Hydrodynamic Lubrication of Journal Bearings by Pseudo-Plastic Lubricants*

(Part 2, Experimental Studies)

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Few experimental investigations on the relation between the nonlinearity of lubricants and the performance of bearings have been published. Theoretical analysis of the relation was presented in the first paper, and experimental analysis of this problem is conducted in this paper.

The initial viscosity and the coefficient of pseudo-plasticity are given by the investigation of the flow rate of pseudo-plastic fluids obtained by adding polyisobutylene to spindle oils. The coefficient of pseudo-plasticity is a scale to indicate the behavior of this lubricant.

Analyzing the experimental results of the performance of bearings using this lubricant, it is found that the film pressure and the load capacity of pseudo-plastic fluids are smaller than those of Newtonian fluids whose viscosity is equal to the initial viscosity. These values become smaller with an increase in nonlinear factor and eccentricity ratio of bearing.

1. Introduction

Few experimental investigations of the effects of non-Newtonian characteristics of lubricants on bearing performance have been published, and discussions of the theoretical and the experimental results have also been few in number.

The authors have investigated the lubrication of pseudo-plastic fluids during studies of the lubrication of non-Newtonian fluids. Theoretical results were already presented in the first paper[1], in which the influences of the non-Newtonian characteristics on the performance of journal bearings are clarified analytically.

The flow characteristics of pseudo-plastic fluids are studied, after adding polyisobutylene, which is used in practice as a viscosity index improver, to spindle oil. The values of the nonlinear factor, which is a scale to indicate the behavior of such a lubricant, are obtained experimentally from the flow characteristics. The performance of the journal bearing using such lubricants, such as the film pressure, the

load capacity and the attitude angle is clarified experimentally, and discussed in comparison with the theoretical results described in the first paper. The results are presented in this paper because the experimental results agree well with the theories.

2. Nomenclature

C : radial clearance
D : diameter of journal
L : submerged depth, or bearing width
M : torque of inner cylinder
N : revolutions per time unit
P : bearing pressure
R : radius of journal
R_b : radius of inner cylinder
R_c : radius of outer cylinder
S : Sommerfeld number
T : temperature
U : circumferential velocity of journal
W : load capacity
k : " coefficient of pseudo-plasticity "
p : film pressure
Ω : angular velocity of outer cylinder
α : ratio between inner and outer cylinder $= \hat{R}_c / \hat{R}_b$
β_1, β_2 : instrument constant of rheometer
$\dot{\gamma}$: shear rate
$\dot{\gamma}'$: apparent shear rate

δ : "nonlinear factor" $=\hat{k}(\hat{\mu}\hat{U}/\hat{C})^2$

ε : eccentricity ratio

θ : bearing angle

λ : fluidity

μ : "initial viscosity"

τ : shear stress

ϕ : attitude angle

Subscript

$\wedge$: dimensional quantity

3. Flow characteristics tests

The flow characteristics of the lubricants formed by adding polyisobutylene to spindle oils are studied here because such lubricants have the behavior of pseudo-plastic fluids.

3·1 Experimental apparatus and procedure

The rheometer shown in Fig. 1 is used for measuring the relationship between the shear stress and the shear rate of lubricants in order to study the non-Newtonian characteristics. The tested fluid fills the space between the outer and the inner cylinder of the rheometer, and the inner cylinder is twisted because of the viscous resistance when the outer cylinder rotates at a constant speed. Calculating the torque at the surface of the inner cylinder by measuring its angular displacement, the shear stress is obtained from the following equation.

$$\hat{\tau}=\hat{M}/(2\pi\hat{R}_b{}^2\hat{L})\cdots\cdots\cdots\cdots\cdots(1)$$

Considering that the tested fluid is non-Newtonian, the shear rate is calculated by the single bob method[2], as follows.

$$\hat{\dot{\gamma}}=\hat{\dot{\gamma}}'\left\{1+\beta_1\frac{d\ln\hat{\lambda}}{d\ln\hat{\tau}}+\beta_2\left(\frac{d\ln\hat{\lambda}}{d\ln\hat{\tau}}\right)^2\right\}\cdots\cdots(2)$$

where

$$\hat{\dot{\gamma}}'=\frac{2\hat{\Omega}}{1-1/\alpha^2}\cdots\cdots\cdots\cdots\cdots(3)$$

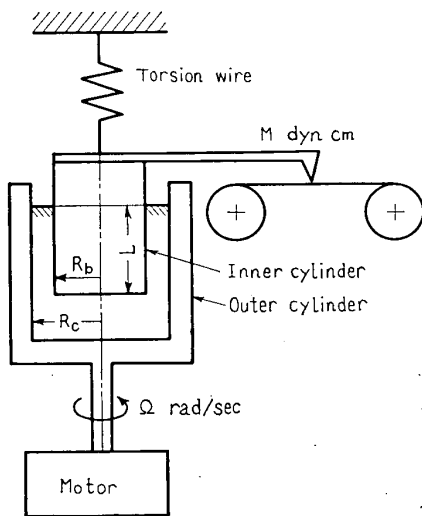$$\hat{\lambda}=\hat{\dot{\gamma}}'/\hat{\tau}\cdots\cdots\cdots\cdots\cdots(4)$$

