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Performance of Thrust Bearings at High Operating Speeds

As part of a continuing research program, a standard $10^1/_2$ in. dia thrust bearing, of the tilting-pad, self-equalizing type, was tested at shaft speeds up to 11,000 rpm and bearing loads ranging up to 400 psi. The bearing and lube oil system were instrumented to measure bearing performance under laminar and turbulent operating conditions. The effects of varying the oil feed rate on bearing temperature and power loss are discussed in this paper. Some observations on the laminar to turbulent transition region are included.

Significance

The significant contribution of this paper is the publication of experimentally measured values of bearing power loss and pad temperatures under variable load, speed, and oil flow. The operating conditions range from laminar to turbulent, and information of this nature for this popular bearing-type has been heretofore unavailable to designers and analysts.

Introduction

This paper presents the initial results of a current research program investigating the performance of a standard $10^1/_2$ in. Kingsbury double thrust bearing operating at shaft speeds in excess of 15,000 surface ft per min. A petroleum-based, light turbine oil with a viscosity of 150 SUS at 100 deg F was used as a lubricant for all test work. During the course of this experimental study, shaft speeds ranged from 4000 to 11,000 rpm, and bearing loading was varied from "no load" to 400 psi, based upon a bearing area of 55.1 sq in. The tests were performed in a new research and development facility recently constructed to investigate all aspects of high speed bearing performance.

The results reported in this paper are considered incomplete since additional testing is currently under way at still higher shaft speeds. These additional data will be published at some future time. However, it was felt that information on the critical laminar to turbulent transition region would be of particular value if presented now. Several interesting examples of bearing power loss in the transitional region are discussed in this text. Moreover, the considerable effect of a variable oil supply rate on both bearing power loss and bearing temperature is described in detail. This paper is intended solely to present new test data as a contribution to understanding the phenomenon of turbulence in bearings. Analysis and theoretical predictions are postponed until the entire series of tests are complete.

Most, if not all experimental work in the field of bearing turbulence has been performed on single element thrust bearings. It was felt that testing of a double thrust bearing would be more truly representative of actual machine applications. The double thrust bearing consists of a loaded, or "active," thrust bearing designed to absorb the thrust load imposed by the parent machine. On the other side of the shaft collar is the slack-side, or "inactive," thrust bearing which serves to carry any transient loads that possibly might develop in the other direction. The two bearings (loaded and slack) that comprise the double thrust bearing undergoing test are identical in design and size. One of the single element thrust bearings utilized to assemble the double thrust bearing is shown in Fig. 1. It is a conventional design with standard dimensions and centrally pivoted pads.

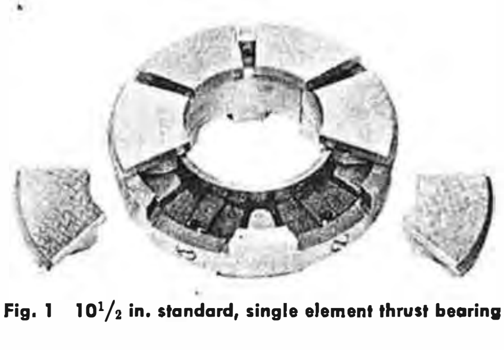

Fig. 1 $10^1/_2$ in. standard, single element thrust bearing

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During the course of normal machine operation, the loaded bearing absorbs the imposed thrust load and operates with a relatively thin film thickness, on the order of 0.001 in. or less. Under this condition, the slack side bearing operates with only the internally generated hydrodynamic load due to collar rotation, and experiences a large film thickness, equivalent to the hot end play of the bearing installation less the film thickness of the loaded bearing. For the test bearing, this slack side film thickness is on the order of 0.017 in. or more. Using one critical value of Reynolds number (as discussed below) for the criterion of transition into the turbulent regime, it can easily be shown that the slack side bearing will encounter turbulence at a much lower shaft speed than a loaded bearing, owing to its thicker film thickness.

Test Apparatus

Mechanical Arrangement. A schematic drawing of the test machine is shown in Fig. 2. A variable speed gas turbine with a rated output of 1100 horsepower is the prime mover. The controllable test speed range is 4000 to 14,000 rpm. The turbine is connected to the test shaft by means of a flexible coupling. Two identical bearing housings contain the bearing components undergoing tests. The forward housing adjacent to the turbine is firmly secured to the foundation while the aft housing is restrained, but free to slide axially. Each housing is equipped with separate lube oil supply and drain lines. Each housing contains a journal bearing to support the test shaft as well as a $10^1/_2$ in. double thrust bearing.

