Measurements of Maximum Temperature in Tilting-Pad Thrust Bearings©

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One of the important restrictions governing the design of tilting-pad thrust bearings is the maximum allowable white metal temperature. However, adequate guidance on typical pad operating temperatures has not been available in established lubrication handbooks. New pad temperature data from 12 test bearings have been combined with previously published work to provide a source of information covering a diverse range of proven bearing types and sizes.

The collected information has been presented as the difference between the maximum pad and oil supply temperatures at fixed values of bearing pressure against mean sliding speed. Clear trends have been found in the data which have enabled curves of typical bearing performance to be drawn through the test results. The differences between the type of lubricant feed ("flooded" or "low loss"/"directed") and the pivot position (center or offset) have been highlighted.

INTRODUCTION

Tilting-pad thrust bearings are a very efficient means of transferring large thrust loads from rotating shafts to surrounding structures. Since the generation of the lubricating film is entirely automatic, and its presence eliminates wear, tilting-pad units have proved to be the most appropriate type of bearing in many areas of industry.

As with all technologies, there are a number of fundamental limitations that define a tilting-pad's range of suitable applications. One such limit is the maximum safe working temperature of the traditional white metal face; hence, the assessment of pad temperature is an essential part of bearing performance prediction. Consequently, a statement of performance must be considered incomplete if it does not contain an estimate of pad temperature.

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The successful introduction of tilting-pad technology was largely due to the good agreement between the operation of modest size pads and the predictions of simple hydrodynamic theory. With the general trend for larger bearings and more arduous duties, the prediction of pad temperature has gained importance. Although the laws of thermodynamics are well understood, there is no complementary simple thermodynamic model of bearing performance for the prediction of operating temperatures. Historically, experimental results have been the only reliable source of pad temperature information. Early formulae for predicting pad temperatures were based on bearing observations, and even today's computer models rely heavily on test data for setting up a number of parameters used in the performance calculations; hence, the consolidation of experimental results into coherent pad temperature information is of prime importance to lubrication engineers.

For many years, various technical journals have published temperature data obtained from bearing tests, but very few articles have drawn together the results from different bearings to see if there is a consistent overall pattern to the reported results. This paper uses previously unpublished test results from the authors' company in combination with other published work, to form a general source of information on typical bearing temperatures.

APPLICATION OF PAD TEMPERATURE DATA

In general, the performance of a thrust bearing is bounded by a limiting duty curve incorporating three basic concepts. Figure 1, as suggested by Martin (*1*), shows a typical limiting curve sketched on a bearing duty diagram. The left-hand segment of the curve represents a minimum acceptable film thickness condition. Since the film thickness is a function of the lubricant viscosity, the location of this part of the boundary curve depends upon oil grade and bearing operating temperature. The horizontal segment of the boundary curve is derived from limiting stress levels around the pad pivot. This stress condition, which is not affected by temperature considerations, fixes the maximum bearing load. The final

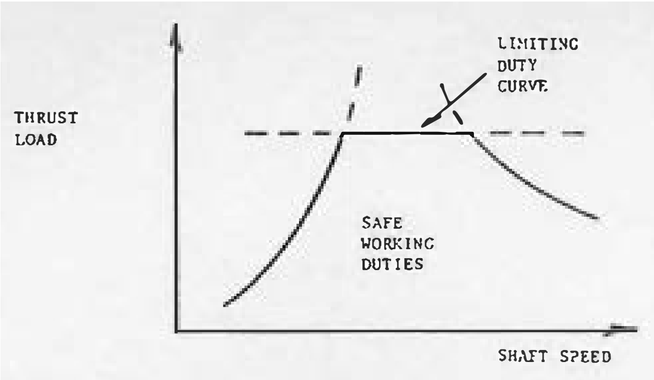


Fig. 1—Limits to thrust bearing operation

segment of the boundary curve, to the right-hand side of the bearing duty diagram, arises from the high pad temperatures generated at high sliding speeds. White metal has a sufficiently low melting point for its yield strength to cause concern in applications where high speeds and loads are expected.

Thus, pad temperature information is required by designers to keep bearing duties within the temperature limit imposed by the white metal surface. In addition, pad temperature data have a much wider application, since it can be used as a guide to oil film temperatures and, hence, power loss over the whole range of operating duties.

PUBLISHED PAD TEMPERATURE INFORMATION

A review of published test results has revealed a number of papers (*2*)–(*11*) containing suitable data on steel thrust pads with white metal faces. These particular references were chosen because they give temperatures for pads typical of designs in general use. The pad shapes tested are either sector-shaped or closely approximate to a sector shape, with pad length-to-width ratios in the range 1.0 to 1.45. Readings of maximum pad and oil supply temperatures are common to all the references (*2*)–(*11*), and sufficient data are given to describe each test condition by mean sliding speed and bearing pressure. Table 1 summarizes the important pad details and range of operating conditions covered by the tests.

The measurement of maximum pad temperature needs some discussion since the results can be affected by experimental technique. New (*5*) shows that there is a considerable variation in temperature across the full extent of a pad face, about 40°C at a pressure of 2.0 MPa and sliding speed of 62 $m \cdot s^{-1}$. Therefore, it is important to place thermocouples in the hottest part of the pad. Gregory (*4*) compares readings taken from two relatively well-separated locations near the pad trailing edge. The position of these points on the mean pad diameter and adjacent to the outside diameter are shown approximately in Fig. 2(a). The results show temperature differences of less than 8°C between the two points for mean pad pressures between 0 and 2.8 MPa, at a sliding speed of 110 $m \cdot s^{-1}$.