Fig. 1 Rheometer

$$\beta_1=\frac{\alpha^2-1}{\alpha^2}\left(1+\frac{2}{3}\ln\alpha\right)\cdots\cdots\cdots\cdots(5)$$

$$\beta_2=\frac{\alpha^2-1}{6\alpha^2}\ln\alpha\cdots\cdots\cdots\cdots\cdots(6)$$

$$\alpha=\hat{R}_c/\hat{R}_b\cdots\cdots\cdots\cdots\cdots\cdots(7)$$

The values of $d\ln\hat{\lambda}/d\ln\hat{\tau}$ in Eq. (2) are given by graphical differentiation in general procedure. If the tested fluid is Newtonian, then the shear rate is equal to the apparent shear rate because $d\ln\hat{\lambda}/d\ln\hat{\tau}=0$, and, if non-Newtonian, it is troublesome to calculate its values by graphical differentiation. Its values are obtained easily in this paper by the following numerical differentiation.

Let us assume that the relationship between the shear stress and the apparent shear rate can be approximated by the following equation.

$$\hat{\dot{\gamma}}'=\sum_{i=1,3}^{\infty}\hat{a}_i\hat{\tau}^i\cdots\cdots\cdots\cdots\cdots(8)$$

from which the expression of $d\ln\hat{\lambda}/d\ln\hat{\tau}$ is as follows.

$$\frac{d\ln\hat{\lambda}}{d\ln\hat{\tau}}=\frac{\hat{\tau}}{\hat{\dot{\gamma}}'}\frac{d\hat{\dot{\gamma}}'}{d\hat{\tau}}-1=\frac{\sum_{i=1,3}^{n}(i-1)\hat{a}_i\hat{\tau}^i}{\sum_{i=1,3}^{n}\hat{a}_i\hat{\tau}^i}\cdots\cdots(9)$$
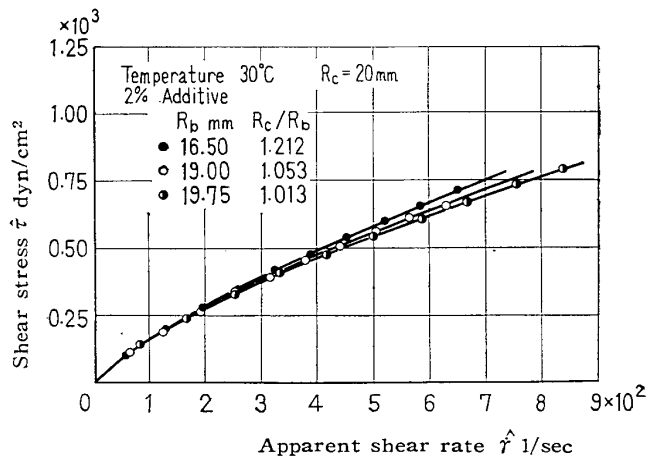

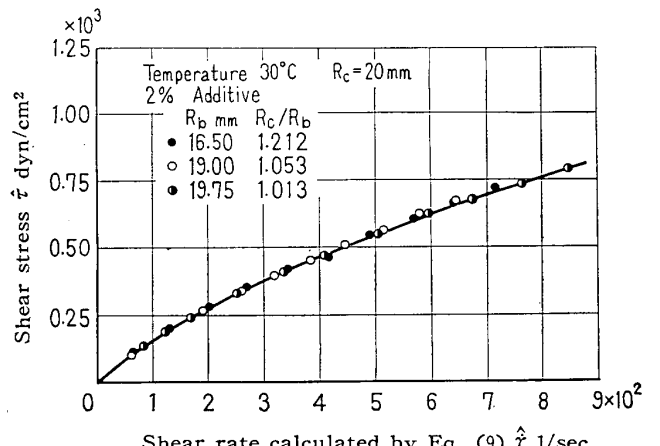
Fig. 2 Flow curve using the apparent shear rate

Fig. 3 Flow curve calculated using Eq. (9)

where the coefficient $\hat{a}_i$ can be calculated by applying the method of least squares to the experimental results.

Figures 2 and 3 indicate the results obtained from the experiments on inner cylinders with different diameters. The flow curves differ in spite of using the same lubricant and the same temperature, as is shown in Fig. 2 whose abscissa is the apparent shear rate. The curves in in Fig. 3 are the results calculated by Eq. (9) which $n=3$. This figure gives the same flow curves using the same lubricant and the same temperature, in spite of different diameters of the inner cylinder. Thus, the shear rate obtained in this experiment is calculated by the above mentioned method, which is an easy procedure and gives good results.

3·2 Non-Newtonian behavior of oils with commercial additives

The flow characteristics of spindle oils with a commercial viscosity index improver added are shown in Fig. 4. Considering that the average shear rate is $\hat{U}/\hat{C}$ in the testing equipment used on the journal bearing described later, the tests are performed in the range of the shear rate. As shown in Fig. 4, lubricants with high molecular-weight polymers added as a viscosity

index improver have pseudo-plastic behavior. The molecular beakdown of the commercial additives, however is not thoroughly known. Therefore, spindle oils with 0.3, 1 and 2% polyisobutylene added by weight are used in this paper as pseudo-plastic fluids and the spindle oil alone is used as Newtonian fluid.

