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Research on Discontinuity of Lubricant Film in Journal Bearings*

(1st Report, Bearing Characteristics and Lubricant Film Rupture)

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There have been presented various boundary conditions for the trailing edge of the lubricant film of journal bearing in the hydrodynamic lubrication theory. They are, however, not so clearly investigated physically in connection with the lubricant film rupture.

In this paper, eccentricity ratio and attitude angle are measured, and, at the same time, the generation and growth of the lubricant film rupture are observed by using a transparent journal bearing bushing. The results are investigated with theoretical solutions under various boundary conditions.

The shaft center is found to travel along Sommerfeld's theoretical locus in the case of a continuous lubricant film. Then, with occurrence of the lubricant film rupture, the shaft jumps rapidly to the theoretical locus defined with Reynolds' or Gumbel's boundary condition.

The theoretical pressure distribution shows that the pressure becomes almost atmospheric in the discontinuous film region.

From these results, the conditions of the lubricant film rupture are investigated theoretically.

1. Introduction

In the theoretical approach to fluid lubrication problems of a journal bearing, the solution of Reynolds equation may be much affected by the boundary condition adopted for the pressure distribution at the trailing edge of the lubricant film, yielding particular bearing characteristics.

For this problem, several boundary conditions have been presented, but they are mostly for the practical convenience without considering the relation between the film rupture at the trailing edge and the pressure distribution profile. The mechanism of the film rupture in the journal bearing, however, might govern the trailing boundary condition, so that this should be necessarily analyzed from the theo-

Fig. 1 Bearing characteristics for various boundary conditions

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retical and experimental standpoints.

For the observation of the lubricant film behavior at the trailing edge, some researches using transparent full journal bearing bushings have been reported⁽¹⁾, but they did not concern the relation between the film rupture and the bearing characteristics such as the locus of shaft center.

In this study, transparent journal bearing bushings are used so that the film rupture may be observed directly, and, at the same time, eccentricity, attitude angle, load capacity, and pressure distribution may be measured. The experimental results are investigated comparing with theoretical solutions under various boundary conditions.

Hereinafter, the film rupture is investigated by considering the cavitation and reverse flow of the fluid in the journal bearing.

2. Boundary conditions

The basic equation for the theory of the hydrodynamically lubricated journal bearing, that is Reynolds equation, can be written as follows:

$$\frac{1}{r^2} \frac{\partial}{\partial \theta} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial \theta} \right) + \frac{\partial}{\partial z} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = \frac{6U}{r} \frac{\partial h}{\partial \theta} \quad \dots\dots\dots(1)$$

where

θ : angular coordinate measured from maximum film thickness point in the direction of shaft rotation

z : axial coordinate measured from bearing center

h : lubricant film thickness

p : lubricant film pressure

r : radius of shaft

U : surface velocity

μ : coefficient of viscosity

For journal bearings, there are two conventional methods to solve Eq. (1). One is to neglect the second term of the left hand side of Eq. (1) by ignoring the axial flow comparing with the circumferential flow under the approximation of an infinitely long bearing. The other, contrary to the former, neglects the first term of Eq. (1) under the approximation of an infinitely short bearing.

Fig. 1 shows the extensions of the lubricant film, the pressure profiles and the loci of shaft center under various known boundary conditions, respectively.

Sommerfeld⁽²⁾ assumed that the lubricant film was continuous over whole bearing clearance, and integrated Eq. (1) with boundary conditions;

$$p=0 \text{ at } \theta=0 \text{ and } 2\pi$$

Hereby, the pressure profile becomes symmetric about $\theta=\pi$, namely, the region of negative

pressures is identical with that of positive pressures, in consequence, the displacement of the shaft is always at right angles to the load direction, as shown in Fig. 1 (I). His solution has been often considered impractical and has not been so often adopted for the bearing design theory, because, in

Table 1 Dimensions of test journals and test bearings

Test journals	a	b	c	d
Outer diameter <i>d</i> mm	29.802	29.852	29.904	29.960
Test bearings	A	B	C	
Inner diameter <i>D</i> mm	29.970	30.036	30.030	
Width <i>L</i> mm	60.	50.	30.	
Width/Diam. ratio <i>L/D</i>	2.0	1.67	1.0

(a) Schematic view of apparatus for shaft center locus

(b) Developed view of bearing for measurement of pressure distribution

Fig. 2 Experimental apparatus

the practical bearing, the negative pressure for $\pi \leq \theta \leq 2\pi$ which has the same magnitude as the positive pressure can not possibly be maintained and the locus of shaft center will never be a straight horizontal line.

