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Tilting pad thrust bearing tests — Influence of three design variables

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The purpose of this paper is to report the results of an extensive series of tests on a special tilting pad thrust bearing. The primary interest was to determine the effects on bearing performance of using circumferential pivot locations outside the normal range of 50 to 60% of the pad arc, from the leading edge. Secondary results were also obtained from tests with pad support disks of various diameters, and with pads of different thicknesses.

1 INTRODUCTION

The test work reported here was prompted by results of design analyses of large tilting pad thrust bearings for vertical shaft hydro-electric turbine-generator applications. Although the primary purpose of this paper is to present results from laboratory tests on a special tilting pad thrust bearing, some background with respect to the analytical work leading to these tests is in order.

The program used for this design work incorporates finite difference forms of the Reynolds and energy equations for the oil film, plus closed-form equations for the pressure and thermal distortions of the pad. These are all iteratively related, including factors to compensate for the carryover of heat from pad to pad. This program was first written in the mid 60's and has been revised numerous times since then to incorporate advances in bearing analysis and to include empirical correction factors from test results.

The specific pad and pad support configuration of interest here is shown in Figure 1. The back face of the thrust pad is simply a flat plane finished by grinding. This surface is then supported by a disk of high strength steel with a spherical pivoting surface on one face to transmit the thrust load to the machine base, and a raised annular ring (also finished flat by grinding) at the outer edge of the opposite face in contact with the flat back face of the pad. This construction has been used for large thrust bearings by the company with which the author is associated and by others, and has been the subject of previous publications, including (1), (2) and (3).

The closed form solutions used in the design program for the pad distortions resulting from pressure and thermal loads are similar to those developed in (1). For this specific construction (Fig. 1) and with pad aspect ratios of approximately 1.0, this method of calculating pad distortions (and ultimately film shape) appears reasonable.

The influence of the carryover of heat from pad to pad has been found to be significant in obtaining analytical solutions representative of actual bearing operation. The method used in this design program to account for heat carryover is based on the work of Ettles in (4) with modifications indicated by test data.

Numerous large tilting pad thrust bearings have been designed with the aid of this program and have been successfully applied in the field.

Through the use of this design program, it is convenient to study the effects on bearing performance of pad and/or support geometry variations. For machines that rotate primarily in one direction, circumferentially offset pivots are often used to improve bearing performance. Historical theoretical analysis ((5), for example) indicated that a pivot placed circumferentially about 60% of the pad arc length from the leading edge provided maximum

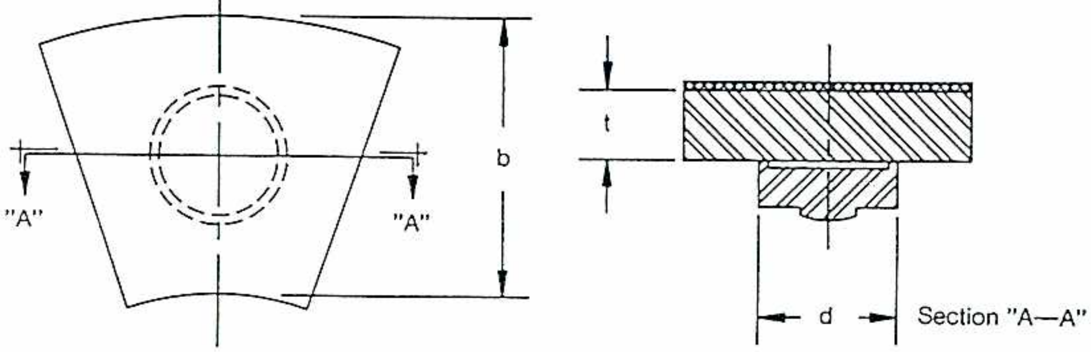

Fig. 1 Thrust pad and support configuration.

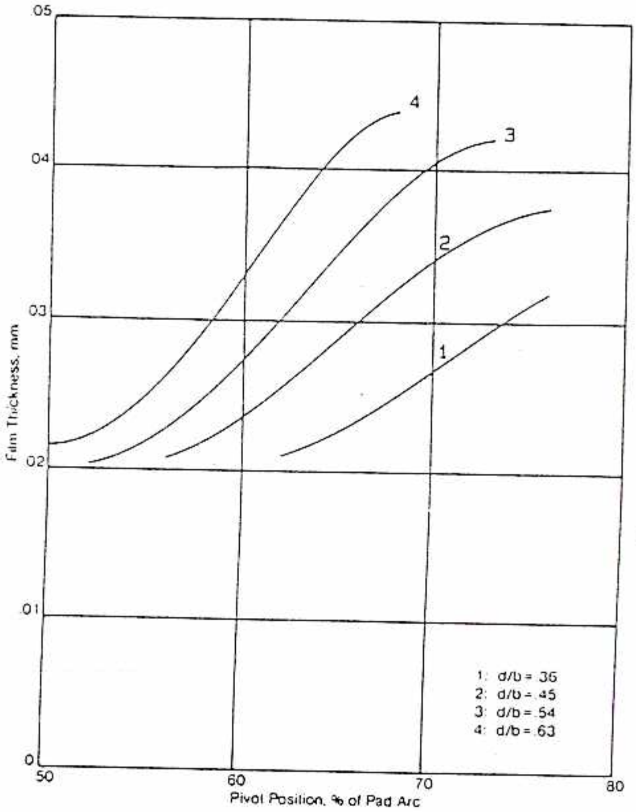

Fig. 2 Calculated minimum oil film thickness vs. circumferential pivot position.