3·3 Flow characteristics of pseudo-plastic fluids and empirical equation

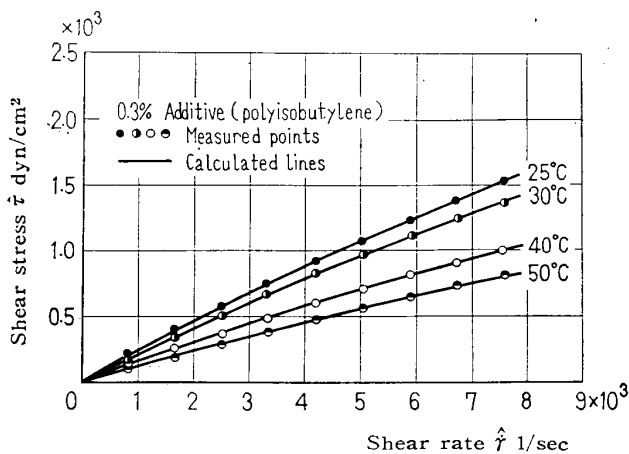In general, the flow characteristics of pseudo-plastic fluids can be given as shown in Fig. 5,
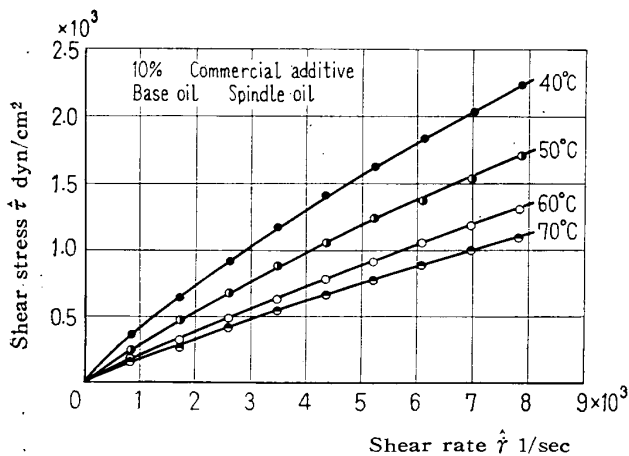
Fig. 6 Flow curve at 0.3% additive

Fig. 7 Flow curve at 1% additive

Fig. 4 Flow curve of oil with commercial additive
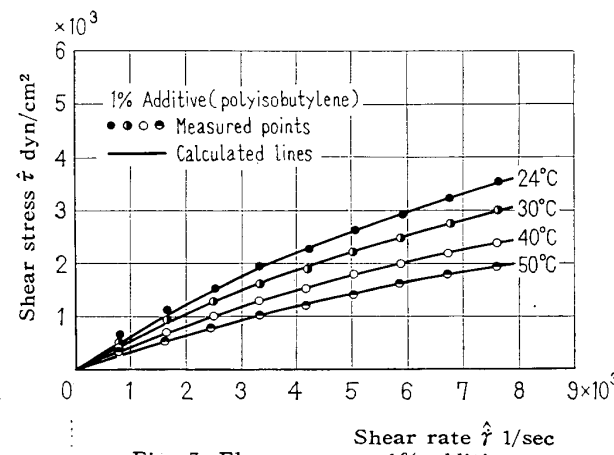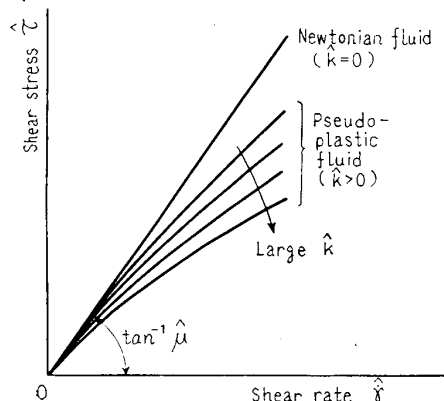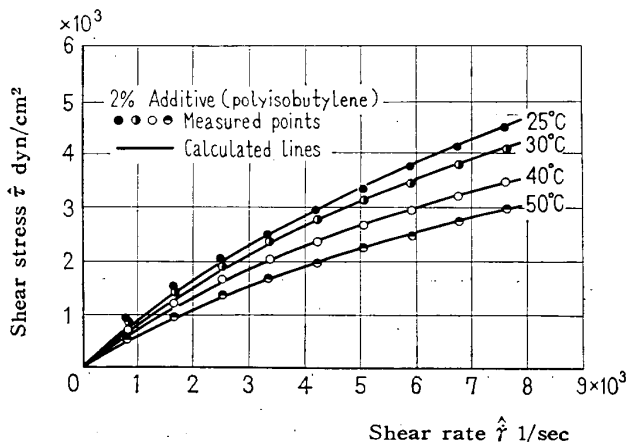
Fig. 5 General flow curve data