In accordance with normal industry practice for high speed thrust bearings, an oil control ring surrounds the collar to divert the bearing discharge oil away from the collar periphery. The construction of the oil control ring is shown in Fig. 3. Teflon oil seal rings are used to contain the oil in the thrust bearing area and prevent leakage either into the journal bearing cavity or out of the housing. This type of bearing arrangement is typical of many actual machine designs. All horsepower loss values reported in this paper include those for a double $10^1/_2$ in. thrust bearing plus the rotating seal rings and stationary oil control ring. Each thrust bearing is a standard 6-pad Kingsbury equalizing type, with a babbitt O.D. of $10^1/_2$ in. and an I.D. of $5^1/_4$ in. Pad arc length is approximately 51 deg and the total bearing area is 55.1 sq in.

Load is applied to the test bearings by means of an external hydraulic system which transmits a force directly to the sliding housing. This load applicator consists of a manually operated pump, hydraulic cylinder and force multiplier arm, with an oil reservoir, accumulator, pressure relief valve, solenoid valve, and pressure gage completing the system. The mechanism of loading can best be understood by referring to Fig. 4. The instrumented journal bearings are of the tilting pad type, 5.0 in. dia $\times$ $2^1/_4$ in. long, and serve only to support the test shaft. A force input is applied to the aft end of the sliding housing causing the housing to move forward a small amount within the limits of bearing end play. This housing movement forces thrust bearing "D" against the test shaft collar, thus applying a load to the shaft. Thrust bearing "C" moves along with the housing since it is not restrained and it operates as an unloaded, or "slack side," bearing. The movement of the test shaft is limited by thrust bearing "A," located in the fixed housing. As a result of the single applied force, both thrust bearing "A" and "D" experience the same loading, while thrust bearings "B" and "C" remain unloaded.

This duplicate arrangement permits a check of repeatability when two identical bearings are installed in the test rig. Conversely, the installation of two dissimilar thrust bearings permits a fair comparison to be made under identical operating conditions. In any event, the flexible coupling at the turbine end of the test shaft is relatively soft and prevents load transmittal to the turbine bearings, precluding any possibility of "sharing" the applied test load.

Instrumentation. The lube oil supply line to each of the six bearings is equipped with a turbine flow meter, throttling valve,

