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Evaluating Tilting-Pad Thrust Bearing Operating Temperatures©

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An analysis of tilting-pad thrust bearing operating temperatures is described. The analysis includes an evaluation of various temperature locations based on their ability to reflect bearing operating conditions. Experimental results for a 267-mm (10.5-in) 8-pad thrust bearing indicates that while a babbitt location on the trailing edge of a pad is often the hottest, the 75/75 percent location provides temperature data on the high-pressure region of the pad. Babbitt temperatures also provide a convenient indicator of the transition from laminar to turbulent flow in the oil film.

INTRODUCTION

Bearing temperatures generated during the operation of a tilting-pad thrust bearing can provide a convenient means of assessing bearing performance. Widespread use of thermocouples and resistance temperature detectors (RTD's) has made possible the collection of this potentially valuable temperature data. Unfortunately, selecting the proper location(s) to place these devices is complicated by the numerous candidates within the bearing or lubricant. Unless the proper location(s) is monitored, the temperature measurements made will prove to be very poor indicators of overall bearing performance. Selection should be based on the ability of the selected location to reflect actual operating conditions, and not on the ease of installation of the measuring device.

This paper presents the results of a series of tests conducted on a tilting-pad thrust bearing to investigate the temperature behavior of various locations. Thermocouples were employed to measure the temperatures of the bearing pads and the lubricant.

The bearings were evaluated using a light turbine oil with a viscosity of 0.027 Pa·s @ 37.8°C and 0.006 Pa·s @ 98.9°C (150 ssu @ 100°F and 43 ssu @ 210°F) supplied at 46°C (115°F), for applied loads ranging from 0–3.45MP_a (0-500 psi) and shaft speeds ranging from 4000 to 13 000 rpm. The influence of lubricant supply method and flow rate are also evaluated. By reporting the influence of operating conditions on various locations within the bearing and lubricant, it is hoped that this paper will provide the information necessary to select the proper location to monitor bearing performance.

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TEST BEARING DESCRIPTION

This discussion is based on bearing temperature data which resulted from a series of tests conducted using a 267-mm (10.5-in) tilting pad, equalizing, double-thrust bearing arrangement.

The primary test bearing had eight, babbitted and steel-backed, heavily instrumented pads or "shoes" on each side of a rotating collar for an (8 × 8) double-thrust bearing configuration. The pads had a babbitt OD of 267 mm (10.5 in) and a bore of 133 mm (5.25 in), for a total bearing area of 356 cm^2 (55.1 in^2). Each pad subtended approximately 38° of arc and had a centrally located (both radial and circumferential) spherical radius support. The tests were conducted using a light turbine oil with a viscosity of 0.027 Pa·s @ 37.8°C (150 ssu @ 100°F). Additional information regarding test apparatus and arrangement of the two bearing elements in the housing can be found in Ref. (*1*).

All tests were conducted with a restriction on the discharge. The bearing collar was shrouded with an oil control ring that was bored with a 3.97-mm (5/32-in) radial clearance over the collar diameter, and fitted with a 25.4-mm (1.0-in) tangential discharge port.

Thermocouples puddled in the babbitt itself, approximately 0.8 mm (1/32 in) below the actual babbit surface were used to measure the pad operating temperatures. Thermocouples were also placed in the lubricant supply and drain lines to measure these oil temperatures. Thermocouple location across the babbitt face of the pad is shown in Fig. 1.

BEARING OPERATING TEMPERATURES

One device used to measure the temperature performance of a thrust bearing is the thermocouple junction.