Fig. 8 Flow curve at 2% additive

from which the relationship between the shear rate and the shear stress may be approximated by the following empirical equation.
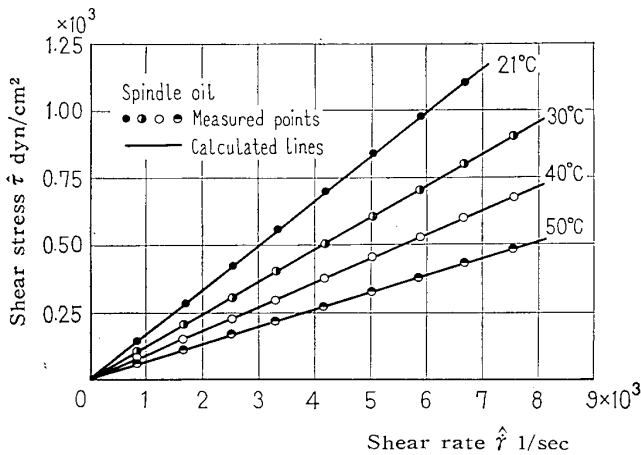

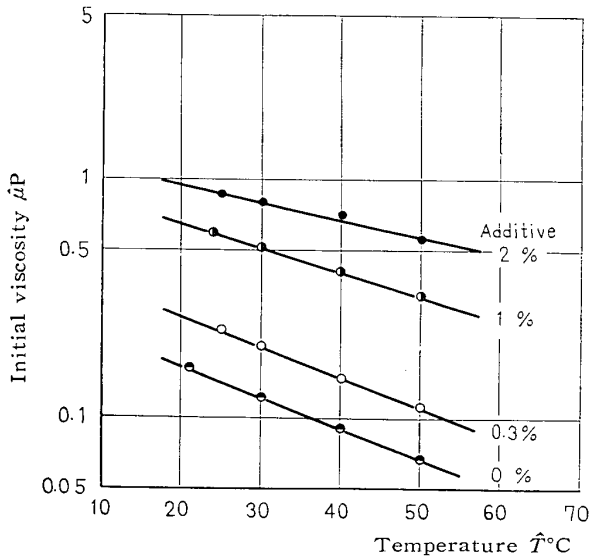Fig. 9 Flow curve for spindle oil only
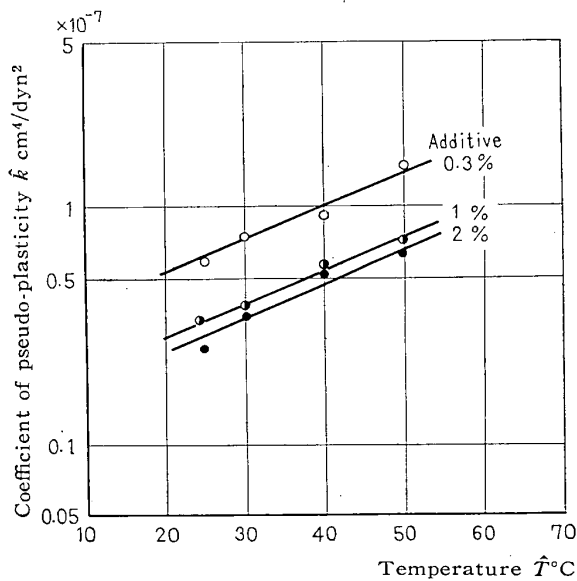

Fig. 10 Initial viscosity versus temperature

Fig. 11 Coefficient of pseudo-plasticity versus
temperature

$$\hat{\mu}\hat{\gamma}=\hat{\tau}+\hat{k}\hat{\tau}^3 \quad\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots(10)$$

where $\hat{\mu}$ is tangent at the original point of the flow curve and is called "initial viscosity" in this paper. If the value of $\hat{\mu}$ does not vary, then the nonlinearity of the flow curve increases with $\hat{k}$, which is called " coefficient of pseudo-plasticity ". In Newtonian fluids, the initial viscosity is equal to the viscosity by Newton's law, because $\hat{k}=0$. The values of the initial viscosity and the coefficient of pseudo-plasticity can be calculated by applying the method of least squares to the experimental flow curves.

The flow curves of spindle oils with 0.3, 1 and 2% polyisobutylene added by weight are shown in Figs. 6, 7 and 8. In these figures, the plotted points represent the experimental results and the thick lines indicate .the relationship between the shear rate and the shear stress, which is calculated from Eq. (10) by the method of least squares. The flow characteristics of pseudo-plastic fluids can be expressed by Eq. (10) because the experimental results agree well with the calculated ones, as shown in these figures.

The flow curves of spindle oil in Fig. 9 show it is Newtonian, because the shear stress is directly proportional to the shear rate.

The characteristics of the initial viscosity dependent on temperature are shown in Fig. 10, which shows that the initial viscosity becomes larger and is insensible to temperature as the amount of the polyisobutylene increases. The viscosity index of the pseudo-plastic fluids can not be defined because the viscosity can not exist in the meaning of Newtonian fluids. However it is interesting that the initial viscosity index appears to increase with the additives as shown in Fig. 10.