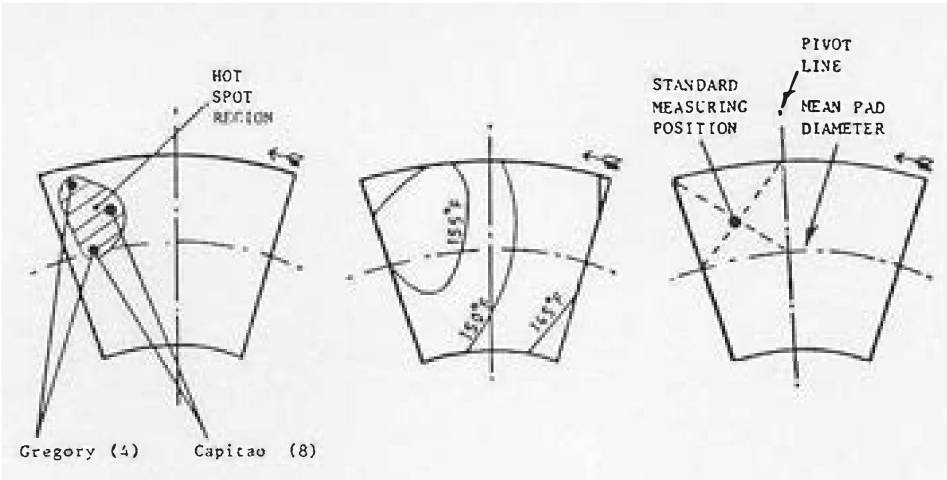
In a later paper, Capitao, Gregory and Whitford (*8*) also compare readings from two measuring points, one near the trailing edge and the other half way to the pad pivot. For these two points, a difference of less than 5°C was found over the pressure range 0 to 2.8 MPa at sliding speeds of 48 and 120 $m \cdot s^{-1}$. As shown in Fig. 2(a), these two references compare information from three locations, and the hottest part of a pad seems to be in a region defined by these three points. This result agrees with the findings of Kettleborough, Dudley and Baildon (*12*). Their comprehensive measurements of pad temperatures, see Fig. 2(b), show shallow temperature gradients around the hottest part of a pad operating at a pressure of 3 MPa and sliding speed of 12 $m \cdot s^{-1}$. Consequently, placing a thermocouple in this hot-spot region can be expected to give a good indication of maximum pad temperature.

In addition to the temperature differences over the white metal face, the temperature gradient through the thickness of the pad must be considered. The white metal surface, being in contact with the source of heat, is hotter than the back of the pad. Typical industrial pads are rather thin for accurate measurement of the transverse thermal gradient,

Table 1—Previously Published Test Results

Ref. Number —	First Author —	Pad Width mm	Pivot Posn. —	Oil Feed —	ISO VG —	Sliding Speed $m \cdot s^{-1}$	Mean Pressure MPa
(*2*)	Elwell	184	C	F	100	2.5–10	0.3–3.4
(*3*)	Bielec	28	C	F+L	68	10–76	1.4–5.5
(*4*)	Gregory	67	O+C	F	32	110	0.0–2.8
(*5*)	New	29	C	F+L	68	62	1.0–4.0
(*6*)	Gardner	38	C	F	32	50–63	0.7–3.5
(*7*)	Capitao	95	C	F	32	25–120	0.0–3.5
		108	C	F	32	20–140	0.0–2.8
(*8*)	Capitao	67	C	F	32	40–146	0.0–2.8
		76	C	F	32	48–160	0.0–2.8
(*9*)	Leopard	40	C	F	68	10–100	1.0–7.0
(*10*)	Mikula	67	C	F+L	32	45–144	3.45
(*11*)	Neal	33	C	F	32	19	0.7–2.8

Pivot position: C = Center pivot O = Offset pivot
Oil Feed: F = Flooded housing L = Low loss (Directed)



a) MEASUREMENT POINTS USED IN PUBLISHED WORK

b) TEMPERATURE CONTOURS BY KETTLEBOROUGH ET AL (12)

c) STANDARD POSITION USED FOR NEW DATA

Fig. 2—Comparison of thermocouple locations and hot-spot regions

but Elwell, Gustafson, and Reid (*2*) report temperature readings at three locations through the thickness of a reasonably sized pad in a marine application. At a load of 3.4 MPa and a sliding speed of 10 $m \cdot s^{-1}$, the temperature difference across the hot-spot region of the pad is about 30°C. This is approaching 40 percent of the temperature difference between the hot-spot and oil supply temperatures. In percentage terms, this is comparable to the differences around the pad face reported by New (*5*). Hence, locating thermocouples close to the pad surface is as important as placing them in the hot-spot region. Most of the references report placing thermocouples up to 1.6 mm from the white metal surface. The most notable exception is Neal (*11*), who states that his thermocouples were positioned in the steel backing 2.5 mm (about 20 percent of the pad thickness) from the pad surface.

Table 1 shows that most information is available for center-pivoted pads operating in a flooded housing. Much less data are available for center-pivoted pads operating with a low loss (directed) type of oil supply. With the exception of Gardner (*6*), who does present some test results, offset-pivoted pads are only mentioned in passing, so this important class of bearing is very poorly represented. References (*2*) to (*11*) contain sufficient information to establish the trends for center pivot pads in flooded housings. Consequently, of the new data presented in this paper, only one set of test results is concerned with this type of bearing; much more information is presented for offset pivot and low loss types. The new data originate from test work carried out in the authors' company over a period of years. The basic details of the tests performed are shown in Table 2 with apparatus and instrumentation being described in the next two sections.

APPARATUS

Two completely separate test rigs were involved in the work. The horizontal shaft arrangement shown schematically in Figs. 3 and 4 was employed for tests "i" to "x" of Table 2. Tests "xi" and "xii" were carried out with the vertical shaft arrangement shown in Fig. 5.

In every case, loading of the test thrust pads was achieved by a series of hydraulic pistons in the retaining ring of an opposing tilting-pad thrust bearing. For tests "i" and "ii," and "iv" to "ix", the opposing bearing was located in the same housing as the test bearing on the opposite side of the thrust collar. This configuration is shown by the solid lines in Figs. 3 and 4. In contrast, tests "iii," "x," "xi," and "xii" utilized separate housings for the loading system and the test bearing, respectively. This is shown for tests "iii" and "x" by the broken lines in Figs. 3 and 4.

The horizontal rig was capable of testing either flooded or low loss lubricated bearings. Thus, tests "i" to "iii" used flooded lubrication with oil entry and exit points as shown in Fig. 3. Tests "iv" to "x" had the low loss arrangement shown in Fig. 4. In these latter cases, the oil was supplied via galleries machined in the retaining ring, a typical example of which is given by the photograph, Fig. 6. Jets of oil emerge from the holes between the pads seen in Fig. 6 and impinge directly on the collar. Oil leaves the housing by the very large drain in its lower half. This arrangement ensures that the collar is free to rotate without incurring punitive energy losses due to churning of oil retained in the housing.

In all the horizontal shaft tests, the temperature for oil supply to the housing was in the range 43 to 50°C. This temperature was measured by a thermocouple in the pipework immediately prior to entry in the housing as shown in Figs. 3 and 4.

