Measurements of Maximum Temperature in Tilting-Pad Thrust Bearings©

D. HORNER, Michell Bearings, Newcastle upon Tyne, United Kingdom
J. E. L. SIMMONS, University of Durham, Durham, United Kingdom
S. D. ADVANI, Michell Bearings, Newcastle upon Tyne, United Kingdom

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.

Presented at the 41st Annual Meeting
in Toronto, Ontario, Canada
May 12–15, 1986
Final manuscript approved January 31, 1986

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

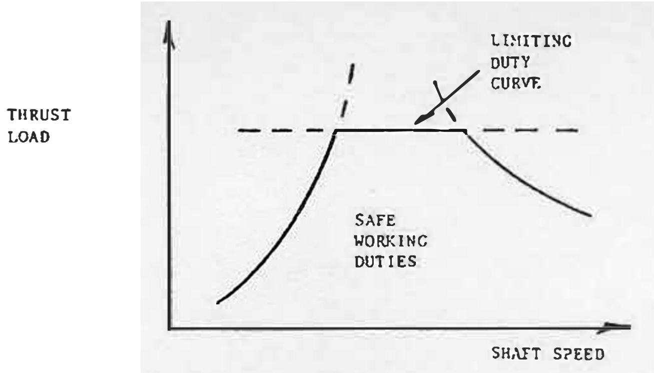

Fig. 1—Limits to thrust bearing operation

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 I 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 I—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)

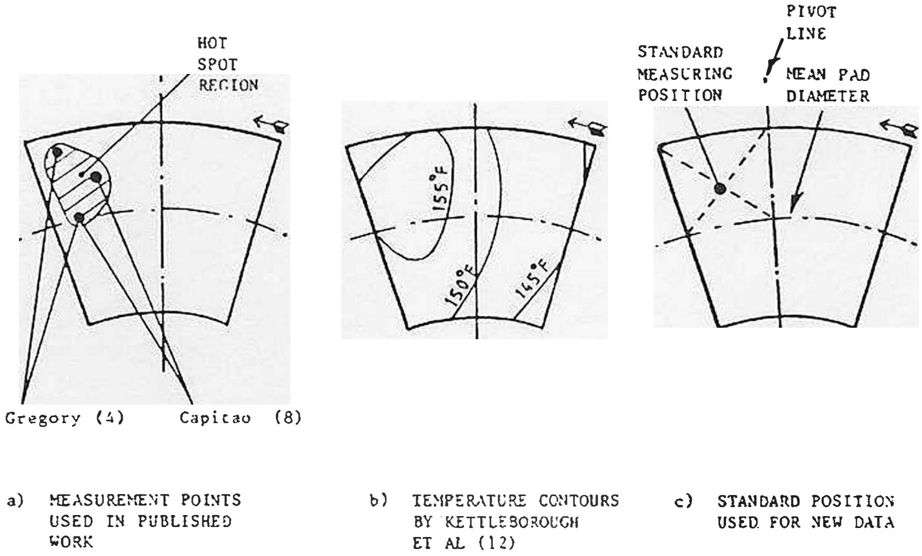

Fig. 2—Comparison of thermocouple locations and hot-spot regions

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 ms^{-1}, the temparture 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 tempera-tures. In percentage terms, this is comparable to the dif-ferences around the pad face reported by New (*5*). Hence, locating thermocouples close to the pad surface is as im-portant 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 thick-ness) from the pad surface.

Table 1 shows that most information is available for center-pivoted pads 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 impor-tant class of bearing is very poorly represented. Refer-ences (*2*) to (*11*) contain sufficient information to establish the trends for center pivot pads in flooded housings. Con-sequently, 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 car-ried out in the authors' company over a period of years. The basic details of the tests performed are shown in Ta-ble 2 with apparatus and instrumentation being described in the next two sections.