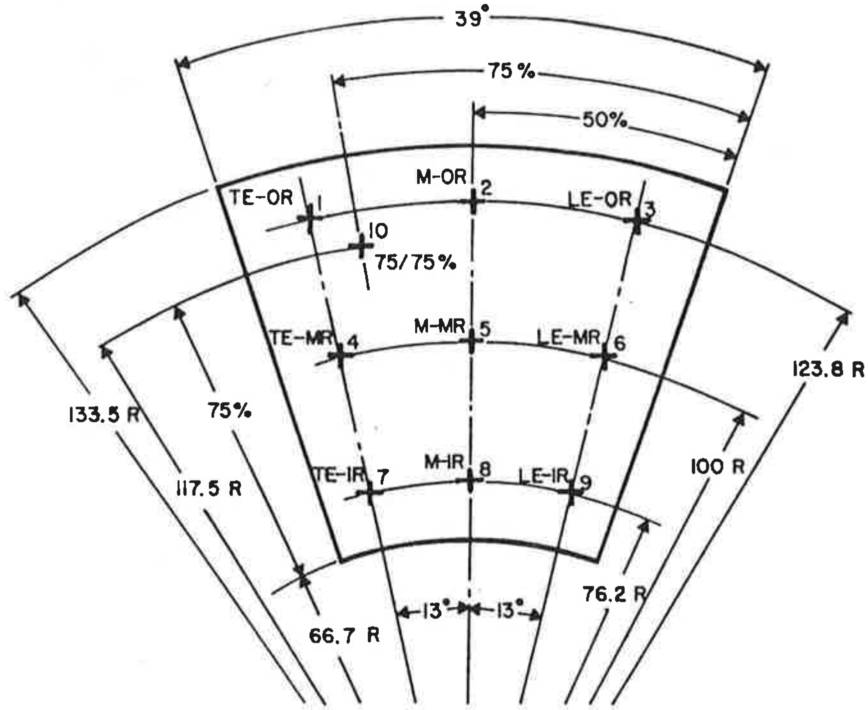

Fig. 1—Location of bearing pad thermocouples puddled in the babbitt

The thermocouple junction is ideally suited to the very tight geometry of the typical thrust bearing installation. The small size and inherent flexibility of the wires associated with the junction makes possible a very comprehensive temperature monitoring system for a thrust bearing.

The purpose of any thrust bearing temperature monitoring system should be to provide data on operating temperatures that effectively gauge operating risk. To accomplish this, the placement of the measuring device must be such that it is located in an area on the pad surface that will likely be exposed to maximum temperature and pressure. This location is necessary because the high-tin-content babbitt used on the working surface of the bearing pad loses its tensile and compressive strength, and is subject to creep at elevated temperatures.

Figure 2 is representative of the radial and circumferential temperature gradients that occur across the pad surface. This temperature profile is a good indicator of the relative temperature values associated with location on the pad surface. In order to reduce clutter, only four out of the ten locations shown in Fig. 1 will be used to monitor bearing performance—these are:

Location Number	Location Name	Abbreviation
4	Trailing edge middle radius	TE-MR
5	Middle-middle radius	M-MR
6	Leading edge middle radius	LE-MR
10	75% from leading edge and 75% up from the pad inner radius	75/75%

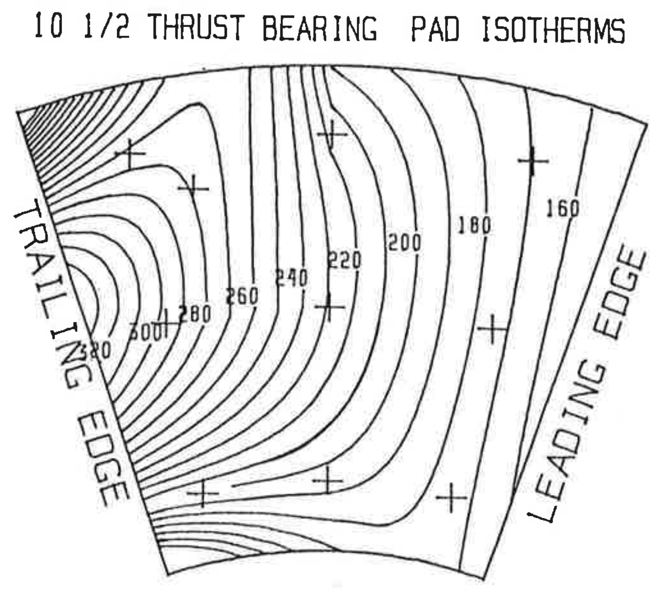

Fig. 2—Bearing pad isotherm showing the thermal gradient across the pad at a shaft speed of 10 000 rpm and a load of 3.45 MPa (500 psi) with 100 percent of recommended flow rate. Temperatures shown are in °F.

