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Oil Flow in Short-Bearing with Circumferential Groove*

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A study was carried out on the oil flow in a short-bearing with a circumferential oil groove, in consideration of oil film disintegration and restoration on the unloaded side.

(1) The ratio between dimensionless oil flow coefficient of the circumferentially grooved bearing (q_T) and the hydrodynamic flow coefficient obtained by the short-bearing approximation ($q_H=2n$) was found to be a function of the oil feed capacity number (C_{npf}) only so far as the present experiment is concerned ($C_n=2\times 10^{-3}\sim 10^{-1}$, $C_{npf}=5\times 10^{-2}\sim 2$). Results of the experiment coincided well with the theoretical values.

(2) $q_T/2n$ value of the circumferentially grooved bearing converged to zero as C_{npf} increased to a large value, while, in bearings with an axial groove or an oil hole, $q_T/2n$ converged to finite values.

(3) In the case of a circumferentially grooved bearing, unlike the other types of bearings, the oil flow could not be divided into two components resulting from the oil-feed pressure and the hydrodynamic film pressure, because the hydrodynamic component itself was affected by the oil-feed pressure.

1. Introduction

Circumferentially grooved bearings are frequently used in internal combustion engines as main bearings or crank-pin bearings, because these bearings demand oil supply through crank shafts. However, research on the oil flow characteristics of these bearings has not been carried out thoroughly as yet.

Shaw and Macks⁽¹⁾ have proposed a theoretical equation based upon the assumption that the whole surface of the bearing is filled with oil film. Although this equation involves the oil-feed pressure and the eccentricity ratio, oil film disintegration in the clearance space on the unloaded side is not taken into consideration. McKee and Washington⁽²⁾ have separated the oil flow into two components by experimental procedures, one of which is an amount related to the oil-feed pressure and the other is a quantity related to the shaft rotation at the eccentric position in the bearing. They have stated, from their experiment, that the oil flow rate of the circumferentially grooved bearings is more dependent upon the oil-feed pressure than that of axially grooved bearings and bearings having one oil hole. As a tendency, this idea would be correct. However, separation of oil flow into two components, one of which is an amount

related to the oil-feed pressure, namely the film flow (FF), and the other a quantity related to the shaft rotation, namely the hydrodynamic flow (HDF), would be applicable only when the HDF component is independent of the oil-feed pressure. Therefore, as will be shown later, applicability of this idea to bearings with a circumferential groove in which the HDF component is affected by the oil-feed pressure is doubtful, though, in the case of axially grooved bearings, this idea would be applicable, because oil is assumed to fill the clearance space completely at the maximum oil film thickness point. Wilcock and Booser⁽³⁾ have combined Shaw's equation with the HDF quantity derived from the short-bearing approximation by DuBois⁽⁴⁾ et al. However, this method is also questionable for the same reason as mentioned above.

The writers compared the oil flow characteristics of a circumferentially grooved bearing with those of an axially grooved bearing in their experiment, and observed that, in the case of the circumferentially grooved bearing, the oil flow rate decreased so much with a decrease of the feed pressure that even the HDF component could not be attained. In the case of the axially grooved bearing, no such a trend was recognized. The writers have reached the conclusion that, in the case of circumferentially grooved bearings, special attention should be paid to disintegration and restoration of oil film on the surface of the unloaded side of the bearings.

As to the behavior of oil film on the surface of

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the unloaded side, observation by a transparent bearing test rig has been reported by Cole and Hughes⁽⁵⁾, and the shape of cavity of the bearings with the circumferential groove has also been reported by Ishii⁽⁶⁾. However, there is no report on the characteristics of oil flow of circumferentially grooved bearings with the oil film disintegration and restoration on the unloaded side surface of the bearings taken into account. The writers, therefore, have made comparison of the oil flow characteristics of short circumferentially grooved bearings between the value obtained by theoretical treatment, with the oil film behavior on the unloaded side surface taken into account, and the value obtained by experimental procedures. As a result, the writers have reached the conclusion that the theoretical values sufficiently coincide with the experimental values and that the HDF itself is affected by the oil-feed pressure.

