A Study on Fundamental Characteristics of Oil Seal**
(Temperature Dependence of Characteristics)

Toshiro YAMADA, Kouki KANI, and Kunihiro NISHIJIMA*
Department of Mechanical Engineering

(Received December 28, 1966)

In order to clarify the sealing mechanism of an oil seal, it is necessary to know its fundamental properties and movement now in action.

There are two subjects in this study. One is to reveal the temperature dependence of fundamental properties, mainly wringing force, average wringing pressure and lip contact width, and the other is to know the lip temperature now in action and to know how the lubrication is carried out at the lip portion. The measurements were carried out with our devised instruments for the former subject and with a thermocouple for the latter. A simple analysis of the lip temperature obtained was made in order to know a mechanism of the lubrication.

The results indicate that with the rise of temperature of oil seal, the wringing force and the average wringing pressure decrease because of the reduction of rubber elastic modulus and of thermal expansion of spring, and the lip contact width hardly varies, and also indicate that under a constant temperature of lubricating oil, the lip temperature never rises so high as in the case of free rising of oil temperature, and that the fluid lubrication is carried out at the lip.

§ 1. Introduction

Sealing characteristics of radial lip-type oil seal have been much investigated, but there are as yet few papers on the fundamental properties of oil seal itself.

This study has been undertaken to clarify a temperature dependence of wringing force, lip contact width and average wringing pressure, which are thought to be closely related with sealing characteristics of radial lip-type oil seal.

Furthermore, by carring out the oil seal tests with a rotating test instrument, it has been revealed how high the interface temperature between the shaft and seal rises with the increase of shaft speed and wringing force of oil seals.

It has been analysed from the results of the increase of lip temperature obtained in those experiments how lubrication is carried out between the shaft surface and the seal lip.

§ 2. Specimen and devices

2.1 Specimen

Only radial lip-type (SC-type) oil seals have been used as the specimen, which were made from synthetic rubber and for shaft 40 mm in diameter.

The chemical composition of synthetic rubber

Table 1. Chemical composition of synthetic rubber.

element	ratio of component
raw rubber	47.2 %
carbon	42.7
allyl amine	2.8
T. C. P.	2.8
phenol	2.1
sulphur & stearin	2.4

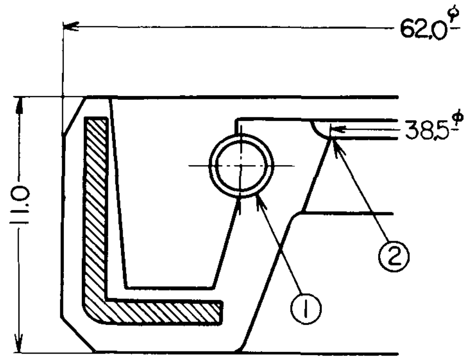

Fig. 1. A part of cut section of radial lip-type oil seal with the plane including the axis. 1, Spring. 2, Lip.

** This work was reported at the Conference of the Japan Society of Precision Engineering at Hiroshima, 20th, November, 1966.

* Uchiyama Kogyo Kaisha, Ltd. Okayama

and the shape of specimen are shown in Table 1 and Fig. 1 respectively.

2.2 Device for wringing force measurement

The wringing force of radial lip-type oil seal is made up of three sorts of forces; the first, force owing to the circular elongation of lip, the second, due to the bent waist of seal, and the third, produced by a garter spring set in the seal.