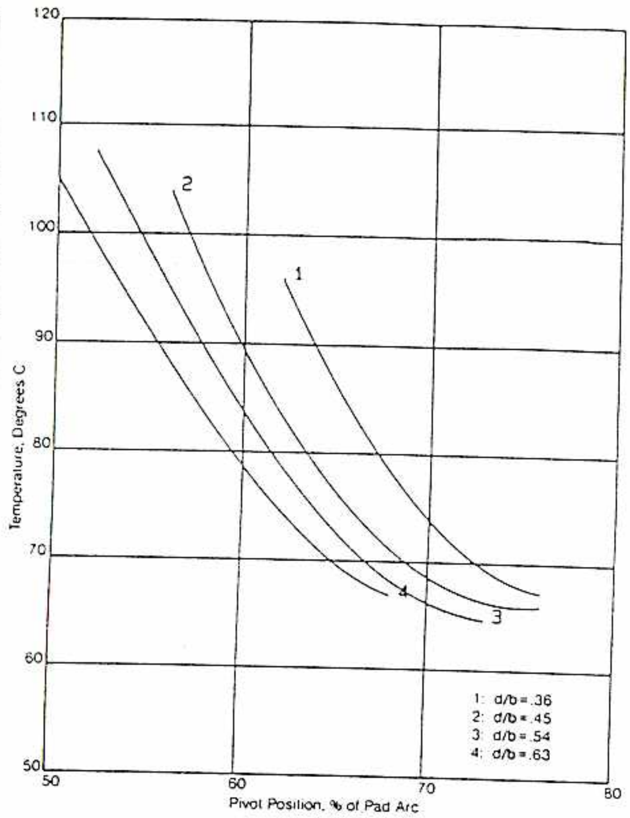

Fig. 3 Calculated maximum pad temperature vs. circumferential pivot position.

load capacity, and that, in fact, a pivot at the center (50% location) resulted in zero load capacity. This situation is true for perfectly flat pads and constant viscosity fluids, but is not true in general since there are numberless machines operating satisfactorily with center pivoted tilting pad thrust bearings.

The ability of such bearings to, in fact, provide significant load capacity is explained when pad deformations and viscosity variations are included in the analysis. This is well accepted today, and these factors are included in design analysis programs, (6) and (7) for example, for tilting pad thrust bearings. Early recognition of the influence of pad deformations on the load capacity of center pivot pads was provided by Raimondi (8).

In the design analysis work which prompted the test work reported here, the effect of circumferential pivot position on bearing performance was studied. This resulted in data as given in Figures 2, 3 and 4. A second item of interest was the effect of support disk diameter, and this is also reflected in these figures.

Pad thickness is the third design variable studied, and Fig. 5 gives representative calculated temperature data from the design work.

In these plots of calculated bearing performance, as with the later plots of measured bearing performance, dimensionless ratios (d/b and t/b) are used for convenience. This is not to imply that the data presented is valid for all designs where such ratios are obtained,

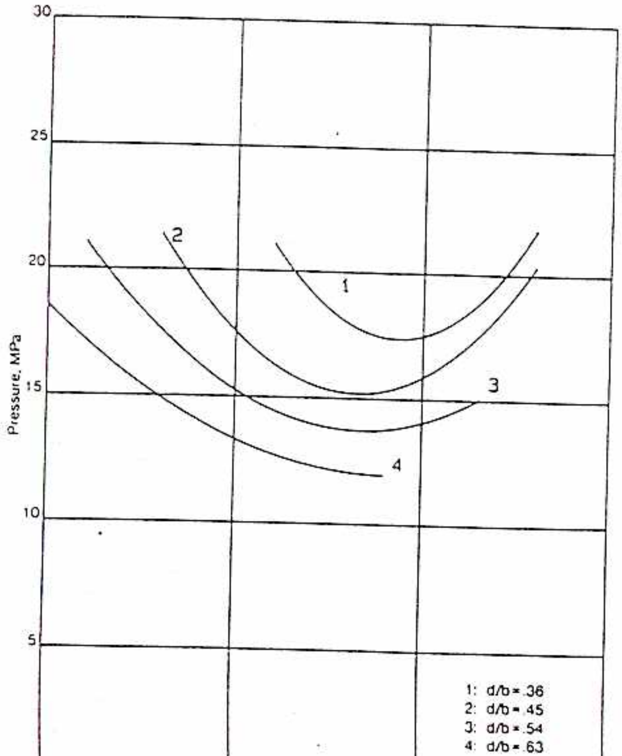

Fig. 4 Calculated maximum oil film pressure vs. circumferential pivot position.

