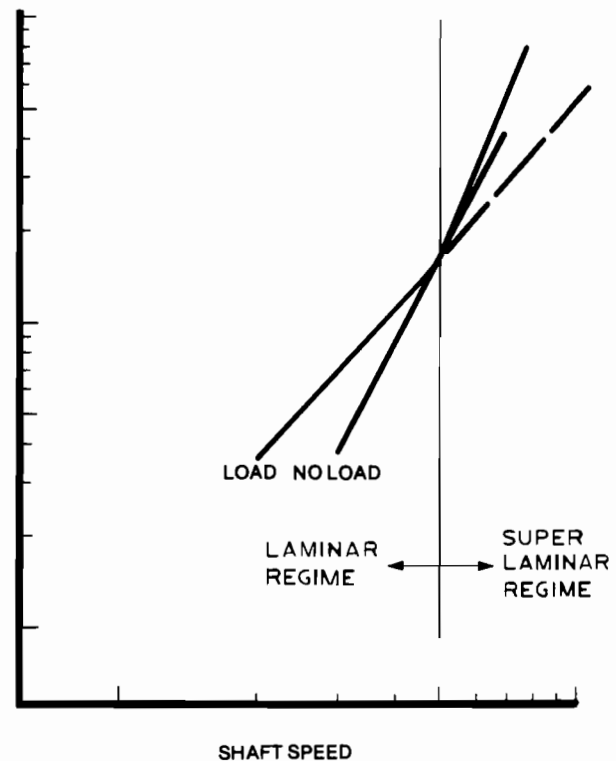


Figure 4. Effect of Load and Shaft Speed on Total Power Loss.

much needed application data on thrust bearing power loss, especially for bearings operating in the superlaminar flow regime.

Fig. 4 shows typical test results for a single element bearing, which illustrate some basic conclusions. The power loss is seen to be somewhat load dependent at the lower speeds (laminar flow), but essentially independent of load at higher speeds (superlaminar flow). Mechanical drive turbines utilize a double element bearing capable of accepting thrust loads in either direction, and losses on the unloaded as well as the loaded sides must be considered. The data shows that in the superlaminar flow regime the losses on the loaded side and slack side are equal. Furthermore, the losses increase much more rapidly with speed above the transition point.

Bearing manufacturers' catalog data gives power losses based on frictional losses in the oil film for laminar flow. Other kinetic losses, such as windage and oil churning, are excluded. The catalog data for laminar flow gives losses for both the loaded and slack sides. The slack side loss is a small percentage of the loaded side loss.

Figs. 5, 6, and 7 compare test data on power loss with catalog data for 17-inch, 15-inch, and 12-inch bearing sizes, respectively. The catalog data understates the actual power loss at all speeds, and at speeds above the transition point the difference becomes greater. The magnitude of the error in predicting power loss is indicated by Fig. 8 which plots the error versus speed for the three bearing sizes. The error increases dramatically with speed and bearing size, and this point is further emphasized in Fig. 9 which gives the percent error in power loss. The dramatic increase in the power loss in the superlaminar flow regime is indicated by the discontinuities in the slopes of

