Journal Bearing Operation at Low Sommerfeld Numbers

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Tests were conducted to determine experimentally the conditions under which the transition from full hydrodynamic film to boundary lubrication occurs in journal bearings. These tests were made with two-grove, sleeve type and a tilting-pad type journal bearing, with a 13-inch diameter shaft. Theoretical considerations dictated transition at lower Sommerfeld numbers for a particular tilting pad arrangement as compared to the sleeve bearing. Experimental confirmation of this was found. Additional tests in the boundary lubrication area indicated general limits for loading here, too.

INTRODUCTION

The work reported here was pursued as portions of two test programs of larger scope on bearings intended primarily for radial support of marine main propulsion shafting. One of these programs investigated the operating characteristics of a two-grove, sleeve-type, disk-lubricated, babbitt-lined journal bearing, as commonly used for the support of the inboard sections of the propulsion lineshafting between the drive equipment and the propeller shaft. The other program included similar test work on a tilting-pad type journal bearing designed for use as the aftermost bearing, on the propeller shaft, supporting the weight of the propeller and associated shafting. This was a five pad bearing, babbitt-lined, and flood-lubricated.

These particular applications require that such bearings operate satisfactorily over a wide range of speeds. On large ocean and lake vessels, maximum shaft speeds are often in the range of 80 to 200 rpm. Operation of these ships requires shaft speeds from these maximum conditions down to maneuvering speeds, which may be in the area of 20 to 50 rpm. An operating condition of recent development is that of single-point mooring of large tankers while loading or unloading cargo at an offshore location. Under these conditions, shaft speeds of 5 to 15 rpm can be encountered. For turbine driven ships, turning gear speeds are also an integral part of the total operation. These are normally in the range of \( \frac{1}{4} \) to \( \frac{1}{2} \) rpm for propulsion shafting, reduced from the turbine turning gear speeds by the reduction gear.

At maximum shaft speeds, full film hydrodynamic conditions are developed within the bearings. At the turning gear speeds noted above, the hydrodynamic film is not developed but contact between the shaft and the bearing occurs. If this metal-to-metal contact continues at increased shaft speeds (surface velocities), bearing wiping and failure will result. Thus, it is of interest to determine the conditions under which the transition from full film operation to boundary lubrication occurs.

The nomenclature used to describe the several conditions of operation of journal bearings is not consistent in the literature. One condition is termed full-film, perfect, fluid, or hydrodynamic and describes the condition in which shaft and bearing are separated by the lubricant film. When the operating conditions are such that this full film separation does not develop, but shaft to bearing contact occurs, it is termed mixed-film, partial, or boundary lubrication. This condition of operation, where

\[
S = \text{Sommerfeld number} = \frac{\mu N}{D/2C} \left( \frac{b L}{d} \right)^2, \text{ dimensionless}
\]

\[
W = \text{Bearing load, lb}
\]

\[
b = \text{Disk axial length, in}
\]

\[
d = \text{Disk diameter, in}
\]

\[
f = \text{Coefficient of friction, dimensionless}
\]

\[
h_{bf} = \text{Min. film thickness, in}
\]

\[
c = \text{Eccentricity ratio, dimensionless} = \frac{1 - h_{bf}}{C}
\]

\[
\mu = \text{Lubricant viscosity, lb sec/in}^2
\]
contact of the shaft and bearing occurs, is sometimes further divided into two areas which are commonly termed boundary and mixed-film. The mixed-film term corresponds to the area of rising coefficient of friction to the left of the minimum point in Fig. 1, while the boundary term then refers to that portion of essentially constant coefficient of friction to the left of this.

![Graph](image)

**Fig. 1—General relationship between journal bearing coefficient of friction and Sommerfeld number.**

In this paper, the full film separation condition was always self-generated, so the term hydrodynamic is preferred. When some contact between shaft and bearing also develops, the term boundary lubrication appears to be an appropriate descriptive term. For this work, lubricant was always present.