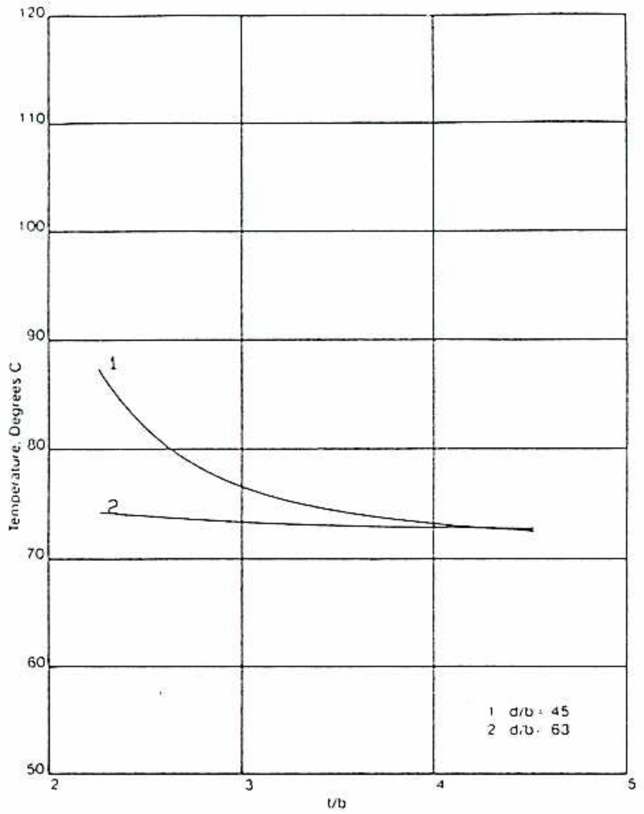

Fig. 5 Calculated maximum pad temperature (62% pivot) vs. pad thickness/pad radial length, t/b.

regardless of bearing size, geometry or operating conditions.

For the large thrust bearing (calculated data presented in Figures 2 through 5), the basic dimension, b, is 561 mm. For the test bearing, this same dimension is 95 mm. The tests were conducted to determine variations in performance due to changes in the three design variables, and there was no attempt or intent to model the film conditions calculated for the large bearing.

Of primary interest was the analytical indication that bearing performance could be improved (reduced film temperatures and pressures, and increased film thicknesses) by moving the pivot beyond the 60% position to somewhere in the area of 70 to 80%.

Because the use of a pivot in this area (75%) deviates from experience (at least that of the author) and because the literature offered limited data, it was concluded that laboratory tests specifically directed at this point were needed to provide additional guidance.

Reference (9) includes test data on elastomer faced thrust pads over a range of circumferential pivot positions from 55 to 85%. These tests showed an increase in film thickness as the pivot position was increased over this range, and also an increase in the peak film pressure. The thrust pads of interest in the work reported here are significantly different in both materials and support geometry from those in (9).

The calculated data in Figures 2, 3 and 4 indicate improved performance for pivots beyond 60% for all three items; film thickness, temperature, and pressure. In the tests reported here, pad temperature was the only item of these three performance characteristics that was measured, and it is thus used as the basis for conclusions regarding performance. For babbitted bearings, a combination of local temperatures and pressures may also be of value in judging bearing capabilities due to the strength-temperature relationship of babbitt.

2 TEST BEARING AND SETUP

A special 381 mm (15 inch) diameter tilting pad thrust bearing was designed and manufactured for these tests. It is shown in Figures 6 and 7.

Fig. 6 Test bearing with one instrumented pad.

Fig. 7 Test bearing components.

It is a two pad bearing in which the pad and support construction described earlier, and shown in Fig. 1, is used. The two pad configuration was chosen to allow the arrangement shown for testing various support disk diameters. The disk retainer plate could be rotated within a circumferential slot in the base ring. This allowed the insertion of the disks to be tested into their corresponding holes in the retainer plate, which was then rotated to place these disks into position for support of the pads.

In addition, a series of thirteen notches associated with each pair of support disk holes was used to position the support disk at the desired circumferential pivot position. A key was used to align and maintain this relationship between the retainer plate and the base ring. The thirteen notch positions correspond to pivot positions from 0.20 to 0.80 in 0.05 steps.

