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The Effect of Lubricant Supply Temperature on Thrust Bearing Performance©

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This paper compares and analyzes the influence of lubricant supply temperature on the performance of a tilting-pad, equalizing, thrust bearing. The paper presents experimental data for a 267-mm (10½-in OD) bearing, operating at shaft speeds up to 13 000 rpm with loads ranging up to 3.45 MPa (500 psi). The data presented demonstrate the relative effect that changes in lubricant supply temperature have on bearing power loss and babbitt temperature. Some conclusions are drawn based upon the trend in relative performance for each lubricant supply temperature tested.

INTRODUCTION

One of the parameters that influences the performance of a fluid film thrust bearing is the lubricant supply temperature. Decisions regarding the lubricant supply temperature are not only reflected in the initial cost of a project, but also in bearing performance and, ultimately, operating cost. Expensive energy has given bearing power loss added importance when evaluating design alternatives. Unfortunately, except for specialized bearing designs (*1*), (*2*), most techniques employed that reduce bearing power loss in a conventional thrust bearing design do so at the expense of the bearing pad or "shoe" operating temperatures. Increasing the lubricant supply temperature to reduce the effective viscosity is no exception, but the question of just how much the power loss and babbitt temperature will be affected remains unanswered. The purpose of this paper is to provide the information necessary to evaluate the effect of lubricant supply temperature changes on bearing power loss and maximum babbitt temperature, based on actual performance data.

The effect of lubricant supply temperature was evaluated on a tilting-pad, equalizing, (6×6) double-thrust bearing arrangement. The bearing was tested using a light turbine oil which had a viscosity of 0.027 Pa.s @ 37.8°C and 0.005 Pa.s @ 98.9°C (150 SSU @ 100°F and 43 SSU @ 210°F – ISO VG32). The viscosity-temperature curve for this lubricant is shown in Fig. 1. The lubricant supply temperature was varied from 43.3°C to 65.6°C in 5.6°C (110°F to 150°F in 10°F) increments. The shaft speed ranged from 5000 rpm to 13 000 rpm and the load ranged from a "no-load" condition to 3.45 MPa (500 psi) in increments of 0.345 MPa (50 psi).

TEST BEARING DESCRIPTION

The test bearing was a 267-mm (10.5-in) tilting-pad, equalizing, double-thrust bearing. Each individual element of the double-thrust bearing consisted of six steel-backed and babbitt-faced heavily instrumented pads or shoes on each side of a rotating collar for a (6×6) configuration. The shoes had a babbitted 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). Details of the arrangement of the two elements in the bearing housing can be found in Ref. (*3*).

Lubricant was supplied to the bearing by the conventional pressurized or controlled flow method. A detailed description of this lubricant supply method can be found in Refs. (*3*) and (*4*). The oil flow rates supplied to the bearing were

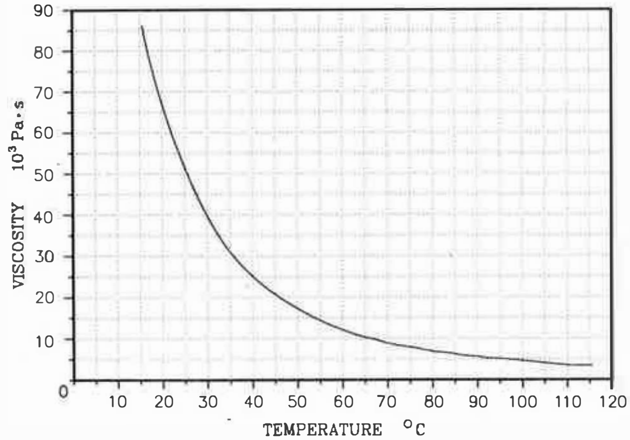

Fig. 1—Temperature-viscosity curve for the ISO VG32 lubricant

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the "standard flow rates" as described in Ref. (5). In addition, it should be noted that for all the lubricant supply temperatures tested, the standard flow rate was maintained by varying the lubricant supply pressure.