The vertical rig tests were carried out with test bearings in which the oil was completely contained within the casing and free from any external lubricant supply. The natural pumping action of the thrust face is used to circulate the lubricant in the casing and past a water cooler which absorbs the heat generated by the bearing. Figure 5 shows the way in which the oil flow path is constrained by the housing, and the point at which oil temperature was measured just before entry to the thrust chamber. In preparing the results

Test Number —	Pad Width mm	Pivot Posn. —	Oil Feed —	ISO VG —	Sliding Speed $m \cdot s^{-1}$	Mean Pressure MPa	Pad O/D mm	No. of Pads per Bearing	Test Rig	Test Posn.
i	68	O	F	46	14–77	0.5–5.5	240	6	H	I
ii	12	O	F	46	6–64	0.5–3.5	133	24	H	I
iii	48	O	F	32	21–26	1.0–3.5	175	8	H	E
iv	40	C	L	46	15–73	0.5–5.5	176	8	H	I
v	40	C	L	46	15–74	0.5–5.0	176	8	H	I
vi	43.5	C	L	46	16–81	0.5–4.0	191	8	H	I
vii	22.5	C	L	46	44–89	0.5–5.0	191	16	H	I
viii	40	O	L	46	15–73	0.5–5.5	176	8	H	I
ix	30.5	O	L	32	14–62	1.5–3.5	175	12	H	I
x	29	O	L	46	37	4.2	162	12	H	E
xi	40	C	F	46	7–22	1.5–7.0	176	8	V	E
xii	40	O	F	46	7–22	1.5–7.0	176	8	V	E

Table 2—Bearings Tested for New Temperature Data

Pivot position : C = Center pivot; O = Offset pivot
Oil feed : F = Flooded housing; L = Low loss (Directed)
Test rig : H = Horizontal; V = Vertical
Test position : I = Test and loading pads in one casing; E = Loading pads external to test pad casing

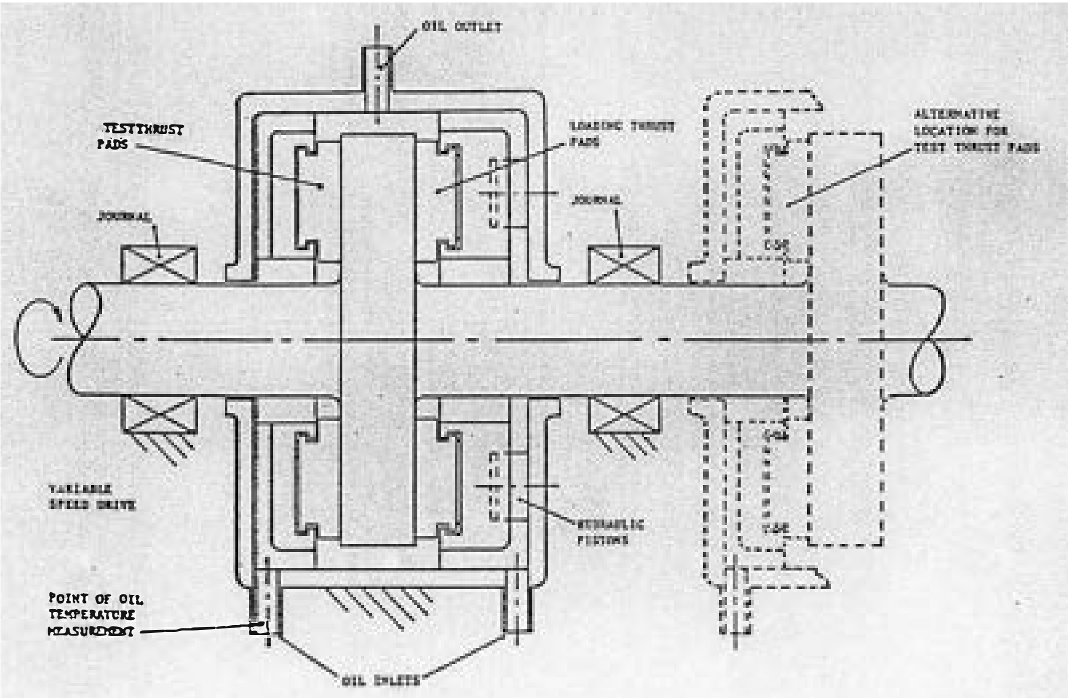


Fig. 3—Schematic arrangement for tests i to iii (flooded lubrication)

which follow, it is this value which has been taken as supply temperature for tests "xi" and "xii." During these two tests, the supply temperature varied between 40 and 60°C, depending on the speed and load applying in each case.

INSTRUMENTATION

Pad maximum temperatures were recorded using thermocouples embedded in the steel backing material at a position which has remained standard throughout. This position is at the center of the trailing outboard quadrant of a pad which is defined, as shown in Fig. 2(c), by the radial line passing through the pivot and the mean pressure diameter. This position is in agreement with experience gained from very early tilting-pad experiments in the authors' company, and with the optimum location suggested by other investigators such as Capitao, Gregory and Whitford (*8*) and Elwell (*13*). Comparison with the test results shown by Figs. 2(a) and 2(b) further confirms the suitability of the chosen location.

The thermocouples were placed in test thrust pads between 1 mm and 2 mm beneath the lubricated surface, depending on the overall size of the pad and thickness of its white metal layer. These depths are in general agreement with those used by other investigators and described earlier.