While each of the four babbitt locations is capable of reflecting changes in operating conditions, only two of these—75/75 percent and TE-MR are located in the area of maximum pad temperature. Although the maximum babbitt temperature occurs more often at the TE-MR location, the 75/75 percent location, because of the oil-film pressure-temperature combination (*4*), provides a more accurate picture of bearing risk.

LOAD VARIATIONS FOR A CONSTANT SHAFT SPEED

Figure 3 graphically illustrates the effects of not only location, but also bearing load on the pad temperatures. The LE-MR location is the lowest temperature recorded, while each location change toward the minimum oil film location of TE-MR reflects the higher temperatures associated with a reduced film thickness condition. Also evident are the increased temperatures for all locations as the hydrodynamic wedge changes to compensate for increased bearing load. Changes in the LE-MR temperature occur because of changes in the temperature of the preceding pad's TE-MR temperature. Increased TE-MR temperatures are reflected in higher LE-MR temperatures because as the increased temperature of the oil carried by the collar (or runner) is brought to the LE-MR of the next pad, it is mixed with constant temperature inlet oil, resulting in higher oil mixture temperatures. The nearly perfect parallel curves seen in Fig. 3 representing the M-MR, 75/75, and TE-MR locations as a function of bearing load and temperature clearly demonstrate the effects of location and load on temperature reported.

Each bearing pad location is represented by its own separate curve as a function of temperature and bearing load. At each and every bearing load, a definite ranking occurs based on temperature:

Coolest — LE-MR (*6*)
2nd coolest — M-MR (*5*)
2nd hottest — 75/75% (*10*)
Hottest — TE-MR (*4*)

This ranking, as mentioned before, is a result of the oil-film thickness at those particular locations (see Fig. 1). On each individual curve, the effects of increased load also becomes apparent; as the load is increased, the oil film is decreased—which results in higher bearing pad temperatures. The resulting uniform increase of 44 percent in temperature over the load range of 100 to 500 psi for each of the three locations—M-MR, 75/75, and TE-MR—is indicative of the relative movement of each location that would be associated with each change in the supporting hydrodynamic wedge.

SHAFT SPEED VARIATIONS FOR A CONSTANT LOAD

Shaft speed changes for a bearing under a constant load condition affect the general level of bearing temperatures by virtue of the changes in the relative rate of viscous shear associated with changes in velocity.

Figure 4 shows bearing pad temperature changes as a function of shaft speed changes. From 4000 to 9000 rpm, everything is as one might expect—a gradual increase in temperature with the increase in the rate of viscous shear associated with changes in shaft speed. The temperatures reach a peak at 9000 rpm and then experience a sharp drop until 11 000 rpm, from which point a much reduced rate of increase is resumed.

The sharp reduction in the M-MR, 75/75, and TE-MR temperatures after 9000 rpm might, at first glance, cause some problems for the analysis advanced so far; but within the framework of that analysis also lies the answer for falling rather than rising temperatures with increasing shaft speeds. The explanation lies in the transition from laminar to turbulent flow within the bearing. Shaft speed, oil viscosity, and oil flow are all factors in the change from laminar to turbulent conditions. A more detailed explanation of the mechanics of laminar vs turbulent flow in tilting-pad thrust bearings can be found in Refs. (*3*), (*7*) and (*8*).

When the flow changes from laminar to turbulent, the different layers, each with their own rate of viscous shear (source of heat generation), start to mix with each other, bringing down the overall temperatures of the bearing pad.