The characteristics of the coefficient of pseudo-plasticity vs. the temperature are shown in Fig. 11, from which the coefficient of pseudo-plasticity is known to decrease with an increase of additives. The coefficient of pseudo-plasticity seems to increase with the amount of the polyisobutylene, but this value decreases in practice, as mentioned above. The polyisobutylene affects both the viscosity and the nonlinearity of lubricants, so that the initial viscosity increases and the coefficient of pseudo-plasticity decreases with additives.

3·4 Stability of flow characteristics

The tests of the flow characteristics described in the above sections were performed as soon as the polyisobutylene was dissolved in spindle oils. It is said that the characteristics of lubricants

mixed with additives are unstable and vary as time goes by.

The tests are carried out in order to examine the stability of the flow characteristics of pseudo-plastic fluids. Results are shown in Figs. 12 and 13, in which symbol ▽ indicates the results measured after journal bearing tests and symbol ○ indicates the results measured before tests described later. Symbol × is the result measured for the same lubricants after a month. It is clear from Fig. 12 that the flow curves are identical in the above three tests. The initial viscosity and the coefficient of pseudo-plasticity also have the same values as shown in Fig. 13, therefore the flow characteristics of lubricants
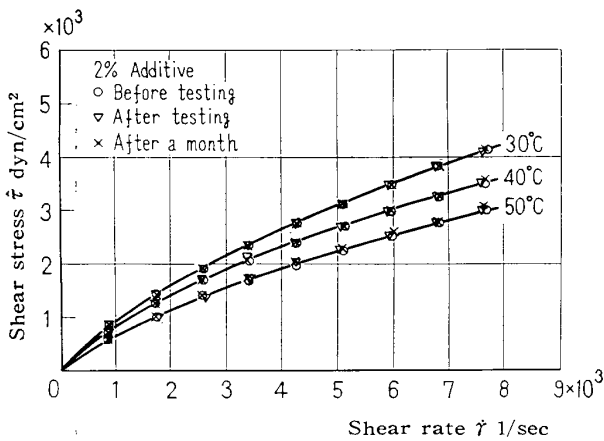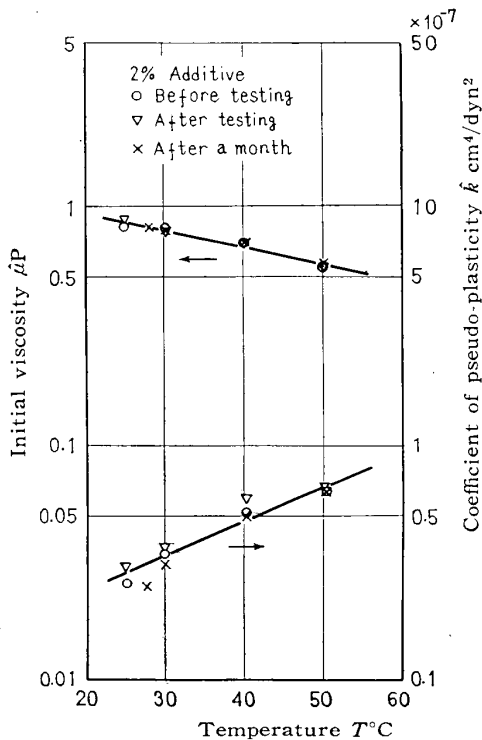
used in this experiment are stable within the period covered by the tests.

4. Experiment on journal bearing

The performance of the journal bearing using pseudo-plastic lubricants will be obtained experimentally and the effects of non-Newtonian characteristics on the performance will be clarified.

4·1 Experimental apparatus and procedure

The testing equipment used on the journal bearing is illustrated schematically in Fig. 14, in which the rotating shaft is supported by ball bearings and the bearing metal is floating on the journal. In order to firmly support the rotating shaft, two ball bearings are used on each side. The shaft then rotates within $1\,\mu$ accuracy. The diameter and the width of the bearing metal are about 50 mm and the radial clearance ratio is about 2.3/1 000. The loads are applied to the bearing by dead weights hung on the wire. The range of weights is from 5.7 to 37.3 kg. Dimensions of the bearing and the journal are given in Table 1.

The displacement of the bearing is measured by two dial gauges set at right angles to each

Fig. 12 Flow curve showing time dependence

Fig. 13 Initial viscosity and coefficient of pseudo-plasticity versus temperature
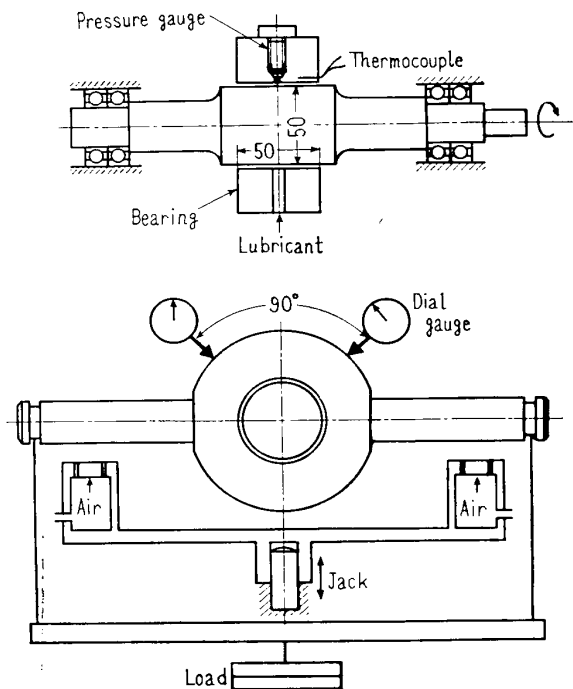
Fig. 14 Journal bearing tester schematic

