Inlet Orifice Type of Lubrication for a Tilting Pad Thrust Bearing," JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 96, No. 1, Jan. 1974, pp. 22–27.
11 Abramovitz, S., "Turbulence in a Tilting Pad Thrust Bearing," TRANS., ASME, Vol. 78, 1956, pp. 7–11.
12 Gregory, R. S., "Performance of Thrust Bearings at High Operating Speeds," JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 96, No. 1, Jan. 1974, pp. 7–14.
13 Gardner, W. W., and Ulschmid, J. G., "Turbulence Effects in Two Journal Bearing Applications," JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 96, No. 1, Jan. 1974, pp. 15–21.
14 Capitao, J. W., "Turbulent Flow in the Tilting-Pad Thrust Bearing," MS thesis, Worcester Polytechnic Institute, Worcester, Mass., 1971.
15 Capitao, J. W., "Influence of Turbulence on Performance Characteristics of the Tilting Pad Thrust Bearing," JOURNAL OF LUBRICATION TECHNOLOGY, TRANS. ASME, Series F, Vol. 96, Jan. 1974. pp. 110–117.
16 Capitao, J. W., Gregory, R. S., and Whitford, R. P., "Effects of High Operating Speeds on Tilting Pad Thrust Bearing Performance," to be presented at ASME Joint Fluids Engineering and Lubrication Conference, Minneapolis, June 1975 published in this issue pp. 73–80.
17 Elwell, R. C., "Instrumentation of Bearing Systems," ASME Paper No. 69-WA/Lub-13, 1969.

DISCUSSION

W. W. Gardner[2]

The type of bearing on which the tests reported in this paper were made is widely accepted and used by rotating machinery designers and manufacturers. Such bearings are frequently being required to operate at increased speeds, primarily in turbomachinery, so the data presented is a valuable addition to this particular area of machine design.

One area of interest in the application of these bearings relates to the details of the oil flow paths into, within, and out of the bearing and its associated housing. The author indicates the use of both overhead and bottom drain lines on the test rig and these are shown schematically in Fig. 3. The use of these drains, either both or only the top drain, was one of the primary test variables and additional information on the specific arrangement would be helpful. Could the author describe these drains including size and orientation. That is, were they radially or tangentially positioned with respect to the bearing?

In this same area, it is indicated in the introduction that the influence of the "flooded" or "evacuated" bearing chamber on temperatures and power loss was investigated. All of the data presented is, it appears, for the "evacuated" condition. Would the author comment on the change found using the "flooded" condition as compared to the data presented?

The decrease in pad temperature associated with the onset of turbulence, as shown in Figs. 5 and 6, is of interest in that it can be considered to confirm the theoretical prediction of higher load capacity (as the author states) and it also provides direct evidence that this decrease occurs in thrust bearings as was previously reported as occurring in journal bearings (Reference [13]). Reference was also made to this temperature decrease in thrust bearings with the onset of turbulence in the author's closure of Reference [12].

The use of this temperature decrease to provide higher load capacity in a practical application is largely discounted by the author due to the relatively high temperature level at which it occurs.

For the specific bearings tested the discusser agrees, but would note that it may be possible to take advantage of this phenomenon with bearings of different materials (as high strength copper backed pads) or different geometries (as offset pivots), which operate at lower temperature levels. Or, combinations of higher temperature capability bearing materials (as copper-lead) and high temperature synthetic lubricants, may permit this increase in load capacity under turbulent conditions to be used.

One of the author's conclusions is that the ultimate design criteria is the maximum pad metal temperature in the loaded area of the bearing. The discusser has no argument with this but would like to add an additional comment to emphasize a point. This being that the maximum allowable temperature is a variable depending on the specific bearing design and operating conditions. For example, bearing failure will occur at relatively lower pad temperatures (and higher loadings) at lower shaft speeds (discounting any turbulence effects). This was demonstrated in the work previously reported.[3]

Two final questions to assist in the use of the data presented in this paper; what was the location of the bearing metal thermocouples with respect to the pad surface, and, would the author comment on the "zero" applied load condition. Is there a fixed value of end clearance associated with this condition?

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R. S. Gregory[4]

This current paper is one of a series of several published and proposed papers by this same author dealing with thrust bearing performance under very specific operating conditions, including the unusual super laminar flow regime. The test procedures outlined in this paper are thorough and result in a large data base that fully delineates bearing performance under the various parametric conditions tested. The author is to be commended for the quality of his experimental technique, and further complimented for his restraint in publishing only a sample of the most significant data lest the complexity of the presentation confuse the issue.

However, there are a number of topics, briefly mentioned in the text, which deserve some amplification simply because they are pertinent to overall bearing performance.