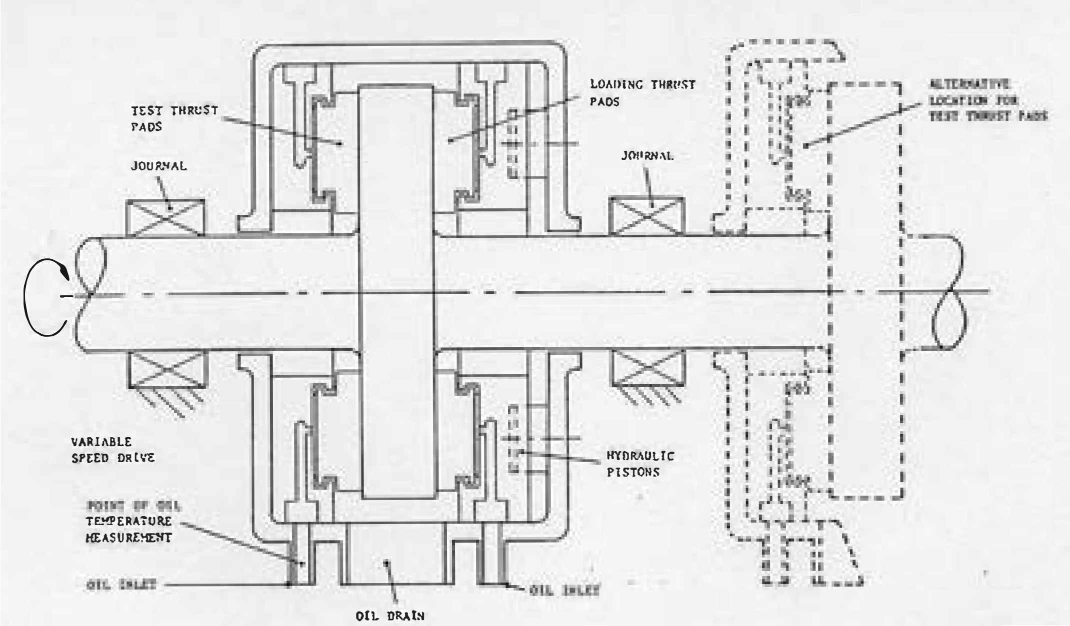


Fig. 4—Schematic arrangement for tests iv to x (low loss lubrication)

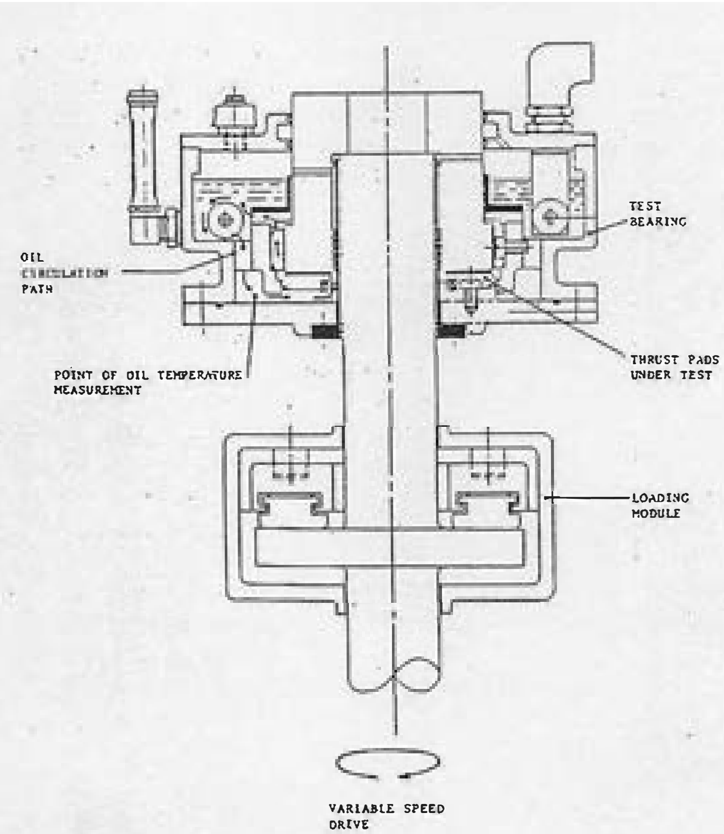


Fig. 5—Schematic arrangement for vertical bearing tests

METHOD OF COMPARING DATA

The sets of test results to be compared are listed in Tables 1 and 2. Since this information has been gained from a variety of bearing designs, tested in different facilities to different procedures, an appreciable amount of scatter can be expected. Furthermore, the test units originate from different design offices and there are substantial differences between the various pad and housing geometries, but all the bearings represent types that have been in production. Thus, the conclusions to be drawn must be considered as guides to the performance of bearings in general rather than the performance of a specific device.

In addition to the bearing geometry, there are a number of operational parameters that have a substantial effect on the running temperature; hence, the means of comparing results must isolate the most significant parameters. The presentation of absolute temperature values is unsatisfactory since the maximum pad value can be expected to change with the general level of bearing temperature. The maximum pad temperature is closely related to the temperature of the oil as it enters the lubricating film at the leading edge of the pad; however, this oil temperature is not often measured during the course of bearing tests and is rarely measured in working bearings. For the results of the study to be of practical use, the analysis must be restricted to temperature measurements that are readily available.

Since all the reports mention the supply temperature to the bearing housings, this oil supply temperature has been used to overcome the problem of overall bearing temperature level. Thus, all the pad temperature data have been presented in this report as the difference between the maximum pad and oil supply temperatures. The oil feed temperatures given in Refs. (*2*) to (*11*) are all between 46 and 50°C, and the range 43 to 50°C covers the bulk of tests

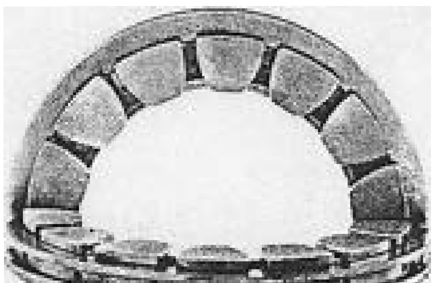


Fig. 6—Example low loss bearing

listed in Table 2; hence, most of the results have been obtained at similar oil feed temperatures, and this variable will have little influence on the comparisons.

The two operating parameters that can be expected to have most influence on the results are mean sliding speed and pad pressure. To emphasize the effects of these two parameters, the collected information is presented at fixed mean pressure values against an axis of sliding speed. Contours of 1.4, 2.8, and 4.1 MPa (200, 400, and 600 psi) have been chosen since they cover common operating conditions, while avoiding a confusing scatter of points. In most cases, interpolation between test points has been necessary to extract information at the required operating conditions. Unfortunately, some valuable information is lost in this process where there is an insufficient range of test results to perform the interpolation. One example of this loss is the work of Mikula and Gregory (*10*) whose results are all at 3.4 MPa (500 psi).

There are two major design features that are known to have a strong influence on pad performance. The information in each of Refs. (*3*), (*5*), and (*10*) has been presented so as to highlight the advantages of low loss oil feed over the traditional flooded housing. These three references agree that the difference in maximum pad temperatures can be expected to rise to about 15°C at a sliding speed of 60 $m \cdot s^{-1}$. Only Gardner (*6*) details the substantial differences between offset-and center-pivoted pads, showing a 20 to 25°C difference at a sliding speed of 50 $m \cdot s^{-1}$. Since these two design features do have a strong influence on pad temperature, the test results have been divided into four groups according to pivot position and feed type.