When the shaft is inserted into the oil seal, the stretch force (P_c) of lip acting in the circumferential direction correlates with the total wringing force (P_r) acting in the radial direction as shown below

$$P_r = 2\pi P_c. \tag{1}$$

The relation between P_c, P, and p are shown in Fig. 2. The information directly obtained from the measurement of oil seal is P_r, and this can be expressed as follows;

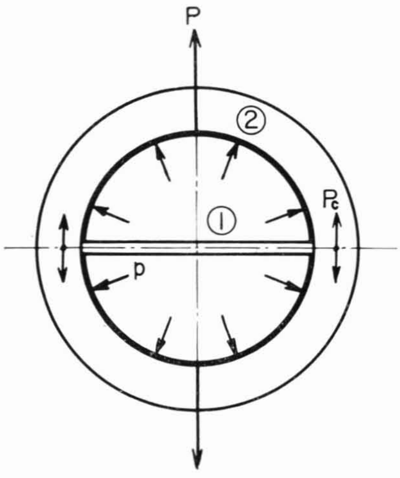

Fig. 2. Schematic illustration of the relation between Pc, P and p. 1, Half-cylinder. 2, Oil seal.

$$P_r = 2\pi rbp, \tag{2}$$

where r is radius of shaft, b lip contact width with shaft, and p average wringing pressure acting in the radial direction.

In this study, P_r has been measured with a balance-type instrument[1]. Fig. 3 shows our device. The device consists of two half-cylinders; one of the pair is fixed and the other can move in radial direction. The deviation of the movable one is measured with dial gauge. After the measurement of the position of which they constitute a perfect cylinder together, the cylinder becomes imperfect because of the wringing force of oil seal if the oil seal is set on the perfect cylinder.

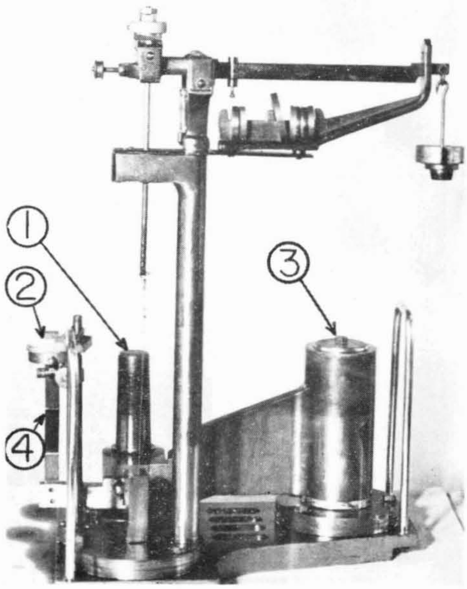

Fig. 3. Instrument for measurement of wringing force. 1, Two half-cylinders. 2, Dial gauge. 3, Center of rotation of two half-cylinders. 4, Fixed bar to movable member.

By applying the force (P) with balance weight to the movable member, the distorted cylinder is restored to the previous perfect situation.

Here, P has the following relation with other values;

$$P_r = \pi P, \tag{3}$$

$$P = 2rbp. \tag{4}$$

The instrument is placed in an oil bath to vary the temperature of oil seal. The oil is heated with a nichrome heater from about 30°C to 120°C. The friction between the half-cylinders and the oil seal lip, and the elastic after-effect of synthetic rubber waste long time before the two half-cylinders are in the perfect cylinder position. Therefore a light vibration is given to the oil bath in order to shorten the time for the balancing at the perfect cylinder position.

2.3 Device for measurement of spring force and elastic modulus of rubber

The spring balance has been remodelled to hold a spring and rubber specimen with one jaw, and the other jaw has been fixed at the bottom of oil bath. Various temperatures required have been given for the specimens by changes in oil temperature.

2.4 Device for measurement of the lip contact width

Fig. 4 shows the device. The device consists of a hollow transparent glass cylinder and a