Seven support disk diameters were used, varying in 9.5 mm (0.375 inch) steps from 19.1 mm (0.75 inch) to 76.2 mm (3.00 inch). This provided d/b ratios of 0.2 to 0.8 in 0.1 steps. Pads of three thicknesses were made for test, 25.4 mm (1.00 inch), 19.1 mm (0.75 inch), and 12.7 mm (0.50 inch). In terms of ratios to the pad radial length (or arc length at mid-radius since the pad aspect ratio was 1.0) the values are 0.27, 0.20, and 0.13.

The base ring of the test bearing was pivoted on its back face at 90 degrees to the pad locations. This provided for equal loading of the two pads.

Copper-constantan thermocouples were embedded in the babbitt facing in all six of the pads tested, at the following locations:

85-85

75-75 75-25

65-65

50-85 50-65 50-50 50-25

These thermocouples were located approximately 0.75 mm (0.03 inch) below the pad face in the 1.5 mm (0.06 inch) thick babbitt facing.

The tests on this bearing were made in the thrust bearing test stand described in (10). Briefly, this is a hydraulically loaded, D.C. motor driven facility capable of accommodating thrust bearings in the 250 mm (10 inch) to 500 mm (20 inch) range of outside diameters. The 750 kw (1000 HP) geared drive has a maximum test stand speed of about 10,000 RPM. The test and slave thrust bearings are enclosed in separate housings, and each thus operates in conjunction with a separate thrust collar.

3 TEST CONDITIONS AND PROCEDURE

As noted previously, the area of primary interest was the influence of circumferential pivot position on bearing performance. The test bearing allowed thirteen distinct locations for this variable. Although the locations of most interest were from 50% to 80%, the bearing design readily permitted tests in the 20% to 50% range, also.

The seven support disk diameters, the three pad thicknesses, and the various pivot location possibilities translated into a large number of possible geometry combinations. In order to keep the test program within manageable proportions, several limits were initially set. These were:

a) Tests would be made only at pivot locations at which the support disk was fully within the pad trailing (or leading) edge. This condition limited the largest support disk to a range of pivots from 40% to 60%. Only the three smallest disks could encompass the full range from 20% to 80%.
b) Tests would be run at speeds up to 4000 RPM. Higher speeds were avoided to prevent effects from non-laminar film conditions. This was consistent with the field applications of the particular bearings that prompted this work.
c) Thrust bearing loadings would be limited to 4.14 MPa (600 psi). The goal was to collect data, not to fail bearings.
d) In this same respect, a maximum pad temperature of 121C (250F) was set.
e) ISO VG32 turbine oil was used for all tests at an inlet temperature held between 48.6C (119.5F) and 49.2C (120.5F).
f) A constant flow of 57 liters/min. (15 GPM) was used for all tests reported here.

The test procedure was simply to set the load and speed conditions desired and maintain this for a minimum of ten minutes. A reading was taken at that time if the oil supply temperature was within limits. If not, adjustments were made to bring it within limits, and data was then recorded. The test data collected consisted of the temperatures from the eight thermocouples embedded in each of the two pads plus the oil inlet and drain, plus the oil flow, bearing load, and shaft speed.

4 TEST RESULTS

Even with the limits noted above imposed, a large amount of data resulted from these tests. The use of a data base program to sort and compare the information was invaluable. The data acquisition program averaged the two temperatures at a specific location on each of the two pads, and also recorded the difference. This difference was typically less than three degrees centigrade and often less than one degree. This gave confidence in the load equalization between pads and in the consistency of the thermocouple installations.

The temperature data used in all of the plots presented here is the average value of the two readings from corresponding locations on the two pads. The term "maximum pad temperature" is used, and it is, of course, the maximum <u>recorded</u> pad temperature. The thermocouple locations were chosen to cover the area where the highest pad temperatures have normally been found, both by analysis and test. However, only a limited number of distinct locations on the pad face can be sensed. In the majority of cases, the highest temperature was recorded at 85-85. However, depending on load, support disk diameter, and pivot location, the maximum temperature moved to

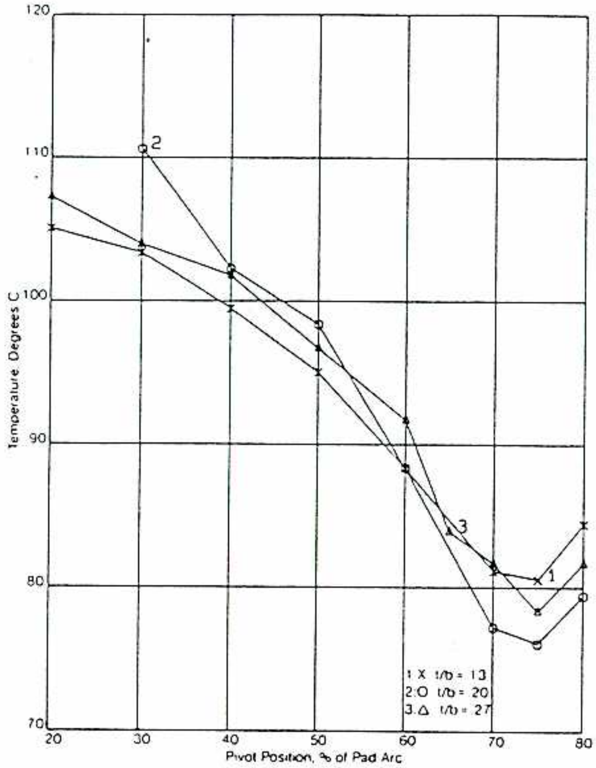

Fig. 8 Measured maximum pad temperature vs. pivot position, 2000 RPM, 4.14 MPa loading, d = 38.1 mm.