Gümbel⁽³⁾ assumed that the lubricant film existed only for $0 \leq \theta \leq \pi$, and the film pressure was zero for $\pi \leq \theta \leq 2\pi$, where a cavitation occurred. Thus, the boundary conditions imposed were;

$$p=0 \text{ at } \theta=0 \text{ and } \pi, \text{ and for } \pi \leq \theta \leq 2\pi$$

Under these conditions, the pressure profile is identical with the positive pressure wave given by Sommerfeld but the locus of shaft center becomes like a semi-circle as observed actually, which is shown in Fig. 1 (II). However, the gradient of the pressure distribution is discontinuous at $\theta=\pi$ which is assumed as the film end. Therefore, the continuity of flow rate is not satisfied at this point.

Reynolds' boundary condition⁽⁴⁾ assumes no negative pressure and its requirement is that the pressure gradient also become zero at $\theta=\pi+\alpha$.

[Fig. 1 (III)]

Namely,

$$p=0 \text{ at } \theta=0 \text{ and } \pi+\alpha, \text{ and for } \pi+\alpha \leq \theta \leq 2\pi$$

$$\frac{\partial p}{\partial \theta}=0 \text{ at } \theta=\pi+\alpha$$

Thus, the continuity of flow rate is improved and the defects of Sommerfeld's theory are almost corrected but a theoretical calculation is rather complicated.

On the other hand, in the short bearing approximation by DuBois and Ocvirk⁽⁵⁾ which assumes an infinitely short bearing, they assumed that the pressure in z -direction was zero at both bearing ends, consequently, in θ -direction Sommerfeld's or Gümbel's boundary condition must be taken as a mathematical requirement. They adopted the latter and solved with the boundary conditions that;

$$p=0 \text{ at } z=\pm \frac{L}{2}$$

$$\text{at } \theta=0 \text{ and } \pi, \text{ and for } \pi \leq \theta \leq 2\pi$$

where L is the bearing width. This pressure distribution is parabolic in z -direction and is like Gümbel's one in θ -direction, as shown in Fig. 1 (IV).

For finite bearings, Raimondi and Boyd⁽⁶⁾ have reported the numerical results of Eq. (1) using mainly Reynolds' boundary condition.

3. Experimental apparatus

The schematic view of the experimental apparatus is shown in Fig. 2(I). The shaft can be driven up to 3'000 rpm by a variable-speed electric motor, and by giving different diameters to several test journals, bearing clearances can be taken between 60 and 218 microns. The test bearings are made of transparent Acryl resin which enables one to observe the lubricant film behavior. The load is applied up to 13kg, through the rings fixed at bearing ends by a dead weight, as shown in Fig. 2 (I). Dimensions of these test shafts and bearings are listed in Table 1.

As the lubricant, water is used and some fluorescent dye is added to make the observation clearer. By keeping a constant level of the lubricant in a tank, the supply pressure p_f is held at 0.03 kg/cm^2 . The angular position of supply hole θ_{pf} is 225° measuring from the load line in the direction of shaft rotation.

The displacement of bearing is

Bearing B: $L/D=1.67$, $2Cr=190\mu$, $N=1500 \text{ rpm}$, $W=6 \text{ kg}$
Fig. 3 Transitional profile of film rupture

measured by two dialgages fixed to the bed, whose apexes contact horizontally and vertically with the outside of the bearing at the center.

Thereafter, the film pressure is measured by observing the aspect of the film rupture at the same time. The test bearings used in this experiment have 41 holes of 0.5mm diameter to measure the film pressure, which are arranged as shown in Fig. 2 (II).

4. Experimental results

4.1 Qualitative investigation of the film rupture

The aspects of the film rupture were observed under an increasing bearing load at a constant rotating speed of shaft.

Under light loads, consequently at small eccentricities, the lubricant film extends over whole bearing clearance, but when increasing the loads, it

ruptures unstably on the unloaded side of the bearing under a certain load. The aspects of this developing film rupture in the diversing side are shown in Figs. 3(I) to 3(VI). In these figures, the region of film rupture is seen as dark area, and the shaft rotates from down-side to up-side.