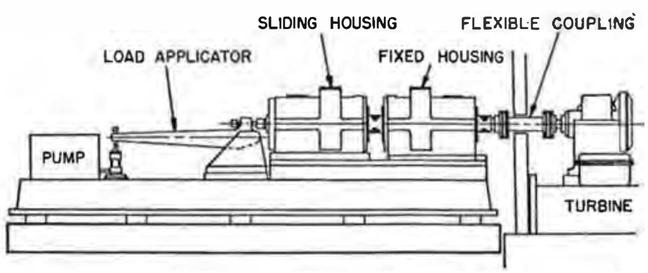

Fig. 2 Schematic of thrust bearing test apparatus

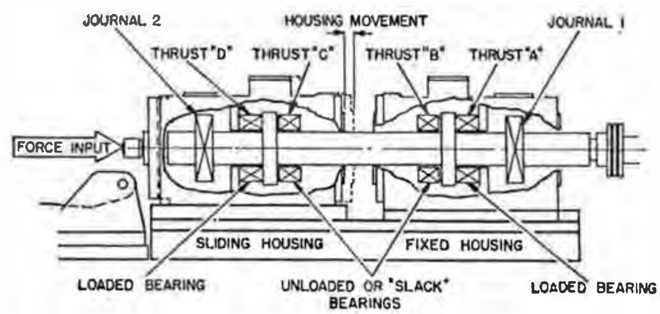

Fig. 4 Description of method of applying thrust load

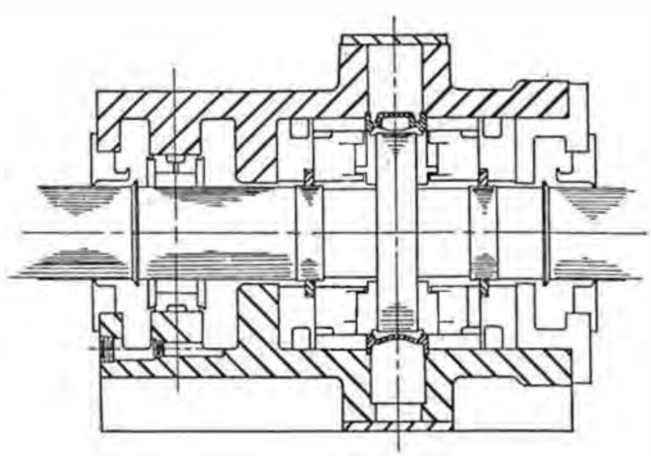

Fig. 3 Cross-sectional view of housing interior

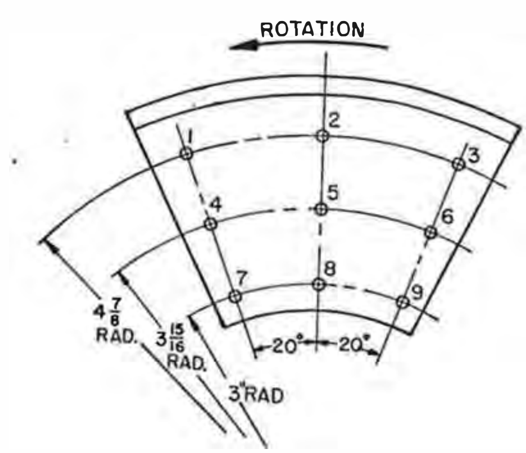

Fig. 5 Thermocouple locations on thrust pad

thermocouple and pressure transducer. All drain lines are similarly equipped with thermocouples and throttling valves, making it possible not only to determine the bearing power losses by an energy balance method, but also to independently vary the oil flow conditions to each separate bearing.

Small thermocouples are imbedded in the babbitt metal of the journal bearing and thrust bearing pads. The actual thermocouple junction is positioned within $^1/_{32}$ of an in. of the babbitt surface. An array of nine thermocouples is installed in one pad each of bearing "A" and bearing "D" in order to determine the temperature gradient in both the radial and circumferential directions of an operating thrust bearing pad. Fig. 5 shows these thermocouple locations.

Additionally, one pad in loaded bearing "A" is equipped with an array of nine eddy-current proximity probes for the purpose of measuring oil film thickness. Electronic load cells are installed in several of the bearing leveling plates to measure the actual applied thrust load carried by each pad of the loaded bearing and to give an indication of the degree of pad equalization. A larger load cell is mounted on the end of the sliding housing for the purpose of measuring the total applied thrust load. All transducer outputs are continuously recorded on multipoint, strip chart recorders.

Test Procedure. A typical test run was performed at a constant shaft speed. The bearing loading was varied from "no load" thru 400 psi in increments of 50 psi. The oil flow rate was set at the normal value as recommended by the author's company, then varied 50 percent above and 50 percent below this normal value for each load condition. The test series reported here included shaft speeds of 4000 thru 11,000 rpm at increments of 1000 rpm. The test runs were not performed in consecutive order, nor has any run been made at 6000 rpm to date. Additional tests at shaft speeds above 11,000 rpm are planned for the future. As a result of the completed testing, large amounts of data have been amassed. Only a limited amount of the most important results are presented in this paper due to space and time restrictions. Future papers will discuss further results.

Test Results

Effects of Oil Flow Rate on Bearing Power Loss. The experimental data are presented in a manner that facilitates information retrieval. To accomplish this, it was necessary to define a new parameter, called the "Oil Flow Ratio." Figs. 6 and 7 depict power loss as a function of this oil flow ratio, as well as bearing loading and shaft speed. This oil flow ratio is computed by dividing the total oil flow (in gallons per minute) supplied to both loaded and slack side bearings, by the total measured horsepower loss for the same two bearings. The bearing loss is computed for the double thrust bearing by an energy balance method whereby power loss is a direct function of measured oil temperature rise, measured oil flow and lubricant specific heat. Thus the oil flow ratio may be regarded as an inverse function of the oil temperature rise across the bearing assembly. Losses due to radiation from the housing are ignored in this analysis.

A unique value of bearing power loss can be experimentally determined for each flow condition tested. When plotted on a log-log scale, the three values of power loss consistently plot as straight lines for each shaft speed. These straight lines have been extrapolated in Figs. 6 and 7 to cover additional flow ratios which may be of interest. Comparisons can be made of bearing power loss with either constant or variable oil flow ratios.

It is important in any subsequent analysis to specify the particular oil flow rate involved, since the effect of changing the oil flow ratio on total bearing power loss is quite significant. Consider this example of a $10^1/_2$ in. double thrust bearing operating at 10,000 rpm with 100 psi load: when supplied with an oil flow of 54 gallons per minute, the total bearing power loss will be 202 horsepower. Reducing the oil flow by 50 percent (to 27 gallons per minute) will result in a measured loss of 150 horsepower or a reduction of almost 26 percent. However, this economy is not achieved without paying a price in higher bearing operating temperatures. Both values of power loss in the example are valid and correct for their respective oil flow ratio of 0.267 and 0.180, emphasizing the fact that the pertinent oil flow ratio must be stipulated in any analysis of bearing performance.

