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# Factors Influencing Power Loss of Tilting-Pad Thrust Bearings

*Several recent technical papers have discussed the advantages of various designs of thrust bearings by comparing the power losses of the different type bearings. However, great care must be exercised to ensure that the comparisons are fair. There are many external factors that influence loss, such as oil flowrate, clearance, supply temperature and so on. Unless compensation for these external factors is included in the analysis, the power loss comparisons may be misleading. This paper attempts to show both qualitatively and quantitatively the influence that various external factors have on bearing power loss. It has been determined experimentally that oil flowrate adjustment can vary power loss by as much as 150 percent. The choice of radial or tangential discharge can reduce power loss by 60 percent, while the actual size of the discharge can influence power loss by 50 percent. Varying the bearing end play has little effect on measured power loss.*

## Introduction

It is often necessary to compare dissimilar types of thrust bearings for the purpose of evaluating competitive designs and determining which is superior from the standpoint of performance. Due to the energy crisis, power losses are now of paramount importance in any such comparison—especially for high shaft speed applications above 3600 rpm. As a first approach, this paper will be limited to a thorough discussion of comparative power loss values. While the significance of other bearing factors, such as turbulence, equalization, temperature, and so on, are not to be denied, it is felt that each is a complex topic in its own right, deserving individual treatment in depth. Hence the focus on power loss in this paper.

In many instances, comparison of published (or vendor-supplied) data on different bearings is complicated by the fundamental differences between installations and operating conditions. These differences are magnified as the size of the bearings and shaft speeds increase into areas where little data are available. Often the significance of an oil drain configuration is overlooked in the haste to "compare apples with oranges" and match numbers. Another criticism is that too little descriptive information is used to qualify published data. For example, stipulation of a bearing power loss level normally includes the lubricant viscosity, bearing load and shaft rpm for that one data point. But the inlet oil temperature, oil flowrate, discharge configuration and end play should also be included for the sake of accuracy.

In order to demonstrate these principles, the approach used in this paper is to take one, standard, bearing design and perform every conceivable test on *it* to show clearly the broad range of power loss values attainable from that one basic design. The different tests performed will involve manipulation of external parameters only—there will be no change in the basic thrust bearing design. Each test is treated independently, and it should be clearly understood that if several design improvements are employed at the same time, the individual beneficial effects resulting from each are not necessarily cumulative. By reporting the effect of each parameter separately, it is hoped that this paper will demonstrate both the need for full disclosure of pertinent details, as well as the versatility of a proven, standard, bearing design.

## Test Bearing Description

This entire discussion is based upon a portion of the power loss data resulting from an extensive series of tests conducted over a period of 5 years 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 Fig. 1, one of which normally carries thrust load and is termed the loaded or active bearing, while the other element (on the other side of the thrust collar) is called the slack side or inactive bearing because it merely serves to position the shaft. Fig. 2 shows the arrangement of the two elements in the bearing housing.

The primary test bearing has six, babbitted, heavily instrumented pads or "shoes" on each side of the collar for a (6 × 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 × 8) double thrust bearing arrangement. In either case, the shoes have a babbitt O.D. of 267 mm (10.5 in.) and a bore of 133 mm (5.25 in.) for total bearing area of 356 cm$^2$ (55.1 square in.). The shoes of the 6-pad design subtend 51° of arc while each shoe of the 8-pad design subtends approximately 38° of arc.

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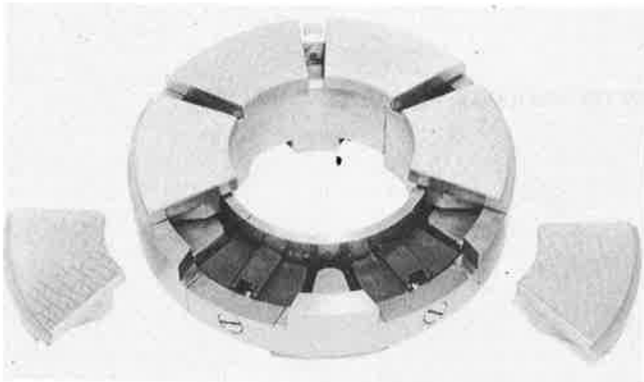

Fig. 1 Single element thrust bearing

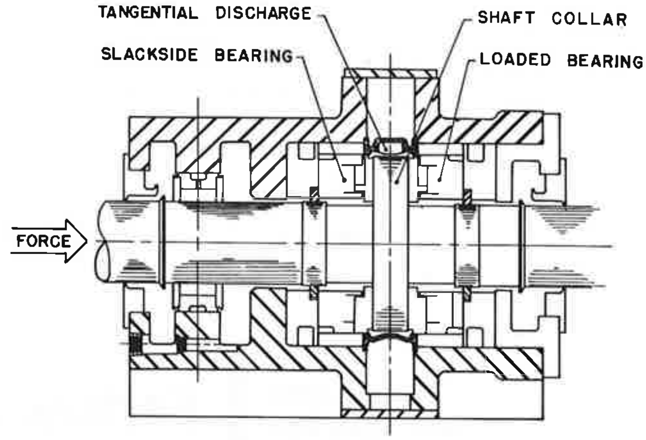


Fig. 2 Cross-sectional view of housing interior showing location of loaded and slackside bearing elements on either side of shaft collar

Power loss values are obtained by an 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 analysis. Additional information regarding instrumentation and test apparatus can be found in references [1, 2, and 3]. The majority of these tests were performed using a light turbine oil with a nominal viscosity of 150 SSU @ 100°F ($32 \times 10^{-6}$ $m^2/s$ @ 37.8°C) except where otherwise noted. Base power loss values were established using a 25.4 mm (1.0 in.) diameter, tangential discharge port, with a radial clearance of 4 mm (0.156 in.) around the collar.