Table 1 Experimental conditions

Diameter of journal	mm	50.010
Diameter of bearing	mm	50.235
Radial clearance	μ	112.5
Width-to-diameter ratio		1
Revolution of journal	rpm	308

other, in which the reference point in measuring is the concentric point formed by the bearing and the journal. To obtain the concentric position, the bearing is floated by pneumatic pressure which is controlled by moving the jack until the indications of the two dial gauges do not vary no matter what the rotational directions of the shaft.

There are thirteen holes, each 1 mm in diameter, used to measure the film pressure at the middle of a cross section of the bearing. The film pressure is detected by a small pressure transducer with a self compensated foil strain gauge to prevent error due to temperature rise. The error may be neglected as the change of the temperature in experiments is only within 2°C. It is recognized preliminarily that the relationship between the strain of the gauge and the pressure is completely linear and that its characteristics are also stable.

The temperature of the bearing metal in the vicinity of the minimum film thickness is considered the temperature of the lubricants, which is measured by the thermocouple. Lubricants are drop-fed through an oil hole at the unloaded side of the bearing. The diameter of the hole is 8 mm.

Spindle oils with 0.3, 1 or 2% polyisobutylene added by weight are used as pseudo-plastic lubricants and spindle oil alone is used as a Newtonian lubricant. The flow characteristics of these lubricants were described in the previous

Table 2 Lubricants

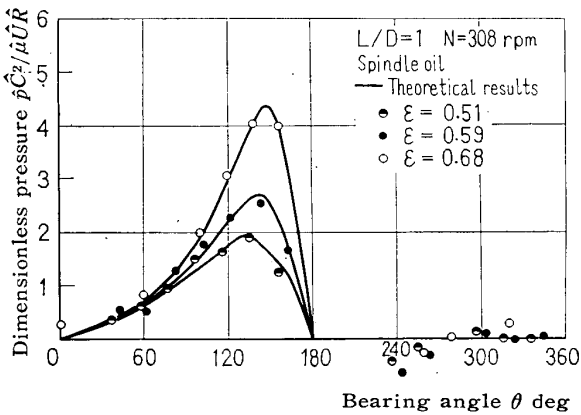| Additives %	$\bar{T}$ °C	$\bar{\mu}$ P	$\bar{k}$ cm⁴/dyn²	δ
0	25	0.140	0 $\times 10^{-7}$	0
0.3	22	0.250	0.565	0.18
1.0	22	0.610	0.305	0.58
2.0	25	0.865	0.295	1.13

Fig. 17 Measured and calculated pressure using 1% additive

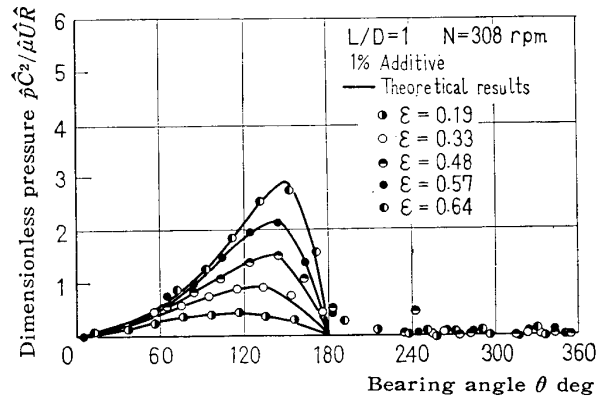Fig. 15 Measured and calculated pressure using spindle oil only

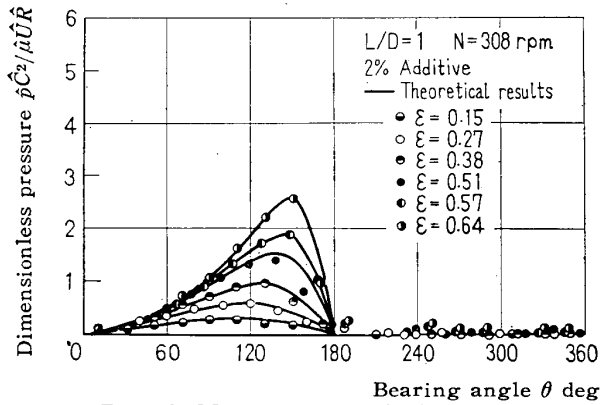Fig. 18 Measured and calculated pressure using 2% additive