Figure 1 shows the familiar general relationship of the bearing coefficient of friction to the bearing Sommerfeld number ($S$). Hydrodynamic conditions exist in the portion to the right of the “knee”, while boundary conditions account for the increased friction to the left of this portion. In test work, this point (or area) of change is normally readily determined, as the friction forces, from shaft to bearing contact, increase rapidly to values well above those in the adjacent hydrodynamic regime.

Although this particular work involved bearings intended for a specific area of application, the data and results are applicable to similar bearings which may be used in numerous other areas.

**TEST EQUIPMENT**

The setup for testing the sleeve type journal bearing is shown in Fig. 2. The 13-inch bore by 11-inch long bearing is contained within the housing on the left end of the test stand. This housing is loaded upwards by a hydraulic jack between it and the base. Measurement of the oil pressure to this jack indicated the load on the bearing. The load is transferred back to the base, from the shaft, through the two pillow-block (rolling contact) bearings mounted on pedestals at both ends of the test bearing housing. The mounting arrangement provided for self-alignment of the bearing to the shaft.

The drive is a hydraulic motor to provide for speed variation over a wide range. A shaft mounted torque meter is located between the drive motor and the bearing. The oil pressure to the drive motor was also used to measure shaft torque. Thermocouples placed into the babbitt surface of the bearing measured the bearing temperature. The test bearing shell is shown in Fig. 3.

As noted above, this test bearing was disk lubricated. This is simply a flanged disk secured to and rotating with the shaft, the lower portion of which extends into the oil contained in the lower portion of the bearing housing. As this rotates, oil is carried on the disk flange where it is scraped off at the top and directed into the bearing. This arrangement is clarified in Fig. 4 which shows the upper half of the housing removed and the scraper held in position by an added bracket.

The tilting-pad journal bearing was tested on the same stand with the same 13-inch diameter shaft. This bearing in its test housing mounted on the stand is shown in Fig. 5. The housing end cover, seal, and adjacent pedestal and pillow-block bearing have been removed. The top

![Image 2](image)

**Fig. 2—Sleeve bearing test setup**

![Image 3](image)

**Fig. 3—Test sleeve bearing shell**
The housing containing this pad bearing was flooded by circulating oil in at the bottom and overflowing through a top outlet, back to a sump tank.

As shaft finish is an important factor in determining the transition from hydrodynamic to boundary conditions, the same shaft was used for tests on both types of bearings, as previously noted, and the results can then be compared. The shaft drawing specified a 16 microinch finish (grind) and actual measurements showed it to be from 14 to 22 microinches in the bearing area. The shaft material was low carbon steel (forging).

The initial babbitt finish is not as critical, as these surfaces “polish in” after a period of operation including a number of starts. Both bearings had been tested rather extensively prior to these low Sommerfeld number tests, and so were well “broken in.” High tin babbitt of the same composition was used in both bearings.

Thus, no variations, of significance, between the two bearings can be traced to differences in finish or material of either the shaft or bearings.

**TEST PROCEDURE**

In all cases, in any one specific test the variation in Sommerfeld number was obtained by changing the shaft speed. That is, the bearing loading was held constant, as was the lubricant viscosity. The latter was not specifically controlled but rather was a result of the test conditions. At the low surface velocities generated in these tests no detectable temperature rise was noted in the bearing over that of the oil. This temperature was used to determine the viscosity value.

Variations, from test to test, were made in the bearing loading and in the specific grade of oil used. These are noted in the test results.

These tests all went from the hydrodynamic condition to the boundary condition, this being accomplished by reducing the shaft rpm. A typical test would start at a shaft speed giving hydrodynamic operation, say 5 rpm.
as determined by shaft torque readings. This speed would then be reduced in increments, holding at each speed for several minutes to establish steady operation and to take data. After conditions were reached at which some contact was occurring, further reduction in rpm was limited, due to the sharp increase in frictional torque which developed. However, it was the determination of the conditions at which this occurred that was of specific interest and this was obtained.