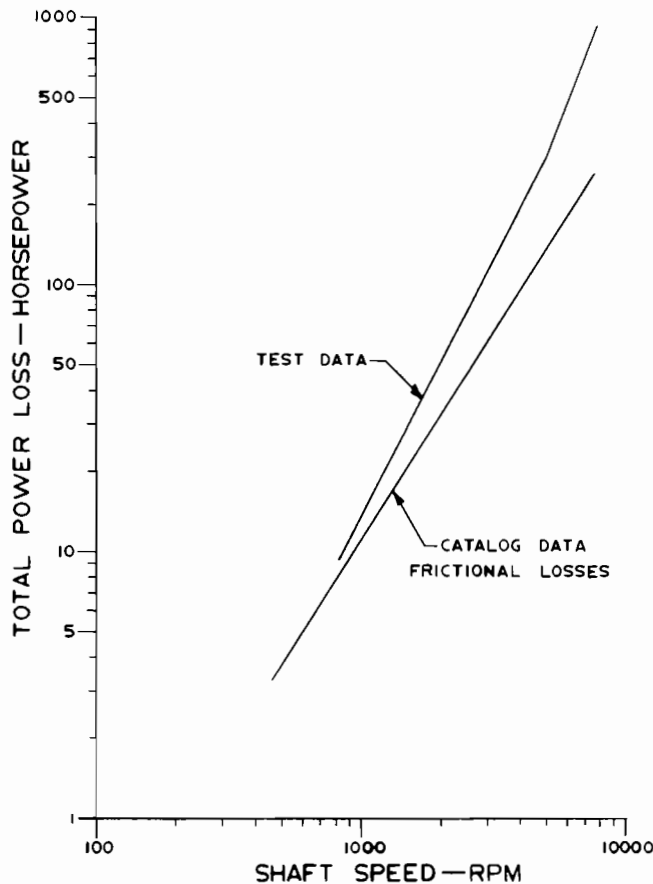


Figure 5. Effect of Shaft Speed on Total Power Loss for 17 inch, 6 X 6 Pad, Double Element Thrust Bearings.

the curves. In order to demonstrate the impact of additional bearing losses on turbine efficiency, typical application of the three bearing sizes were considered. Table 2 shows the percentage decrease in efficiency attributable to error in estimating power loss from catalog data. The change in efficiency is computed by means of the following simple formula:

$$\Delta \% \eta = \frac{\Delta \text{ Power Loss}}{\text{Horsepower Rating}} \quad (1)$$

TABLE 2

Effect of Additional Total Power Loss on Efficiency

Bearing Outside Diameter inch	Application Speed RPM	Range HP HP	Δ Power Loss HP	Δ % Decrease in Efficiency Percent
17	6,000	40,000	320	0.8
15	7,000	30,000	185	0.62
12	10,000	20,000	188	0.94

The impact indicated in Table 2 is typical, and it might be expected that the decrease would range between 0.5% and 1.0%.

It can be seen that the larger value of Δ power loss for the 17-inch bearing is associated with a larger turbine rating so that the percentage change in efficiency is smaller as compared to the 12-inch bearing application. However, the increased steam

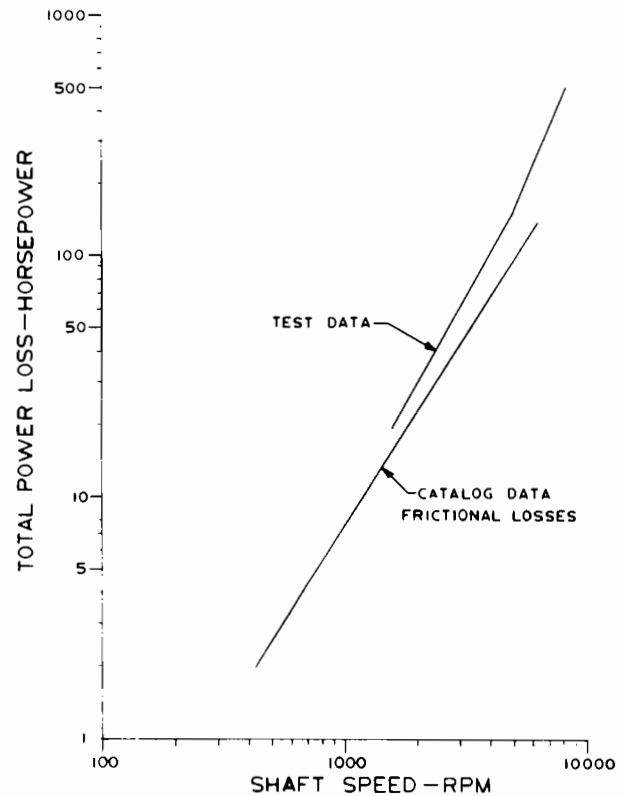


Figure 6. Effect of Shaft Speed on Total Power Loss for 15 inch, 6 X 6 Pad, Double Element Thrust Bearings.