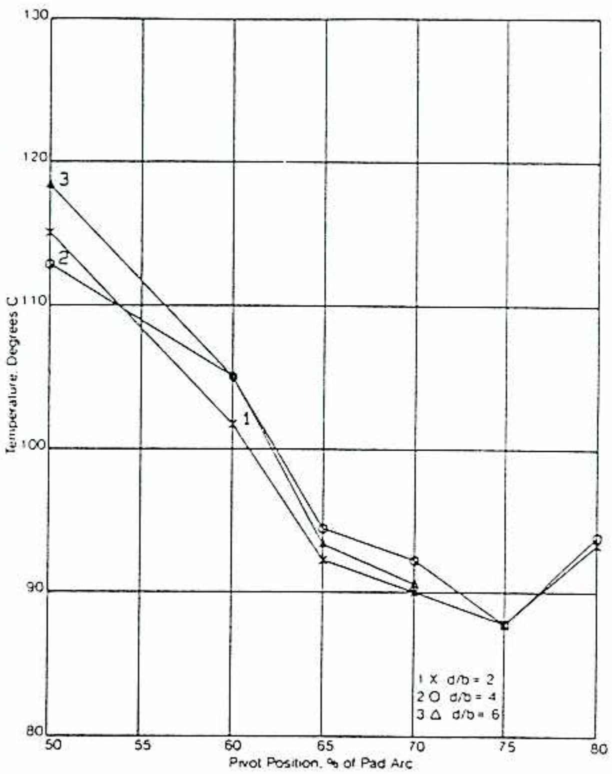

Fig. 9 Measured maximum pad temperature vs. pivot position, 3000 RPM, 4.14 MPa loading, t = 25.4 mm.

other locations. There was even one test run, with the pivot at the 20% location, where the maximum temperature was recorded at 50-25.

Tests with pivots in the 20 to 50% range were run primarily to determine what performance could be expected from an "offset" pivot bearing running backwards. Reference (11) provides additional recent data in this area.

Figures 8 through 16 provide a representative view of the test results obtained.

5 DISCUSSION

A two pad bearing was used for these tests primarily because it allowed the special design arrangement for changing support disk sizes and pivot position. It is recognized that the performance of a thrust bearing with less than a full complement of pads (which would be eight here) is not reduced in proportion to the reduction in total bearing area. However, for the purpose here of investigating relative performance due to other geometrical changes, the two pad design was deemed satisfactory.

A size effect was also recognized (Ref. 3), but the pad thickness values chosen for test were believed to cover a range sufficient to result in measurable changes in bearing performance.

Figure 8 shows the effect of pivot position on maximum pad temperature for the three pad thicknesses at a specific operating condition. This plot covers the full range of pivot positions. It should be noted that for pivots near the leading edge (20%-30%), the hot spot may well be in the leading edge half of the pad. This portion of the pad had a limited number of thermocouples so the possibility of the maximum recorded temperature being representative of the maximum bearing temperature is somewhat less than for other pivot locations (40% and higher). A clear trend with respect to the influence of pivot position is shown. This is not the case for the pad thickness.

Figure 9 shows the effect of pivot position on maximum pad temperature for three support disk diameters (d) at a specific operating condition. Here, also, the effect of pivot position is clear, but the same is not true for support disk diameter. This plot is data from tests on the thickest pad, however, where support disk diameter would be expected to have least influence.

Figure 10, then, is temperature data plotted against pad thickness for three support disk diameters. For the thin pad, the advantage of increasing the support disk diameter is evident.

It became evident during the analysis of the test data that the effect of the pivot position was clear and consistent. The lowest temperatures were found in the tests with the pivot at 75%. Evidence with respect to the effect of support disk diameter and pad thickness was not as clear, but it appeared that some form of three dimensional plot might be helpful.