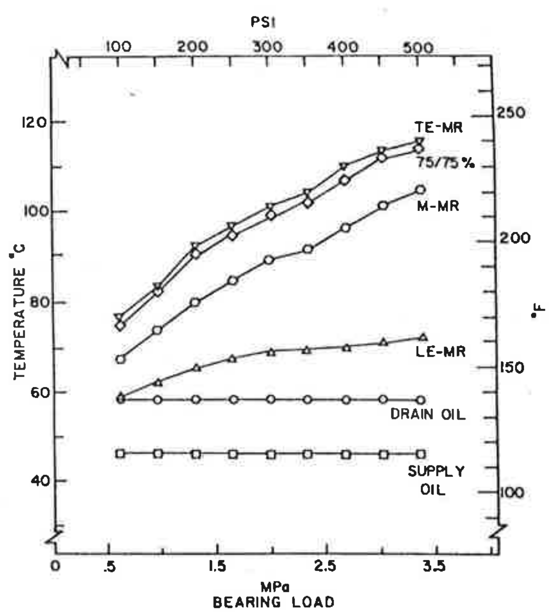

Fig. 3—A comparison of the temperatures recorded at selected locations for a bearing operating at a shaft speed of 4000 rpm for loadings that ranged from 0 to 3.45 MPa using a lubricant supplied at 100 percent of recommended flow rates at a temperature of 46°C.

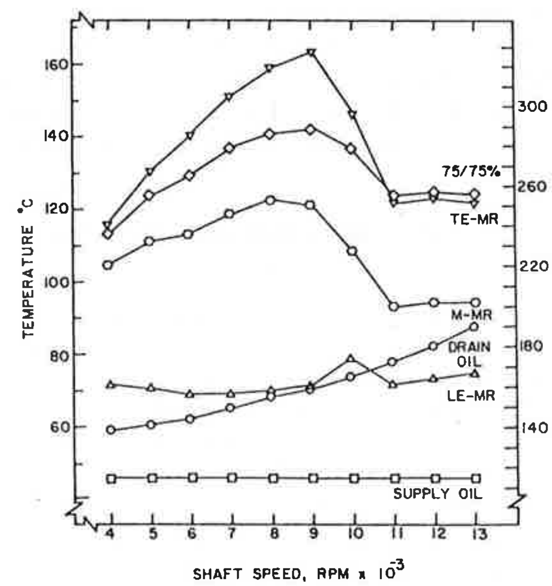

Fig. 4—A comparison of the temperatures recorded at selected locations for a bearing operating at a load of 3.45 MPa for shaft speeds that ranged from 4 to 13 000 rpm using a lubricant supplied at 100 percent of recommended flow rates at a temperature of 46°C.

At some point, in this case 11 000 rpm, the rate of viscous shear and the mixing reach a point where the temperatures stop falling and start climbing again, but at a much reduced rate.

Another point to be made is the changing, at 11 000 rpm, of hottest bearing pad location from the TE-MR position to the 75/75 position. This move of minimum film thickness location from the TE-MR inboard to the 75/75 location is caused by pad deflection at the trailing edge, causing an increase of oil film at that location. It should be noted that the pad distortions that shift the maximum temperature position toward the 75/75 percent location can be a function of bearing size. The shift can be more pronounced in large bearings.

VARIATIONS IN OIL FLOW RATE

Of the total quantity of oil that is supplied to a thrust bearing, only a small portion will actually find its way into the oil film that supports the applied load (*5*). The remainder of the oil is utilized to provide beneficial cooling of the bearing components. This additional cooling oil, which is not utilized by the oil film, extracts a price in the form of overall power losses by virtue of pumping and churning losses. Because of this tradeoff, a compromise must be reached to balance the cooling benefits with the power loss penalty. The standard recommended oil flow rate is felt to be such a compromise. The standard recommended oil flow rate for the tilting-pad thrust bearings used during the tests was based on maintaining a suitable temperature rise from supply to discharge as determined by theoretical power loss calculations. A discussion of oil supply flow rate determinations can be found in Ref. (*2*), and in the discussion of drain oil temperatures in this paper.