For the tests at the very low turning gear speeds, a reducing gearbox was installed in the drive line. Tests were made in the range of 0.012 to 0.2 rpm. This was well into the boundary regime and no torque data was taken, as the setup and instrumentation were not suitable for determination of these relatively large values.

**TEST RESULTS**

The test data for each of these bearings is of interest independent of the other. However, it is also of interest to compare the operation of the bearings. Two differences, other than bearing type, existed which need to be clarified to permit this comparison. First, the sleeve bearing was disk-lubricated and the pad bearing was flood-lubricated. Second, the sleeve bearing L/D ratio was 0.85, while that of the pad bearing was 1.0.

Tests of the disk-lubricated bearing included collection of considerable data on the oil flow from the disk-scraped arrangement. At the lower shaft speeds (of specific interest here), where oil is not thrown off the disk by centrifugal force, the flow was found to vary linearly with a parameter including the disk width, disk diameter, rotational speed, and viscosity. Figure 7 is a plot of oil flow data vs this parameter for two different size disks and four different viscosity oils. This is included here simply to indicate the consistent and predictable nature of this type of an oil supply device at these lower speeds.

For the sleeve bearing test data to be of value for comparison to the pad bearing, the oil flow to the sleeve bearing (from the disk) must at least meet the minimum requirements for hydrodynamic lubrication. Fuller (1) gives data on oil feed requirements for journal bearings, and comparison shows that the oil flow supplied by the disk is more than sufficient, even at turning gear speeds as low as \( \sqrt{0.1} \) rpm. Hydrodynamic operation of the bearing was not obtained at the turning gear speeds. However, this was due to the low Sommerfeld value and not, then, due to oil starvation.

The difference in L/D ratios of the two bearings is small and this has a minor effect on film thickness, particularly at the low Sommerfeld numbers where transition from hydrodynamic operation occurs (2). Figure 8 is a comparison of theoretical eccentricity ratios for a two-groove sleeve bearing and a five-pad (load between pads) journal bearing, both with 1:1 L/D ratios, which was developed from data in (3). It should be clarified that this refers to pad eccentricity ratio, from which the minimum film at each of the two loaded pads can be determined. The point of interest here is that the sleeve bearing will have a larger film thickness at higher Sommerfeld numbers, but the pad bearing has the advantage in this respect at lower numbers. For a sleeve bearing with an L/D ratio of 0.85 rather than 1.0, as in Fig. 8, the crossover point of the two plots would be slightly to the right.

![Fig. 8—Comparison of eccentricity ratios for two types of bearings tested.](image)

The advantage of the pad bearing over the sleeve bearing at the lower Sommerfeld numbers is a direct result of the bearing load being carried in two locations (two areas of minimum film), rather than one. At higher Sommerfeld numbers, the sleeve bearing, with one arc of approximately 150 degrees, is more effective in generating a load carrying hydrodynamic film than are the two 60 degree arc pads (each carrying 0.618 of the total load). However, at the low Sommerfeld numbers the effective portion of the 150 degree arc of the sleeve bearing is reduced (as the bearing attitude angle approaches zero), and the two 60 degree supporting arcs become more effective.

Figure 9 is a plot of data from the sleeve bearing tests at higher Sommerfeld numbers, as compared to the theoretical curve, and is included primarily to indicate the general agreement obtained using the test equipment described here.
The test data obtained from the sleeve bearing at the low Sommerfeld numbers in the area of transition from hydrodynamic to boundary operation are plotted in Figs. 10, 11, and 12. This data is separated into three plots primarily for clarity. It should also be pointed out that the instrumentation and setup used at these low shaft speeds did not permit the coefficient of friction values to be determined with a high degree of accuracy. However, the Sommerfeld numbers are accurately determined, and the change in the friction coefficient is significant in going from hydrodynamic to boundary conditions, so the information of primary interest is clearly obtained.

