Traction Characteristics of Lubricating Oil under High Pressure*

(1st Report, Traction Coefficients at Mean Pressures up to 2.7 GPa)

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The authors made a high performance disk machine to clarify traction characteristics of lubricating oils, and succeeded in obtaining the traction coefficient curves of the typical lubricating oils at appreciably higher pressures than in earlier experiments.

By analyzing the traction coefficients obtained in experiments, the authors made empirical equations for calculating traction coefficients. Unknown factors in the equations could be accurately decided by experiments conducted under low/medium pressure conditions where performance of experiments was comparatively easy. Using the equations, it is possible to estimate the traction coefficient curves at high pressure ($p_{\text{mean}} = 2.0 - 2.7 \text{ GPa}$) where performance of experiments is very difficult due to metallic contacts between surfaces even when test rollers with very smooth surfaces are used.

Key Words: Lubricating Oil, Traction Coefficient, Friction Coefficient, EHL Oil Film, Specific Sliding, High Contact Pressure

1. Introduction

Power transmission capacity of a traction drive is smaller than that of the corresponding gear drive with much the same size in most cases. However, the running noise and vibration of traction drives are much lower than those of gear drives. Due to this reason, recently, much attention has been paid to traction drives in order to achieve reduction in the running noise of power transmission in the high speed range.

The power transmission capacity of a traction drive becomes higher when the contact pressure between the rolling elements is increased. However, the rolling elements will fail due to the fatigue of material when the contact pressure exceeds a certain limit (pitting limit). Recently, the authors succeeded in appreciably increasing the pitting limit of rollers made from high hardness steels⁽²⁾⁽³⁾ by introducing the elastohydrodynamic lubricating (EHL) oil film between the contact surfaces, as in the case of the low hardness and medium hardness steels⁽¹⁾. Therefore, it is suggested that the traction drives can be used under appreciably higher contact pressures than those used by earlier designers of traction drives.

When the mean contact pressure exceeds 1.3 GPa (about 13000 times the atmospheric pressure), it is very difficult to measure accurately the shear stress of EHL oil film due to occurrence of an appreciable amount

of metallic contacts even when a sufficient amount of lubricating oil is flooded on the test rollers with very smooth surfaces. Therefore, the traction coefficients clarified in earlier experiments are in the pressure range less than $p_{\text{mean}} \approx 1.4 \text{ GPa}$ (143 kgf/mm^2)⁽⁴⁾⁽⁵⁾. Recently, the authors succeeded in appreciably increasing the pitting limits up

Fig. 1 Main parts of traction-coefficient testing machine

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to a Hertzian pressure of about 2.6 GPa (265 kgf/mm²) by preventing metallic contacts between the surfaces almost completely⁽³⁾. The technique achieved in this experiment was developed further, and it became possible to measure the traction coefficients of EHL oil film at higher pressures up to 2.6 GPa. Moreover, it was made possible to estimate easily the traction coefficients at extremely high pressures, at which the conduction of reliable experiments is impossible, using the empirical equations proposed by the authors.

2. Testing machine and test oils

Figure 1 shows a plan view of main parts of a testing machine used in the present investigation to examine the traction characteristics of lubricating oils. This testing machine was designed and made by the authors, and it is capable of applying test loads up to 50 kN. The most important characteristics in the testing machine is that changes in the traction characteristics can be measured after confirming the formation of EHL oil film between test rollers during running.

Figure 2 shows test rollers used in the experiments at specific slidings higher than $|\epsilon_s| = 2\%$. These test rollers have three distinguished features; (a) the length of boss is 2.5 times the effective width (10 mm), (b) the tapered mounting hole makes it possible to mount the test rollers without any clearance, and (c) the side key ways can prevent a deterioration of accuracy of test rollers during loaded running.

When the specific sliding is lower than 2%, a slight difference is given in the diameters of two mating rollers and the gear ratio of the power circulating gears is made equal to unity.

Test rollers were made from a Cr-Mo steel (SCM435). They were hardened and tempered to a Brinell hardness of about 540 and then ground to a roughness of $R_{\max} = 0.2 \mu\text{m}$ using a precision cylindrical grinder made by M. Co.