Any oil flow rate greater than the minimum required by the oil film will have only a limited effect on pad babbitt temperatures. Increases in oil flow rates over this minimum serve to create a more stable overall cooling effect within the bearing. On the other hand, bearings operating under less than minimum film conditions are "oil-starved," and any increase in oil flow rate would cause a significant reduction in pad temperatures with increased film thickness.

FIFTY PERCENT OF RECOMMENDED OIL FLOWS

Contrasting Fig. 3 with Fig. 5 demonstrates the effects of restricted oil flows on pad babbitt temperatures. The same almost-perfect parallel curves for M-MR, 75/75, and TE-MR are found with the reduced flows as were found with the full recommended flows. Even the 44 percent increase in the three temperatures from 100 psi to 500 psi bearing load was identical—the only difference being general level of increase for all temperatures of about 7–9°F. This would tend to support the contention that as long as a minimum amount of oil is being supplied, any additional oil only serves to cool bearing components and does not significantly affect pad babbitt temperatures.

A comparison of the 50 percent of recommended flow data in Fig. 6 with that of the recommended flow data in

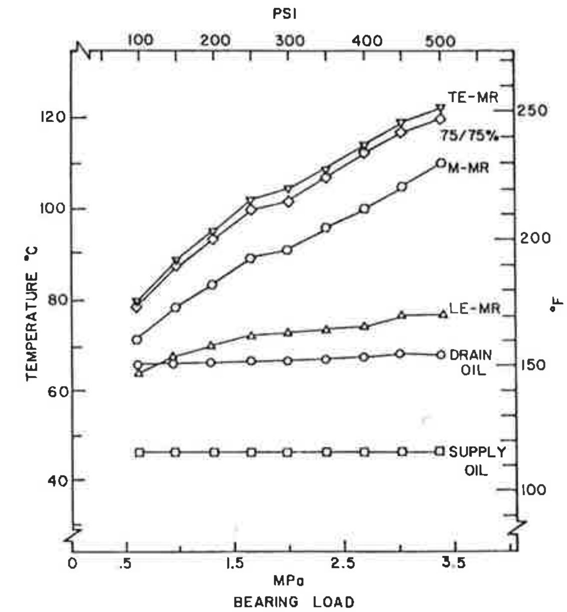

Fig. 5—A comparison of the temperatures recorded at selected locations for a bearing operating at a shaft speed of 4000 rpm for loadings that ranged from 0 to 3.45 MPa using a lubricant supplied at 50 percent of recommended flow rates at a temperature of 46°C.

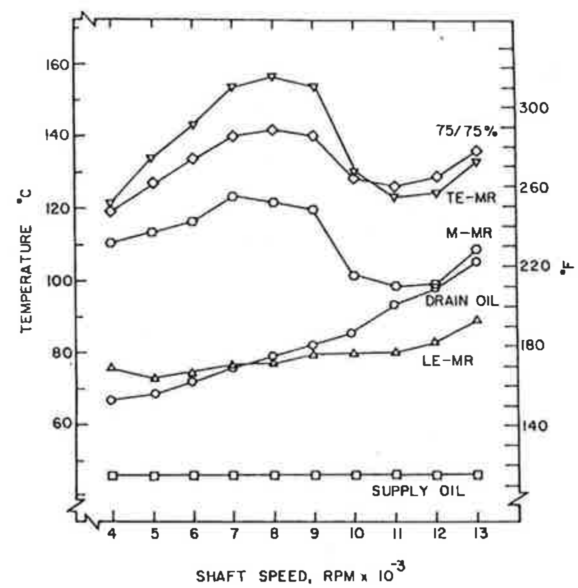

Fig. 6—A comparison of the temperatures recorded at selected locations for a bearing operating at a load of 3.45 MPa for shaft speeds that ranged from 4 to 13 000 rpm using a lubricant supplied at 50 percent of recommended flow rates at a temperature of 46°C.