Similar data from the pad bearing is shown in Fig. 13. A general comparison of this data with that from the sleeve bearing can be made by reference to Figs. 10, 11, and 12, and it can be seen that the transition occurs at lower S values with the pad bearing. This comparison is more clearly made in Fig. 14.
In addition to these tests, the pad bearing was subjected to a series of high-loading, low (turning gear) shaft speeds. Loadings to 1,000 psi and shaft speeds from 0.012 to 0.2 rpm were used. A set of conditions was established and the test then run for a period of several days. The bearing was inspected after each such run. No differences in bearing operation were noted as a result of speed changes within the range tested. The effect of loading was determined simply by visual inspection and the results are thus necessarily subjective. However, the following general conclusions were derived.

1. Above 750 psi, definite movement of the babbitt metal (ripping) was observed.
2. Up to 300 psi, only light polishing of the babbitt developed.

This obviously leaves a substantial range between these two figures for which the results are not well defined. These test results generally confirm previously published information in this respect (4).

DISCUSSION

Reference (5) discusses the low-speed limits of hydrodynamic lubrication, using the criteria of 10 times the surface finish as the value of the calculated minimum film thickness at which this limit is reached. For oil-lubricated babbitt bearings and a steel shaft, it is pointed out that the shaft finish is the limiting value as the babbitt polishes during initial operation. Using the Fig. 2 plot of this reference and the above criteria applied to the test sleeve bearing, a check of this prediction was made. Considering the maximum surface finish of the test shaft of 22 micro-inches (as measured), this plot indicates that the lower limit of hydrodynamic lubrication will be at a Sommerfeld number of about 0.005 to 0.006. This agrees well with the test data presented here for the 13 inch sleeve bearing.

General agreement is also obtained with Fig. 4 of reference (6). This is a plot of the data of Burwell (7) of Sommerfeld number at the minimum coefficient of friction vs journal roughness.

Theoretical and experimental work (8) has been conducted with particular emphasis on bearings for marine main propulsion shafting, specifically the stern tube (aft most) bearing. With fixed-type bearings at this location, shaft to bearing misalignment is inevitable due to the large overhung weight of the propeller and resultant shaft deflection. This particular work emphasized the effect of such misalignment on low speed operation. As can be anticipated, the transition area is shifted to significantly higher Sommerfeld numbers as the misalignment increases. General agreement is also found with the data reported in (8) for aligned conditions, and the test results given here. This comparison also emphasizes the advantage of using a self-aligning type bearing in applications where some misalignment is difficult or impossible to avoid, as marine propeller shaft bearings.

SUMMARY

Data extracted from tests on journal bearings designed for marine main propulsion shafting has been presented to indicate the operation of the bearings at low Sommerfeld numbers. In particular, the conditions for transition from hydrodynamic to boundary lubrication were shown by the plots of test data. Good agreement with previously published data in this area was found. The advantage of a particular pad bearing arrangement as compared to a conventional sleeve bearing was shown.

REFERENCES


DISCUSSION

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The significant emphasis on experimental work is very welcome in Mr. Gardner's paper on low speed bearing operation. His comparison of performance between the sleeve bearing and tilting pad bearing was, however, carried out using dimensionless groups and this may tend to mask the reader's appreciation of the true situation in the absence of further knowledge of actual operating conditions and clearances. Many variables make up the Sommerfeld number and a higher value of this does not automatically indicate a higher speed condition.

To obtain a "feel" for the situation, performance
predictions have been made of a tilting pad bearing and a sleeve bearing, both 13 inches in diameter and 13 inches long. The oil inlet temperature at various speeds, assumed in this study, is given in Fig. A1 and the corresponding inlet viscosity is shown for a heavy medium oil. At very low speed, the bearing operating viscosity will be very near to oil inlet conditions and the Sommerfeld number will be dependent on this viscosity together with the speed. Mr. Gardner has indicated that the limit to hydrodynamic lubrication for his particular bearings may in theory occur at an oil film thickness of 0.00022 inches* and this is the limit used herein. At typical bearing clearances this limit is reached at a few revolutions per minute.