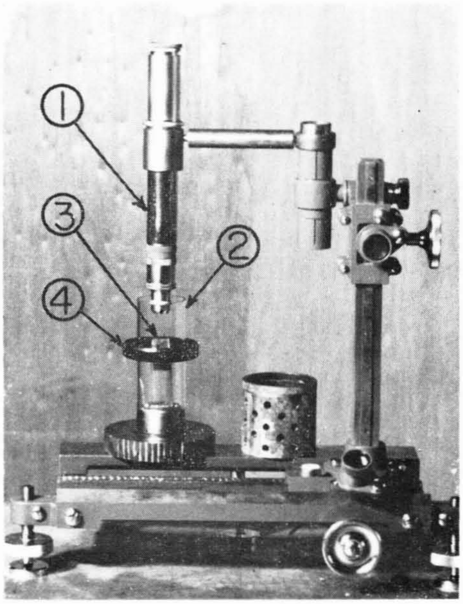

Fig. 4. Instrument for measurement of lip contact width. 1, Travelling microscope. 2, Hollow glass cylinder. 3, Mirror. 4, Oil seal.

small mirror which is set at a point on the axis and inclined 45° to the axis of the glass cylinder in order to see the contact portion from inside. When the glass cylinder is inserted into the oil seal, the contact portion with each other is seen

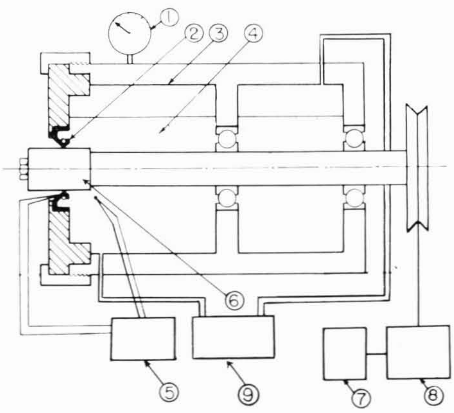

Fig. 5. Schematic diagram of rotating test machine. 1, Pressure gauge. 2, Oil seal. 3, Oil cavity. 4, Lubricating oil. 5, DC amplifier for thermocouple. 6, Shaft for test. 7, Motor. 8, Speed change gear box. 9, Oil temperature control tank.

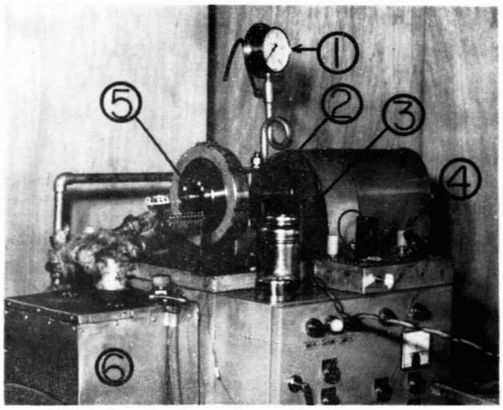

Fig. 6. General view of rotating test machine in use. 1, Pressure gauge. 2, Oil cavity. 3, Vacuum bottle for ice. 4, DC amplifier for thermocouple. 5, Shaft for test. 6, Oil temperature control tank.

to be a black narrow band. Consequently the width of the band (that is, lip contact width) can be observed and measured with the set mirror and a travelling microscope.

2.5 Rotating test machine

The rotating test machine designed by the

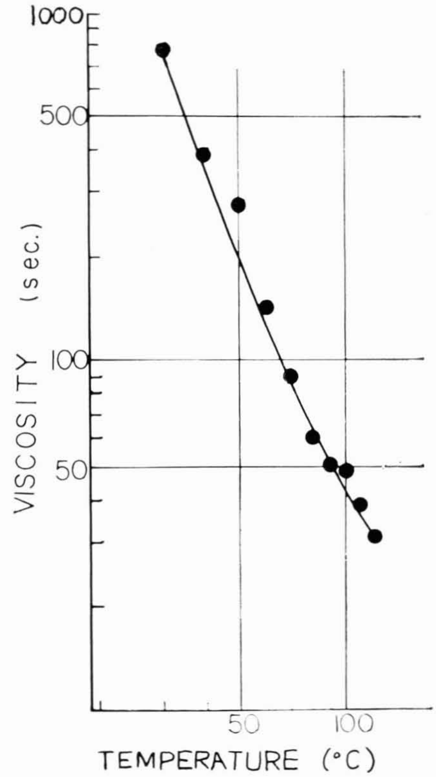

Fig. 7. Temperature dependence of viscocity of lubricating oil (moter oil #20). Viscocity is plotted in Redwood Second.

authors is used. Its scheme and general view are shown in Fig. 5 and Fig. 6 respectively. Lubricating oil used for this test is motor oil # 20 (SAE20), and the relation between its viscosity and temperature is shown in Fig. 7.