Fig. 3(I) shows the aspect of film rupture just when the load reaches 6kg. At beginning it is so unstable that the cavities change their forms in various ways by combining together and separating themselves. Sometimes, it was clearly recognized that the ambient air was introduced into cavities as bubbles from both bearing ends when the lubricant film just broke. This phenomenon was not observed when both bearing ends were dipped in lubricant, so that the air seemed to be introduced into the bearing clearance and to fill cavities only when the bearing was surrounded by atmospheric air.

The aspects of film rupture in this unstable state are shown in Fig. 3(II) to 3(V). This unstable state disappears in a very short time and the aspect of film rupture becomes stable, as shown in Fig. 3(VI), with the discontinuous region of the film rupture extending in axial and circumferential directions.

The air bubbles in the bearing clearance travel against the shaft rotation and toward the bearing center region where a reverse flow may take place according to the hydrodynamic theory. As mentioned later, this reverse flow of lubricant which has great

Fig. 4 Eccentricity ratio vs. Sommerfeld number

Fig. 5 Attitude angle vs. Sommerfeld number

influence on the introduction of the air will be experimentally proved.

A violent vibration of the bearing was observed before the film began to rupture. Idehara and others⁽⁷⁾ have reported that the journal can not be stable when the film extent is complete. Therefore, it seems that this vibration is attributed to the transition from the unstable equilibrium in the continuous film state to the stable one in the state where the film ruptures, and that the dynamic instability of the bearing caused by some external disturbances leads to a film rupture.

(I) Bearing A $L/D=2$, $2Cr=117, 170\mu$

(II) Bearing B $L/D=1.67$, $2Cr=82, 134, 190\mu$

So the authors made it possible to keep the continuous film state until considerably large eccentricities by experimenting carefully and statically.

4.2 Locus of shaft center and film rupture

Experimental relationships between eccentricity ratio and Sommerfeld number are shown in Figs. 4 (I) to 4 (III) for a comparison with theoretical ones. In these figures, designations, such as, DuBois and Ocvirk, Sommerfeld, Gumbel and Reynolds ($L/D=\infty$) indicate respectively theoretical results by their boundary conditions, as mentioned previously, and Reynolds($L/D=1$) means the numerical solution⁽⁶⁾ using Reynolds' condition for the bearing of $L/D=1$.

Generally, the experimental results fairly well agree with Sommerfeld's, Gumbel's and Reynolds' ($L/D=\infty$) curves. In Fig. 4 (I), where $L/D=2$, they agree well with Reynolds' ($L/D=\infty$) curve. When L/D ratio decreases, Sommerfeld number increases a little at the same eccentricity ratio, as shown in Figs. 4 (II) and 4 (III), so that the experimental results shift to Reynolds' ($L/D=1$) case. Yet they rather coincide with Reynolds' ($L/D=\infty$) curve than with Reynolds' ($L/D=1$).

DuBois and Ocvirk's curve has qualitative deviation from the experimental results, and, quantitatively, it agrees with them for small eccentricity ratios and large L/D ratios.

Experimental relationships between attitude angle and Sommerfeld number for $L/D=2$ are shown in Fig. 5. In this figure, the solid lines connecting experimental points represent the film rupture process from the completely continuous film state to the final state of regular and stable rupture after passing through the transitional state with irregular, unstable and local film rupture.

It is clearly observed that attitude angle is

(III) Bearing C $L/D=1$, $2Cr=76, 130\mu$

Fig. 6 Locus of shaft center

near 90° , that is, the shaft moves according to the theoretical locus given by Sommerfeld's condition at the continuous film state. In the beginning of film rupture, attitude angle decreases rapidly and the shaft shifts discontinuously, as shown in Fig. 5, to the theoretical loci of which the film rupture is considered. This is remarkable at larger L/D ratios.

Experimental results of shaft center locus are shown in Figs. 6 (I) to 6 (III) with theoretical curves. In them, the theoretical locus given by DuBois and Ocvirk is neglected for it coincides with Reynolds' ($L/D=1$) curve.

In Fig. 6 (I) for $L/D=2$, at comparatively

Fig. 7 Film rupture angle vs. Sommerfeld number

Fig. 8 Film rupture angle vs. Eccentricity ratio

light load, the film is completely continuous and the displacing direction of the shaft is 90° to the loading direction as indicated by Sommerfeld's theory. Further increasing the load, the film begins to rupture and after transitional state it turns into the final discontinuous film state. At that time, the shaft center shifts remarkably, which depends mainly on the decrease of attitude angle, as mentioned previously. This is also shown as the solid straight lines connecting plotted points for the film rupture generation. Thus, in the finally ruptured film state, the shaft center falls on the semi-circular locus as given by Reynolds' ($L/D=\infty$) and Reynolds' ($L/D=1$) results.