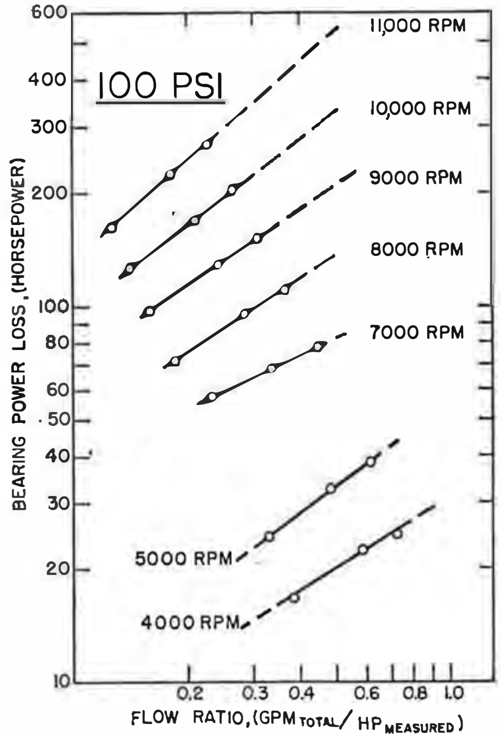

Fig. 6 Accumulated results of bearing power loss versus oil flow ratio at 100 psi load

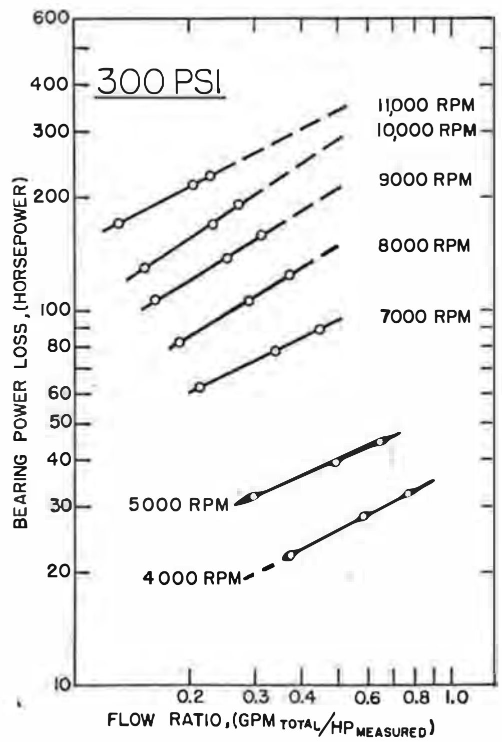

Fig. 7 Accumulated results of bearing power loss versus oil flow ratio at 300 psi load

Effect of Oil Flow Rate on Bearing Metal Temperatures. The array of thermocouples imbedded in the loaded pad babbitt permits one to report a multitude of temperature values to describe the temperatures across a bearing pad. However, each thermocouple location should not be given equal weight. Trailing edge temperatures are obviously more important than leading edge values in limiting bearing loads at a given speed. Single thermocouple values taken at one specific pad location are subject to some inaccuracy, partly because of the different manner in which the pad equalizes each time the test rig is started, but also due to shifts in the location of the maximum temperature. Therefore this paper adopts "pad average temperature" as the best estimate of true bearing performance. This pad average temperature is merely the average of the four hottest thermocouples on the pad, which are always located at positions 1, 2, 4, and 5 in Fig. 5. Of course, the actual trailing edge temperatures are somewhat higher, but the pad average temperature does give a good approximation of pad temperature in the quadrant that is often instrumented in a bearing pad.

Under the circumstances noted in the example above, the pad average temperature was 195 deg F with an oil flow rate of 54 gpm. Reducing the oil flow to 27 gpm results in a higher pad average temperature of 206 deg F. While this particular temperature is not too severe, owing to the relatively light loading of the bearing, operation at higher loads will cause significantly higher pad temperatures. Fig. 8 shows the changes in pad average temperature and drain temperature due to an increase in thrust load at a constant operating speed and relatively constant oil flow rate. Note the lack of response of the drain temperature to large load changes. Two pad trailing edge temperatures are also shown to give some indication of typical temperature levels, and illustrate the problems that could occur if the pad average temperature was not used.

The influence that the oil flow ratio alone exerts on pad temperature can be seen from Fig. 9 plotted for a constant 100 psi load. Note that a flow rate change affects pad temperature much more significantly at the higher shaft speeds.

The Laminar-Turbulent Region. It has been reported in the literature for several years that bearing power loss will exceed laminar predictions after a certain shaft speed is reached and surpassed. Traditionally this phenomenon has been termed turbulence and the transition point is one of some interest to bearing designers. It is possible in many instances to detect this point of transition by plotting values of bearing horsepower loss against shaft speed on a log-log grid. The change in slope of the resulting curve is indicative of the increased power losses associated with the turbulent regime after passing through the transition point.