PRESENTATION OF PAD TEMPERATURE INFORMATION

Figures 7 to 10 show the collected maximum pad temperature data presented according to pivot position and oil feed arrangement. Points representing data derived from published work are annotated with their reference list number. The remaining points are taken from the tests listed in Table 2. As expected, there is a considerable amount of scatter on most of the four figures. A certain amount of scatter is always expected since the results obtained from individual pads are critically dependent upon the maintenance of fine tolerances. All the new data presented show the average of hot-spot temperatures in the ring of pads making up a thrust face. In the authors' experience, in common with Leopard (*9*), differences of 20°C between the hottest and coldest running pads are typical for most applications. Since it is common practice to instrument only a limited number of pads in a complete ring, the differences between pads can explain part of the 30°C spread of results shown on Figs. 7 and 8. Despite the scatter of points on Figs. 7 to 10, there are clearly discernible trends, and curves characteristic of the bearing types have been drawn through each set of data. Since the spread of results is generally greater than expected, the test results have been examined for systematic causes of these variations. Within the limits imposed by the available data, the effects of bearing geometry, oil supply condition, and experimental practice are considered in the next three paragraphs.

Many aspects of bearing geometry are not quantifiable, but are characteristic of different styles of design. Although the differences between two styles of bearings may be quite marked, there is generally insufficient data available to assess the benefits in pad temperature. One easily quantified parameter that is normally considered important is the pad size. Figure 11 has been drawn using the data for center-pivoted pads operating in a flooded casing. The figure shows that for pads up to 150 mm long there is no obvious trend of rising temperature with pad size, in the sliding speed range of 40 to 50 $m \cdot s^{-1}$. Repeating this exercise at other speed ranges up to 80 $m \cdot s^{-1}$ also gives the same result. This is in agreement with the theoretical work of Ettles (*14*) who shows that at typical operating duties, the trend of increasing temperature with pad size should only start for pads larger than about 150 mm.

The viscosity grade and flow rate of oil supplied to test bearings can be expected to have an influence on pad temperatures. Using Table 1 with Fig. 7, it can be seen that there is no consistent linking between thick oils (high-viscosity oil grades) and high temperatures; however, this result has been demonstrated for only the limited range of oil viscosity grades present in this study. The effect of oil quantity has been shown in Refs. (*3*), (*4*), (*5*), (*7*), (*8*), and (*9*). The result in each reference has been that provided sufficient oil is passed through the bearing to adequately cool the pads, the pad temperature is relatively insensitive to changes in the flow rate. This condition is satisfied if the rise of oil temperature as it passes through the housing is kept to less than about 17°C, depending on the type of bearing. Since it is normal manufacturing practice to specify an oil flow sufficiently large to ensure adequate pad cooling, it is reasonable to assume that the data given in the references relates to adequately cooled pads and is insensitive to flow rate.

One of the most important aspects of experimental technique is the placing of the thermocouples that measure maximum pad temperature. It has already been shown that thermocouples have to be placed in the hot-spot area. Neal (*11*) has used thermocouples placed further away from the pad surface than other workers, but the temperatures he presents are not significantly less than the other published results.

The conclusion to be drawn at this stage is that there is insufficient data to state how much the spread of results can be attributed to systematic causes. The indications are that pad size and oil viscosity grade, within the ranges for which data are available, are not major factors governing pad temperatures in typical bearing installations when compared with the pivot position and the type of lubricant feed.

DISCUSSION OF RESULTS

The sets of mean performance curves drawn through the data presented on Figs. 7 to 10 are essentially similar, demonstrating a coherent pattern of behavior. Their chief feature is that the constant pressure contours diverge and become flatter with increasing speed. This type of bearing

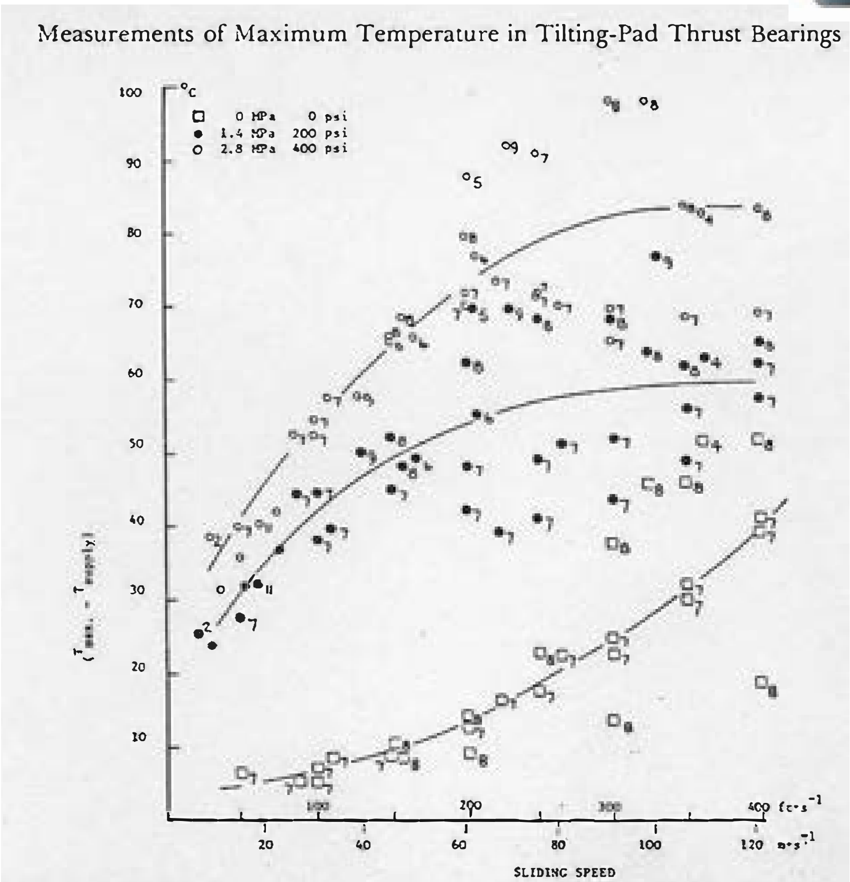


Fig. 7—Temperature difference parameter vs sliding speed for center pivot pads, flooded lubrication. (The suffixes attached to data points on this and subsequent graphs refer to the list of references given at the end of the paper. Data points without suffixes are new information stemming from the present study.)