All tests were conducted with the bearing collar shrouded by an oil control ring bored to a 3.97-mm (5/32-in) radial clearance and fitted with a 24.5-mm (1.0-in) diameter tangential discharge port.

The bearings were instrumented with 43-J-type thermocouples puddled in the babbitt itself, approximately 0.8 mm (1/32 in) below the actual babbitt surface. Thermocouples were also placed in the lubricant supply and drain lines to measure these temperatures. The location of the thermocouples across the babbitt face of the pads is shown in Fig. 2.

TEST DATA

The data are presented in relative rather than absolute terms in the form of percentage changes from a benchmark value established at 48.8°C (120°F). The 48.8°C (120°F) benchmark was established because it is a value representative of most industrial applications. The percentage changes are calculated in the following manner:

$$Y = \frac{V - V_B}{V_B} \times 100$$

Y = percentage change (in power loss or temperature) from 48.8°C (120°F) for common load and shaft speed combinations.

V = power loss or maximum babbitt temperature recorded for a specified load, shaft speed and lubricant supply temperature combination.

V_B = power loss or maximum babbitt temperature at the same load and shaft speed as V for a lubricant supply temperature of 48.8°C (120°F).

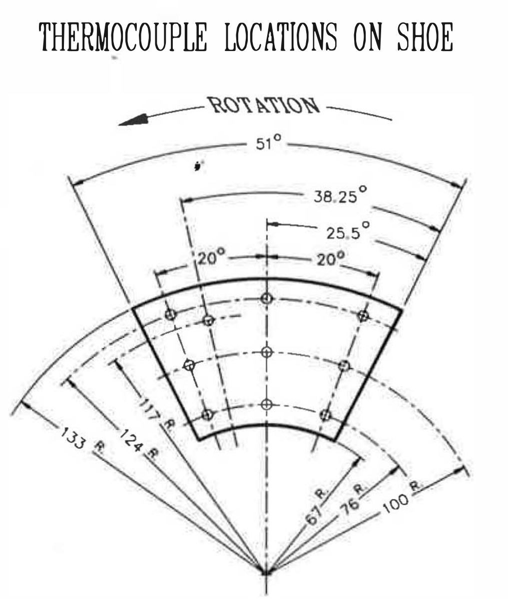

Fig. 2—Location of thermocouples puddled in the shoe babbitt

OPERATING TEMPERATURES

Comparisons of operating temperatures are made between maximum measured babbitt temperatures regardless of location on the bearing. Figures 3 through 5 compare the relative response, on a percentage basis, of the maximum measured babbitt temperature to changes in the lubricant supply temperature for bearing loads of 0.69, 2.07, and 3.45 MPa (100, 300, and 500 psi). Each increment of oil supply temperature change is 5.6°C (10°F) and represents an excursion of 8⅓ percent from the 48.8°C (120°F) benchmark. The excursions range from 8⅓ percent below to 25 percent above the benchmark.

The relative response of maximum babbitt temperature to changes in oil supply temperature for a bearing with a 0.69 MPa (100 psi) load at shaft speeds of 5, 7, 8, 9, 11, and 13 000 rpm is shown in Fig. 3. Although the extent of the oil supply temperature excursions ranges from 25 percent above to 8⅓ percent below the benchmark, the response of the maximum babbitt temperature is limited to 17.6 percent and −6 percent, respectively. Maximum babbitt temperature is not only influenced by oil supply temperature but also shaft speed, and both of these factors influence the onset of turbulence in the bearing's oil film. The various shaft speed and oil supply temperature combinations result in different laminar to turbulent transition points for each oil supply temperature. This is reflected in the erratic response of the maximum babbitt temperatures for both 8 and 9000 rpm. The remainder of the responses (5, 7, 11, and 13 000 rpm) were very predictable for the oil supply temperatures above the benchmark; the lower the shaft speed, the greater the influence on maximum babbitt temperature. The responses for the oil supply temperature below the benchmark demonstrated no discernible pattern.