During the tests, the temperature of oil is controlled by the circulation between oil cavity of the test machine and the control tank. The range of shaft speed of this test machine is 550 ~3000 *r. p. m.* The eccentricity of shaft (2ε) is less than 0.03 mm. The roughness of shaft surface has the values of Ha 0.14~0.19 μm.

§ 3. Experimental procedure and results

3.1 Temperature dependence of wringing force

The measurement has been made for the five springs whose spring constants are different respectively. For all springs, the diametral interferences of seals and shaft are 1.5 mm at 25°C. After the oil seal is set on the instrument, enough time is needed (about one hour), before measurement, so that the seal may perfectly reach the temperature required in the oil. Thereafter the load is applied little by little, and finally the load equilibrates with the wringing force of the seal whose inside diameter becomes that of a true circle. The load finally applied is P.

The temperature dependence of wringing force of the seal at several temperatures is shown in Fig. 8. The percentage value of wringing force to the initial (at 25°C) fairly decreases with the increase of temperature.

3.2 Temperature dependence of the spring force and the elastic modulus of the synthetic rubber.

The spring and rubber specimen are kept at the temperature required in the oil bath, and then the force applied to elongate them to required length is measured. For the rubber specimen, the load is removed while the seal temperature is changing.

Springs used as specimem in this experiment are the same ones that are used for the measurement of wringing force. All the springs have the same length as 130 mm at 25°C. The rubber test pieces are rectangular strips which are made from the same quality as oil seal and 100 mm in length between jaws, 10 mm in width and 2 mm in thickness. The spring specimens are stretched to 140 mm in length and the rubber one to 108 mm respectively. The results are

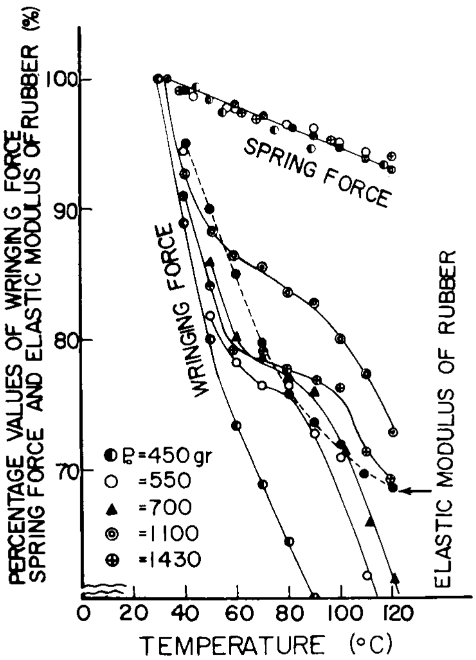

Fig. 8. Temperature dependence of wringing force, spring force and elastic modulus of synthetic rubber. They are plotted in percentage scale to the initial at 25°C. $P_0 = P$ at 25°C.

shown in Fig. 8 plotted in the percentage scale of elongation to the value at 25°C in order to illustrate the correlation with wringing force. The elastic modulus of the rubber decreases with the increase of its temperature. The spring force hardly decreases with changes in temperature and has little corelation with the sort of springs.