Contour plots of measured maximum tempera-

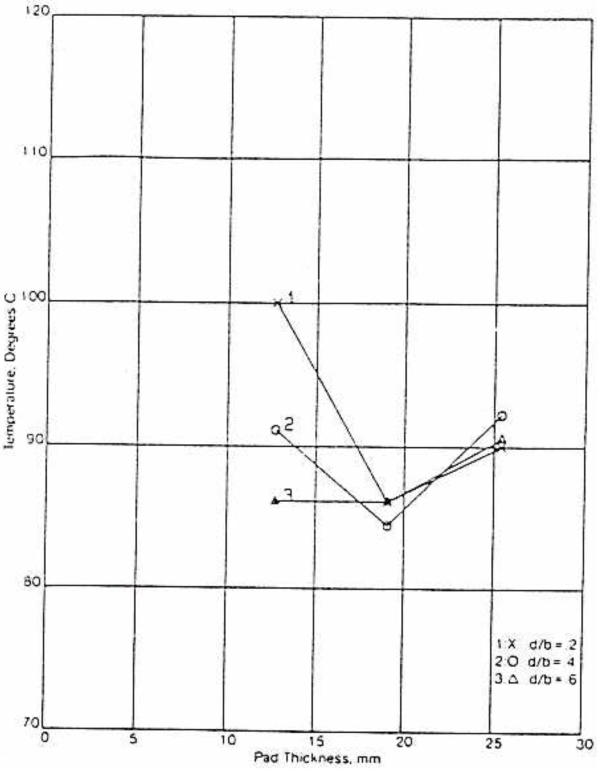

Fig. 10 Measured maximum pad temperature vs. pad thickness, 3000 RPM, 4.14 MPa loading, 70% pivot.

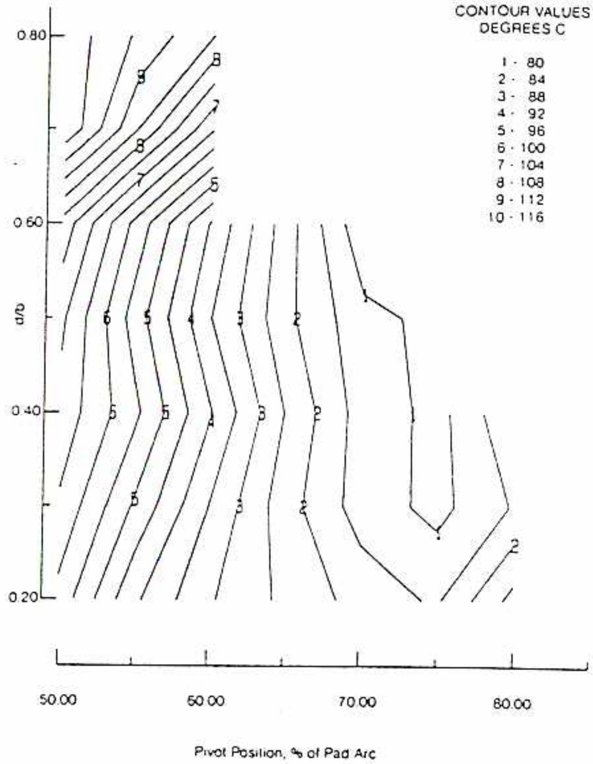

Fig. 12 Measured maximum pad temperature, 3000 RPM, 2.76 MPa loading, t = 12.7 mm.

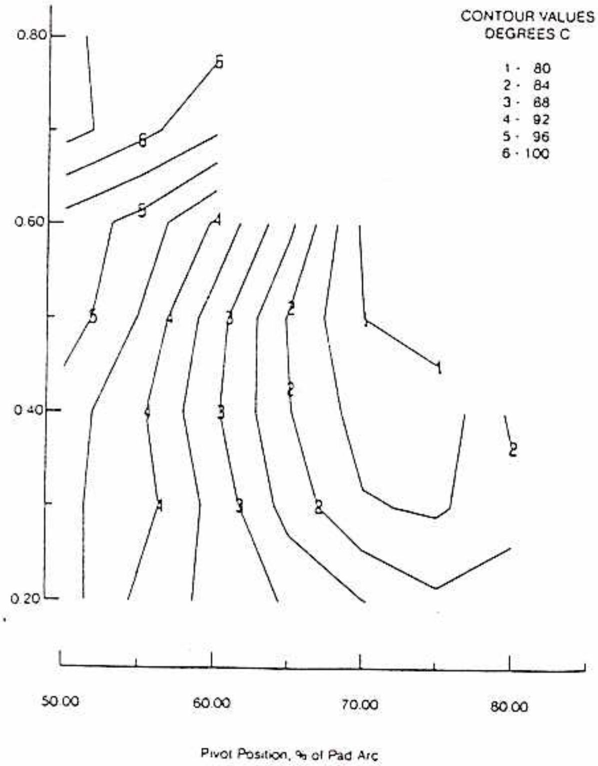

Fig. 11 Measured maximum pad temperature, 2000 RPM, 4.14 MPa loading, t = 12.7 mm.

ture vs. d/b ratios and pivot position are shown in Figures 11 through 16. In all of these, the influence of the pivot position is to be seen, and the influence of support disk diameter and its relationship to pad thickness is more evident.