2. Nomenclature

R :	radius of shaft	cm
D :	diameter of shaft	cm
c :	radial clearance	cm
C :	diametrical clearance	cm
L :	width of each bearing part equally divided by circumferential groove	cm
n :	eccentricity ratio= e/c ,	
e :	eccentricity	cm
U :	sliding velocity	cm/sec
N :	shaft speed	rps
P :	load per unit projected area	kg/cm ²
p_f :	oil-feed pressure	kg/cm ²
p :	oil film pressure	kg/cm ²
μ :	oil film viscosity	kg sec/cm ²
Q_T :	total volume flow rate	cc/sec
Q_H :	hydrodynamic flow rate	cc/sec
q_T :	dimensionless total oil flow coefficient	
q_H :	dimensionless hydrodynamic flow coefficient	
C_n :	load capacity number= $(L/D)^2(D/C)^2\mu N/P$	
C_{npf} :	oil feed capacity number= $(L/D)^2(D/C)^2\mu N/p_f$	

3. Theoretical analysis

Referring to Fig. 1, assume a half of the bearing surface to be divided by the circumferential groove. The origin of the coordinates is determined on the border line between the surface and the groove and at the position of the maximum oil film thickness. The x -axis is determined in the sliding direction of the shaft and the z -axis in the direction of the bearing width. Here, the region denoted by the relation of $\theta_2 \leq \theta \leq \theta_1$ where oil film fills the whole bearing width of L is defined as the pressure region, and the region $\theta_1 \leq \theta \leq \theta_2$ as the cavitation region, where θ means the angle measured from the maximum oil film thickness

point to the direction of shaft sliding. Since a circumferentially grooved bearing is practically divided by the oil groove into two bearings and each is an extremely short bearing whose L/D ratio is usually smaller than 0.5, the short-bearing approximation is applicable reasonably. The oil film pressure gradient in the circumferential direction $\partial p / \partial x$ can be neglected compared with that in the axial direction $\partial p / \partial z$, and the Reynolds fundamental equation for an incompressible fluid under static loading is given by the next equation.

$$\frac{\partial}{\partial z} \left(\frac{h^3}{\mu} \frac{\partial p}{\partial z} \right) = 6U \frac{dh}{dx} \quad \dots\dots\dots (1)$$

Assuming that μ is constant, and as h is independent of z , Eq. (1) can be integrated twice by z .

$$p = \frac{6\mu U}{h^3} \frac{dh}{dx} \frac{z^2}{2} + C_1 z + C_2$$

With the relation of $h=c(1+n \cos \theta)$ and $dh/dx = -(c/R)n \sin \theta$, the oil film pressure is shown by the next equation.

$$p = -\frac{6\mu U}{c^3(1+n \cos \theta)^3} (c/R)n \sin \theta \frac{z^2}{2} + C_1 z + C_2 \quad \dots\dots\dots (2)$$

Introducing the boundary conditions that $p=p_f$ at $z=0$ and $p=0$ at $z=L$, the oil film pressure distribution and its gradient can be obtained by the next equations.

$$p = \frac{3\mu U n}{c^2 R} \frac{\sin \theta}{(1+n \cos \theta)^3} (Lz - z^2) - \left(\frac{p_f}{L} \right) z + p_f \quad \dots\dots\dots (3)$$

$$\frac{dp}{dz} = \frac{3\mu U n}{c^2 R} \frac{\sin \theta}{(1+n \cos \theta)^3} (L - 2z) - \left(\frac{p_f}{L} \right) \quad \dots\dots\dots (4)$$

Since the side flow becomes zero at the position denoted by $\theta = \theta_1$, the next equation can be obtained by putting this boundary condition into Eq. (4).

$$\left. \frac{dp}{dz} \right|_{z=L} = -\frac{3\mu U n \sin \theta_1}{c^2 R (1+n \cos \theta_1)^3} L - \left(\frac{p_f}{L} \right) = 0 \quad \dots\dots\dots (5)$$

Using the relations $D=2R$, $C=2c$ and the shaft speed N (rps), Eq. (5) can be reduced to the next form.