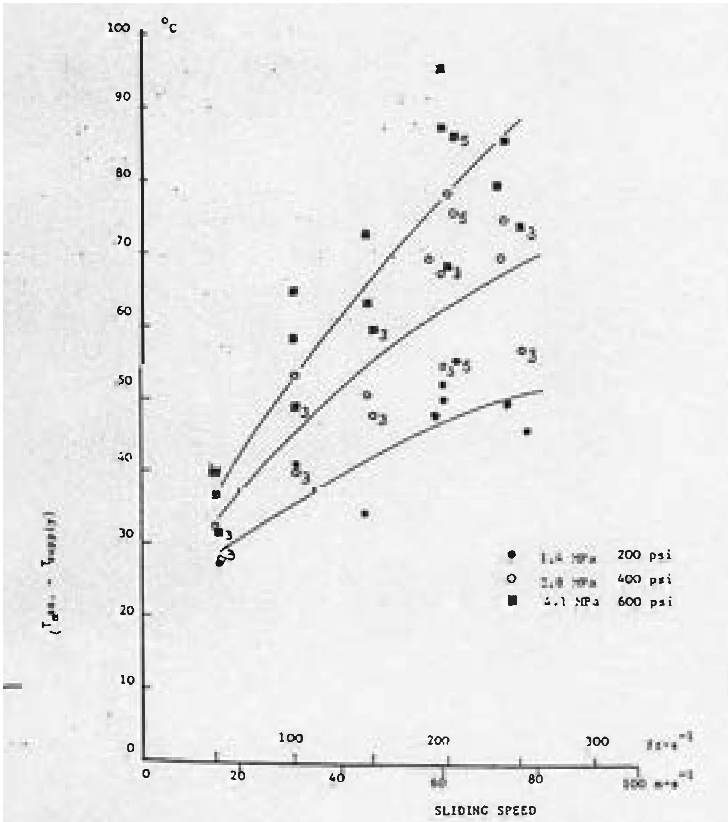


Fig. 8—Temperature difference parameter vs sliding speed for center pivot pads, low loss lubrication.

behavior can be compared with the characteristics predicted by theoretical work. A very simple study f heat transport in a film by Cameron (*15*), suggests that conduction should predominate at low sliding speeds and that convection becomes more important as the sliding speed increases. Further work by Neal (*11*) leads to the conclusion that at low sliding speeds, when conduction is important, velocity is the most significant parameter influencing pad temperatures. At high sliding speeds, however, when convection dominates, pad temperature will be governed by pressure rather than velocity. Since these trends are exhibited by the test results, there is hope that a simple thermodynamic model based upon the theory of heat transport can be developed to predict pad temperatures.

So that the results from the four bearing types can be compared, the mean performance curves on Figs. 7 to 10 have been copied onto the graphs that form Fig. 12. The differences between center- and offset-pivoted pads are shown on Figs. 12(a) and 12(b). Figure 12(a) makes the comparison for flooded casings and shows the expected uniform 20°C temperature difference between the two types. Figure 12(b) is for the low loss type of lubrication and gives a difference which rises from 20 to 30°C with increasing speed. The complementary figures 12(c) and 12(d) compare the flood and low loss types of feed arrangement. Figure 12(c) shows the difference between the feed types to be about 10°C for center-pivoted pads, which is up to 5°C less than expected from work presented in Refs. (*3*), (*5*), and (*10*). Since this disagreement is a significant proportion of the difference between the two feed types, further work in this area would be useful. On the final graph [12(d)], the difference between feed types for offset-pivoted pads is seen t ease with speed up to 15°C, in a manner very similar

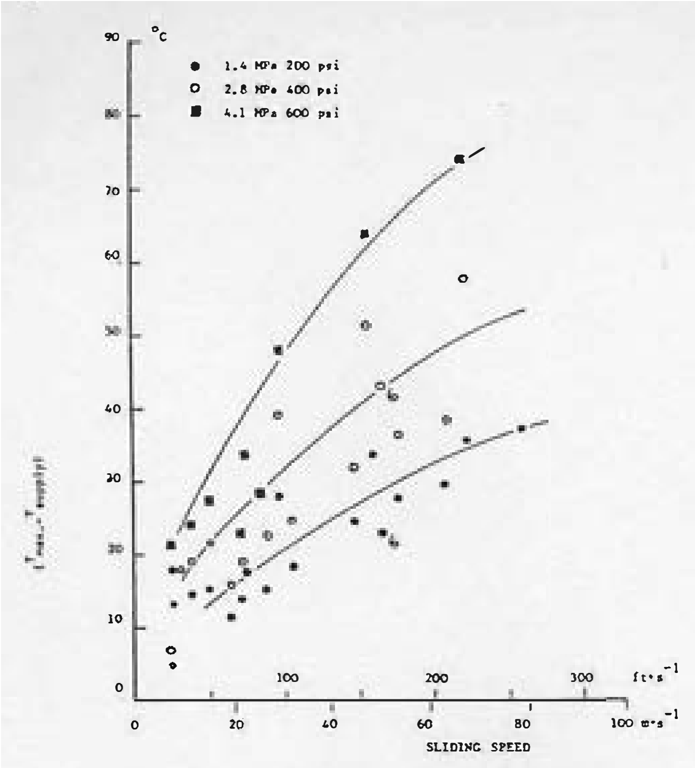


Fig. 9—Temperature difference parameter vs sliding speed for offset pivot pads, flooded lubrication.

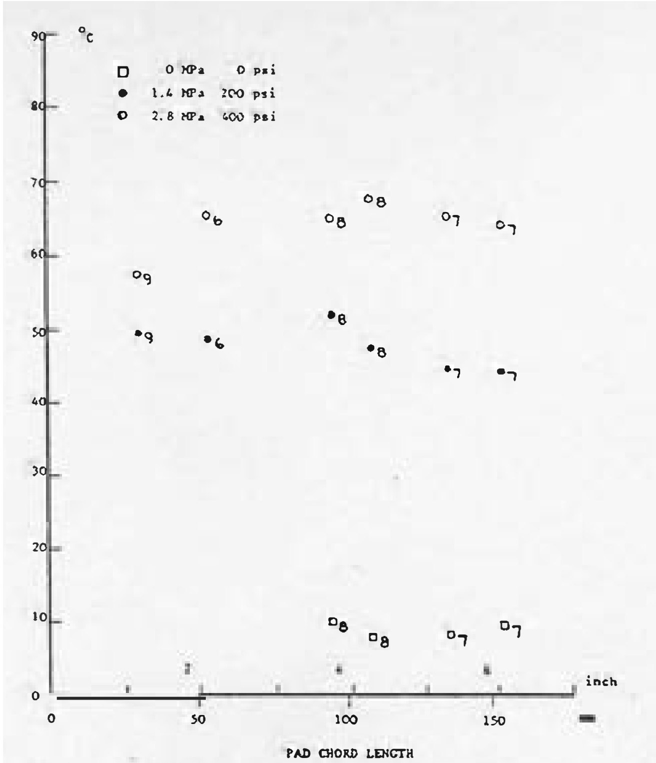


Fig. 11—Effect of pad size on temperature difference parameter: center-pivoted pads, flooded lubrication, sliding speed 40 to 50 m·s⁻¹.