Figure 4 shows the maximum babbitt temperature response for a bearing loaded to 2.07 MPa (300 psi). Differ-

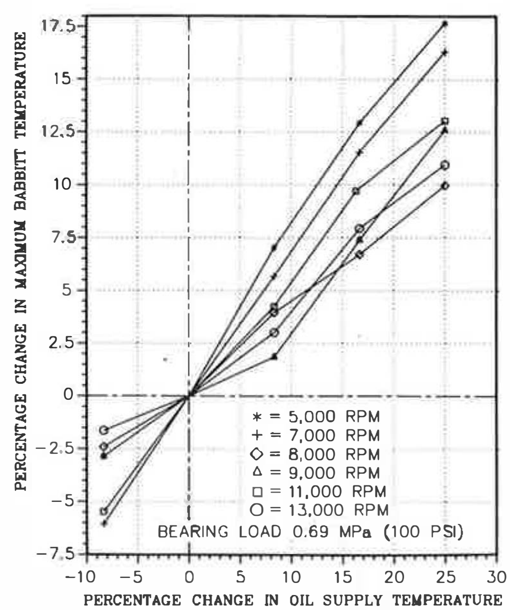

Fig. 3—A comparison of the percentage changes in maximum babbitt temperature for a percentage change in oil supply temperature for a bearing loaded to 0.69 MPa for shaft speeds of 5, 7, 8, 9, 11, and 13 000 rpm.

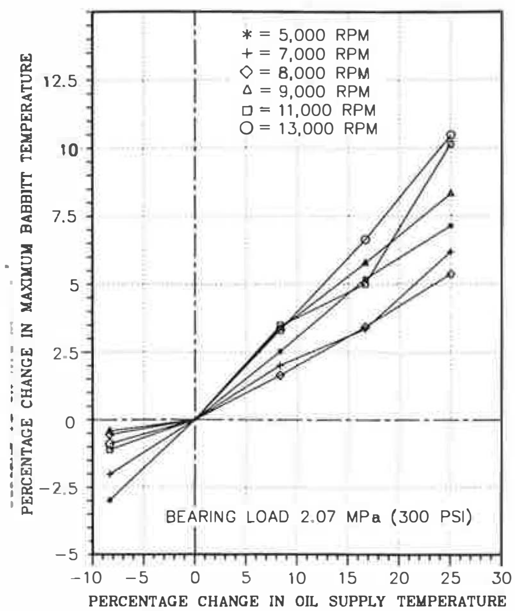

Fig. 4—A comparison of the percentage changes in maximum babbitt temperature for a percentage change in oil supply temperature for a bearing loaded to 2.07 MPa for shaft speeds of 5, 7, 8, 9, 11, and 13 000 rpm.

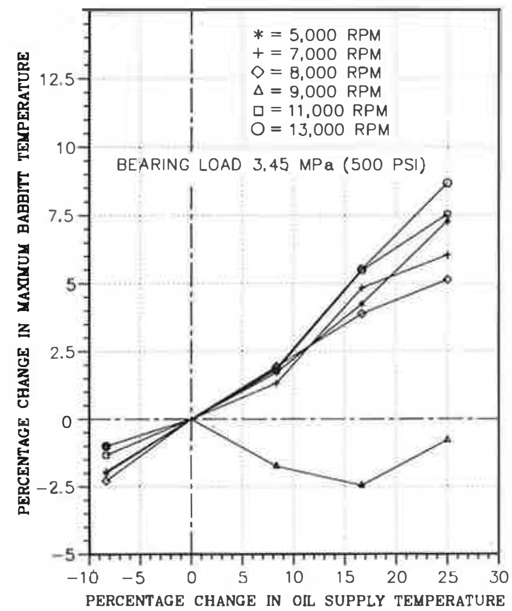

Fig. 5—A comparison of the percentage changes in maximum babbitt temperature for a percentage change in oil supply temperature for a bearing loaded to 3.45 MPa for shaft speeds of 5, 7, 8, 9, 11, and 13 000 rpm.

ences in the relative temperature response attributable to additional bearing load can be seen by comparing Figs. 3 and 4. Pronounced differences occur in the response of the babbitt temperatures for the oil supply temperatures both above and below the benchmark. The higher bearing load has significantly reduced the maximum babbitt temperatures excursions (17.6 vs 10.5 percent) above the benchmark. The overall response below the benchmark oil supply temperature is similar in that the babbitt temperature excursions have also been significantly reduced (-6 vs -3 percent).