Such a plot is presented in Fig. 10 for the "no load" case of the test bearing with constant flow ratios of 0.3 and 0.5. Even though the change in slope is not pronounced, it is evident from Fig. 10 that the transition point is attained at a shaft speed between 8000 and 8600 rpm. Note that only one point of transi-

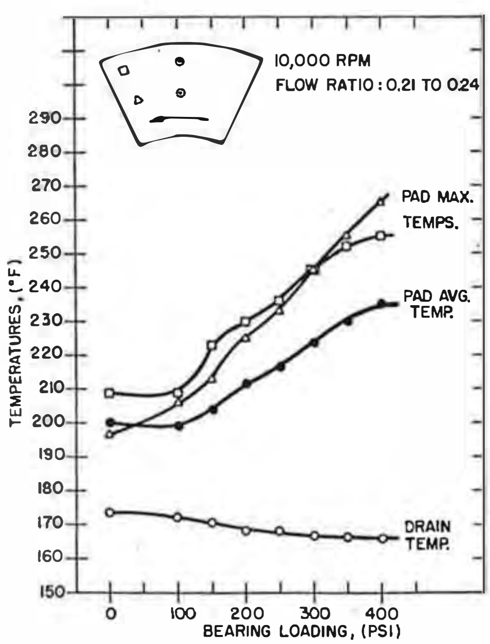

Fig. 8 Bearing temperatures versus bearing psi loading

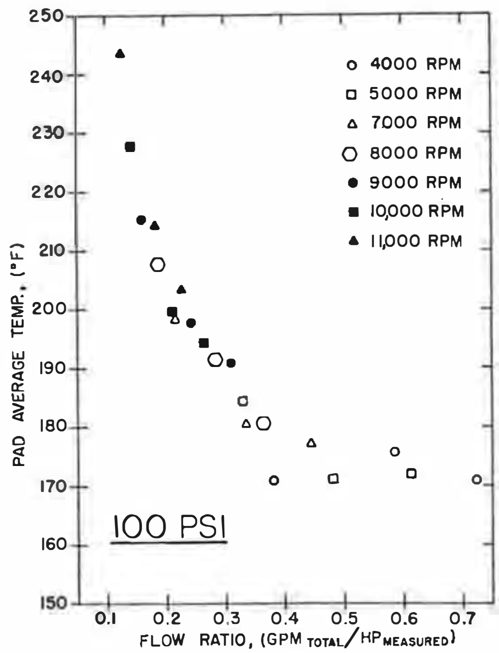

Fig. 9 Pad average temperatures versus oil flow ratio

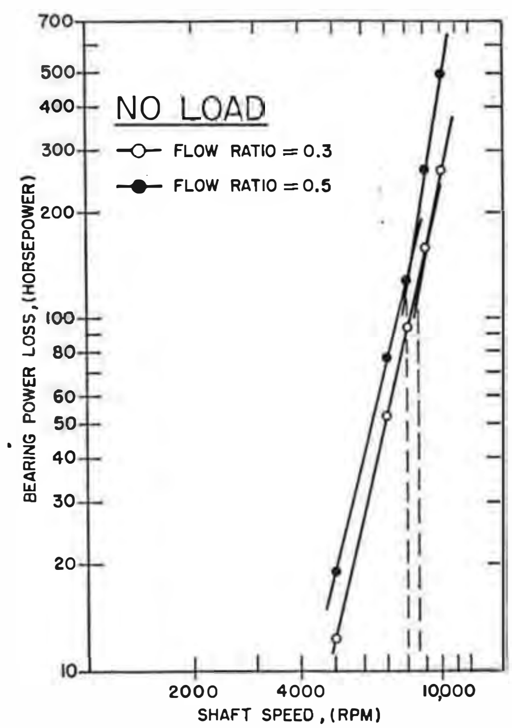

Fig. 10 Bearing power loss versus shaft speed at no load

Table 1 Laminar-to-turbulent transition information

Bearing	Load (psi)	rpm	Re	Predicted status	Experimental status
Loaded	No load	5,000	349	Laminar	Laminar
Slack	No load	5,000	349	Laminar	Laminar
Loaded	No load	7,000	500	Laminar	Laminar
Slack	No load	7,000	500	Laminar	Laminar
Loaded	No load	8,000	580	Transition	Transition
Slack	No load	8,000	580	Transition	Transition
Loaded	No load	9,000	658	Turbulent	Turbulent
Slack	No load	9,000	658	Turbulent	Turbulent
Loaded	100 psi	5,000	212	Laminar	Laminar
Slack		5,000	550	Laminar	Laminar
Loaded	100 psi	7,000	350	Laminar	Laminar
Slack		7,000	721	Turbulent	Turbulent
Loaded	100 psi	9,000	511	Laminar	Laminar
Slack		9,000	870	Turbulent	Turbulent
Loaded	100 psi	10,000	603	Turbulent	Turbulent
Slack		10,000	934	Turbulent	Turbulent
Loaded	300 psi	5,000	146	Laminar	Laminar
Slack		5,000	892	Turbulent	Laminar
Loaded	300 psi	7,000	242	Laminar	Laminar
Slack		7,000	1215	Turbulent	Turbulent
Loaded	300 psi	11,000	477	Laminar	Laminar
Slack		11,000	1824	Turbulent	Turbulent
Loaded	300 psi	13,000	618	Turbulent	?
Slack		13,000	2137	Turbulent	?