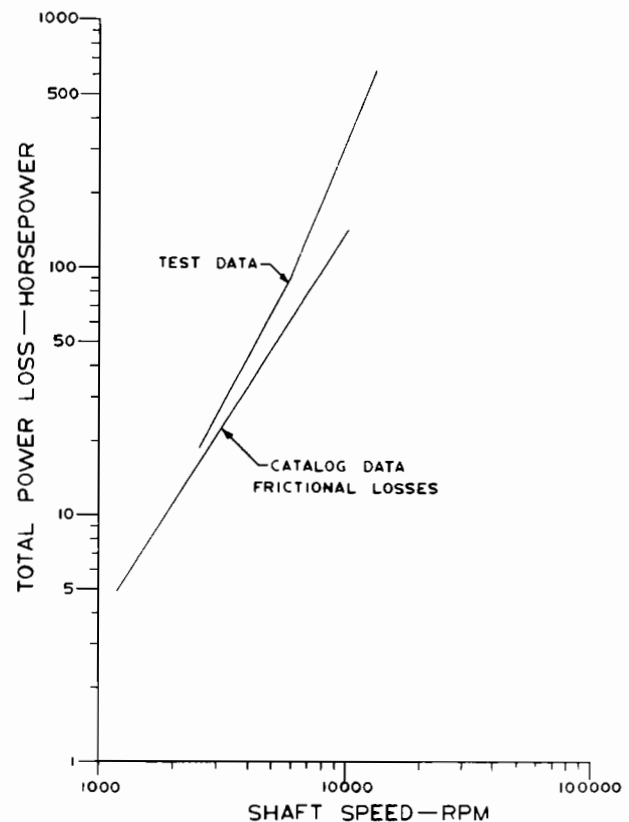


Figure 7. Effect of Shaft Speed on Total Power Loss for 12 inch, 6 X 6 Pad, Double Element Thrust Bearings.

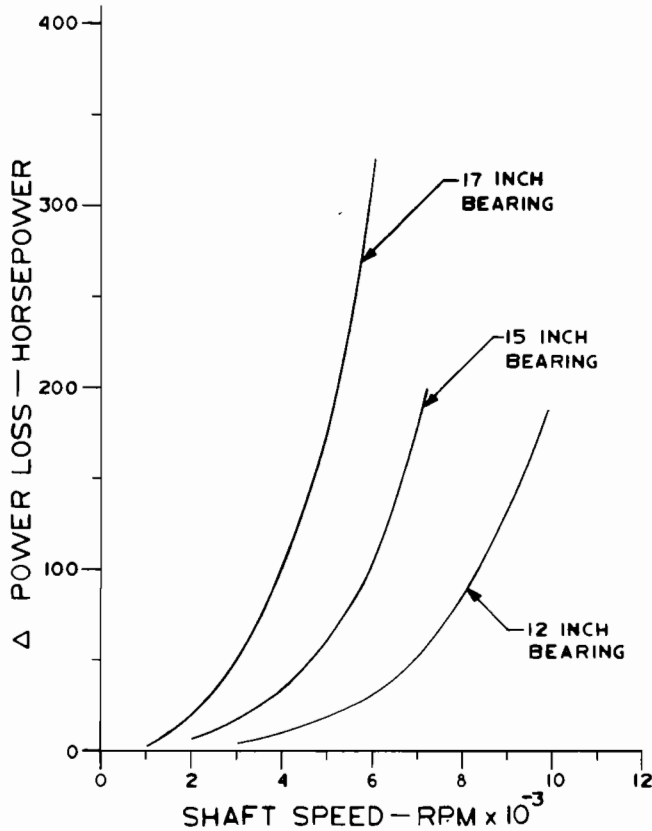


Figure 8. Difference in Total Power Loss Data, Test Minus Catalog Frictional Losses Versus Shaft Speed for 6X6 Pad Double Element Thrust Bearings.

flow and its cost will be greater for larger machines. The authors' company accounts for these increased losses in predictions of turbine performance.

Bearing Load Capability

Proper selection of the thrust bearing size for a given application requires a precise knowledge of the load-carrying capability of the bearing over the operating speed range. If too small a bearing is selected, operating failures due to overloading are likely. This situation cannot be tolerated because of the catastrophic nature of thrust bearing failures on large turbines. If too large a bearing is selected, penalties will be taken in terms of increased power loss and oil flow.