## Test Procedure—Scope of Discussion

This paper will discuss the effects of a series of *independent* tests on thrust bearing power loss. The external parameters evaluated during the course of these tests are summarized below:

(*a*) Total oil flowrate: 50 to 150 percent of normal recommendation

(*b*) Apportionment of flow to each bearing element

(*c*) Lubricant viscosity

(*d*) Thrust bearing end play: 0.3 mm to 0.8 mm (0.011 to 0.030 in.)

(*e*) Type of discharge: tangential, slotted or radial

(*f*) Size of discharge: 12.7 mm to 38.1 mm (0.5 to 1.5 in.)

(*g*) Clearance around collar: 4 mm to 12.7 mm (0.156 to 0.5 in.)

## Total Oil Flowrate

Probably no single factor has greater influence upon thrust bearing performance (temperature, power loss, capacity, etc.) than the rate of oil supply to the bearing. Fortunately, it is one of the easiest to control, or adjust if necessary. The distinction must be made here that this discussion addresses the topic of "pressurized" or "controlled flow" bearings as opposed to "flooded" or "oil bath" bearings which have no oil supply flowrate, per se. In a flooded bearing, the bearing sits in a pool of oil, free to ingest as much as it requires, but some form of cooling must be supplied to keep the oil bath temperature from continually rising due to the losses from oil shear. In a controlled flow mode, each loaded and each slack side bearing receives a continuous supply of fresh oil, fed at relatively low pressure, in an amount deemed suitable for operation. Even low to moderate shaft rpm is sufficient to pump the oil through the bearing to the discharge port, where it is expelled. Thus, the bearing parts are constantly bathed in a fresh supply of "new" or cool oil. Only a small portion of the total quantity of oil supplied will actually find its way into the thin oil film that separates the stationary shoes from the rotating collar. At these speeds, the rotating collar (not inlet pressure) will force a mixture of cool inlet oil and hot carryover oil into the thin oil film between collar and shoe. The major portion of the supplied oil circulates between and around and under the thrust shoes, and even through the bearing components (such as base ring and leveling plates) to provide beneficial cooling of the working surfaces. Without this beneficial cooling, the bearing would fail if the combined service factors of shaft speed and load are severe enough to cause local heating of the shoe (see reference [4]) above the babbitt temperature limit. Because of this need for cooling, the excess oil flow is not superfluous. However, the excess oil will enact a penalty, because even when removed expeditiously, it will contribute to the overall power loss due to pumping and churning losses. Some compromise must be reached to balance the requirement for cooling with the resultant power loss penalty. While it is beyond the scope of this paper to discuss pad babbitt temperatures in depth, it should be noted that increasing the oil flow will not necessarily change the oil film thru-flow or the local heating pattern on the pad babbitt surface, but will only achieve a more subtle, overall cooling effect, as long as the bearing is already supplied with the minimal flow required by the oil film. On the other hand, if the bearing is operating in an "oil-starved" condition, increasing the flowrate will cause a dramatic reduction in pad temperature and more than likely, an increase in oil film thickness.

In any event, the latitude is wide for the selection of a "proper" oil supply rate. Figs. 3, 4, and 5 show the influence that adjustment of the oil flowrate will have on (6 × 6) thrust bearing power loss for unit loadings of 0.7, 2.07 and 3.45 MPa (100, 300 and 500 psi), respectively. The flowrate values plotted as the abscissae represent "total flow" which is the sum of the oil supplied to the loaded and the slack side bearings. The dashed line connects the oil flows which conform to normal or standard recommendations for that particular operating condition. The majority of those data points identified by the symbol ⊙ represent flow values of 50, 75, 100, 115, 125 or 150 percent of the standard recommendation. The standard values (dashed line) are, of course, the manufacturer's preferred flowrates, while the other points are experimental data points.

The standard oil flowrate is determined by calculating the theoretical power loss and then computing the required oil flow to maintain a suitable temperature rise from inlet to discharge. For the tests shown here at standard flow, the temperature rise was 9.5°C to 18.3°C depending on shaft rpm, with a constant supply temperature of 46.1°C. Unless otherwise noted, all tests were performed with light turbine oil (150 SSU @ 100°F). However, since the supply temperature was a constant 46.1°C, it is possible to form lines of constant temperature rise across shaft rpm's by connecting points of equal value formed by the ratio of loss/flowrate.

Based upon Figs. 3, 4, and 5, the effect of changing the oil supply flowrate upon power loss is clearly evident. For example, doubling the oil flowrate will increase power loss 25 percent at 4000 rpm and 38 percent at 13,000 rpm. The different rate of increase is due to the

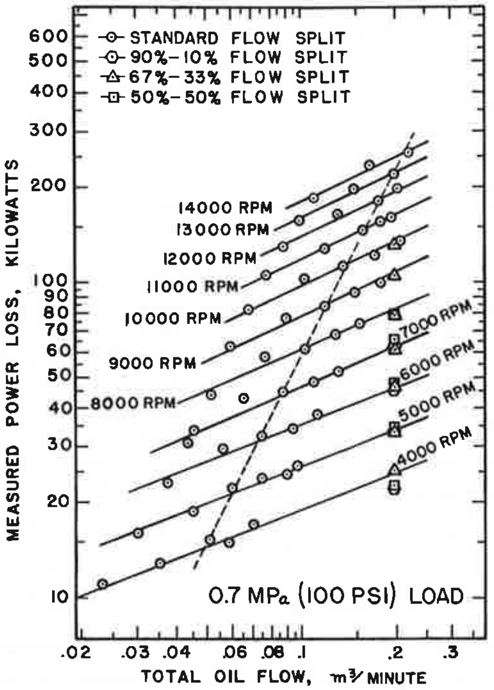


Fig. 3 Measured thrust bearing power loss at 0.7 MPa load for a 267 mm (6 × 6) thrust bearing. Values based upon the use of light turbine oil, supplied at 46.1°C, and a tangential discharge with a 25.4 mm diameter port. Multiply kilowatts by 1.341 to obtain horsepower. Multiply m$^3$/minute by 264.2 to obtain U.S. GPM values.