Figures 11 and 12 are for the thin pad, 12.7 mm thick; Figures 13 and 14 are for the medium thickness pad, 19.1 mm; and Figures 15 and 16 are for the thick pads, 25.4 mm. Note that the lack of data in the upper, right hand portions of these plots is due to the fact that tests were limited, as noted earlier, to combinations of support disk diameters and pivot locations such that the support disk did not extend beyond the trailing edge of the pad.

In Figures 11 and 12 (the thin pad), a small diameter support disk provides reduced pad temperatures for center pivot pads, but at the 75% pivot location, a large diameter support disk is better.

In Figures 15 and 16 (the thick pad), the 28.6 mm diameter support disk (d/b = 0.3) gave the lowest temperatures for center pivot pads, while an even smaller disk is indicated for the 75% pivot position.

Figures 13 and 14 are for the pads of medium thickness, and the indications at the 75% pivot are somewhat between those for the other pads (as might be expected). Best results appear to be with d/b ratios of about 0.4 to 0.5, although the use of the smallest support disk gave temperatures almost as low.

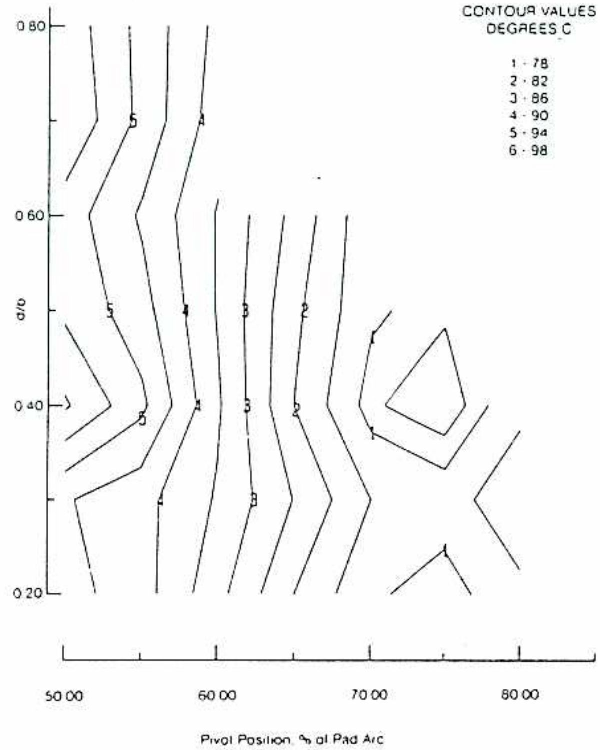

Fig. 13 Measured maximum pad temperature, 2000 RPM, 4.14 MPa loading, t = 19.1 mm.

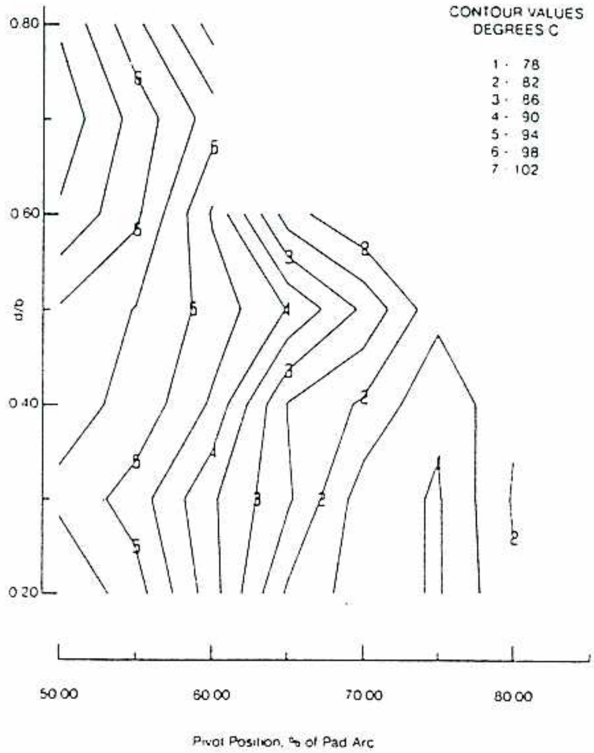

Fig. 15 Measured maximum pad temperature, 2000 RPM, 4.14 MPa loading, t = 25.4 mm.

