

Tilting Pad Thrust Bearing Tests - Influence of Oil Flow Rate on Power Loss and Temperatures

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Oil flow rate to tilting pad thrust bearings influences two important bearing operating characteristics; bearing power loss and pad operating temperatures. Reducing oil flow rates is desirable as this reduces the size and cost of the oil supply system and also can reduce bearing power losses. However, this can increase bearing operating temperatures, reducing the load capacity. Results of laboratory tests of thrust bearings with babbitt faced pads are reported in which oil flow rates were varied and losses and temperatures were measured.

1. INTRODUCTION

Tilting pad type thrust bearings are used in a wide variety of rotating machinery where significant thrust loads must be accommodated. The thrust load is transmitted from the rotor to the stator through hydrodynamic oil films that develop between a rotating collar on the shaft and the stationary pads in the thrust bearing. These oil films, which are typically 0.025 to 0.075mm (0.001 to 0.003 inches) thick, are thus subjected to shear at rates which can be quite high in large and/or high speed machines, and specifically in much turbomachinery. The viscous nature of the lubricant, commonly mineral oil, results in the generation of heat within these films. At high shear rates this heat generation (power loss) can become significant in terms of the overall power rating and efficiency of the machine.

Oil is circulated through these thrust bearings to provide for the development of the hydrodynamic oil films, and also to remove the majority of the heat generated. In the interests of efficiency and economy it is desirable to reduce the losses in these bearings, and also to reduce the oil flow rate to these bearings.

Thrust bearing pads commonly use babbitt (whitemetal) as a facing material to operate against a steel collar. Babbitt is used due to its excellent bearing characteristics, but its primary weakness is a

limited temperature capability. The rate of heat generation in the oil film directly influences the temperature of the babbitt facings. So, two primary factors in the design and operation of these bearings are the power losses and the pad temperatures.

Thrust bearing and associated housing designs have been developed that eliminate a large portion of the parasitic losses that can develop in the oil flow paths leading to and from the thrust pads themselves. These designs inject the oil directly to the leading edges of the thrust pads and then allow this oil to exit the housing through large unrestricted passages. A thrust bearing design of this type is shown in Figure 1. With this more precise control of the oil flow paths to the thrust pads, and an associated reduction in the oil flow that essentially bypasses the pads, total oil flow rates can be reduced. Once the oil leaves the thrust pads it is desirable to provide unrestricted passages for its exit. This helps reduce losses associated with oil flooding the thrust collar outer diameter surface, where losses can be significant at high velocities.

Several authors have presented results of tests of various thrust bearing and housing designs with respect to the influence of oil flow rate, and other factors, on bearing performance. References 1 through 10 are representative. The data here relates to the specific bearing and housing design configurations shown in Figures 1 & 2 and presents the loss and temperature data together on single

plots, which helps to visualize the relationship of these to the flow rate and to each other. Comparisons to calculated hydrodynamic oil flow rates are also given.

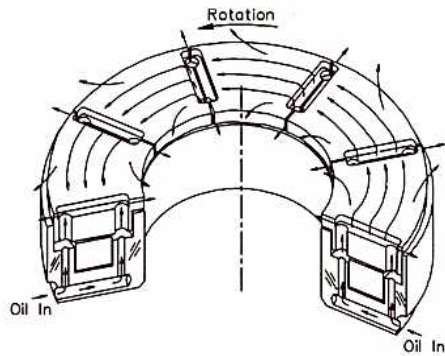


Figure 1. Test Bearing with oil feed directly to pockets between pads.

Reducing oil flow rates is desirable but raises the question of how far they can be lowered without significantly affecting (increasing) bearing pad temperatures. A simplified picture of lubricant flow in and out of a thrust pad film is shown in Figure 3. The net flow into the leading edge of the pad film is identified as Q_1 . The thrust collar drags oil into the leading edge at a rate determined by the velocity of the collar and the thickness of the film at this edge. This flow rate is reduced by the hydrodynamic oil film pressure generated within the oil film, with a net flow in (Q_1). Flow exiting the trailing edge of the pad is labeled Q_3 . This is the result of the thrust collar dragging oil through a reduced film thickness at or near the trailing edge plus flow resulting from the oil film pressure. Q_3 enters the oil film of the leading edge of the following pad, mixing with additional supply oil to form Q_1 for this next pad. The flow exiting the outer edge of the pad (Q_4) is primarily the result of the oil film pressure forcing oil through the film thickness. The inner edge flow (Q_2) is similar except that the net flow rate is considered essentially zero. It is believed that the pumping action of the thrust collar prevents any inward flow from leaving the bearing, but is recirculated into following pads. Thus, the minimum flow required to avoid a 'starved' oil film is Q_4 times the total number of pads in the bearing. This is a value calculated in the course of thrust

bearing design. (Later references to Q_4 are the total for the bearing, not just one pad.)