Fig. 1 Development of a bearing with circumferential oil groove

$$\left(\frac{L}{D}\right)^2 \left(\frac{D}{C}\right)^2 \frac{\mu N}{p_f} = \frac{-(1+n \cos \theta_1)^3}{24\pi n \sin \theta_1} \dots\dots\dots (6)$$

The left side of Eq. (6) includes the oil-feed pressure p_f instead of the load per unit projected area P which is involved in the load capacity number C_n for the short bearing approximation by DuBois⁽⁴⁾ et al. Therefore, it may be named the oil-feed capacity number C_{npf} , and then θ_1 is determined as a function of n and C_{npf} only. In the cavitation region, $l(\theta)$ is defined as the width of oil film where the clearance space is completely filled with oil. In the region of $\theta_1 \leq \theta \leq \theta^*$ where $l(\theta)$ decreases with an increase of θ , disintegration of the oil film will occur successively from the end part of the bearing. $l(\theta)$ of this region, therefore, can be called the boundary line of disintegration, so the following suppositions are made as the boundary condition of disintegration.

$$\left. \begin{array}{l} p_{z=l(\theta)} = 0 \\ \frac{dp}{dz} \Big|_{z=l(\theta)} = 0 \end{array} \right\} \dots\dots\dots (7)$$

Although $l(\theta)$ is a function of θ , the short bearing approximation would be applied reasonably to each part of this region. Using similar calculations shown by Eq. (1)~Eq. (6), the next relation can be obtained with the boundary conditions of $p=p_f$ at $z=0$ and $p=0$ at $z=l(\theta)$

$$\left(\frac{l}{D}\right)^2 \left(\frac{D}{C}\right)^2 \frac{\mu N}{p_f} = \frac{-(1+n \cos \theta)^3}{24\pi n \sin \theta} \dots\dots\dots (8)$$

Equation (8) is adjusted by putting the ratio $l(\theta)/L$ as ψ .

$$\psi = \left[\left(\frac{1}{24\pi C_{npf}} \right) \left\{ \frac{-(1+n \cos \theta)^3}{n \sin \theta} \right\} \right]^{1/2} \dots\dots\dots (9)$$

In the first place, $l(\theta)$ decreases with the increase of θ . However, from half way $l(\theta)$ would begin to increase with an increase of the oil film thickness, because the rate of oil flow to the clearance space of the bearing from the groove is proportional to the cube of the clearance. This border θ , denoted by the symbol θ^* , can be determined as a function of the eccentricity ratio n only by differentiating Eq. (9) and putting it as zero. This is because θ^* means the position of $dl(\theta)/d\theta = d\psi/d\theta = 0$ from its definition.

$$3 + \frac{(1+n \cos \theta^*) \cos \theta^*}{n \sin^2 \theta^*} = 0 \dots\dots\dots (10)$$

In the region of $\theta^* \leq \theta \leq \theta_2$, where $l(\theta)$ increases with an increase of θ , $l(\theta)$ can be called the boundary line of restoration. It may be assumed that a lot of oil strips separate from the boundary line of disintegration and each strip flows into place in the restoring line having the same width of the place in the disintegration line at which the separation occurs, as shown by Fig. 2. This assumption seems to be reasonable from the photographic observation carried out by Cole⁽⁵⁾ et al. and this is based upon the idea that

the viscosity of air or gas in the cavitation region is much smaller than that of the oil film and, accordingly, that the pressure gradient both in the circumferential and in the axial direction can be neglected because of the narrow width of the oil strips. Assuming the triangular part on the restoration line is surrounded by dl and dx , the shear flow which goes into this part and goes out from the same part can be expressed by the next equation.

$$\left. \begin{array}{l} Q_{in} = \frac{U}{2} h \Theta dl \\ Q_{out} = \frac{U}{2} h dl \end{array} \right\} \dots\dots\dots (11)$$

Where $\Theta = h(\beta)/h(\theta)$, and β denotes the bearing angle in the disintegration region which shows the same value of $l(\theta)$ in the restoration region. Applying the continuity law, it may be assumed that the difference between Q_{out} and Q_{in} is compensated by the oil quantity of z -direction which flows into the triangular part through dx due to the pressure gradient. Then, as $dl = L d\psi$, from Eq. (11) and Eq. (12) the next relation can be obtained.

$$Q_{out} - Q_{in} = - \frac{Rh(\theta)^3}{12\mu} \frac{dp}{dz} \Big|_{z=l(\theta)} d\theta \dots\dots\dots (12)$$

$$\left\{ 1 - \frac{h(\beta)}{h(\theta)} \right\} d\psi = - \frac{h(\theta)^2}{6\mu U} \frac{R}{L} \frac{dp}{dz} \Big|_{z=l(\theta)} d\theta \dots\dots\dots (13)$$