The higher bearing load also caused the maximum babbitt temperature response for the shaft speeds above 8000 rpm to converge at about 3.4 percent when the oil supply temperature is increased 8⅓ percent. Additional increases in the oil supply temperature cause the babbitt temperatures to once again diverge. It is interesting to note that the effect of the additional load is almost nonexistent for a shaft speed of 13 000 rpm. Regrettably, the erratic babbitt temperature responses recorded at the other shaft speeds do not seem to exhibit any definitive pattern of behavior.

The response of maximum babbitt temperature to a high bearing load (3.45 MPa-500 psi) is shown in Fig. 5. The difference in the relative temperature response attributable to changes in bearing load can be seen by comparing Figs. 3, 4, and 5. Figure 5 shows that, once again, the effect of additional bearing load is to reduce the maximum babbitt temperature excursions to 8.7 percent above and -2.5 percent below the benchmark oil supply temperature. The additional bearing load has also increased the tendency of the temperature response at all shaft speeds, except 9000 rpm, and all oil supply temperatures to converge. The erratic babbitt temperature responses evident at the lower loads have been further moderated by the additional bearing load for shaft speeds above 9000 rpm. The reason for this can be attributed to the oil film thickness. The high sliding velocity generates high film temperatures which, in turn, reduce the effective viscosity of the oil in the film. Beyond a certain temperature, the temperature viscosity curve becomes fairly flat (See Fig. 2) thereby stabilizing the effective viscosity and, therefore, the oil film's response to changes in bearing load.

The contrast in turbulent transition points between the various oil supply temperatures produces a very interesting result: At a shaft speed of 9000 rpm, the maximum babbitt temperature excursions ranged from -0.75 to $-2\frac{1}{2}$ percent. This occurs because of the differences in babbitt temperature peaks (*6*), (*7*), (*8*). The explanation for this is that babbitt temperatures for the oil supply temperatures above the benchmark peak at 8000 rpm, while, for the benchmark and below, the peak occurs at 9000 rpm. This results in the maximum babbitt temperature of the benchmark being the highest at a shaft speed of 9000 rpm. Therefore, it should be noted that under a certain set of operating conditions (load, speed, oil flow rate, and oil supply temperature), increasing the oil supply temperatures may actually reduce babbitt temperatures. Replotting Fig. 5 so that shaft speed is the independent variable, yields Fig. 6.

The influence of shaft speed on the maximum babbitt temperature excursions from the benchmark oil supply temperature can be seen in Fig. 6. The most significant events on this plot are the turbulence induced "V's" at 9000 rpm. The "V's" result because the various combinations of operating conditions produce the different peak temperature points that have been previously mentioned.

The influence of bearing load on the maximum babbitt temperature excursions for the four oil supply temperatures can be summarized as follows: The low-load (0.69 MPa – 100 psi) bearing exhibited the greatest sensitivity to changes in the oil supply temperature both above and below the

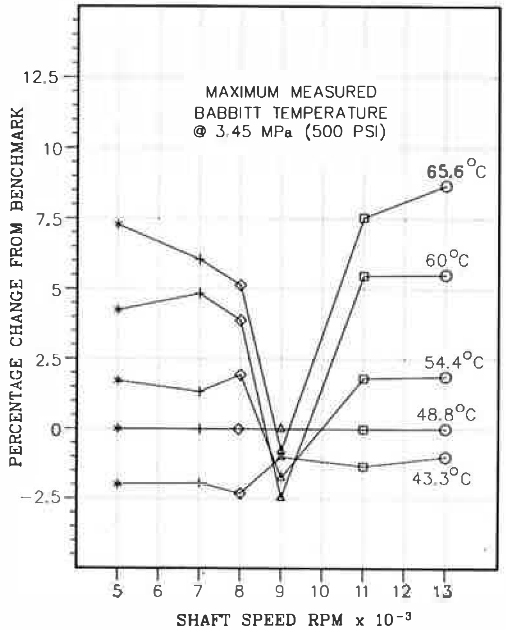

Fig. 6—A comparison of the percentage changes in babbitt temperature to changes in shaft speed for a bearing loaded to 3.45 MPa at oil supply temperatures of 43.3, 54.4, 60 and 65.6°C.