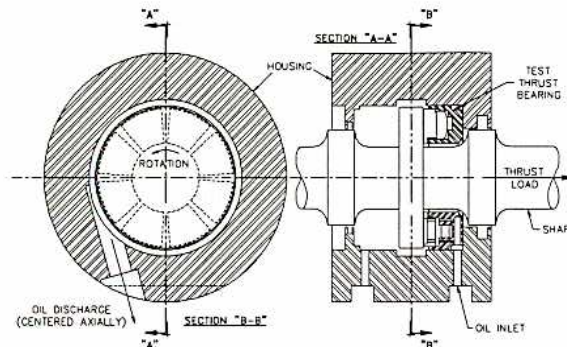


Figure 2. Test bearing and housing with open tangential discharge.

Oil flow rates specified for thrust bearings are commonly based on a calculated power loss and a desired value for the increase in the oil temperature as it flows through the thrust bearing (typically 15 to 20°C). For high speed applications, as most turbomachinery, this flow rate is well in excess of the Q_4 value. Excess flow, beyond what is needed hydrodynamically, acts to provide overall cooling to the bearing.

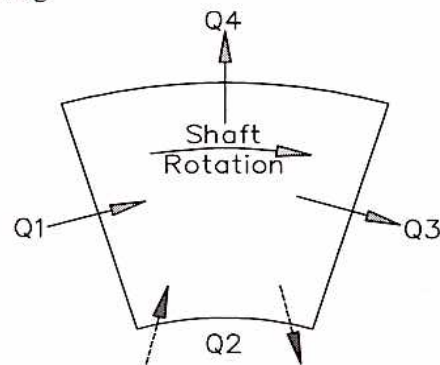


Figure 3. Schematic of oil film flow.

The test program reported here studied the effect of oil flow rate on both bearing power losses and pad temperatures for this specific bearing design. This was done over a range of shaft speeds and thrust loads, with both steel backed and chrome-copper (CrCu) backed, babbitt faced thrust pads. In summary the variables were:

<u>Independent</u>	<u>Dependent</u>
Oil flow rate	Power loss
Shaft speed	Pad temperatures
Thrust load	Oil dT
Pad backing material	

2. TEST EQUIPMENT

The facility used for these tests is described in (9). Briefly, this is a DC motor driven, hydraulically loaded thrust bearing test stand capable of accommodating thrust bearings in the range of 250mm to 500mm (10 to 20 inches) outside diameter. The 750kw (1000 hp) variable speed geared drive has a maximum speed of about 10,000 rpm. The test and slave thrust bearings are enclosed in separate housings, and each thus operates against its own thrust collar, integral with the connected shafts. The test bearing was a 267mm (10.5 inch) eight pad thrust bearing of the design shown in Figure 1. The slave bearing was a 381mm (15 inch) eight pad thrust bearing. The lubricant used was an ISO VG32 turbine oil. The setup for the test bearing in its housing is shown in Figure 2.

The following were recorded for each test point:

1. Shaft rpm
2. Oil flow rate to the test thrust bearing
3. Oil flow rate to the slave thrust bearing
4. Oil inlet temperature
5. Oil discharge temperatures
6. Thrust load
7. Pad temperatures

Oil flow rates were readily varied with the normal test facility controls, as were shaft speed, oil supply temperature and thrust load. Oil supply temperature was held constant at 49°C +/-1°C (120°F +/-2°F) in all of the tests reported here. Power loss values were calculated based on the oil flow rate and the temperature rise of the oil from inlet to drain. As seen in Figure 2, no inactive side thrust bearing was used, and no oil was supplied to that side, so the temperature rise of the oil (dT) was the result of losses only from the test bearing. Pad temperatures were measured by miniature thermocouples embedded in the babbitt facing about 0.75mm (0.03

inches) below the pad faces. Four thermocouples were put into each of four pads in the eight pad bearing (alternate pads). These were located at the 60-75, 75-75, 50-85 and 85-85 positions. (Numbers refer to the radial-circumferential locations on the pad face expressed as percentages of the pad radial length and the pad circumferential arc length, respectively, from the inner-leading edge corner of the pad.) The pad temperatures given in the plots are the high average from any one location for the four instrumented pads.