β can be determined with the same value of ψ as in the case of θ , and therefore, substituting β into Eq. (9) instead of θ , β can be obtained as a function of ψ and C_{npf} . Then, the part inside of { } on the left side of Eq. (13) can be written as shown by the next equation.

$$1 - \frac{h(\beta)}{h(\theta)} = 1 - \frac{(1+n \cos \beta)}{(1+n \cos \theta)} = f_1(\psi, C_{npf}, n, \theta) \dots\dots\dots (14)$$

Substituting the relations $z=l(\theta)$, $U=2\pi RN$ and C_{npf} into Eq. (4), the next relation can be derived.

$$\begin{aligned} - \frac{h(\theta)^2}{6\mu U} \frac{R}{L} \frac{dp}{dz} \Big|_{z=l(\theta)} &= - \frac{1}{2} \left\{ \frac{-n \sin \theta}{1+n \cos \theta} \psi \right. \\ &\quad \left. - \frac{(1+n \cos \theta)^2}{24\pi C_{npf}} \left(\frac{1}{\psi} \right) \right\} = f_2(\psi, C_{npf}, n, \theta) \end{aligned} \dots\dots\dots (15)$$

From Eq. (13) to Eq. (15)

$$\frac{d\psi}{d\theta} = \frac{f_2}{f_1} = f(\psi, C_{npf}, n, \theta) \dots\dots\dots (16)$$

In the ideal case where $p_f=0$ and the clearance space

Fig. 2 Disintegration and restoration of oil film

is filled with oil all over the bearing width at the position of the maximum oil film thickness, n is determined by the next relation as a function of C_n only.

$$C_n = \frac{(1-n^2)^2}{\pi n} \left[\frac{1}{\pi^2(1-n^2)+16n^2} \right]^{1/2} \dots\dots\dots(17)$$

Actually, there might be some error, because p_f is not zero and the clearance space is not filled with oil all over the bearing width at the maximum oil film thickness position. However, since p_f is considerably small compared with P and since the oil film pressure near the position of the maximum oil film thickness is also small, it would be reasonable to regard n as a function of C_n only, where θ_2 is not so large. If so, as C_n is determined by the working condition of the bearing and C_{npf} by the oil feed condition, Eq. (16) can be written as follows when C_n and C_{npf} are given.

$$\frac{d\psi}{d\theta} = g(\psi, \theta) \dots\dots\dots(18)$$

Although there would be some difficulty in solving Eq. (18) directly, it is possible to determine the relation between ψ and θ by the progressive calculation procedure, starting from ψ^* value which is obtainable by introducing θ^* obtained from Eq. (10) into Eq. (9). Then, if θ at which ψ reaches 1 is expressed by $\theta_{\psi=1}$, $\theta_{\psi=1}$ can be shown by the next relation.

$$\theta_{\psi=1} = \varphi(C_n, C_{npf}) \dots\dots\dots(19)$$

This $\theta_{\psi=1}$ is defined as the position where oil film is restored to the whole bearing width and is equal to θ_2 , as defined before. By the procedure stated above, both θ_1 and θ_2 can be found and then the total oil flow quantity through the bearing is shown by Eq. (20).

$$Q_T = -2 \int_{\theta_2}^{\theta_1} \frac{Rh^3}{12\mu} \frac{dp}{dz} \Big|_{z=L} d\theta \dots\dots\dots(20)$$

Equation (20) may be used for the direct computation of the oil flow rate for the circumferentially grooved bearings. It would be more convenient, however, to read just the equation to a form which includes the dimensionless oil flow coefficient. The dimensionless hydrodynamic oil flow coefficient denoted by the symbol q_H is defined as the ratio between the hydrodynamic flow, which is obtainable when the clearance space is filled with oil all over the width

Fig. 3 Comparison of oil flow in a short bearing and in a bearing with circumferential groove

at the maximum oil film thickness point, and the circulating flow in the clearance space when the shaft rotates at the center of the bearing. In the case of a short bearing having a width of L , the next relation comes into existence theoretically.

$$q_H = \frac{Q_H}{\pi DNLC/4} = 2n \dots\dots\dots(21)$$

Equation (21) can be applied to each of the bearing parts divided by the circumferential groove. However, as shown by Fig. 3, the oil quantity returning to the groove from the bearing parts does not act as the oil flow through the bearing. As a result, the oil flow rate through the circumferentially grooved bearing becomes the same value as obtained from the short bearing having a width of L . Although Eq. (20) denotes the total oil flow quantity through the bearing, it would be possible to express Q_T in the same way as Q_H , using the dimensionless oil flow coefficient denoted by the symbol q_T .