In the tilting pad bearing (loaded between pads), the pad bore clearance C is the controlling factor at low speed. The assembled clearance C′ (say) only becomes a significant factor when operating at much higher speeds than considered here. The effect of pad bore clearance on performance is shown in Fig. A2. It is of interest to note that Sommerfeld numbers cannot be used in this case to give an order of merit, because when the pad bore clearance is reduced, the limiting film thickness occurs at a higher Sommerfeld number but at a lower speed. This is because the Sommerfeld number contains a clearance term.

In the comparison of the two types of bearings, given in Fig. A3, a nominal clearance ratio of 0.001 has been used for the sleeve bearing, and the same value (0.001) for the assembled clearance ratio \(C'/D\) for the tilting pad bearing but with a preload (or preset) of 0.4, (i.e., pad bore clearance ratio \(C/D\) equal to 0.00167). The top graph presents results in a dimensionless form of \(f(C/D)\) against \(B \times (\frac{D}{C})^2\), while the bottom graph presents results in absolute values of power loss against speed. The film thickness limit \(P_{min}\) equals 0.00022 inches occurs at a lower Sommerfeld number for the tilting pad bearing than the sleeve bearing, but at the same speed, 1 revolutions per minute in both cases.

It could be argued for the case considered that, if the pad bore radius were reduced, the tilting pad bearing would be superior (likelihood of becoming

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hydrodynamic at a lower speed than for the sleeve bearing) and if the pad bore radius were increased, the sleeve bearing would be superior. In either case, the tilting pad bearing would be associated with a lower Sommerfeld number than that of the sleeve bearing at the limiting film thickness condition. This emphasizes that the use of dimensionless terms may be misleading for comparing absolute values. It is of interest to note that abrupt changes in friction shown in Fig. 14, occurring at Sommerfeld numbers of 0.0016 for the tilting pad bearing, are of the same order as the limiting values in Fig. A3, but this may be just fortuitous.

In order to assess Mr. Gardner’s results in absolute terms, it would be helpful to have further information. In particular:

1) What were the speeds and viscosities at which the abrupt changes in friction occurred, especially those in Fig. 14?
2) What were the clearances in the bearings tested and the preload of the tilting pad?
3) How did oil inlet temperature vary with speed and was this the same for both tilting pad and sleeve bearing?
4) What temperatures were used to determine viscosity and where were these temperatures measured?

The author has stated that the conditions at which the sharp increase in friction torque occurs is of specific interest. Answers to the above questions would help to define these conditions more precisely.

**AUTHOR’S CLOSURE**

The author appreciates the review of this paper by Mr. Martin and Mr. Jones. In response to the specific questions asked he would like to offer the following information.

The temperatures used to determine the viscosity values for calculation of Sommerfeld numbers were those indicated by the thermocouples in the bearing babbitt. As noted in the paper, at the low surface velocities in these tests all thermocouples, those in the bearing babbitt and in the oil system, read essentially the same and varied primarily with ambient room temperature. Thus, the problem of selecting the representative temperature was much simplified. The oil inlet temperature was not controlled to any specific value but again was essentially room temperature. These conditions were true for tests on both bearing types.

The sleeve bearing diametral clearance was 0.0135 inches and the pad bearing had a pad bore diametral clearance of 0.0175 inches. The preload on this bearing was 9.4. The difference in these bearing clearances does affect the values of the Sommerfeld numbers, as pointed out by this discussion, and this is reflected in the data in the paper.

To clarify this point, a look at a specific case will be helpful. For the 200 psi conditions plotted in Fig. 14, the corresponding speeds and viscosities at the point of abrupt change in the friction coefficient are as follows:

<table>
<thead>
<tr>
<th>Bearing Type</th>
<th>Shaft rpm</th>
<th>Oil Viscosity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pad Bearing</td>
<td>1.3</td>
<td>28 microreyns</td>
</tr>
<tr>
<td>Sleeve Bearing</td>
<td>2.5</td>
<td>27 microreyns</td>
</tr>
</tbody>
</table>

Thus, the advantage of the pad bearing is in the order of 2:1, in terms of rpm, for these particular conditions. If a pad bore clearance equal to the sleeve bearing clearance had been used, the advantage of the pad bearing would presumably be even greater.