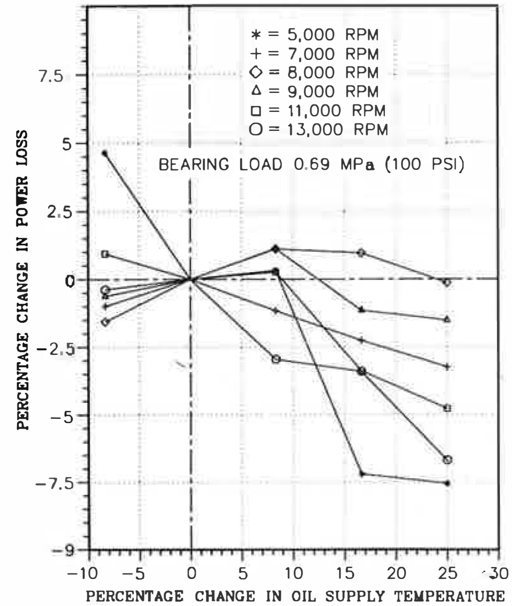

Fig. 7—A comparison of the percentage changes in bearing power loss and for a percentage change in oil supply temperature for a bearing loaded to 0.69 MPa for shaft speeds of 5, 7, 8, 9, 11, and 13 000 rpm.

benchmark. Almost without exception, the greatest excursions were those of the low-load bearing. The medium (2.07 MPa – 300 psi) and high (3.45 MPa – 500 psi) bearing loads were generally less sensitive to changes in the oil supply temperature. The higher bearing loads have a stabilizing effect on bearing temperatures. This is because, at the higher loads, the load-induced changes in oil film thickness are not as severe as the changes when the bearing is only lightly loaded.

POWER LOSS

Bearing power losses were established by the familiar energy balance technique whereby the loss is computed as a direct function of the measured oil temperature rise (supply to discharge), measured oil flow rate, and the specific heat of the lubricant. Omitted from this analysis were radiation losses from the housings and conduction losses via shafting and foundation because they are small and constant for the entire series of tests.

Bearing power losses are influenced by the oil supply temperature because of the temperature-viscosity relationship. (See Fig. 2.) As mentioned before, the higher the oil temperature, the lower the effective viscosity and, therefore, the lower viscous shear and power loss. Comparisons of the power loss responses for the various oil supply temperature and load combinations are shown in Figs. 7 through 9.

The power loss excursions (percentage) for the various supply temperatures at a bearing load of 0.69 MPa (100 psi) are plotted on Fig. 7. The response of all shaft speeds, except 7 and 13 000 rpm, to an 8⅓ percent increase in oil supply temperature is higher, not lower, power loss. An additional increase in oil supply temperature of 8⅓ percent puts all except 8000 rpm into the negative category. Increasing the oil supply temperature 25 percent finally puts the 8000 rpm excursion on the negative side. The maximum power loss excursion is only 7½ percent compared to the 25 percent increase in oil supply temperature. As was the case with the babbitt temperature responses, the power loss responses are erratic and seem to follow no particular pattern, except for 8 and 9000 rpm. Comparing the temperature response of 8 and 9000 rpm in Fig. 3 with the power loss response would suggest that these two shaft speeds are in the turbulent transition range.

The power loss response to a moderate load (2.07 MPA/300 psi) can be seen in Fig. 8. The additional load, with the exception of 5000 rpm, has caused the power loss responses of the individual shaft speeds to group themselves within a range of 3¾ percent. In fact, the response of both 11 and 13 000 rpm remained almost unchanged. Although the maximum power loss excursion has increased to almost –11 percent when the oil supply temperature is increased 25 percent, this increase is attributable solely to the response of 5000 rpm. Unfortunately, as was the case for the babbitt temperature excursions at this load, no definitive pattern seems to be suggested. One possible explanation for the large excursions at 5000 rpm is that, at this low shaft speed, the primary effect on bearing power loss is oil film shear and not pumping and churning losses.