Fig. 10 shows data on the performance of a 15-inch bearing operating with 100% of the manufacturer's recommended oil flow. Lines of constant bearing metal temperatures at the hot spot of the bearing pad are plotted. The shape of the lines indicates that as bearing speed increases, the loading must ultimately be decreased to maintain a constant metal temperature. At higher temperatures, 275°F to 300°F, the onset of turbulence increases the load capacity to some extent. However, these temperatures are higher than the normal application range for this type of bearing and geometric configuration, and no practical benefit is derived. As speed is increased through the superlaminar flow regime, there is a point at which the load-carrying capacity decreases dramatically. At these high speeds the bearing babbitt metal temperature increases much more rapidly with applied load. The shaded area in Fig. 10 represents the normal application range recommended by

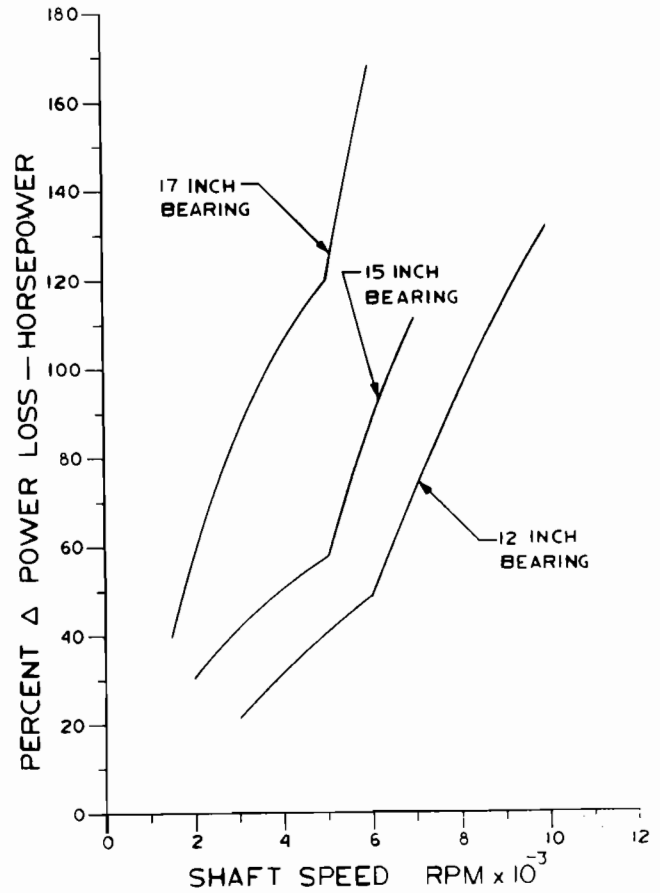


Figure 9. Percent Difference in Total Power Loss Data, Test Minus Catalog Frictional Losses Versus Shaft Speed for 6 X 6 Pad Double Element Thrust Bearings.

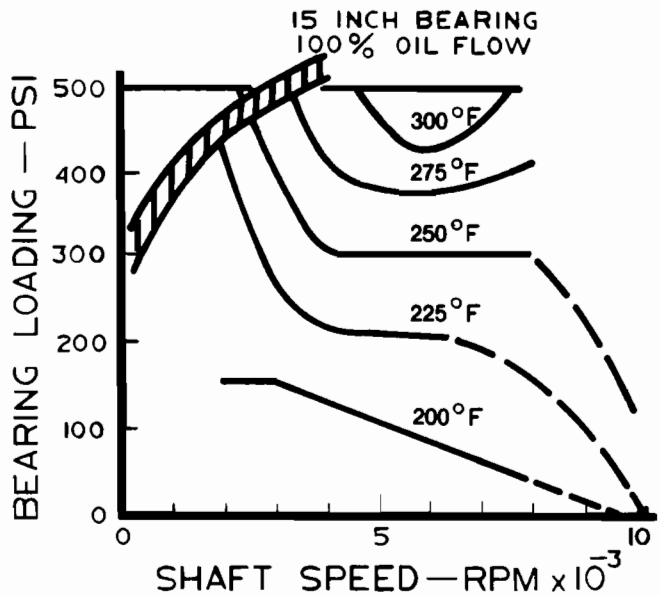


Figure 10. Bearing Performance Curve — 100 Percent Nominal Lube Oil Flow Rate.

bearing manufacturers' catalog data. This limit is based on hydrodynamic conditions sufficient for maintenance of a minimum oil film thickness in the laminar flow regime.

With the growth in ratings of mechanical drive steam turbines, thrust loads have correspondingly increased. Turbine wheel diameters have become larger to accommodate more steam flow. Coupling thrust, which is directly proportional to the horsepower rating, has added to the burden. Consequently, many bearings operate in the superlaminar flow regime, and for such speeds it is necessary to define a reasonable application limit.