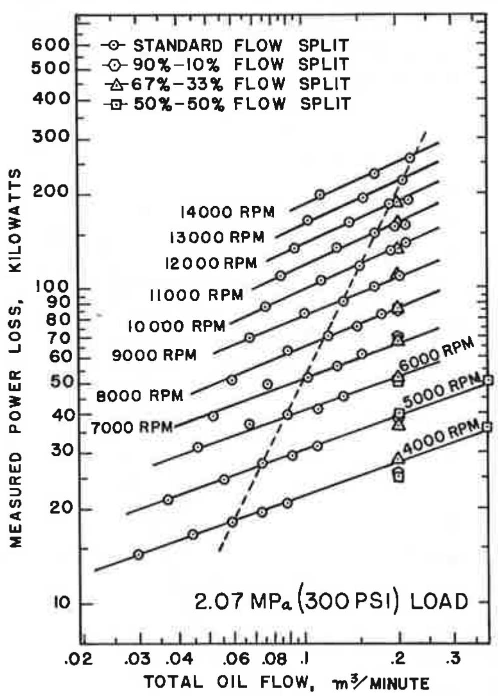


Fig. 4 Measured thrust bearing power loss at 2.07 MPa load for a 267 mm (6 × 6) thrust bearing. Values based upon the use of light turbine oil, supplied at 46.1°C, and a tangential discharge with a 25.4 mm diameter port.

speed effect on churning losses and the influence of film turbulence. Substantial savings can be realized by reducing oil flows, provided a safe minimum is maintained.

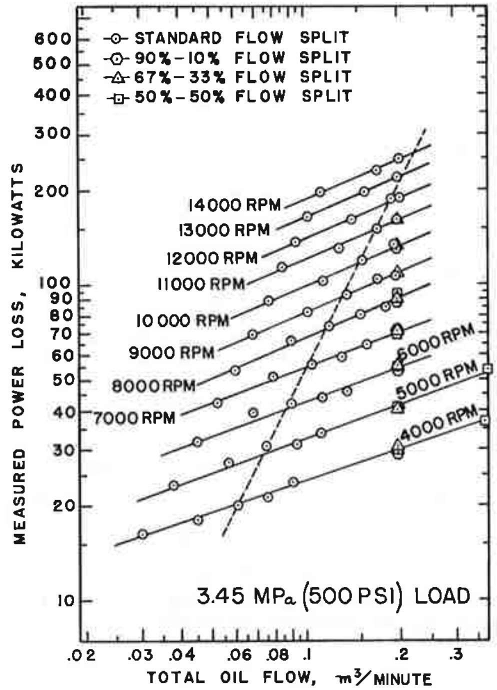


Fig. 5 Measured thrust bearing power loss at 3.45 MPa load for a 267 mm (6 × 6) thrust bearing. Values based upon the use of light turbine oil, supplied at 46.1°C, and a tangential discharge with a 25.4 mm diameter port.

## Apportionment of Flow

All the standard flow data points identified by symbol ⊙ on Figs. 3, 4, and 5 are based upon sharing the total flow between the loaded and slackside bearings proportionally according to the computed, theoretical power loss. This division is shown in Fig. 6 for the various bearing loadings, where the percentage of total flow allotted to the loaded bearing is plotted as a function of shaft speed. The slackside bearing receives the balance of the oil flow.

A series of additional tests were performed to determine whether a different apportionment of total flow would substantially influence the power loss data, perhaps due to uneven cooling, choking of oil passages, and so on. These results have also been plotted in Figs. 3, 4, and 5, for a constant value of total oil flow (0.2 m$^3$/minute) distributed as follows:

90 percent to loaded bearing, 10 percent to slackside bearing
67 percent to loaded bearing, 33 percent to slackside bearing
50 percent to loaded bearing, 50 percent to slackside bearing

With the exception of some data scatter occurring at the 4000 rpm cases, no appreciable deviation from the normal trend was detected for these tests. These were gross overflow conditions for many of the lower rpm tests—for example, 3 to 13 times the standard flow for some cases of the 4000 rpm case. Normally, under standard flow conditions at 4000 rpm and 2.07 MPa load, the loaded bearing receives 0.0503 m$^3$/minute and the slackside bearing 0.0076 m$^3$/minute of oil. With a 90-10 percent split, the loaded bearing was fed 0.176 m$^3$/minute and the slackside bearing 0.020 m$^3$/minute. With a 50-50 percent split, the loaded bearing was fed 0.098 m$^3$/minute and the slackside bearing 0.098 m$^3$/minute.

From these tests, it is concluded that different flow apportionment will not unduly influence resulting power loss, as long as the minimal oil flow requirements have been met. However, it is felt that it would be dangerous to reduce the flow to the loaded bearing much below the typical values shown in Figs. 3, 4, and 5 without careful temperature monitoring because of the risk of possible oil starvation.