The effects of a high bearing load (3.45 MPa/500 psi) on the power loss response for the various oil supply temperatures is plotted in Fig. 9. Once again, the excursions at 11 and 13 000 rpm are almost identical to those of the previous two loads (Figs. 7 and 8). One consequence of the additional load is the onset of the turbulent transition at 9000 rpm. The correlation between minimum babbitt temperature and maximum power loss is excellent (Figs. 5 and 9). The remainder of the shaft speeds tested do not yield very predictable responses. This can only be attributed to a shifting laminar-turbulent transition point.

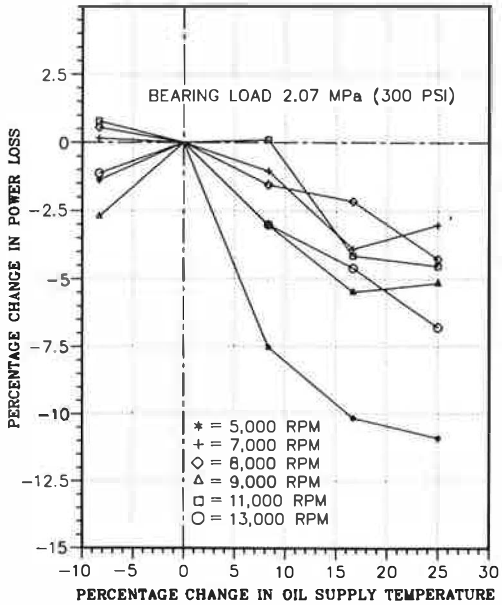

Fig. 8—A comparison of the percentage changes in bearing power loss for a percentage change in oil supply temperature for a bearing loaded to 2.07 MPa for shaft speeds of 5, 7, 8, 9, 11, and 13 000 rpm.

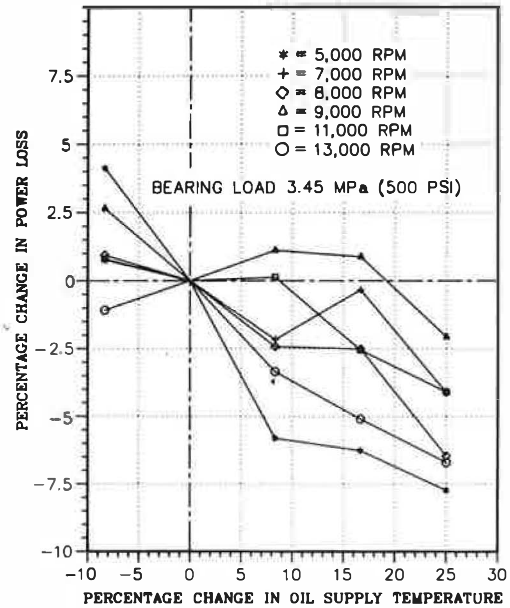

Fig. 9—A comparison of the percentage changes in bearing power loss for a percentage change in oil supply temperature for a bearing loaded to 3.45 MPa for shaft speeds of 5, 7, 8, 9, 11, and 13 000 rpm.

It is hoped that the information herein presented will prove useful to both the designers and users of rotating machinery in making decisions regarding changes to the oil supply temperature.

CONCLUSIONS

1. Oil supply temperature is one of the variables, as well as load and speed, that can change the laminar-turbulence transition point.
2. Oil supply temperature changes that shift the laminar-turbulent transition point make the effect of that change extremely difficult to predict.
3. The influence of oil supply temperature changes can be predicted if the transition point is not shifted. Increases in the oil supply temperature result in higher babbitt temperatures and lower power losses and vice versa.
4. The percentage changes in babbitt temperature and power loss are less than the percentage change in oil supply temperatures.
5. The higher bearing loads moderate the babbitt temperature excursions that result from oil supply temperature changes.

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