(a) It is stated that the tests were performed with both "flooded" and "evacuated" housings. Were there significant differences between the two modes of operation, and were the various oil levels monitored visually or by other means to determine if the bottom oil drain was equally effective at high and low oil flow rates? It has been the discusser's experience that a gravity drain in the housing bottom must be carefully sized in order not to "starve" the bearing at low flow and still remain effective at high flow rates without choking. It seems that a valved or adjustable drain is an optimum solution only if the oil level is monitored to permit operation at the same height, regardless of flowrate.

(b) It is reported that all six pads were equipped with thermocouples at the same 75–75 percent location. What degree of equalization was determinable, based upon the identical thermocouples in each of the six pads? Do Figs. 5 thru 8 represent measured temperature values for the hottest of the six pads? In a similar test on a 10½ O.D. bearing, the discusser has found an average temperature spread of 10–15°F between the coolest and hottest pads of the bearing, with occasional peaks of up to 25°F difference.

[3] Gardner, W. W., "Performance Tests on Six-Inch Tilting Pad Thrust Bearings," ASME Paper No. 74-Lub-13, 1974.

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(c) It is stated that the 75–75 percent location is "near enough the maximum temperature to give good protection...". Fig. 4 indicates that a generous quantity of thermocouples were distributed over the pads, especially the bottom pad. Have the individual temperature values been evaluated to determine the location of the peak temperature occurring on the pad surface? The discusser has found that the highest measured pad babbitt temperature moves with different operating conditions, yet is located somewhat closer to the pad trailing edge. For example, at shaft speeds of 94.2 m/s, one thermocouple (TR86-C89) exceeded the 75–75 percent location (TR75-C75) by 10–15°F for some low load conditions, while another thermocouple (TR50-C89) exceeded the 75–75 percent location by 30–40°F at high loads. This is mentioned so that the reader is aware that local hot spots on the pad surface possibly can exceed even the high temperatures reported in this paper for the 75–75 percent location.

(d) Finally, the discusser is pleased to see some correlation between the evidence of transition as typified by the power loss curves Figs. 13 and 14 and the temperaure curves of Figs. 5 and 6. While more work remains to be done on establishing a definite correlation, it appears that simple temperature measurements can be used to identify the transitional point between laminar and superlaminar flow. The sharp drop in temperature at the supposed transitional point is a dramatic indication of an abrupt change of state in the fluid film.

It is felt that the author's test work contains much information that would prove useful to bearing users and designers, and future technical papers on this topic are awaited with interest.

Authors' Closure

The author would like to thank Messrs. Gardner and Gregory for their comments and for sharing several insights developed in similar test programs.

As indicated in the discussions, the operation of the test stand in the "flooded" and "evacuated" modes was one of the major areas of investigation. Both the overhead and bottom drains were 4 in. in dia and were constructed such that they were tangential to the test housing. The discharge was then tangential with the 17 in. bearing installed in the housing. For the smaller bearings tested (15 and 12 in.), the housing construction remained constant; thus the discharge flow became a combination of radial and tangential flow. It is important to note, that at all times, the periphery of the shaft collars were sealed by teflon seals to reduce parasitic losses. Generally, it was observed that the power loss consumption in the "flooded" mode was appreciably greater and the bearing babbitt temperatures were higher than in the "evacuated" mode. Although the oil levels were not monitored, the size of the drains insured that the housing was either "flooded" or "evacuated" and the author agrees completely that gravity drains must be carefully sized as not to "starve" the bearing.

The second area of interest pointed out in both discussions, was the pad babbitt temperatures. The conclusion that the ultimate design criterion is the maximum pad metal temperature in the loaded area of the bearing is still valid at lower shaft speeds, where bearing failures have occurred at relatively lower pad temperatures. This phenomenon has been observed in several test programs [2, 3] and can be explained by the mechanical properties of the babbitt surface [7]. To achieve true "hot-spot" temperature readings a broad range of thermocouple locations were used. The majority of the thermocouples were puddled directly in the babbitt using a construction previously documented in the literature [8, 9]. Based upon a pad metal temperature distribution, using the $T_{R75\text{-}C75}$ thermocouples on all six pads, the load equalization was approximately ±5 percent from an average. This average temperature has been used for presentation purposes, Figs. 5–8. Further data analysis is in progress, so that it will be possible to plot the isotherms across the pad surfaces from the leading to trailing edges. This analysis will be the subject of future publications. During the test program, it was observed that the "hot-spot" temperature did move with different operating conditions. The author would like to note however, before any direct comparison can be made to other test results, that all the parameters describing the bearing geometry (pad construction details) and operating conditions (oil inlet and discharge configuration) arrangements must be compared to the actual bearings tested.

The "zero" load condition was considered to be a point of zero travel on the loading mechanism and zero pressure in the hydraulic load pistons. No attempt was made to actually measure the end clearance associated with this condition.