## Lubricant Viscosity

Another commonly used method of controlling bearing power loss is to alter the viscosity of the lubricant, either by substitution of a

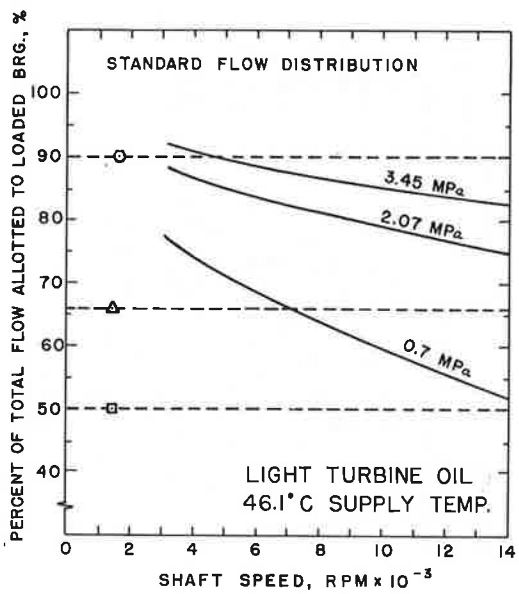


**Fig. 6 Standard flow distribution to loaded element of a 267 mm (6 × 6) thrust bearing. Special, constant flow distributions are indicated with dashed line.**

lighter (or heavier) oil or even by adjusting the inlet oil temperature. Results typical of the latter action are shown in Fig. 7. Inlet oil temperature was held at 46.1°C ± 1° during the entire series of tests discussed in this paper—except for the isolated tests shown in Fig. 7 which were conducted at 35°C and 57.2°C.

Some of the typical small power loss variations @ 2.07 MPa are listed in Table 1.

The minor power loss reduction achieved by raising the inlet oil temperature 11.1°C was accomplished with a 7.2°C increase in local

**Table 1 Influence of supply temperature on power loss @ 2.07 MPa load**

| rpm | 35°C Supply | 46.1°C Supply (base) | 57.2°C Supply |
|---|---|---|---|
| 5,000 | 30.4 kw +8.6% | 28.0 kw | 27.4 kw −2.1% |
| 6,000 | 44.7 kw +12.0% | 39.9 kw | 39.0 kw −2.3% |
| 8,000 | 72.5 kw +2.6% | 70.7 kw | 70.3 kw −0.6% |
| 10,000 | 124.9 kw +5.4% | 118.5 kw | 115.8 kw −2.3% |
| 12,000 | 189.7 kw +2.3% | 185.5 kw | 178.2 kw −3.9% |

pad babbitt temperature. The increase in power loss due to lowering the inlet oil temperature by 11.1°C resulted in a 4.4°C cooler pad babbitt temperature. The fluctuation in pad babbitt temperature due to changing inlet oil temperature is *not* uniform over the entire load and speed range, owing to viscosity/temperature relationships and shifting laminar/turbulence transition point.

A much more effective way to reduce power loss is to substitute a lower viscosity lubricant, if specifications, oil film thickness and bearing loading will permit. The results of substituting light turbine oil (150 SSU @ 100°F) for Navy specified 2190 TEP (average viscosity: $94.2 \times 10^{-6}$ m²/s @ 37.8°C or 430 SSU @ 100°F) is shown clearly in Fig. 8. The power losses for the heavier viscosity oil supplied at its own standard oil flowrate are shown in curve A. Reducing the flowrate to 80 percent of standard value results in some loss reduction, as shown in curve B. Further reduction in flowrate, until the flows matched the standard flows for light turbine oil (approximately 50 percent of the flows used for curve A) will yield the power losses of curve C. Substituting the light turbine oil at its standard flowrates will result in power loss levels shown for curve D. It was necessary to terminate the tests of curves B and C prematurely because of the excessive pad temperatures caused by the reduction in oil flow and the subsequent decrease in cooling. On the other hand, the bearing of curve D is certainly operating with a thinner oil film thickness than that of curve A, so it is extremely important that the bearing loading is not excessive and will safely tolerate the use of the lighter viscosity lubricant, as in this test case.

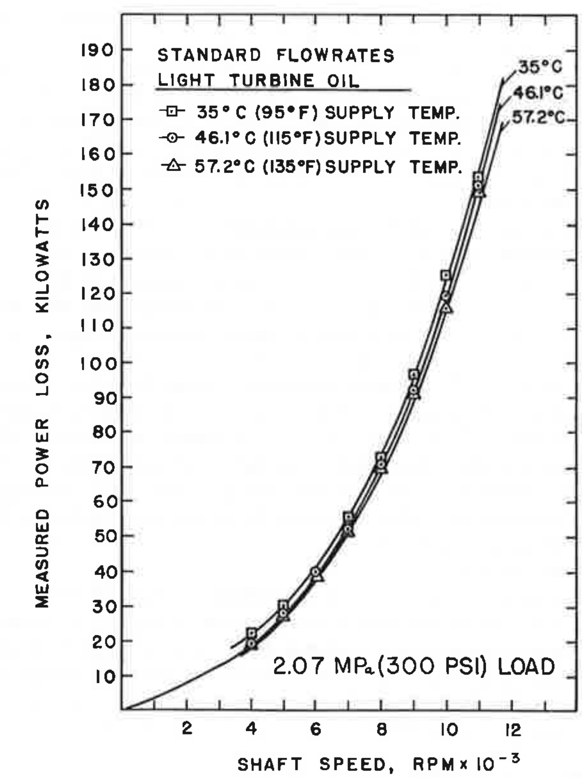


**Fig. 7 Effect of supply temperature on (6 × 6) thrust bearing power loss. Values based upon the use of light turbine oil supplied at standard flowrates, and a tangential discharge with a 25.4 mm diameter port.**