Fig. 4 isolates pad temperature differences attributable to changes in shaft speed. Again, the general level of all the temperatures is slightly higher with the reduced flow, and the rate of increase with shaft speed is almost identical from 4000 to 7000 rpm. After 7000 rpm, the reduced flow bearing starts to enter the turbulence region, causing a much smaller increase in pad temperatures between 7000–8000 rpm, and then actual reductions in pad temperatures starting at 9000 rpm and lasting until 11 000 rpm. This earlier onset of turbulence accounts for the lower value of maximum pad babbitt temperature in the reduced flow bearing.

It should also be noted that the reduced flow bearing experiences higher rates of pad temperature increases after 11 000 rpm, which is indicative of reduced superfluous flow.

150 PERCENT OF RECOMMENDED FLOWS

The curves in Fig. 3, representing recorded temperature data generated during tests run with the standard recommended flows, are almost exactly identical to curves in Fig. 7, which represents the same temperature data collected during tests run with 150 percent of the recommended flows. The only difference to be noted between the two sets of curves is that of the LE-MR oil temperature. The LE-MR location is cooler because 50 percent more oil flow is available to dilute the high-temperature oil carried over from the TE-MR location of the previous pad, as well as providing beneficial cooling to bearing parts.

Differences in temperature attributable to changes in shaft speed (Fig. 4 vs Fig. 8) start to occur at 10 000 rpm due to the onset of turbulence. The 150 percent flow data indicate that the onset of turbulence is just starting at 10 000 rpm, while for the standard recommended flows, the large reduction in temperature would suggest that turbulence is well under way.

Again, the temperature data collected would suggest that changes in oil flow rate above a certain miminum rate required to maintain a minimum oil film have little affect on bearing pad temperatures until the onset of turbulence at the upper end of the speed range.

DRAIN OIL TEMPERATURES

A review of Figs. 3, 5, and 7 provides an interesting insight into the ability of the drain oil temperature to reflect changes in bearing load. Specifically, a change in bearing load is not necessarily reflected by a change in drain oil temperature.