tion can be seen due to the fact that both the loaded and slack bearings are operating with approximately the same value of film thickness. It is thus likely that they enter turbulent regime simultaneously, thru the same transition point.

Using a slightly modified version of a critical Reynolds number as reported by Abramovitz[1] it is possible to make some theoretical estimates of the location of the transition point. Equation (1) shows that the Reynolds number is a function of density, mean operating velocity, minimum film thickness and lubricant viscosity.

$$\mathrm{Re} = \frac{\rho U h_0}{\mu} \tag{1}$$

A calculated value of minimum film thickness is substituted for the mean film thickness used by Abramovitz to formulate the Reynolds number for his water lubricated bearing tests. Alternatively, the value of critical Reynolds number could have been adjusted to compensate for the viscosity difference.

Abramovitz indicated the transition region started at a critical Reynolds number value of 580 to 800. Table 1 summarizes the results of these theoretical predictions for a range of shaft speed and load values. As the table indicates, transition is predicted for the "no load" case at approximately 8000 rpm, based on the use of calculated values of film thickness and viscosity and with an oil flow ratio of 0.47. Good agreement is anticipated for the 0.5 oil flow ratio, and this indeed is the case.

A similar plot for a double thrust bearing operating at 100 psi with constant oil flow ratios of 0.3 and 0.5 is shown in Fig. 11. Each curve shows two, clear transitional points. The first occurs between 5300 and 6400 rpm, the second between 8800 and 9700 rpm. It is suggested that the first transitional point is due to the slack side bearing entering the turbulent regime at a lower shaft speed due its thicker film thickness. The second transitional point is taken to represent the loaded bearing entering the turbulent regime at a somewhat higher shaft speed owing to its much thinner oil film thickness. Referring to Table 1 and the values of Reynolds number reported for 100 psi bearing loadings, it can be seen that the transitional point for the slack side bearing is predicted to lie between 5000 and 7000 rpm while loaded bearing transitional point is delayed until the interval between 9000 and 10,000 rpm.

[1] Abramovitz, S., "Turbulence in a Tilting-Pad Thrust Bearing," Trans. ASME, Vol. 78, Jan. 1956, pp. 7–11.