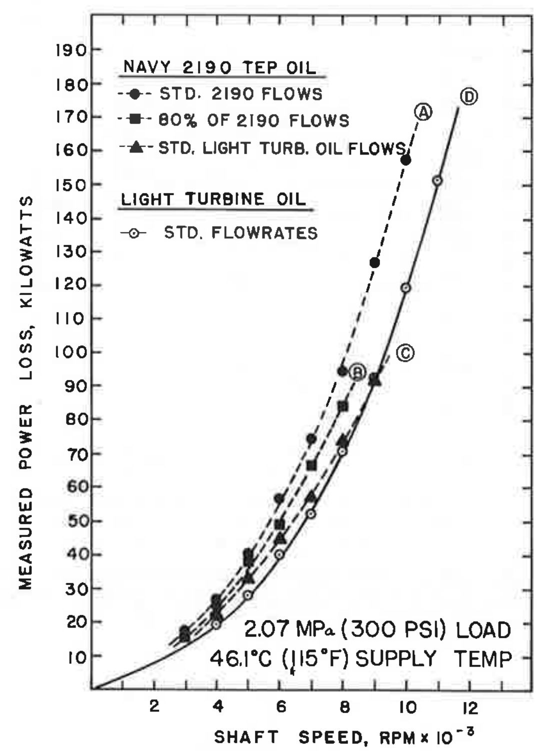


**Fig. 8 Effect of lubricant viscosity on (6 × 6) thrust bearing power loss. Lubricants supplied at 46.1°C. Thrust bearing utilizes a tangential discharge with a 25.4 mm diameter port.**

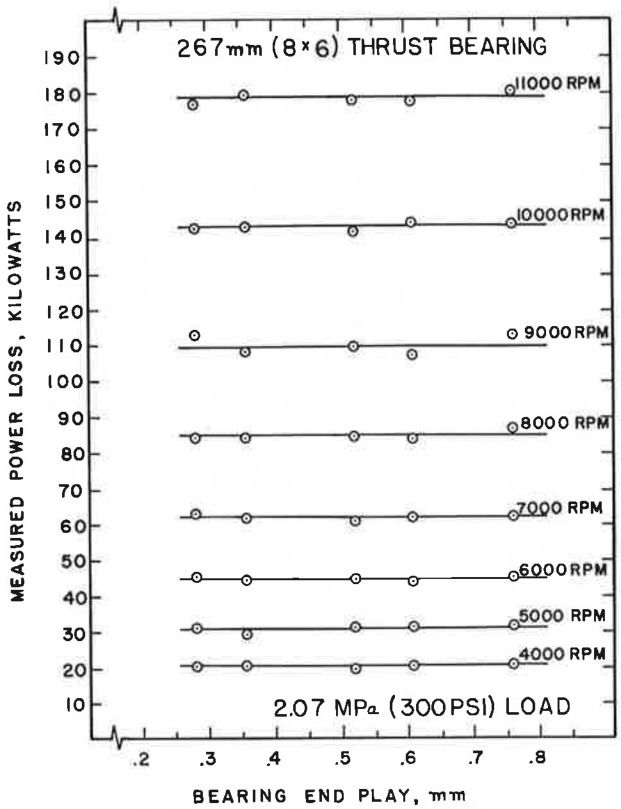


**Fig. 9 Effect of varying end play on (8 × 6) thrust bearing power loss. Values based upon the use of light turbine oil supplied at standard flowrates and 46.1°C, and a tangential discharge with a 25.4 mm diameter port.**

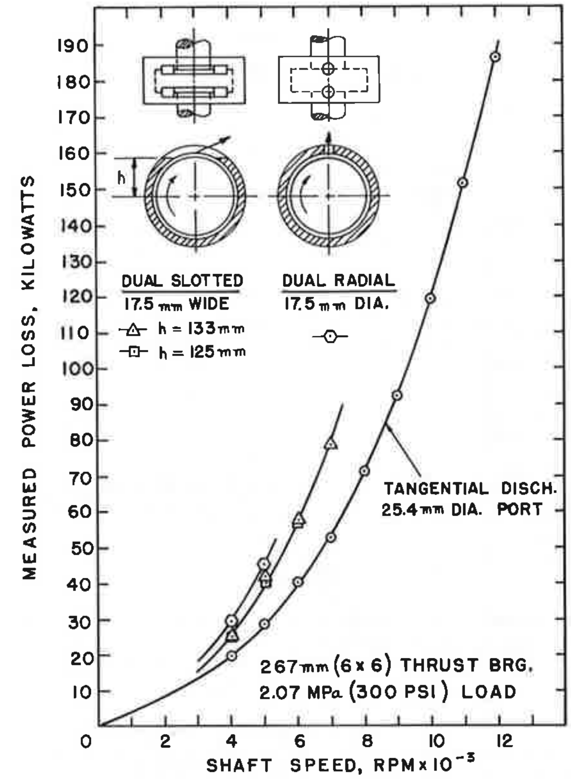


**Fig. 10 Effect of discharge configuration on (6 × 6) thrust bearing power loss. Values based upon the use of light turbine oil supplied at standard flowrates.**

## End Play

The remaining bearing investigations discussed in this paper involve physical changes to the housing that contains the thrust bearing. The first of these tests evaluated the effect that altering the end play would have on bearing power loss. The recommended cold end play for a 267 mm (10.5 in) thrust bearing is 0.4 mm (0.015 in.) to 0.5 mm (0.019 in.), depending on shaft speed. Theoretical power loss calculations would indicate that loss is inversely proportional to the end play, because a larger end play value will reduce the slackside bearing losses.