Decreasing the load from the ruptured film state, the shaft moves back to the bearing center along Reynolds' ($L/D=\infty$) and Reynolds' ($L/D=1$) curves. At this load, the film has begun to rupture, but, the film rupture does not recover. It is at the lower load ($\varepsilon=0.2\sim0.3$) that the film becomes completely continuous again. Therefore, the locus of shaft center shows a kind of hysteresis.

Fig. 9 Pressure distribution and profile of film rupture

This durability of the film rupture seems to be one of the reasons why the rapid shift of shaft center has not been observed evidently.

For the case of $L/D=1.67$ shown in Fig. 6 (II), the experimental results have the same qualitative tendency as that for $L/D=2$. However, the film begins to rupture in smaller eccentricity ratios, and the shift of shaft center, especially the decrease of attitude angle, in the beginning of the film rupture is smaller so that the shaft center locus lies in the larger attitude angle area. The hysteresis is not so notable.

The experimental points scatter for the case of $L/D=1$ but have the same tendencies, as shown in Fig. 6 (III).

From these three figures, it is recognized that as L/D ratio decreases, the film begins to rupture in smaller eccentricity ratios, hence, the hysteresis loop becomes smaller.

Examples of experimental relationships of the film rupture angle θ_{p0} versus Sommerfeld number and eccentricity ratio are shown in Figs. 7 and 8, respectively. There the film rupture angle θ_{p0} is defined as the angle measured in the direction of shaft rotation from load line to the point where the lubricant film ruptures.

A qualitative agreement between experimental results and Reynolds' ($L/D=\infty$) curve is generally established, but the experimental ones have large values, which means that the lubricant film extends more widely than theoretically predicted. It is noted that no smaller rupture angle than Gumbel's prediction has been observed.

4.3 Film pressure distribution and film rupture

Some film pressure distributions are shown in connection with the profiles of the film rupture in Figs. 9 (I) to 9 (IV). The pressure distributions are shown on a developing surface, with isobaric curves in cmHg unit and the profiles of the film rupture are also shown as dotted curves. The angle of coordinate is measured from load line in the direction of shaft rotation.

When the film is completely continuous, a wide negative pressure region is observed after the minimum film thickness point, as shown in Fig. 9 (I). As the film begins to rupture, negative pressure becomes smaller. In this case, the lubricant film ruptures locally, unstably and irregularly in portions of zero or low positive pressure regions. Increasing the load further, the aspect of film rupture becomes stable and regular, and extends in almost all zero pressure regions as shown in Figs. 9 (II), (III) and (IV).

So the pressure in the film rupture region may

be considered to be atmospheric. The film rupture region exists stably about the bearing center and then it is not observed that the air is introduced into the bearing clearance in axial direction by crossing over the streamline of lubricant.

5. Critical conditions for film rupture

When the film is completely continuous, the experimental results agree with Sommerfeld's theory, that is, the pressure distributions are symmetrical and the negative pressure seems to be of the same order as the positive pressure. Hence the cavitation may occur accompanied with a film rupture when the minimum film pressure decreases below the saturated vaporization pressure of lubricant. On the other hand, it is experimentally recognized that under the theoretical reverse flow condition, the ambient air flows into the bearing clearance from both bearing ends and may cause a film rupture.

In the following, these two factors of film rupture, which are the cavitation occurrence and the reverse flow introducing air bubbles, will be investigated theoretically.

5.1 Limiting condition for cavitation

As mentioned previously, when the minimum film pressure $p_{\min}$ becomes equal to the saturated vaporization pressure of lubricant p_s , cavitation should occur. This can be considered to be one limiting condition for the generation of film rupture.

By integrating Reynolds Eq. (1) assuming an infinitely long bearing approximation neglecting the second term of Eq. (1) and using Sommerfeld's boundary condition;

$$p = p_a \text{ at } \theta = 0 \text{ and } 2\pi \dots \dots \dots (2)$$

where p_a : ambient pressure

one obtains the pressure distribution p as

$$p = 12\mu N \left(\frac{r}{c_r} \right)^2 \frac{\pi \varepsilon (2 + \varepsilon \cos \theta) \sin \theta}{(2 + \varepsilon^2)(1 + \varepsilon \cos \theta)^2} + p_a \dots \dots (3)$$