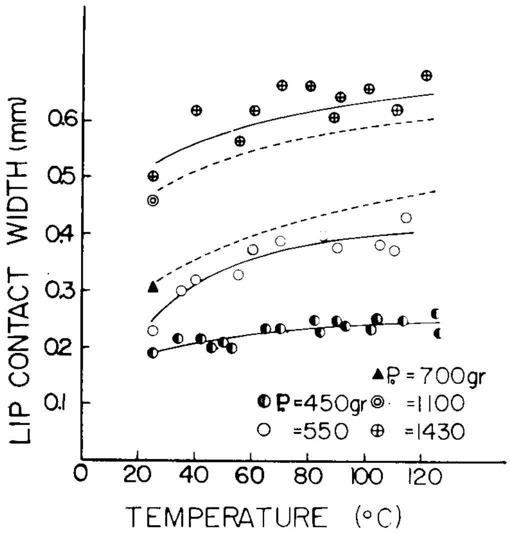

Fig. 9. Temperature dependence of lip contact width. $P_0 = P$ at 25°C. Dotted lines are estimated ones.

3.3 Measurement of the lip contact width

The measurement of the lip contact width is carried out by setting the same spring that has been used in the above measurement. The oil seal has been set on the hollow glass cylinder as the cylinder axis coincides with that of the seal. The temperature is kept at a required point after being placed in oil bath. Since the black line becomes indistinct because of the osmotic effect of oil, the measurements have been done after one gets a clear boundary colorated with oil colour along the contact portion. As shown in Fig. 9, the result indicates that the lip contact width becomes wide with the increase of spring force, but hardly increases with the rise of temperature.

3.4 Average wringing pressure

The average pressure and the pressure distribution in the direction of shaft axis is considered to be closely related with the sealing characteristics of oil seal. For evaluating the average pressure, the results of the subsection 3.1, 3.3 and Eq. 4 are necessary. Substituting P from 3.5 and b from 3.3 in Eq. 4, we obtain p. The relation between the temperature and the spring force dependence of average wringing pressure are shown in Fig. 10.

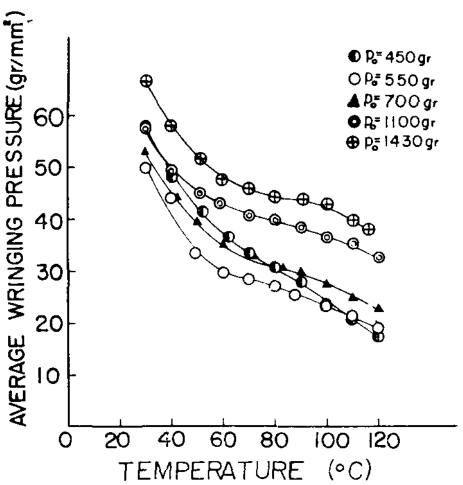

Fig. 10. Temperature dependence of average wringing pressure. $P_0 = P$ at 25°C.

It is seen that the average pressure decreases with the rise of temperature, and except one case, a strong spring produces a high average pressure, but the average pressure is not proportional to the spring force.

3.5 Rise of temperature of the lip owing to the increase of shaft speed and wringing force.

The same springs that have been used in the above experiment are employed in this measurement. The range of shaft speed is 550~3000 r. p. m. and the lubricating oil pressure is equal to a atmospheric one.

The measurement of temperature is done in the following way; a copper-constantan thermocouple whose diameter is 0.05 mm is buried at about 0.5 mm depth under the lip surface in the rubber of oil seal. A high sensitive chopper type DC amplifier is used as the out put voltage of thermocouple is small. This device can measure 1°C variation.

The result of calibration indicates that the temperature at the thermocouple point shows 1~2°C lower value than that of lip contact surface because of the difference of conduction of heat. The lip temperature rises to a saturated value by running the shaft, which is defined as the lip temperature. During this measurement, the room temperature is about 20°C.

3.5.1 Case of a constant temperature of the lubricating oil

The lubricating oil temperature is measured by thermocouple at a point about 5 mm distant from lip portion in oil.

The ripple of oil temperature is about ±2°C. The oil level in oil cavity of the test machine is kept 5 cm above the upper surface of shaft. The measurement has been made by using two spirngs; one is the strongest one, and the other the weakest. Measurement values under this condition is shown in Fig. 11.