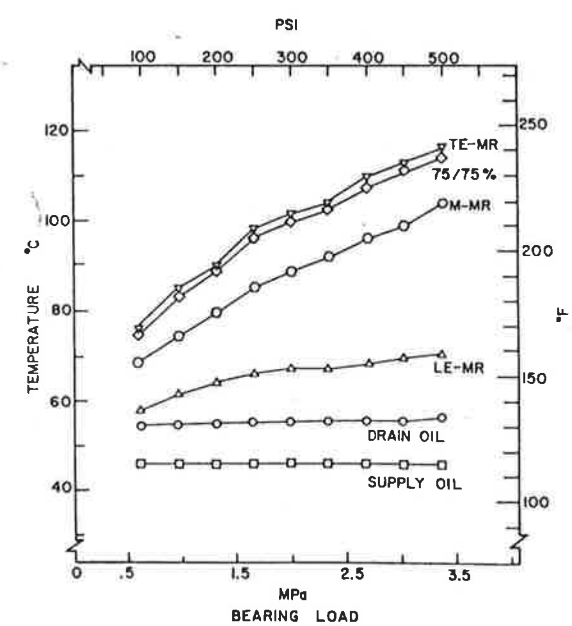

Fig. 7—Same as Fig. 5 except 150 percent

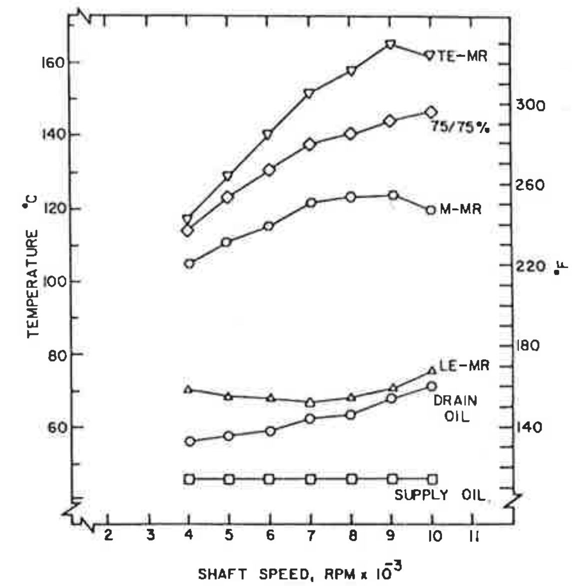

Fig. 8—Same as Fig. 6 except 150 percent and 4 to 10 000 rpm

As mentioned previously, only about 5–10 percent of the total oil supplied to the bearing ever finds its way into the oil film. As a result, changes in oil-film temperature produced by changes in load can be diluted by adjustments to the oil supply flow rate. This thermal balancing of the bulk oil temperature is reflected by the constant drain oil temperature reported by Fig. 3. The flow adjustments necessary to neutralize the increases in oil-film temperature associated with increased load can be found plotted on Fig. 9 (4000 rpm).

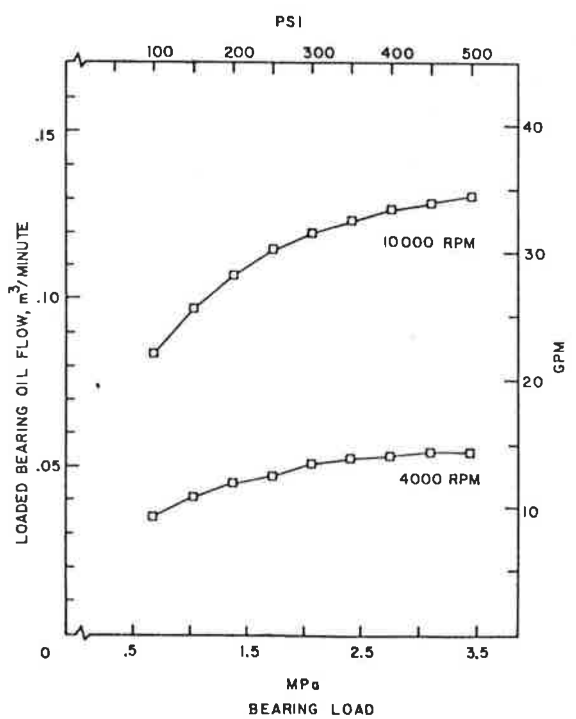

Fig. 9—A comparison of the 4000 and 10 000 rpm shaft speed 100 percent recommended oil flow rates to the loaded bearing for loads that range from 0 to 3.45 MPa using an ISO VG 32 lubricant supplied at 46°C.

Figures 4, 6, and 8 show the influence of surface speed on drain oil temperature—namely, the greater the surface speed, the higher the drain oil temperatures. Unfortunately, at the higher surface speeds, the ability of drain oil temperatures to monitor bearing load remains unchanged. The same relationship between oil flow rate and drain oil temperature that existed at 4000 rpm, also can be found at 10 000 rpm (see Figs. 9 and 10). Comparing the two curves on Fig. 9 reveals that the oil flow change, on a percentage basis, necessary to compensate for load is almost identical.

The influence of oil flow rate on drain oil temperature is illustrated in Figs. 3 to 8. Oil flow rate adjustments of 50 percent produced parallel shifts of the drain oil temperature curves up or down, depending on whether the amount of oil was increased or decreased. This is attributed to only the slight influence that oil film temperature has on bulk oil temperature (*5*).