The common failure mode for a thrust bearing is wiping of babbitt. This kind of failure is related to the unit load and temperature. As load is increased, the bearing metal temperature increases until at some point the babbitt metal begins to yield. In the test program conducted at the General Electric Mechanical Drive Turbine Products Department, the onset of failure was observed to be a ripple pattern in the surface of the babbitt metal. Similar observations have been made by Booser, Ryan, and Linkinhoker (6) where babbitt ripple of .002-.003 inches in height occurred for a journal bearing operating with an oil film thickness of .0005 inch. As those investigators pointed out, the evidence suggested that the action of the fluid film alone caused the flow of the babbitt surface. It was further revealed that comparison of the maximum pressure (babbitt compressive stress) in the bearing zone with the 0.2% offset tensile strength of the babbitt suggested that babbitt creep in this case was related to the yield strength of the babbitt or more likely to the strength of the babbitt in the zone of maximum subsurface shear stress.

For a tilting pad thrust bearing it is conceivable, at least, to relate the unit loading in a similar way to the maximum babbitt stress. An analysis has been conducted on pad loading, including the complexities involved due to pad bending, which indicates that the babbitt stress is a constant multiple of the unit loading. For such a situation the relationship of speed, load, and temperature might be as depicted in Fig. 11. Babbitt strength is shown to decrease with temperature according to the babbitt strength lines. Generalized curves of babbitt strength (multiple of unit loading) versus temperature for a

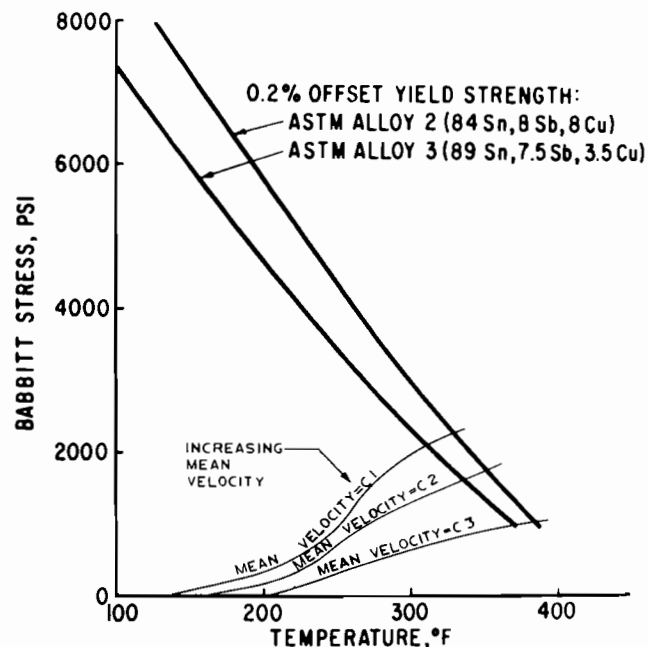


Figure 11. Babbitt Strength Limitations on Maximum Bearing Metal Temperatures for a Given Bearing Size.

given size bearing operating at constant shaft speeds are superimposed. The intersection of these constant speed curves with the babbitt strength line would indicate the onset of babbitt yielding and failure. Failure then becomes a function of babbitt load, speed, and temperature. At lower speeds, failure occurs at lower temperatures and higher loads. Provided an adequate oil film could be maintained, unusually high loads might be possible at low mean velocities. There have been some documented reports — Brandon and Bahr (1), Gardner (10) — of such instances around the industry. However, these surface speeds are well below the practical range of application for thrust bearings for large mechanical drive steam turbines. At higher speeds, failure occurs at higher temperatures and lower loads. For higher speeds where the bearing is operating in the superlaminar flow regime, it is necessary to restrict bearing loads to lower levels than would apply under laminar flow conditions.