The bearing housing had a tangential oil discharge passage leading from the annular groove around the thrust collar. Tests subsequent to those reported here (10) included pressure probes mounted flush with the bore of the discharge annulus. With this open tangential discharge and the oil flow rates used in these tests, no measurable pressure was found in this annulus.

3. TEST RESULTS

The initial tests were made at a constant shaft speed with three oil flow rates but with the thrust load varied over a wide range. This was done to check the consistency of the test data and to insure the satisfactory performance of the bearing. Loss and temperature data from these tests are shown in Figures 4, 5 & 6. The flows of 1.26 and 0.95 liters/sec (20 and 15 US gpm) are in the range that would normally be specified for this size bearing operating at the speed indicated. The resulting temperature rise in the oil flowing through the bearing is 15 to 20°C at the higher loadings (Figure 6). The differences in power loss and/or pad temperatures for these two flow values are small. Reducing the flow to 0.63 liters/sec (10 gpm) results in more pronounced differences (Figures 4 & 5).

Figures 7 through 11 better illustrate the influence of reduced flows on power losses and pad temperatures, and are a representative sample of the data obtained for the variables tested. In all of these plots the calculated Q4 values are indicated by the vertical dashed lines intersecting the associated loss and temperature curves with solid circles.

Figure 7 compares the loss and temperature data for the two pad backing materials tested. For the same conditions the higher losses for the chrome-copper backed pads are due to the reduced oil film (and pad) temperatures, and the resulting increase in the viscosity of the oil in the film.

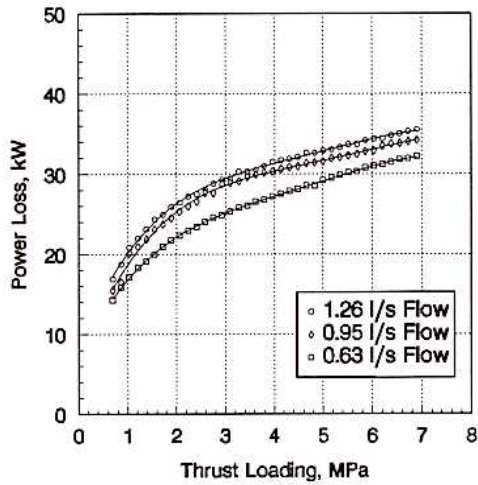


Figure 4. CrCu backed pads, 6000 rpm.

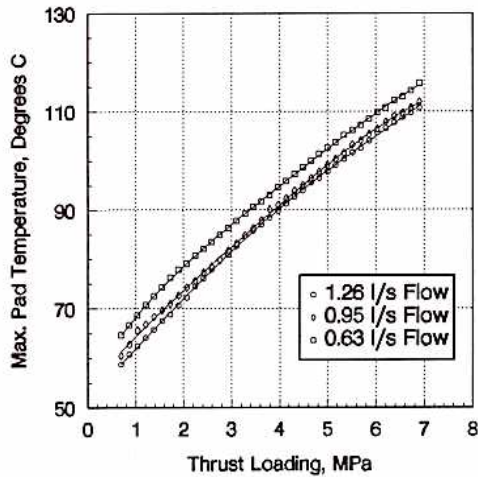


Figure 5. CrCu backed pads, 6000 rpm.

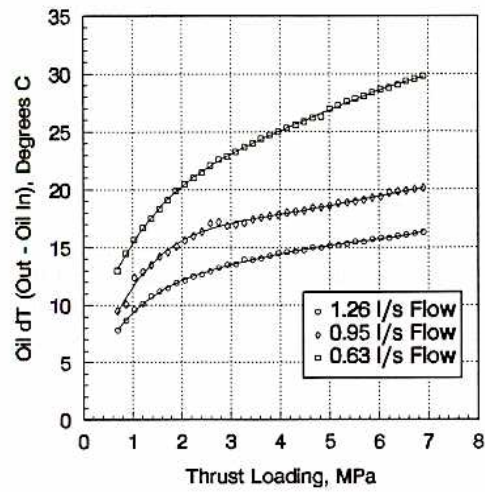


Figure 6. CrCu backed pads, 6000 rpm.