$$\begin{aligned} q_T &= \frac{Q_T}{\pi DNLC/4} = n \int_{\theta_2}^{\theta_1} \sin \theta d\theta + \frac{1}{24\pi C_{npf}} \\ &\times \int_{\theta_2}^{\theta_1} (1+n \cos \theta)^3 d\theta = -n(\cos \theta_1 - \cos \theta_2) \\ &+ \frac{1}{24\pi C_{npf}} \left[\frac{n^3}{12} (\sin 3\theta_1 - \sin 3\theta_2) \right. \\ &+ \frac{3}{4} n^2 (\sin 2\theta_1 - \sin 2\theta_2) + \frac{3}{4} n (\sin \theta_1 \\ &\left. - \sin \theta_2)(n^2 + 4) + (\theta_1 - \theta_2) \left(\frac{3}{2} n^2 + 1 \right) \right] \end{aligned} \dots\dots\dots(22)$$

The ratio between q_T and q_H , namely $q_T/2n$, shows the ratio between total oil flow quantity and the idealized hydrodynamic oil flow. Figure 4 shows the result of theoretical calculations with varied bearing dimensions, running conditions and oil feed conditions corresponding to the experiment described later. As a result, the value of $q_T/2n$ is expressed as a function of almost C_{npf} only, and shows a trend of converging to zero when C_{npf} tends to infinity, or, in other words, when the oil feed pressure becomes zero. Strictly speaking, $q_T/2n$ value might be affected by the load capacity number C_n , because q_T value shown by

Fig. 4 Relation of $q_T/2n$ and C_{npf} (Theoretical value)

Eq. (22) includes both the 2nd and the 3rd power terms of n . However, there would be almost no effect of C_n , because n is always smaller than 1 and, accordingly, the effects of higher power terms of n are smaller than that of the 1st power term. In this case, although calculation has been made in a range of $C_n = 2 \times 10^{-3} \sim 10^{-1}$, or, in other words, n value derived from Eq. (17) changes from 0.5 to 0.9, which is close to 1, the effect of C_n on the value of $q_T/2n$ has not been found. Besides this, the effect of the higher power terms of n on the value of $q_T/2n$ becomes smaller when n becomes smaller than that of the above range. It may be said, therefore, that, instead of being controlled by C_n , the value of $q_T/2n$ would be controlled by C_{npf} only. Figure 5 shows an example of the boundary line of disintegration and restoration obtained by calculation. In this example, since θ_2 is 34° , it means that the oil film does not cover all the width of the bearing at the maximum oil film thickness position. Nevertheless, $q_T/2n$ value is 1.208, which is larger than 1. In other words, in spite of the fact that the pressure region of this example is narrower than π of the pressure region of the idealized hydrodynamic flow, the oil flow rate exceeds that of the idealized hydrodynamic flow. This fact is, of course, due to the effect of the oil-feed pressure. It means that this effect acts to raise the oil film pressure gradient at the end of a bearing in the pressure region of $\theta_2 \sim \theta_1$. From this example, it may also be possible to point out the defect of the method of adding merely the hydrodynamic flow to the film flow estimated separately.

4. Experiments and considerations

Experiments were carried out with four kinds of circumferentially grooved bearings with different

Table 1 Dimensions of the test bearings

Bearing number	Bearing diameter D ϕ	Bearing width L mm	Diametrical clearance C mm	C/D	L/D
No. 1	40.034	13	0.048	1.2×10^{-3}	0.325
No. 2	40.066	13	0.048	2.0	0.325
No. 3	40.075	13	0.089	2.2	0.325
No. 4	40.068	8.5	0.082	2.0	0.212
No. 5	40.090	33	0.104	2.6	0.825