Fig. 9 shows measured power loss levels as a function of end play over a considerable range above and below the recommended end play value. This series of tests, conducted on an (8 × 6) thrust bearing proves that increasing the end play above normal will not reduce the resulting power loss level. In fact, even reducing the end play to 0.3 mm (0.011 in.) (about one-half of the recommended value) did not appear to influence the losses. Evidently, gross adjustments to the oil film thickness do not affect the viscous shear occurring between rotating collar and slackside bearing elements despite the relatively high shaft speeds of these tests. The implication is that the slackside bearing losses are a smaller percentage of the total loss than predicted by theoretical analysis. It is suggested that the bearing end play will not affect power loss unless it is set to a very restrictive value, forcing the slackside bearing to operate with a mean film thickness approaching that of a loaded bearing, which is approximately 0.05 mm to 0.08 mm (0.002 to 0.003 in.). Thermal growth and bearing deflection under load tend to cancel each other out, and need not be considered. Thus, a restrictive value of end play for this bearing might be 0.15 mm (0.006 in.)—a value never attempted during these tests. It must be concluded that adjustment of end play within reasonable limits will not significantly influence overall power loss values.

## Oil Discharge Ports

Substantial increases in bearing power loss are detected when other than a tangential exit is provided for the discharge oil. The discharge port can be cast or machined into the housing itself, or provided as a separate control ring or liner around the collar periphery. It was found to be extremely important that the discharge oil be removed expeditiously from the region around the rotating collar which acts as a pump impeller at these high shaft speeds. However, it is necessary to provide some minimal back-pressure because a completely free discharge, such as a massive bottom drain, could lead to oil starvation or cavitation, and promote pad flutter or spragging[1].

During this series of tests, the tangential discharge was established as the standard against which the other types of discharge were evaluated. Fig. 10 demonstrates these results for a (6 × 6) thrust bearing while Fig. 11 shows similar information for an (8 × 8) bearing. Each tangential discharge was equipped with a 25.4 mm (1.0 in.) diameter circular discharge port. This one size of discharge port was used throughout the entire range of shaft speeds and oil flows, and seemed to be most efficient, with only a slight indication of choking at some high oil flowrates.

In the case of the (6 × 6) thrust bearing, the tangential discharge was replaced with a dual slotted discharge which has rectangular exit areas and an easement to tangential movement of the discharge oil. The sketch indicates the construction details. The immediate result was an increase of 50 percent in the bearing power loss at 7000 rpm and a high discharge pressure that forced leakage past the seal rings that contained oil in the housings.

Similar results were found for the (8 × 8) bearing as shown in Fig. 11 which utilized a single slot of equivalent exit area. When it was obvious that the slotted discharges were not performing as well as their tangential counterparts, the slots were made deeper to increase the exit area, but the results were virtually identical, as shown in Figs. 10 and 11.

[1] A "spragged" pad tips sufficiently for the leading edge to contact the rotating shaft collar due to a divergent film shape. See pp. 5–65 of reference [5].

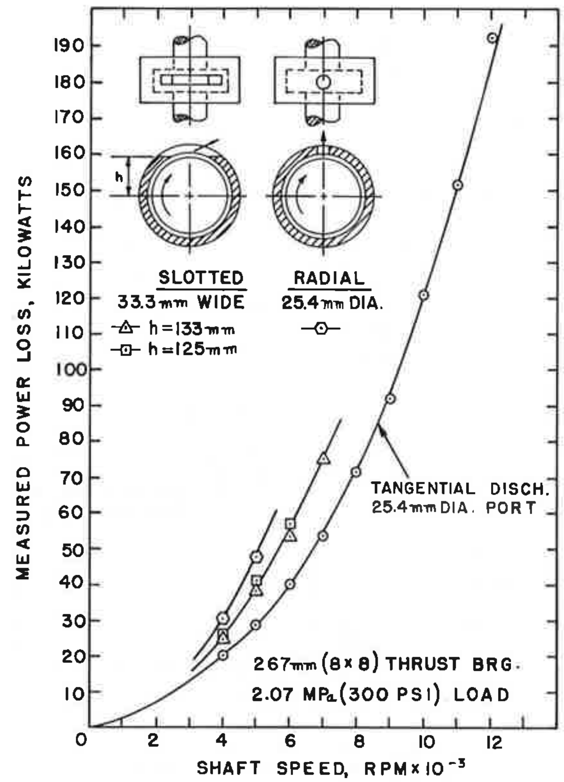


**Fig. 11 Effect of discharge configuration on (8 × 8) thrust bearing power loss. Values based upon the use of light turbine oil supplied at standard flowrates.**