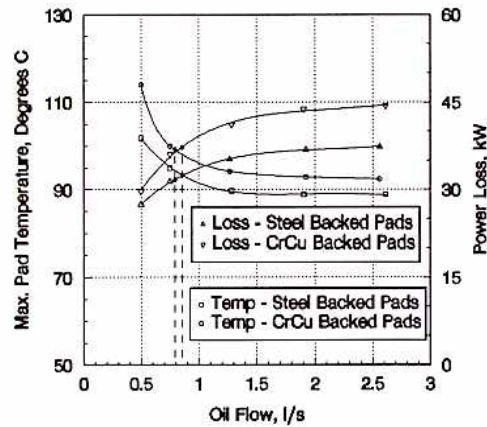


Figure 7. 3.45 MPa loading, 7000 rpm.

Figure 8 gives loss and temperature data for three different shaft speeds. The losses increase significantly with speed, as expected. The highest pad temperatures develop at the highest shaft speed only at lower flow values. At the higher flow values the lower temperatures at 9000 rpm as compared to those at 7000 rpm are believed to be the result of oil film turbulence. (Calculations indicate that turbulence is present over a portion of the oil film.)

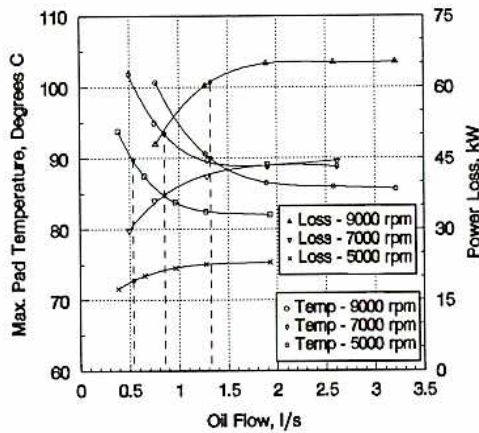


Figure 8. CrCu backed pads, 3.45 MPa loading

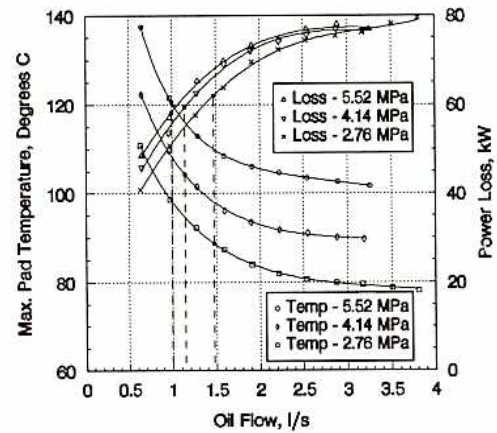


Figure 10. Steel backed pads, 9000 rpm.

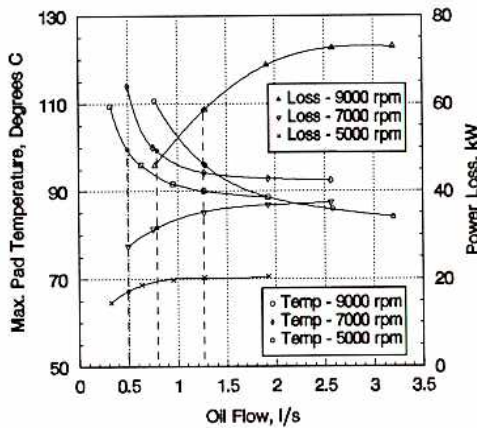


Figure 9. Steel backed pads, 3.45 MPa loading.

Figure 9 is the same as figure 8 except for steel backed pads. At the higher oil flows the pad temperatures are lower at 9000 rpm than at either 7000 or 5000 rpm.

Figure 10 shows losses and flows vs oil flow rate for three different loadings, at a constant shaft speed. Pad temperature is a strong function of load, but the variation in losses is relatively small.

Figure 11 is similar to Figure 10 except for chrome-copper backed pads, and different loadings.

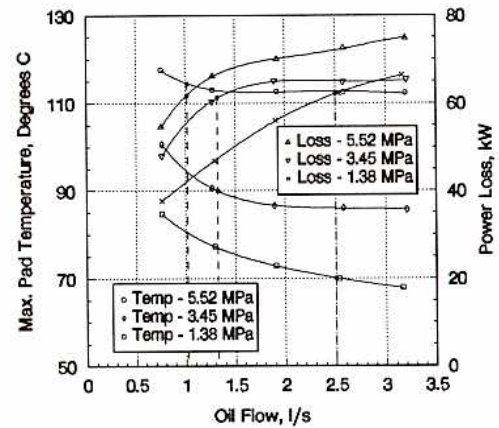


Figure 11. CrCu backed pads, 9000 rpm.