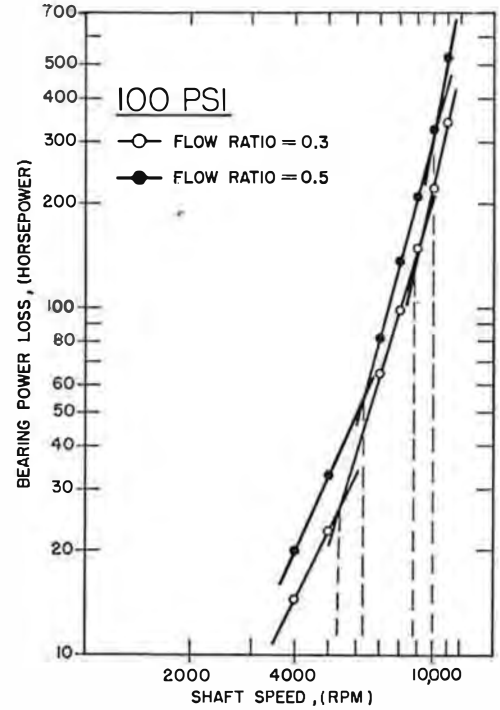

Fig. 11 Bearing power loss versus shaft speed at 100 psi

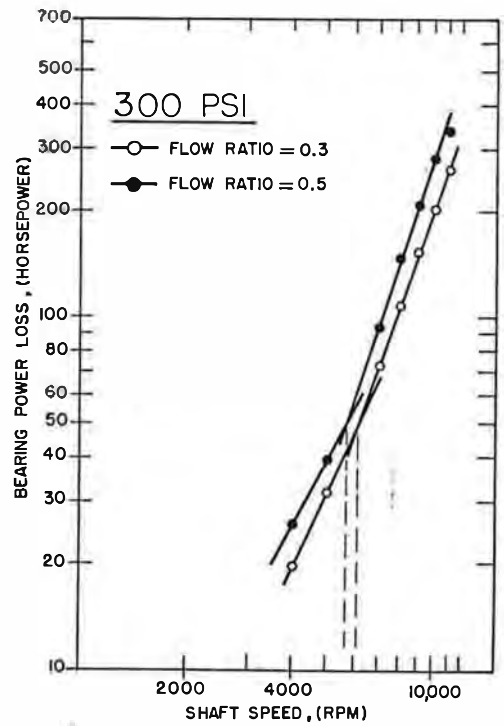

Fig. 12 Bearing power loss versus shaft speed at 300 psi

Fig. 12 shows similar results for 300 psi loading. Experimental evidence would indicate the transition point to occur between 5700 and 6100 rpm. Referring to Table 1, it is evident that there is some conflict, since the calculated Reynolds number for the slack side bearing at 5000 rpm is 892, which exceeds the critical value of 580. However, the experimental evidence shows that the bearing is still operating in the laminar regime at this speed. The onset of turbulence for the loaded bearing is predicted between 11,000 and 13,000 rpm. There is no clear indication in

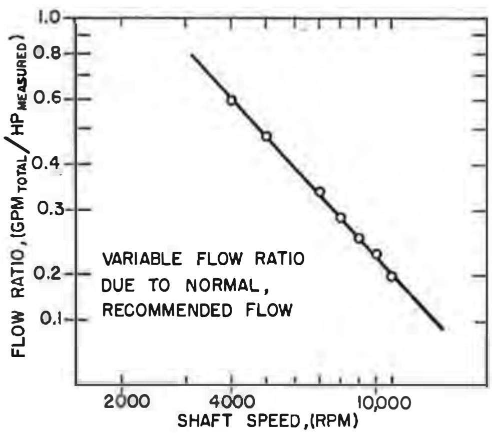

Fig. 13 Variable flow ratio versus shaft speed

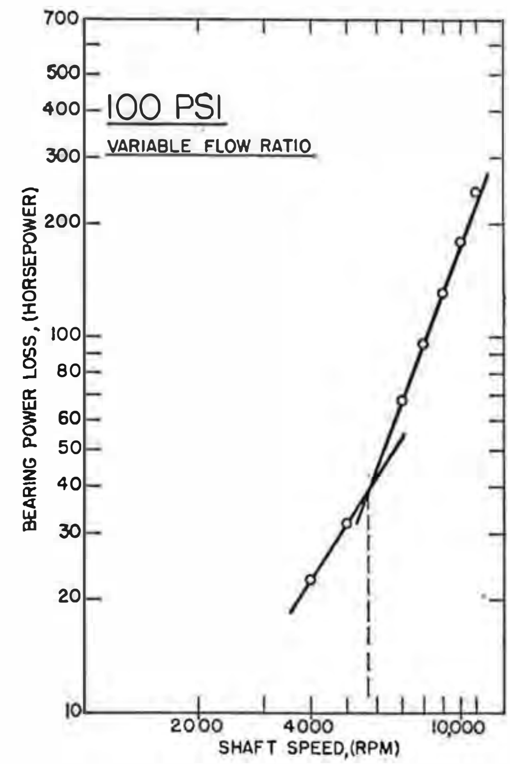

Fig. 14 Bearing power loss versus shaft speed at 100 psi

the test data of a second transitional point up to the maximum test speed of 11,000 rpm, indicating that additional tests are required at higher speeds.

There has been some question raised in the literature[2] whether a single critical Reynolds number value will actually pinpoint the onset of turbulence. No one fixed value of Reynolds number may be adequate to cover this transitional phase. It is apparent from the data presented here that this is indeed the case. It is suggested that a range of Reynolds numbers is required to locate the transition phase with accuracy. Constant oil flow ratios simplify the problem of discussing bearing performance because they remove one variable parameter from the discussion. However, they are not truly representative of actual bearing practice. Bearing manufacturer recommended oil flow rates usually vary according to the operating speed and allowable oil temperature rise.

[2] Missana, A., Booser, E. R., and Ryan, F. D., "Performance of Tapered Land Thrust Bearings for Large Steam Turbines," *ASLE Transactions*, Vol. 14, No. 4, Oct. 1971, p. 304.