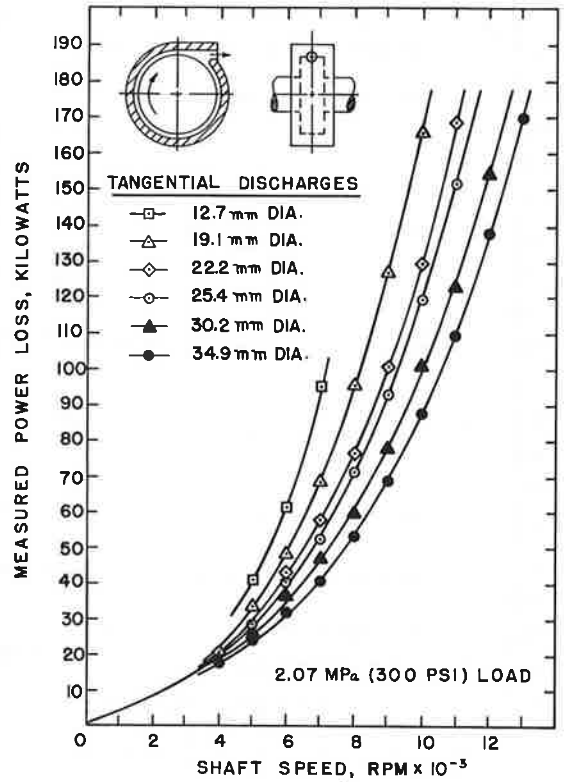


**Fig. 12 Effect of discharge port size on (6 × 6) thrust bearing power loss. Values based upon the use of light turbine oil supplied at standard flowrates and 46.1°C, and tangential discharges.**

Finally, a true radial discharge was tried, via a single 25.4 mm diameter hole for the (8 × 8) bearing and dual 17.5 mm holes for the (6 × 6) bearing. The supply temperature was lowered to 35°C in an attempt at reducing leakage. Even so, the high discharge pressure caused leakage so severe that the testing was terminated after only a few data points were collected. However, the measured power loss was some 61.3 percent over the tangential drain value at 5000 rpm.

During the course of these tests, the superiority of the tangential drain was clearly demonstrated from the standpoint of power loss and sealing considerations. Tangential discharges can be made bi-directional to accommodate either shaft rotation. These tests indicated that radial discharges or abrupt 90° direction changes in drains are to be avoided if at all possible, because of the power loss and pressure effects that will result from such designs.

## Discharge Port Size

Once the tangential discharge was proven to be an efficient type of discharge, a study was undertaken to explore the optimum size of the port. Ideally, the discharge port should be capable of operation over a broad range of shaft speeds and oil flowrates without excessive choking, while at the same time, measured discharge pressure and power loss should be within tolerable levels. The diameter of the port size was varied from 12.7 mm to 38.1 mm (1/2 to 1 1/2 in.) in small increments (usually 1.6 mm) in order to determine best operating conditions. The standard flowrates for light turbine oil (150 SSU @ 100°F) were used for these tests to facilitate comparisons with previously collected data.

A portion of the resulting data, edited for clarity, is presented in Fig. 12. A substantial power loss advantage can be achieved by selecting the size of the discharge port to match the oil flowrate and shaft speed at the operating point. For example at 7000 rpm, the measured power loss will increase 80 percent (from 52.5 kw to 94.3 kw) when a 25.4 mm diameter port is reduced to a 12.7 mm port. Also, the loss will diminish 23 percent (from 52.2 kw to 40.1 kw) when the 25.4 mm port is opened up to a 35 mm diameter at the same 7000 rpm. The improvement in power loss appears to level out at around 35 mm because no further reduction in loss was measured for discharge ports of 38.1 mm. Thus, Fig. 12 graphically demonstrates the penalty in power loss that too restrictive a discharge will impose upon thrust bearing operation. There have been instances of babbitt erosion of the pads adjacent to discharge ports occurring in installations where the discharge was severely throttled. It is advisable to provide a larger discharge in order to avoid these difficulties. However, using too large a discharge will increase the possibility of cavitation and oil starvation of the pads. During the course of testing, there was visual and audible evidence of pad flutter when the discharge holes were opened up to 38.1 mm from a 35 mm diameter. For this reason, a suitable pressure tap at the discharge port, in line with the flow stream, is recommended as a good indicator of discharge port performance. A pressure of 34.5 kPa to 103.4 kPa (5 to 15 PSIG) is considered suitable, but shaft speed effects and oil supply rate will influence the absolute value of discharge pressure for each different installation.

## Collar Clearance

The final area of investigation was the radial clearance between stationary housing and rotating shaft collar. Normally, this value of radial clearance is established to match the bearing size and not changed regardless of service conditions. Now, however, it is recognized that extremely high shaft speeds or abnormal oil flowrates may necessitate adjustment of this clearance to achieve certain economies of power loss. These bearing tests were started with a 4 mm (5/32 in.) radial clearance obtained with a profiled bore as seen in the sketch on Fig. 13. This testing was continued using a straight bore (refer to sketch) of several different radial clearances, with the results also shown in Fig. 13. All these tests were run in conjunction with a 25.4 mm tangential discharge at standard flowrates for light turbine oil (150 SSU @ 100°F).