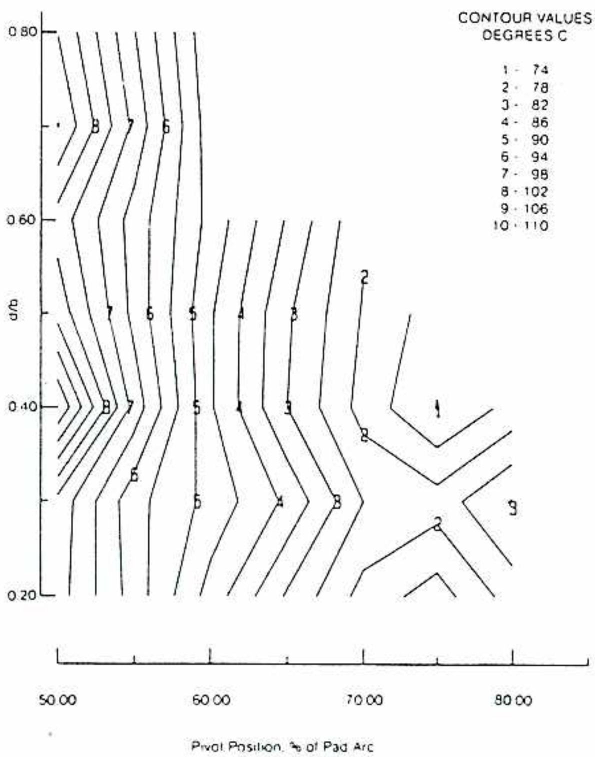

Fig. 14 Measured maximum pad temperature, 3000 RPM, 2.76 MPa loading, t = 19.1 mm.

It can be argued that the variations in temperature are small regardless of the disk diameter (d/b ratio) for any given pivot position. This is true, and apparently the result of too small a variation in the pad thicknesses tested. Trends are indicated, however, particularly for the thin pads (Figures 11 and 12), that support theoretical results which require pad crowning for load capacity with center pivots (thus a small disk is better), but less crowning for offset pivots (thus a larger disk gives better results).

Pad crowning is also believed to be the basis for the improved performance found with the pivot at the 75% location rather than 60%. Crowning, due to pressure and thermal loads, commonly results in a diverging film at the trailing edge. The extent of this divergence directly affects bearing performance. As the pivot is moved further downstream, this divergence decreases and is transferred to the leading edge as an increase in the convergence angle. The decrease in divergence on the trailing edge improves the bearing performance, while the increase on the leading edge somewhat reduces performance. The net result is, however, an increase in performance until the pivot is so far downstream that very high local pressures (and reduced film thicknesses) have to be developed for pad balance.

6 CONCLUSIONS

1- For the bearings and conditions tested, the lowest maximum recorded pad temperatures were consistently found with the pivot at 75%.

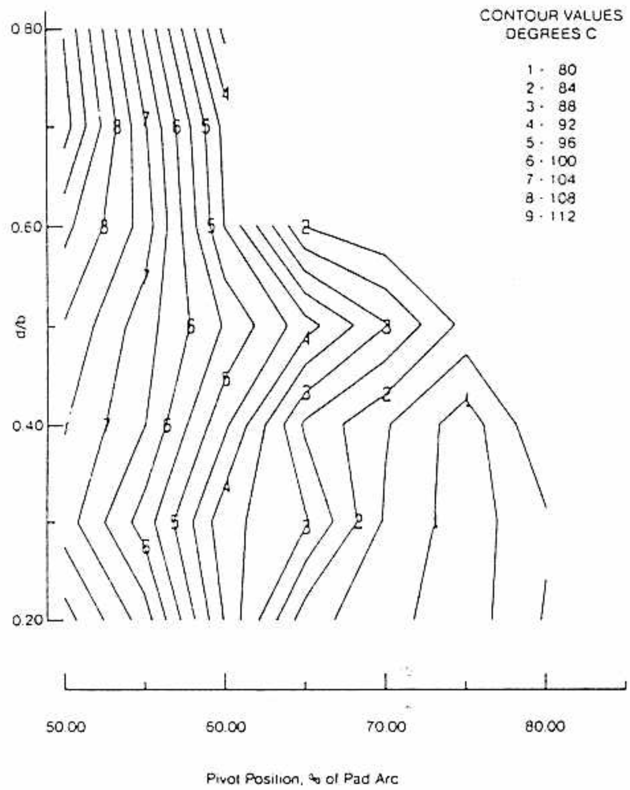

Fig. 16 Measured maximum pad temperature, 3000 RPM, 2.76 MPa loading, t = 25.4 mm.

2- The test bearings operated with pivot positions less than 50%, to as low as 20%. Decreased performance was found, as compared to pivots at 50% or greater, but definite load capacity was present.

3- The influence of support disk diameter on bearing performance was most evident with the thinnest of the three pad thicknesses tested, as anticipated. The contour plots show the need for a small support disk diameter for center pivot pads and the advantage of a larger disk for the 75% pivot location. These plots (thin pad) indicate that an optimum condition exists with the 75% pivot location and the largest support disk possible for that location (d/b = 0.5).

4- As the pad thickness increases, the 75% pivot location remains optimum (at least as determined by pad temperature), but the optimum support disk diameter decreases.

5- Further studies need to be pursued in two areas:
   a) Analytical efforts to correlate these findings with the "size" effect of larger bearings.
   b) Additional test work to determine if the 75% pivot location produces similar results with other pad support configurations.

7 ACKNOWLEDGMENTS

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