Table 2 Typical values of oil flow ratio

Shaft speed rpm	Oil flow ratios Maximum	Minimum	Average
4000	0.630	0.584	0.601
5000	0.491	0.471	0.479
7000	0.340	0.335	0.344
8000	0.290	0.284	0.288
9000	0.258	0.241	0.250
10,000	0.239	0.211	0.224
11,000	0.209	0.181	0.198

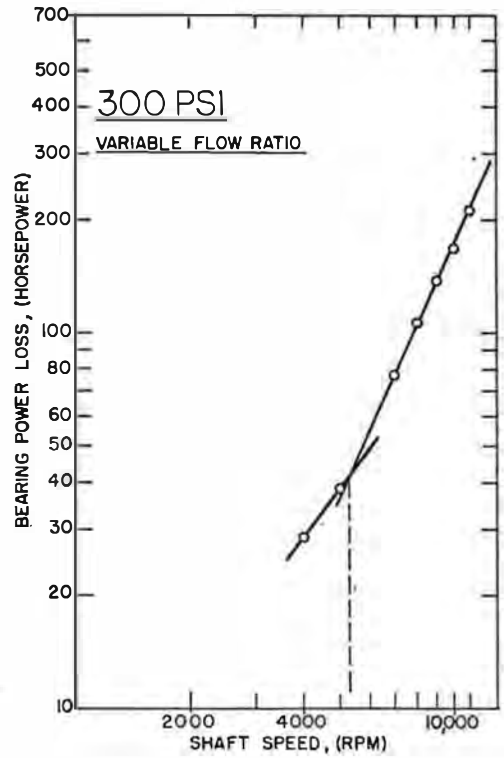

Fig. 15 Bearing power loss versus shaft speed at 300 psi

A typical curve relating oil flow rate and shaft speed is shown in Fig. 13. The flow ratio changes from a value of 0.6 at 4000 rpm to 0.2 at 11,000 rpm. This curve is a good representation of the "normal" flow values used as a basis for experimental work. Table 2 reports values of flow ratio encountered during this experimental work.

As could be expected, use of a variable flow ratio does have some effect on log-log plots of the bearing test results. Figs. 14 and 15 represent power loss expressed as a function of the variable flow ratio for bearing loads of 100 and 300 psi. The changing flow ratio makes the change in slope of the curve at the transition point more apparent. The transitional zones, however, remain at approximately the same locations indicated by the constant flow ratio curves. Again, it is presumed that the single transitional point denotes where the slack side bearing first enters the turbulent regime. There is sufficient data to locate the second transition point, associated with the loaded bearing.

Conclusions. It is intended to obtain more data at higher speeds to locate the turbulent transition region for the loaded bearing with consistency. Additional test runs at intermediate speeds would also be useful in defining the slack side bearing transition zones. This work is currently in progress and will be reported in the near future. It is felt that one of the major contributions of this paper is the experimental evidence that demonstrates the effect of oil flow rate change on the bearing power loss. The magnitude of the effect is such that any comparison or stipulation of bearing power loss must have an oil flow rate clearly stated in order to completely determine the operating performance of the bearing in question. Similarly, bearing temperatures are subject to wide fluctuations for a number of reasons, including variations in the oil flow rate.

The newly defined "oil flow ratio" parameter greatly simplifies the problem of presenting bearing performance data. By combining the power loss and oil flow terms in a simplistic manner, this parameter becomes a valuable tool to both the bearing experimenter and the bearing designer because it establishes a basis for the comparison of results. Moreover, the relationship that the Oil Flow Ratio shares with the oil temperature rise across the bearing can be exploited by making approximations for the lubricant specific heat which is also a function of bulk oil temperature.

The difficulty associated with using a value of Reynolds number, calculated for a specific operating condition, to predict the laminar, transition or turbulent status of the oil film does not appear to be resolved as yet. Further experimental work will help to locate the areas of transition with precision. Perhaps the Reynolds number calc lation itself could be refined by the use of experimentally measured oil film thickness values, better temperature viscosity assumptions and other improvements. For example, the use of hot end play values instead of the normal cold end play will definitely improve the conflict between experimental and measured transition points for the 300 psi operating condition outlined above.

In this series of tests, the peak temperatures achieved under certain flow, load and speed conditions are well in excess of computed or predicted values. They do represent a limiting factor that could prove restrictive to further testing. For example, maximum pad trailing edge temperatures were in excess of 290 deg F during runs made at 9000 rpm under certain flow conditions. However, the bearing operated well and survived undamaged for the short duration of these extreme conditions.

The tests completed thus far show clear evidence of transition from laminar to turbulent fluid flow for both the loaded and unloaded sides of a typical bearing assembly operating in a typical range of design speeds and other conditions. As such, the occurrence of turbulence may be regarded as a phenomenon to be encountered within the present state of the art of machine design.

Acknowledgments

The facilities and personnel of Kingsbury, Inc. were utilized to perform these bearing tests and gather the data presented in this paper. Gratitude is expressed to Kingsbury, Inc. for permission to publish these results. The many helpful comments and suggestions of Mr. R. C. Elwell of General Electric Co. are also gratefully acknowledged.

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