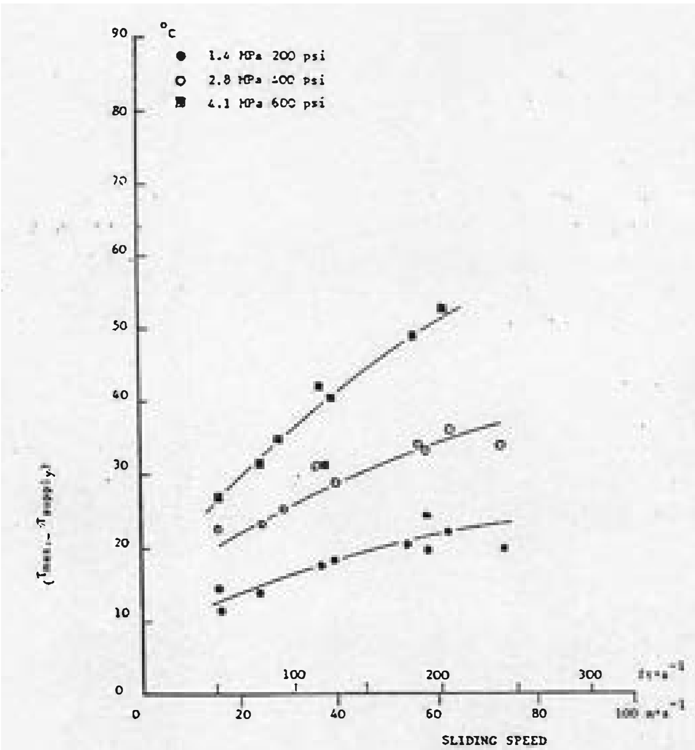


Fig. 10—Temperature difference parameter vs sliding speed for offset pivot pads, low loss lubrication.

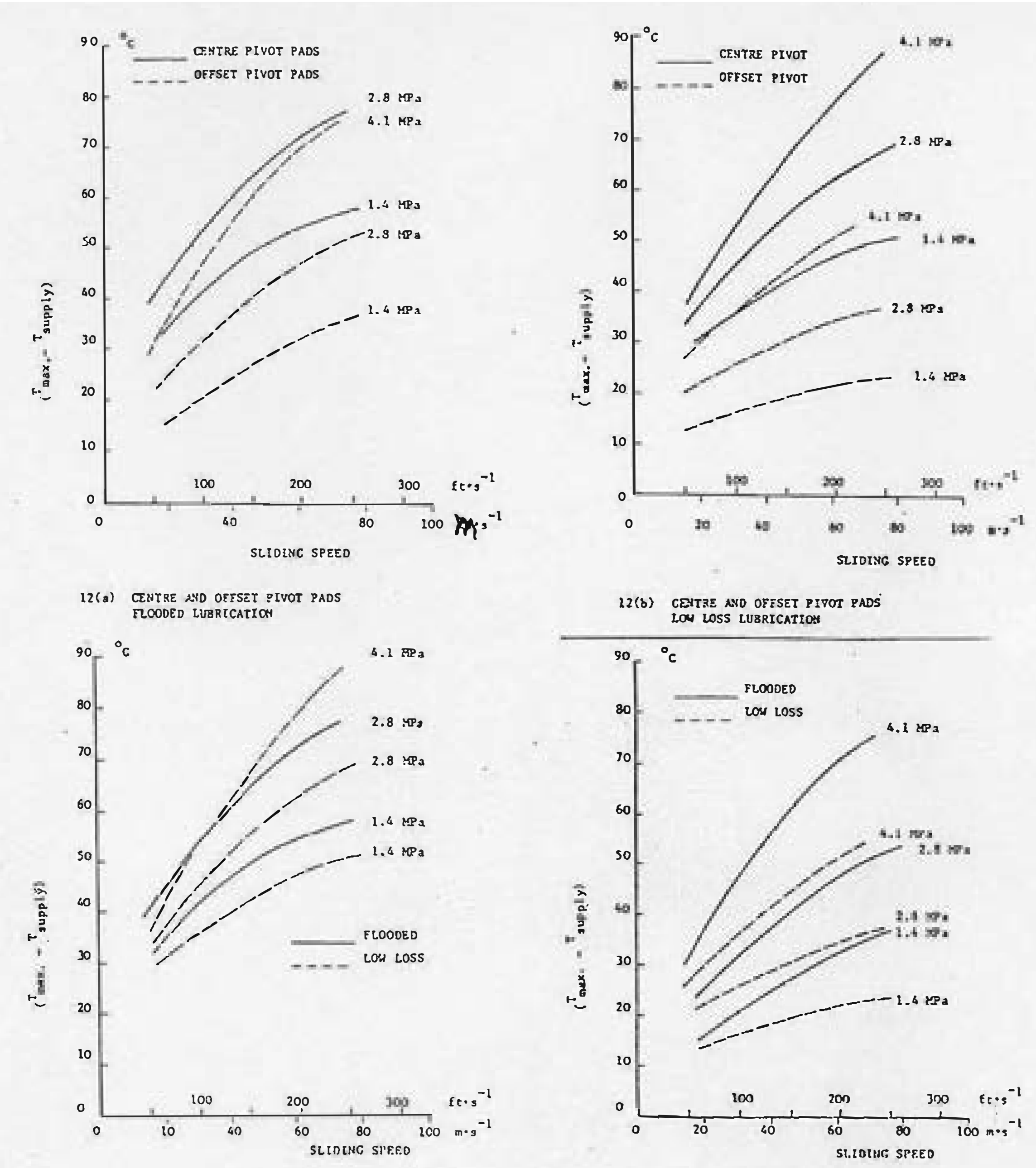
to the published work on center-pivoted pads. Overall, the four sets of mean performance curves can be seen to present a coherent view of pad temperatures covering the normal range of bearing duties.