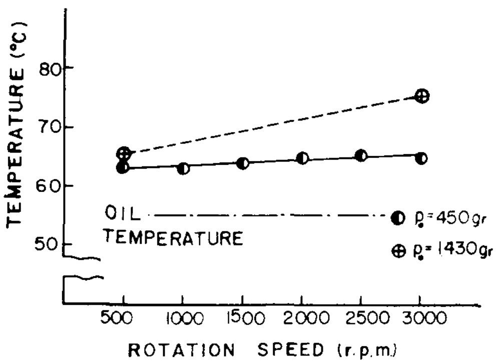

Fig. 11. Lip temperature as the function of rotating shaft speed under a constant lubricating oil temperature.
$P_0 = P$ at 25°C. One dotted line shows the lubricating oil temperature.

3. 5. 2 The case where oil cavity is empty

Since the cavity of test machine is empty, the shaft is rotated while one is pouring oil to the lip portion so as not to vanish oil film between the lip and shaft.

The temperature is measured at a saturated point. The relation between the rise of lip temperature and the increase of shaft speed, and the wringing force are shown in Fig. 12.

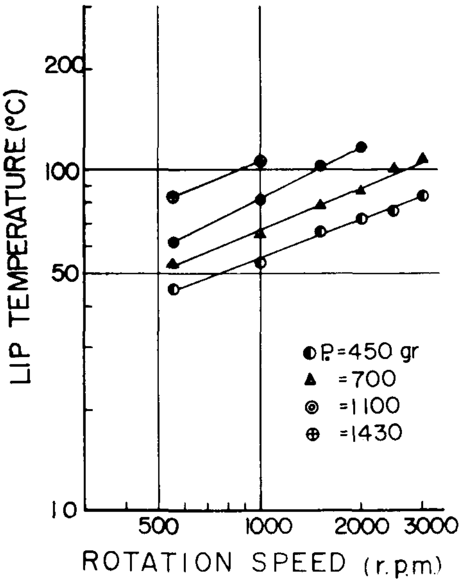

Fig. 12. Lip temperature as the function of rotating shaft speed under free rising of lubricating oil temperature. $P_0=P$ at 25°C.

§ 4. Discussion and conclusion

From the experimental result of subsection 3.1 and 3.2, for each spring, Fig. 8 shows that the rate of reduction of wringing force is great in lower range of temperature, and small in higher range. In the same figure at the low range of temperature, the tendency of elastic modulus of the rubber is the same as that of wringing force, but in higher range, the rate of wringing force change is the same as that of spring force.

Therefore it seems that in low range of temperature the decrease of wringing force with the rise of temperature mainly depends on the decrease of elastic modulus of rubber, and in higher range it is mainly governed by the decrease of spring force caused by the linear expansion with heat.

Next, let's discuss the 3.5.1 results of this experiment. It is thought that the heat which is generated by friction between the lip and the shaft is taken out mainly by the lubricating oil and the shaft. As shown in Fig. 11, the lip temperature does not rise more 20°C than that of lubricating oil, so far as those measurements even with the strongest spring and maximum shaft speed are concerned. Now if the result of 3.5.2 is taken into account, it will be apparent that the heat generated by friction at the lip portion is taken out mainly only by lubricating oil under a constant temperature of the oil. That is, both the experimental conditions are the same in subsection 3.5.1 and 3.5.2 measurement except the condition of existence of lubricating oil. Therefore in both cases, the quantity of the heat flowing out through the shaft may be said to be equal when the temperature of the lips is the same. Furthermore, if the quantity of the heat taken out by the lubricating oil were little, the lip temperatures would be nearly equal in both cases, but when lubicating oil is not used (that is, 3.5.2 measurement) the lip temperature rises much higher than that in the other case. Consequently the conclusion mentioned above is obtained.