Fig. 5 An example of the boundary line of disintegration and restoration (Calculated result)

width~diameter or clearance~diameter ratios, and with a kind of axially grooved bearing for reference. All bearings used had a diameter of 40 mm. Figure 6 shows a diagram of the test rig, Fig. 7 shows the test bearings used and their principal dimensions are shown in Table 1. With these bearings, tests were carried out by changing the running conditions within the range shown in Table 2. The oil film viscosity listed in the table was defined as the average value of eight points corresponding to the temperature measured with copper-constantan thermo-couples embedded equally spaced on the circumference at the middle part of the bearing surface. Figure 8 shows the test results. As had been expected, $q_T/2n$ values agreed well as a function of C_{npf} only, and the experimental values coincided well with the theoretical values. It is observed that the $q_T/2n$ value of the circumferentially grooved bearing has a trend of converging to zero when the C_{npf} value becomes infinity. This is why, in the case of the circumferentially grooved bearing, oil can not be fed from the groove to the bearing surface when the feed pressure becomes zero. Of course, it may not be said that the oil feeding from the groove to the bearing

Table 2 Test ranges

Oil-feed pressure	P_f (kg/cm ²)	0.3~2.0
Load per unit projected area	P (kg/cm ²)	30~40
Shaft speed	N (rpm)	600~2 400
Oil film viscosity	μ (cp)	9~16
Load capacity number	C_n	$2 \times 10^{-3} \sim 10^{-1}$
Oil feed capacity number	C_{npf}	$5 \times 10^{-2} \sim 2$
Lubricating oil	Specific weight 0.883 0 Viscosity ($^{10.6 \text{ CS}+100 \text{ F}}$ $^{2.2 \text{ CS}+210 \text{ F}}$)

Fig. 6 Diagram of the test rig

Fig. 7 Test bearings

Fig. 8 Relation of $q_T/2n$ and C_{npf} (Experimental value)

surface becomes completely zero, because there would be a little negative pressure in the cavitation region. However, as the pressure in the cavitation region of statically loaded bearings is almost equal to the ambient pressure of the bearings, namely the atmospheric pressure, such an oil quantity must be small. On the other hand, in the case of the axially grooved bearing, the trend of converging to a finite value was found, and, in this experiment, this finite value was around 0.5. As the ideal suppositions for the axially grooved bearings, if the oil groove width is equal to the bearing width, and if the position of the groove coincides with that of the maximum oil film thickness, the HDF must be perfectly obtained and, accordingly, it may be said that the $q_T/2n$ value converges to 1. In this experiment, however, the $q_T/2n$ value converged to a value smaller than 1. Two main reasons are conceivable for this phenomenon. One is that the width of oil groove was 80% of the bearing width and the other is that the position of the oil groove did not coincide with the maximum oil film thickness position, because the tests were carried out with the oil groove position set in the opposite direction to the loading. However, the trend that $q_T/2n$ converges to a finite value when C_{npf} increases must be recognized as a character of the axially grooved bearing which is different from the circumferentially grooved bearings. For reference, test results of the axially grooved bearing carried out

by McKee and of the single oil hole bearing by DuBois-Ocviok and by McKee were also compared with this experiment, as shown in Fig. 8. As a result, the dependence on the oil-feed pressure was greater in the order of the circumferential groove, the single oil hole and the axial groove. In the case of the single oil hole bearing, as the ratio between the oil hole diameter and the bearing width was smaller than the ratio between the oil

groove width and the bearing width of the axially grooved bearing, the $q_T/2n$ value corresponding to the same value of C_{npf} showed a middle value between the axially grooved bearing and the circumferentially grooved bearing. Anyway, the trend of the $q_T/2n$ value converging to a finite value with an increase of C_{npf} is similar to that of the axially grooved bearing and is apparently different from that of the circumferentially grooved bearing.

5. Conclusions

The following conclusions can be drawn from the above-mentioned analysis and experiments.

(1) The ratio between the dimensionless oil flow coefficient of the circumferentially grooved bearing (q_T) and the hydrodynamic flow coefficient obtained by the short-bearing approximation ($q_H=2n$) is found as a function of the oil feed capacity number (C_{npf}) only so far as the present experiment is concerned ($C_n=2 \times 10^{-3} \sim 10^{-1}$, $C_{npf}=5 \times 10^{-2} \sim 2$). Results of the experiment coincided well with the theoretical values.

(2) $q_T/2n$ value of the circumferentially grooved bearing converged to zero as C_{npf} increased to a large value, while, in the bearings with an axial groove or an oil hole, $q_T/2n$ converged to finite values.

(3) In the case of a circumferentially grooved bearing, unlike other types of bearings, the oil flow could not be divided into two components resulting from the oil feed pressure and the hydrodynamic film pressure, because the hydrodynamic component itself was affected by the oil feed pressure.

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