A typical speed range of application for the 15-inch thrust bearing is in the range of 4,000 to 8,000 RPM. Referring again to Fig. 10, it is seen that the constant temperature lines are relatively flat for this range, and a given temperature can be associated with a corresponding load. For instance, a load of 300 PSI corresponds to a temperature of around 250°F. This could be taken as a reasonable limit. However, if the design limit were taken as an extrapolation of the manufacturers' catalog data, serious overload could occur for this speed range. For example, consider a 30,000 HP turbine operating at 7,000 RPM with a maximum thrust load of 33,000 pounds. Using a 300 PSI load limit, the required thrust bearing area is 110 square inches. The 15-inch 6 X 6 pad, center pivot tilting pad thrust bearing with an area (each side) of 112.5 square inches would be required. On the other hand, if an extrapolation of catalog data were assumed as the design limit, a 12-inch 6 X 6 pad, center pivot tilting pad thrust bearing with an area (each side) of 72 square inches would be sufficient. However, this size bearing would run considerably hotter than 250°F. Design data similar to Fig. 10 for the 12-inch bearing indicates that the babbitt metal temperature would be in the neighborhood of 300°F. The cost of operating at the cooler temperature has been an increase in bearing size from 12 inches to 15 inches and a corresponding increase in actual bearing loss of 100 horsepower. This would represent an actual decrease in efficiency of 0.33%. Comparison of catalog data for the 12-inch bearing and test data for the 15-inch bearing would show a 160 horsepower increase and a corresponding efficiency drop of 0.53%. In addition, twice the oil flow is needed for the 15-inch bearing and the extra cost for a larger oil supply system must be considered. However, the consequences of one thrust bearing problem is such that the reliability trade-off must predominate in this example.

Fig. 10 also shows that beyond 8,000 RPM the sharp decrease in load-carrying capability limits the application of the 15-inch bearing. This implies a general limit in the thrust load capability of bearings of this type and geometry in terms of a maximum load-speed envelope.

The effect of variations in oil flow on bearing performance was also investigated in the recent test programs (16, 17). Increases in oil flow from manufacturers' recommendations does not result in significant changes in bearing performance. Decrease in oil flow results in diminution of performance. Fig. 12 depicts bearing performance when 50% of the recommended oil flow is employed. Comparison with Fig. 10 shows lower load capability for a given bearing metal temperature. While reduction in oil flow decreases the bearing power loss for a given bearing size (16, 17), larger bearings are required because of the performance degradation. The increased power

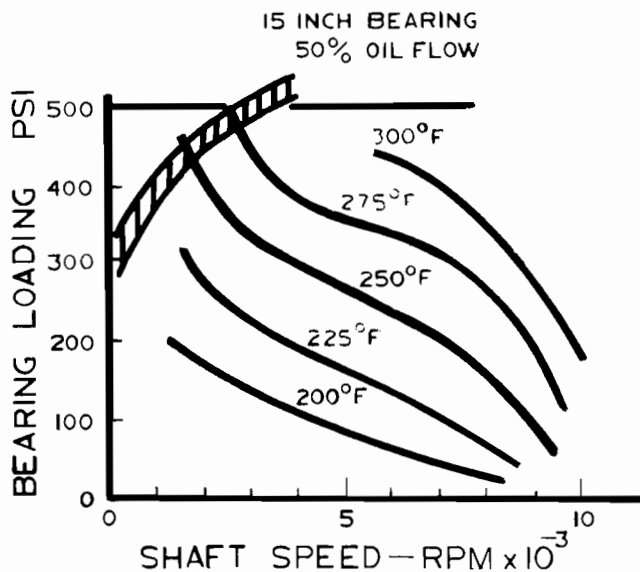


Figure 12. Bearing Performance Curve — 50 Percent Nominal Lube Oil Flow Rate.

loss for larger bearings more than offsets the gain due to reduction in oil flow. It therefore becomes apparent that the bearing manufacturers' recommendations of oil supply flow rate do reflect an optimum, when bearing power loss and pad metal temperatures are both considered.

CONCLUSIONS

The prototype tests reported here have led to changes in application data which had affected the design and estimated efficiency of mechanical drive steam turbines. The following effects are evident.

1. Power loss is higher than predicted by extrapolation of manufacturers' catalog data. Recognition of this increased loss must be accounted for in efficiency predictions. Decreases on the order of 0.5% to 1.0% are typical.
2. Bearing pad metal temperature and loading are the parameters upon which application limits must be based. The use of extrapolations of manufacturers' catalog data to define application limits could result in bearing overload and failure.
3. Larger thrust bearings are required by the new application limits. The increased bearing power losses can add substantially to fuel costs. The increased oil flow requirements also increase size and cost of the oil supply system.
4. Manufacturers' recommended oil flows represent an optimum when bearing power loss and metal temperature are considered.
5. Because of the rapid reduction in load-carrying capability at high surface velocities, there exists a maximum thrust load-speed envelope for this kind of bearing construction. Improvements in bearing design and reductions in applied load are being pursued to prevent these limits from being reached.

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