Bearing in mind the discrepancy noted above, the mean performance curves can be used as a guide to the temperature levels to be expected in a typical bearing. From the required operating condition and basic bearing geometry, the sliding speed and mean pad pressure can be calculated. Using the mean performance curves, the temperature difference between pad and oil supply can be found, and hence the maximum expected pad temperature. Predicted pad temperatures of up to 120°C are usually quite acceptable if the behavior of the bearing is well understood. If greater temperatures are found, the curves can be used to indicate the remedial action to be taken. One course of action is to reduce the mean pressure by increasing the pad area, and suffer the penalty of higher sliding speeds. At the high end of the sliding speed range shown on Figs. 7 to 10, the performance curves are relatively widely spaced and a reduction in temperature is normally obtained despite the penalty associated with the increase in the sliding speed.

REVERSE ROTATION

Having discussed the normal mode of operation for offset- and center-pivoted pads, a point of importance for some users is the performance of offset-pivoted pads during periods when a machine is running in reverse. Despite the prediction of simple lubrication theory, Kettleborough, Dudley, and Baildon (*12*) have shown that offset-pivoted pads do have a substantial reverse running load-carrying capacity. Further test results are presented here to demonstrate this aspect of performance.

Figure 13 shows steady-state temperatures obtained from offset-pivoted pads operating in both directions of rotation, with results gained from center-pivoted pads of the same



12(c) CENTRE PIVOT PADS, FLOODED AND LOW LOSS LUBRICATION

12(d) OFFSET PIVOT PADS, FLOODED AND LOW LOSS LUBRICATION

Fig. 12—Mean performance curves

size. All tests were conducted on the horizontal test rig shown in Fig. 4, with pads 48 mm wide made up into two rings (one offset- and one center-pivoted) giving a bearing outside diameter of 271 mm. Each bearing was driven at speeds from 3500 to 10 000 rpm and loaded to give 2.0, 3.0, and 4.0 MPa mean pressure on the pads. Since the test bearings were designed to operate at high sliding speeds, low loss type of lubrication was employed. For normal operation of both sets of pads, oil was supplied at a rate calculated to give a 17°C temperature rise across the bearing. When the offset pivoted pads were tested in reverse rotation, the quantity of oil supplied to the bearing was held at the same value as for the normal direction of rotation, to model the usual reverse running condition.

Figure 13 shows that offset pivoted pads running in reverse reach a higher steady-state temperature than center-pivoted pads operating at the same duty. However, the difference between backwards running offset-pivoted pads and center-pivoted pads is surprisingly small compared to the difference between the normal operation of offset- and center-pivoted pads. For the classes of machine that do not require reverse operation for extended periods of time, offset-pivoted pads give satisfactory results at reverse running duties typical of normal operating conditions for center-pivoted pads. This advantage has been recognized and used for decades by several manufacturers of rotating machinery.

CONCLUSIONS

This paper has presented maximum pad temperature results obtained from a variety of bearings using pads up

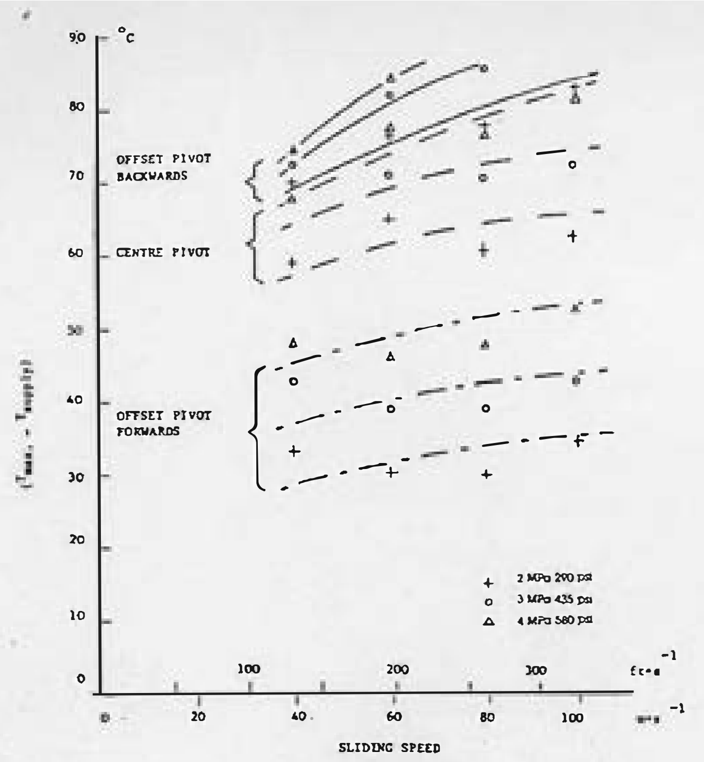


Fig. 13—Comparison of offset-pivoted pad reverse rotation, with center and offset-pivoted pads forward rotation, low loss lubrication.

to 184 mm (radial width) in size, at sliding speeds up to 80 $m \cdot s^{-1}$ (262 $ft \cdot s^{-1}$) and mean pad pressures up to 4.1 MPa (600 psi). Not only does the pad temperature vary with operating condition (speed and load), but consistent trends have been found in the data for different bearing geometries. In a generalized form, the principal conclusions drawn from the study are as follows.

1. At low sliding speeds (less than 30 $m \cdot s^{-1}$), pad temperature is relatively insensitive to mean pad pressure. This indicates that conduction of heat energy through the lubricating film is more significant than convection.
2. At higher sliding speeds (40 to 80 $m \cdot s^{-1}$), pad temperature is much more sensitive to pressure and, over the range covered, sliding speed becomes less important. This is consistent with heat dissipation by convection being dominant at high sliding speeds.
3. Offset-pivoted pads run approximately 20°C cooler than center-pivoted pads.
4. Offset-pivoted pads operate satisfactorily in reverse.
5. Low loss (directed) lubrication offers a temperature advantage over a flooded housing, which rises to 15°C at high sliding speeds (60 to 80 $m \cdot s^{-1}$).
6. Pad size has been shown to have very little effect on maximum pad temperature, for approximately square pads up to 184 mm radial width.
7. Lubricant viscosity grade has also been found to be of minor importance over the range of typical oils used in tilting-pad bearings (ISO grades 32 to 68).

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