About the result of the measurement in subsection 3.5.2, the clear linear relation between the lip temperature and shaft speed for each spring exists in log-log scale graph as shown in Fig. 12. A simple analysis of these results will be done as follows; It is considered that the power consumed with the friction between the oil seal and the shaft converted to heat, and when the rise of lip temperature stops, this generated heat is equal to the heat flowing out. If it is assumed that the oil film between lip and shaft has constant thickness around the shaft as the eccentricity of shaft is small, and the fluid lubrication is carried out there, viscous frictional force (T) per unit area of the shaft surface caused by running of shaft and viscosity of oil is expressed by the equation

$$T=\eta V/\delta, \tag{5}$$

where η is coefficient of viscosity of oil, V sliding speed of the shaft surface and δ thickness of oil film. Consequently, total frictional force (F) around the shaft surface is

$$\begin{aligned} F &= 2\pi rb\cdot T \\ &= 2\pi rb\cdot\eta V/\delta \\ &= 2\pi rb\cdot\eta\cdot 2\pi rN/\delta. \end{aligned} \tag{6}$$

In this expression, r is a radius of shaft, b a lip contact width and N rotating number per unit time.

The power (W) consumed by the viscous friction is

$$W = 2\pi N \cdot rF$$
$$= 2\pi N \cdot r \cdot 4\pi^2 r^2 b\eta N/\delta. \qquad (7)$$

Now, one can express the heat (Q) generated at lip per unit time in the following form

$$Q = k \cdot 8\pi^3 r^3 b\eta N^2/\delta, \qquad (8)$$

where k is a proportional constant. On the other hand, by assuming that the lip temperature is kept constant by the flowing out of this heat (Q)

$$Q = \alpha\ (\theta - \theta_0). \qquad (9)$$

In this expression, α is a proportional constant, θ is lip temperature and θ_0 is room temperature. And by assuming the following relation according to Figs. 7 and 10

$$\eta = A\ (\theta - \theta_0)^{\beta}, \qquad (10)$$
$$p = B\ (\theta - \theta_0)^{\gamma} \qquad (11)$$

and furthermore

$$\delta = C\ (\theta - \theta_0)^{-\varphi} p^{-1}, \qquad (12)$$

where A, B, C, ß, γ and φ are characteristic constants, we obtain the next Eq. 13 from Eqs. 8, 9, 10, 11 and 12

$$\alpha(\theta - \theta_0)^{1-\beta+\varphi-\gamma} = DN^2. \qquad (13)$$

In the Eq. 13, although D includes b, it hardly varies with lip temperature, as mentioned in 3.3, D is regarded as constant here. By using experimental value and Eqs. 10 and 11, the values of β and γ can be obtained. Then, by substituting the values of β and γ obtained above in Eq. 13 and by using the values of line slopes in Fig. 12, the φ values are obtained as shown in Table 2. In Table 2, the φ values are fairly in good agreement. Consequently, those processes of development of equations assumed here are thought to be correct. Furthermore, by use of Eq. 12, the thickness of oil film is calculated from values of γ and φ.

Table 2. φ values for each spring calculated from values of γ, β and the slope of each line in Fig. 12.

$P_0(gr)$	γ	φ
1430	−0.23	0.88
1100	−0.22	0.64
700	−0.30	0.73
450	−0.38	0.55

$\beta = -1.54$

The result of this evaluation indicates that oil film thickness decreases with the rise of lip temperature. That seems reasonable. Thus it can be seen that the results obtained in this experiment are qualitatively in accordance with the above fluid lubrication assumption. That is, it may be concluded that the fluid lubrication is mainly carried out at lip portion.

Acknowledgements

The authors wish to express their gratitudes to H. NAKAE and K. NISIZAKI for their earnest help in the experimental works.

Reference

1) H. Ishiwata, & Y. Fujiwara: Journal of the J. S. M. E., 64 No. 512, (1961) 48.