where N : rotating number of shaft

Then the load capacity per unit width W is given by⁽⁸⁾

$$W = 24\mu N r \left(\frac{r}{c_r} \right)^2 \frac{\pi^2 \varepsilon}{(2 + \varepsilon^2)\sqrt{1 - \varepsilon^2}} \dots \dots \dots (4)$$

so that the mean pressure p_m is

$$p_m = \frac{W}{2r} = 12\mu N \left(\frac{r}{c_r} \right)^2 \frac{\pi^2 \varepsilon}{(2 + \varepsilon^2)\sqrt{1 - \varepsilon^2}} \dots \dots \dots (5)$$

and the minimum film pressure $p_{\min}$ is given by

$$p_{\min} = p_a - 3\mu N \left(\frac{r}{c_r} \right)^2 \frac{\pi \varepsilon}{(2 + \varepsilon^2)} \left(\frac{4 - \varepsilon^2}{1 - \varepsilon^2} \right)^{\frac{3}{2}} \dots \dots \dots (6)$$

By considering the case of $\varepsilon = \text{const.}$ in Eqs. (5) and (6), it is easily recognized that

$$p_{\max} - p_a = |p_{\min} - p_a| \alpha p_m \dots \dots \dots (7)$$

At the critical state where the film begins to

rupture, the minimum film pressure $p_{\min}$ is considered to be the saturated vaporization pressure of lubricant p_s , namely,

$$p_{\min} = p_s \quad (8)$$

Thus,

$$\frac{p_m}{p_a - p_{\min}} = \frac{p_{\text{merit}}}{p_a - p_s} = \frac{p_{\text{merit}}}{p_a} \left(\frac{p_a}{p_a - p_s} \right)$$

where p_{merit} : mean pressure at critical state

Then,

$$\frac{p_{\text{merit}}}{p_a} = \left(\frac{p_a - p_s}{p_a} \right) \left(\frac{p_m}{p_a - p_{\min}} \right) \quad (9)$$

Now, introducing the new bearing characteristic number S_0 defined by the following equation,

$$S_0 = \frac{\mu N}{p_a} \left(\frac{r}{c_r} \right)^2 \quad (10)$$

The S_0 has a similar expression to Sommerfeld number $S = \frac{\mu N}{p_m} \left(\frac{r}{c_r} \right)^2$ using ambient pressure p_a instead of mean pressure p_m . Thus, S_0 is independent of bearing load and it depends only on design parameter r/c_r and operating condition p_a , μ and N .

Substituting Eq. (10) into Eq. (5)

$$S_0 = \left(\frac{p_m}{p_a} \right) \frac{(2 + \varepsilon^2) \sqrt{1 - \varepsilon^2}}{12\pi^2 \varepsilon}$$

The critical characteristic number $S_{0\text{crit}}$ is related with the eccentricity ratio $\varepsilon_{\text{crit}}$ at the critical state of the film rupture generated by cavitation, then, using Eqs. (5), (6) and (9),

$$S_{0\text{crit}} = \left(\frac{p_a - p_s}{p_a} \right) \frac{2 + \varepsilon_{\text{crit}}^2}{3\pi \varepsilon_{\text{crit}}} \left(\frac{1 - \varepsilon_{\text{crit}}^2}{4 - \varepsilon_{\text{crit}}^2} \right)^{\frac{3}{2}} \quad (11)$$

Thus, the eccentricity ratio at which the lubricant film begins to rupture, $\varepsilon_{\text{crit}}$, depends only on the newly defined characteristic number S_0 expressed by Eq. (10).

5.2 Limiting condition for reverse flow

The circumferential velocity of lubricant u in the bearing clearance is given by⁽⁸⁾

$$u = \frac{U(h-y)}{h} - \frac{y(h-y)}{2\mu r} \frac{dp}{d\theta} \quad (12)$$

where

y : radial coordinate vertical to journal surface

Fig. 10 Bearing characteristic number vs. Eccentricity ratio at film rupture

U : surface velocity of shaft

h : lubricant film thickness

The limiting condition for the beginning of reverse

flow is $\frac{\partial u}{\partial y} \Big|_{y=h} = 0$. In this case, it is clearly known

that reverse flow occurs at $\theta=0$ at first because the pressure distribution is symmetric with respect to the line connecting the shaft and bearing centers in the case of a completely continuous film. Then, the limiting condition for the beginning of reverse flow is given by

$$\frac{\partial u}{\partial y} \Big|_{y=h} = 0 \quad (13)$$

This condition is calculated from Eqs. (3) and (12) as,

$$\frac{\partial u}{\partial y} \Big|_{y=h} = -\frac{U}{c_r(1+\varepsilon)} \times \left\{ 1 - 3 \left(1 - \frac{2(1-\varepsilon^2)}{2+\varepsilon^2} \frac{1}{1+\varepsilon} \right) \right\} = 0$$