One of the most critical factors influencing bearing power loss is the oil supply flow rate. Oil supplied to the bearing, which is not utilized directly in the formation of the load-supporting film, enacts a penalty in the form of increased power loss due to churning and pumping of the excess oil. Not surprisingly, thrust bearing designs have emerged that utilize significantly reduced oil flow rates. As demonstrated previously, this increases the drain oil temperatures, but tells you nothing about babbitt temperatures.

Figure 11 contrasts the maximum measured babbitt and drain oil temperatures of a conventional and LEG [leading edge groove—see Ref. (*6*) for a description] design thrust bearing. What should be noted is that the relative drain oil temperatures do not provide a very accurate picture of the relative babbitt temperatures, which reinforces the findings of other studies (*5*).

CONCLUSIONS

1. Drain oil temperature, of the five locations monitored, was found to be the least likely to reflect bearing operating conditions.

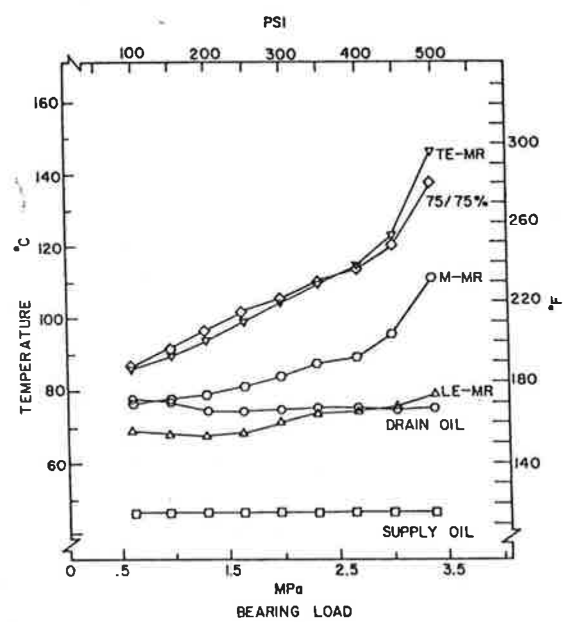

Fig. 10—Same as Fig. 4 except 10 000 rpm

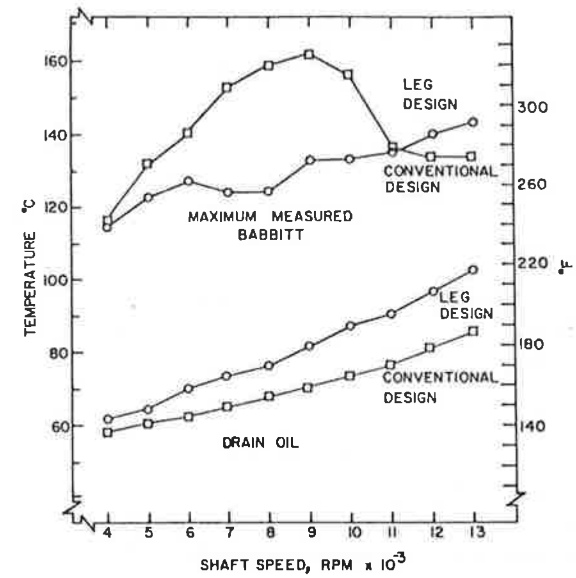

Fig. 11—A comparison of maximum measured babbitt temperatures and drain oil temperatures of conventional and leading-edge-groove bearings operating at a load of 3.45 MPa for shaft speeds ranging from 4 to 13 000 rpm using an ISO VG 32 lubricant supplied at 46°C. The conventional design supplied with 100 percent and the leading-edge groove design supplied with 35 percent of the recommended oil flows.

2. Drain oil temperatures respond more to changes in the oil flow rate than to changes in the operating conditions.
3. Monitoring the 75/75 percent babbitt location temperature is an effective way to evaluate bearing operating risk.
4. Babbitt temperatures provide a convenient indicator of the transition from laminar to turbulent flow in the oil film.
5. Oil flows greater than the minimum required by the oil film will have only a limited effect on pad babbitt temperatures until the onset of turbulence.

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