Substantial power loss economies are evident in this data. For example, at 7000 rpm the power loss can be reduced from 52.2 kw with

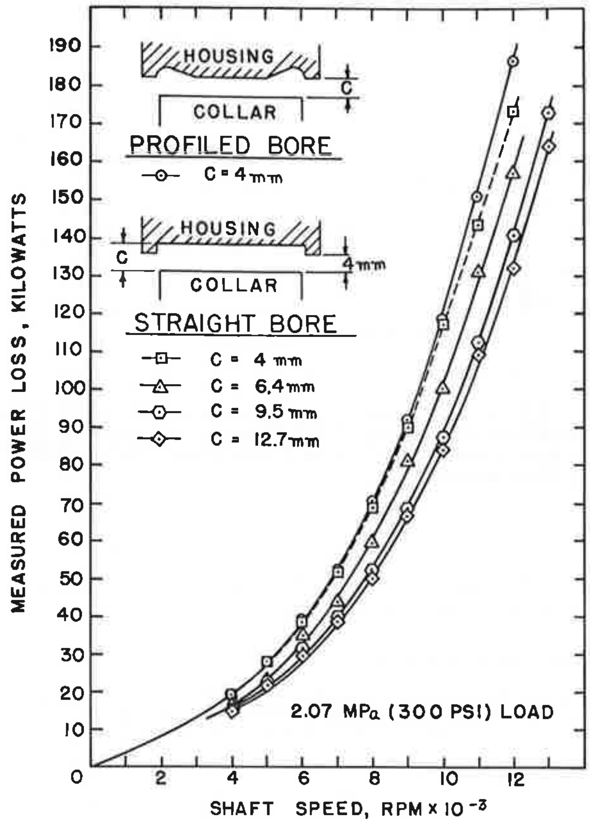


**Fig. 13 Effect of collar radial clearance on (6 × 6) thrust bearing power loss. Values based upon the use of light turbine oil supplied at standard flowrates and 46.1°C and a tangential discharge with a 25.4 mm diameter port.**

a 4 mm clearance, to 43.7 kw with a 6.4 mm clearance, or to 38.4 kw with a 12.7 mm radial clearance. This reduction amounts to 16 and 26 percent, respectively. These economies do not appear to be speed-dependent because at 10,000 rpm, the straight bore loss is 117.0 kw with 4 mm clearance, 100.3 kw with 6.4 mm clearance, and only 83.9 kw with 12.7 mm radial clearance, for percentage reductions of 14 and 28 percent, respectively. These power loss values are quoted to demonstrate the power loss improvement that can be achieved by altering the discharge configuration surrounding the collar. Even changing from a profiled to a straight bore resulted in some beneficial power loss reduction (about 5 percent) so there is every indication that some care and attention in the design of this region of a bearing housing will yield the desired results.

The bearing user is cautioned that it may be dangerous to increase the radial clearance to a value so large that insufficient discharge pressure is generated, and the oil flows between the bearing pads without restriction and without actually lubricating the pads. Bearing failures will result. The radial clearance should not be so large that oil is flung off the collar by centrifugal force and the collar rim spins unwetted with oil. The collar rim must be oil-wetted in order to provide the necessary sealing and produce sufficient backpressure of the discharge.

## Conclusions

The experimental data presented in this paper proves that substantial power loss variations can be obtained by merely adjusting the external parameters that affect thrust bearing performance. The following summary indicates the relative influence of each factor evaluated in this study:

(*a*) Oil supply flowrate: loss variations up to 150 percent
(*b*) Flow apportionment: no measurable loss variation
(*c*) Lubricant viscosity: lighter oil cuts losses by 25 percent
(*d*) End play: no measurable loss variation
(*e*) Type of discharge: loss variations 60 percent higher for radial discharges
(*f*) Size of discharge port: losses reduced 50 percent by increasing size of discharge
(*g*) Collar radial clearance: losses reduced by 25 percent with increased clearance

Several conclusions can be drawn from the information presented above. First, this type of oil film lubricated, slider bearing is a versatile performer, capable of operation over a wide range of different conditions with a minimum of risk. Second, adjustment of various external parameters to unreasonable values will precipitate bearing wipes or failures because of the inability to maintain an adequate film thickness. This is not the fault of the bearing, but the responsibility of the bearing user or housing designer. Third, more complete documentation of actual operating conditions would facilitate the comparison of published data and the evaluation of different designs. Fourth, claims of superior performance for new designs can often be discounted when the competing bearings are compared on an equal basis.

The topics of bearing safety and longevity were indirectly referenced when high risk areas of operation, such as large discharge port sizes or collar clearances, were noted. The length of this paper does not permit a full discussion of critical pad babbitt temperatures, although this data is available.

This type of experimental study becomes particularly useful for improving the accuracy of theoretical power loss predictions for a wide range of bearing designs. Since the total bearing power loss is comprised of several elements (loaded and slack bearing shear, seal ring losses, churning losses between the pads and at the collar rim, pumping losses at the discharge port, turbulent effects and others), it is extremely difficult to quantify the contribution of each component. Yet a computed bearing power loss value must include all these factors if it is to be used with confidence. Therefore, detailed knowledge about the influence of parametric variations can only aid in the further study, and the eventual reduction of the individual parasitic losses, by indirectly defining the magnitude of the problem, and suggesting viable remedies.

It is hoped that the information presented above will prove useful to those who try to design more efficient rotating machinery without sacrificing reliability. Perhaps these suggested areas of design improvement will help to conserve energy by avoiding needless waste.

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speed bearing designer/user should be to design an overall bearing system to ensure that the bearing receives an *adequate* supply of oil for generation of the load-carrying film as well as cooling, and can discharge the spent oil with a minimum of extraneous pumping or churning loss. Hence the term "controlled flow".

The term "flooded" bearing has the negative connotation of intentionally restricting the discharge to maintain a full cavity and high discharge pressure, yet it is known that too restrictive a discharge may erode adjacent babbitt areas and boost power loss unnecessarily. From the standpoint of safety, even a "flooded" high speed bearing will pump out all the lubricant and fail if the pressurized flow is interrupted. At the other extreme, too free a discharge will result in pad instability and ultimate bearing failure from lack of adequate lubrication. The goal, then, is to provide an efficient discharge which provides a minimal backpressure by properly designing the discharge area to meet the specific operating conditions.