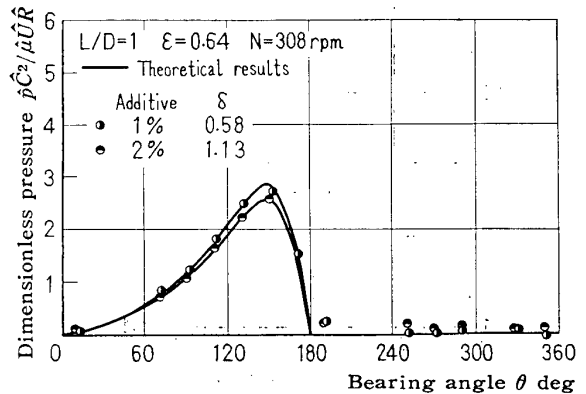Fig. 16 Measured and calculated pressure using 0.3% additive

Fig. 19 Measured and calculated pressure using different δ values

section. The nonlinear factor is calculated in Table 2 by using the data of the initial viscosity and the coefficient of pseudo-plasticity from Figs. 10 and 11. Since the change of the temperature is within 2°C, the nonlinear factor can be evaluated for an average temperature in experiments, as shown in Table 2.

The film pressure is measured and the eccentricity ratio and the attitude angle are calculated from the measurement of the displacement of the bearing when applied loads vary.

4·2 Experimental results and discussions

The experimental results of testing the journal bearing performance using pseudo-plastic fluids are presented here and compared with the theory which has been described in the first paper.

4·2·1 Pressure distribution

The pressure distributions are shown in Fig. 15 to Fig. 19. In these figures, the plotted points represent experimental values and the thick lines theoretical values. It is assumed in the theoretical analysis that the film pressure develops at $0 \leq \theta \leq \pi$ and not in the diverging films, because of the half Sommerfeld boundary conditions.

The pressure distribution in spindle oil is shown in Fig. 15, in which the theoretical lines are calculated as Newtonian fluids because the nonlinear factor is zero. A negative pressure developed slightly in experiments denotes that the half Sommerfeld boundary conditions are not exactly applicable. The pressure distributions of lubricants with 0.3, 1 or 2% polyisobutylene added are shown, respectively, in Figs. 16, 17 and 18. Negative pressure is not observed in diverging films when pseudo-plastic fluids are used, but a small positive pressure develops beyond the angle of the minimum film thickness. Its value is small as compared with that of converging films. The film pressure of pseudo-plastic fluids in diverging region is different from that for Newtonian fluid. The difference of the pressure shapes is not caused by non-Newtonian characteristics, but by the film breakdown described later.

Film pressures with different nonlinear factors are shown in Fig. 19 where the same eccentricity ratio holds. The film pressure of oil with 2% additive is larger in dimensional quantities than that of 1% additive, because the viscosity increases with the amount of additives. However, the dimensionless pressure decreases with an increase of additives, as shown in Fig. 19. Thus if the initial viscosities are the same,

then the film pressure decreases as the nonlinear factor increases.

4·2·2 Load capacity, Sommerfeld number and attitude angle

The relationship between the load capacity and the eccentricity ratio is shown in Fig. 20, and the relationship between the eccentricity ratio and the Sommerfeld number is given in Fig. 21. In general, the viscosity in pseudo-plastic fluids cannot exist in the same meaning as that of Newtonian fluids, and, in this paper, the Sommerfeld number of pseudo-plastic fluids is defined by using an initial viscosity as follows.

$$S = \frac{\hat{\mu}\hat{N}}{\hat{P}}\left(\frac{\hat{R}}{\hat{C}}\right)^2 \quad \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots(11)$$

where $\hat{P} = \hat{W}/\hat{L}\hat{D}$

In Figs. 20 and 21, the experimental results are indicated by plotted points and the thick lines represent the theoretical results.

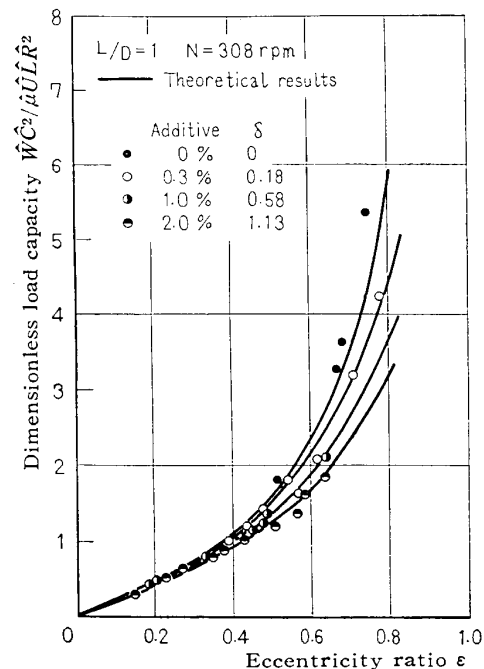The pressure distribution of pseudo-plastic

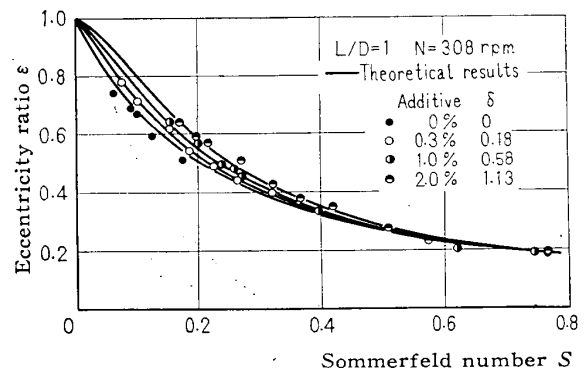Fig. 20 Measured and calculated load capacity