4. DISCUSSION

The primary interest here was to measure the effects of reduced oil flows on thrust bearing pad temperatures and on the bearing power losses, knowing that losses can be reduced by reducing oil flows but realizing that pad temperatures can suffer at some point in this process. In addition, a comparison of calculated oil requirements (Q4 values) with the test results was of interest.

As the flow rate is increased, pad temperatures and power losses tend to reach levels where further increases in flow result in small to no change in either of these. This is reflected in Figures 7 through 11. As the flow is decreased below a certain level, however, the slopes of both the temperature and loss curves change at an increasing rate.

It could be anticipated that the Q4 oil flow values would intersect these curves at the start of the "knee" of each curve. That is, as flow is reduced below the minimum required for a full hydrodynamic film, the film thickness will then decrease and the pad temperatures will increase. The losses will decrease as a result of the increased film temperatures (decreased oil viscosities) and also due to an oil film that is less than complete over the face of the pads. In all of these plots, however, the Q4 values intersect the loss and temperature curves within the "knee" of each curve. This can be the result of one or more factors including; 1) a lack of accuracy in calculating the Q4 values, 2) the flow in excess of Q4 provides cooling to reduce the pad temperatures and similarly increase the losses, and 3) even with the design of this bearing where oil is fed directly to the leading edges of the pads there are still leakage paths that prevent a flow of Q4 from being used with 100% efficiency.

The calculated Q4 value increases with shaft speed but decreases with thrust load. The latter is most clearly shown in Figure 11 which covers the broadest range of loadings. At the lowest loading (1.38 MPa) the Q4 value is 2.5 l/sec, a large flow for this bearing and these operating conditions, but still to the left of the knee of both the temperature and loss curves. This is not to imply that higher flows should be supplied to lightly loaded bearings since even at flows well below Q4 the losses and temperatures are less than those at higher loadings. Flutter of unloaded pads is also flow related and can be a problem (11).

From a design standpoint, some relationship between the calculated Q4 value and the specified flow for a specific application would be helpful. Pad temperatures are commonly monitored in

turbomachinery thrust bearings and upper limits set, typically as alarm and shutdown points. Although it is the combination of local oil film pressures and corresponding temperatures that determines the limit of the babbitt material (12), this combination is not readily measured, and babbitt temperatures are relied on for judging the operating condition of babbitt bearings. (Temperature limits can vary depending on the application, however. High speed, lightly loaded bearings can safely operate at higher pad temperatures than lower speed, heavily loaded bearings, for example.)

In the data here, as in all of the tests in this investigation, the bearings operated without failure to loadings of 6.9 MPa (1000 psi) and speeds to 9000 rpm with flows at the Q4 levels and somewhat below. In all cases, pad temperatures were lowered by increasing the oil flow above the Q4 values. It is suggested that flows for a specific application be set considering Q4 (at the rated load for the bearing and at the design speed) as a minimum, and use a multiplier on this to set the design flow. For the specific bearing design tested here, a multiplier of about 1.5 is suggested, to move the operating point towards the flat portion of the temperature curve. This increases the losses somewhat but provides a higher margin of safety.

5. SUMMARY

Lubricant supplied to a tilting pad thrust bearing is used to generate the hydrodynamic fluid film between the faces of the pads and the thrust collar, to carry away the majority of the heat energy developed in these films, and to provide additional cooling of the bearing parts. Tilting pad thrust bearing designs that direct the lubricating oil to the leading edges of the thrust pads make more efficient use of the oil flow than designs that essentially flood the inlet areas of the bearing. These bearing designs allow some reduction of oil flow rates, which reduces the size and energy requirements of the equipment needed to supply this oil, and also reduces the energy losses associated with this oil flowing through the bearing itself. Reducing the oil flow rate can, however, increase the bearing operating temperatures.

Laboratory tests were made to measure these loss and temperature levels, with various flow rates, for one type of thrust bearing design that provides oil directly to the leading edges of the thrust pads. The results of these tests show small to no changes initially in both temperatures and losses as flows are reduced from relatively high levels. Significant changes in the slopes of both the loss and temperature curves (vs flow) develop at further reduced flow rates. The reduction in losses is beneficial but the increase in pad temperatures erodes the safety margin of the bearing.

Comparison of the calculated hydrodynamic flow (minimum flow required to provide full oil films at the pad faces) with the test flows, and the associated loss and temperature curves were made.

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