The critical condition caused by reverse flow is determined only by eccentricity ratio, and this critical eccentricity ratio $\varepsilon_{\text{crit}}$ is

$$\varepsilon_{\text{crit}} = 0.303 \quad (14)$$

For $\varepsilon > 0.303$, the reverse flow must take place theoretically in a certain circumferential region and this seems to contribute to the film rupture by introducing air bubbles.

In this sense, Eq. (14) may possibly give one critical condition for the film rupture.

5.3 Experimental verifications

The experimental eccentricity ratios at which the film ruptures are shown in Fig. 10, with theoretical results given by Eqs. (11) and (14). It was observed that the lubricant film begins to rupture mostly at an angular position of $90^\circ \sim 100^\circ$ measured from the load line in the direction of shaft rotation. Its region extends widely in a moment, just after the film ruptures.

It should be noted that for the range in which S_0 is smaller than 8.3×10^{-2} , at which the curves given by Eqs. (11) and (14) intersect each other, experimental results fall within the domain enclosed

by these two curves with only a few exceptions. The film never ruptures if ε is smaller than 0.303, which gives the critical condition for reverse flow. Thus, the theoretical results given by Eq. (14) may be considered the minimum eccentricity ratio where the film may possibly rupture for small values of S_0 . Meanwhile, it is clearly observed that the film can not be completely continuous for larger eccentricity ratios than that given by Eq. (11), which represents the critical condition for cavi-

tation. Thus, the theoretical results given by Eq. (11) may be considered the maximum eccentricity ratio where the film can not be completely continuous for small S_0 .

It should be also noted that these conditions may be applicable for a finite bearing, though Eq. (11) is derived under an assumption of infinitely long bearing. In this connection, the theoretical value of ϵ_{crit} calculated with short bearing approximation becomes smaller than that of Eq. (11).

Some experiments were undertaken using light oil as lubricant. As shown in Fig. 10, they have the same characteristics.

For the range of $S_0 > 8.3 \times 10^{-2}$, theoretical results become different from the above case and indicate that cavitation always may take place before reverse flow occurs.

In order to increase S_0 , turbine oil was used as lubricant for this range. But contrary to the theoretical prediction, Eq. (11), which gives the critical condition for cavitation, can not be recognized as a limiting condition for the film rupture but experimental film rupture limit seems to depend upon the occurrence of reverse flow. For this problem, further investigations may be undertaken in the future.

6. Conclusions

In this paper, the lubricant film rupture in journal bearings is investigated theoretically and experimentally.

The results obtained are as follows;

(i) A comparison of the experimental results with the theoretical solutions of Reynolds' fundamental equation for the so far proposed boundary conditions, shows that in the state of continuous film the shaft center moves in accordance with Sommerfeld's theory. When the film begins to rupture due to the extreme decrease of attitude angle, the shaft shifts discontinuously to the theoretical loci given by the theories based upon the boundary conditions of Reynolds or Gümbel. As the load is decreased gradually after the rupture of the film,

the shaft center begins to move back to the bearing center following the above loci. However, once the film ruptures, it does not recover easily. Thus the shaft locus shows a kind of hysteresis.

(ii) The experimental results for eccentricity ratio, attitude angle and film rupture angle show a fairly good agreement with the theoretical solutions based upon the boundary conditions of Reynolds for an infinitely long bearing, but show considerable quantitative discrepancies for a finite bearing. Also the theory of DuBois and Ocvirk of the short bearing approximation shows qualitative discrepancies.

(iii) The results of the measurement of film pressure show that the pressure in the region of film rupture is equal to the ambient pressure and that there is present a low negative pressure in the area after the minimum film thickness point.

(iv) The phenomenon of film rupture is influenced by two conditions—the cavitation due to low pressure and the reverse flow; and the condition of the former changes in accordance with the newly introduced bearing characteristic number S_0 . For small values of S_0 , the critical condition for cavitation becomes the upper limit of eccentricity ratio of the film rupture, and that for the reverse flow does the lower limit. More investigations are necessary for the case of larger values of S_0 , but it may be concluded that the reverse flow evidently aggravates the film rupture.

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