Fig. 21 Measured and calculated Sommerfeld number

fluids decreases below that of Newtonian fluids whose viscosity is equal to the initial viscosity, so that the load capacity is smaller than that of a Newtonian fluid. Therefore, the Sommerfeld number is larger than for the Newtonian fluid at the same eccentricity ratio. These tendencies become larger as the nonlinear factor and the eccentricity ratio increase, as shown in Figs. 20 and 21. If the initial viscosity is the same, the coefficient of pseudo-plasticity increases with the nonlinear factor. And the increase of the eccentricity ratio indicates a large shear rate in converging films, even if the journal velocity is the same. As shown in Fig. 5, the flow curve indicates clearly the nonlinear characteristics under the condition mentioned above; therefore, the effects of the non-Newtonian characteristics on the bearing performance, such as the load capacity and the Sommerfeld number, become larger.

The experimental and theoretical relationships between the eccentricity ratio and the attitude angle are shown in Fig. 22. The experimental results for Newtonian fluids are larger than the theoretical ones at the small eccentricity ratio and approach the theoretical values as the eccentricity ratio increases. Although the half Sommerfeld boundary conditions have been used in theory, a continuous film exsits in practice over a wide range of bearing clearances when the eccentricity ratio is small and a negative pressure is observed in diverging films as shown in Fig. 15. Therefore, the experimental results are somewhat larger

L/D = 1   N = 308 rpm
———— Theory (δ = 0)
− − − − Theory (δ = 1.13)

Additive δ
●   0 %   0
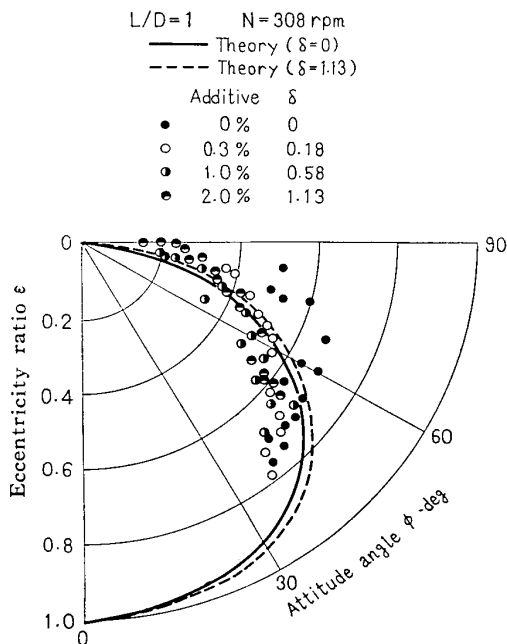○   0.3 %   0.18
◑   1.0 %   0.58
◉   2.0 %   1.13

Fig. 22 Measured and calculated attitude angle

than the theoretical results. As the eccentricity ratio increases, the film breakdown occurs in diverging films and the film pressure in this region becomes the same as that of atmosphere. Under these conditions, the half Sommerfeld conditions are satisfied, therefore the experimental results agree approximately with calculated ones.

When pseudo-plastic fluids are used, the film is easy to break because air is apt to be absorbed into such lubricants. The film breaks at even a small eccentricity ratio and then a negative pressure does not develop in diverging films as shown in Fig. 16 to Fig. 19 and the film pressure becomes the same as that of atmosphere in the vicinity of the minimum film thickness. Thus, the theoretical results agree relatively well with experiments. The effects of the boundary conditions of the film pressure on the attitude angle are large, and so the influences of non-Newtonian characteristics on the attitude angle can not be clarified within the limits of this experiment. As the film pressure developed in diverging film is very small, the theory applying the half Sommerfeld boundary conditions agrees well with the experiments in pressure distribution, load capacity and Sommerfeld number.

5. Conclusions

The flow characteristics of pseudo-plastic fluids were studied experimentally and the performance of a journal bearing using such lubricants was obtained and compared with the theoretical results. The following results were obtained.

(1) Lubricating oils with viscosity index improver and viscosity thickener added have the same behavior as pseudo-plastic fluids.

(2) It is clarified experimentally that the flow characteristics of pseudo-plastic fluids are approximately expressed by Eq. (10).

(3) The film pressure of pseudo-plastic fluids in a journal bearing is smaller than that of Newtonian fluids whose viscosity is equal to the initial viscosity. The load capacity becomes smaller and the Sommerfeld number larger at the same eccentricity ratio. These tendencies become larger with an increase in the nonlinear factor and the eccentricity ratio.

(4) The experimental results agree well with the theoretical ones with respect to film pressure, load capacity, Sommerfeld number and eccentricity ratio.

References

(1) Wada, S. and Hayashi, H., This Bulletin, p.268.
(2) Krieger, I. and Maron, S., Jour. Appl. Phys., Vol. 25, No. 1 (1954), p. 72.