Two typical oils A and B with very different viscosities were used as test oils. The oil A is a conventional gear oil (Heavy Medium) for cylindrical gears. The oil B is a high viscosity lubricating oil (Cylinder oil). Changes in the viscosity of the oils at atmospheric pressure are shown in Fig. 3.

3. Experimental conditions

3.1 Lubricating conditions and specific sliding. In most experiments, the lubricating

Fig. 2 A pair of test rollers (for higher specific slidings)

oil was regulated to a temperature of $(40^\circ \pm 1^\circ\text{C})$ and flooded at a point above the contact region of the rollers at a rate of about 1.5 l/min using a nozzle with a diameter of 7 mm. In some experiments, the oil was regulated to a temperature of 70°C . Solid particles in the lubricating oil were removed by a high performance filter in the lubrication system of the testing machine. In order to produce a stable EHL oil film at high contact pressures, it was necessary to use a high performance oil filter.

The specific sliding between the rollers is calculated from $\epsilon_s = (U_2 - U_1) 100 / U_2 (\%)$, where U_1 and U_2 are peripheral speeds of the driving and following rollers, respectively.

3.2 Theoretical oil film thickness. The theoretical minimum oil film thickness $h_{\min}$ between the rollers was calculated using Dowson's equations⁽⁶⁾. When calculating the shear strain (velocity gradient) in the oil film, it is better to use the theoretical oil film thickness h_c , instead of $h_{\min}$, at the parallel portion between the two contacting rollers. The film thickness h_c was calculated using the equations derived by Dowson and Toyoda⁽⁷⁾. In the present experiments, the values of h_c are larger than those of $h_{\min}$ by about 30%.

3.3 Correction of losses at bearings, etc. The traction torque at the driving roller shaft includes the additional torques produced by the bearings of back-up rollers, etc. when it is directly calculated from the measured strain on the shaft. Therefore, the traction coefficient T/P between the rollers is calculated from Eq. (1), and the effect of the losses is removed.

$$\frac{T}{P} = \frac{2T_r}{DP} - \mu_s \dots \dots \dots (1)$$

where, T is the driving force (tangential force) on the outer surface of the roller, P the radial force between a pair of test rollers, T_r the driving torque measured with

Table 1 Characteristics of test oils

Test oil	Oil A	Oil B
Specific weight 15/4 °C	0.878	0.903
Flash point °C	200 <	280 <
Viscosity mm ² /s (cSt)	40 °C 60 100 °C 9	370 29
Viscosity index	95 <	-

Fig. 3 Changes in viscosity of test oils (at atmospheric pressure)

strain gauges, and D the outside diameter of test rollers. μ_B is the equivalent friction coefficient on the outer surface of the test roller and it is calculated from the sum of two kinds of losses; the one at the rolling bearings of shafts of the back-up rollers and driving roller, and the other at the contact region of the outer surface of the back-up rollers. The values of μ_B can be calculated from the friction torque under pure rolling conditions. The numerical values of μ_B for the testing machine used in the present experiments are shown in Fig. 4. The effect of μ_B on the traction coefficients is very small in the high pressure region in which traction coefficients are measured in the present experiments.

4. Experimental results

In the design of a traction drive, pitting which occurs on the contact surfaces is the most important problem. The upper limit of contact pressure at which no pitting occurs can be expressed in terms of Hertzian contact pressure p_{max} in most cases, and therefore, the experimental results will be shown in terms of the Hertzian contact pressure.

It was extremely difficult to produce a full EHL oil film at Hertzian contact pressures higher than $p_{max} = 2.0$ GPa at which no experiments were conducted in earlier investigations. For example, when the surface of test rollers was case-hardened to a higher hardness than $H_v = 700$ in order to prevent the rolling contact fatigue, it was impossible to produce a sufficient EHL oil film between surfaces. In contrast to this, when steel rollers with a lower hardness than $HB = 500$ and with a better running-in capability were used, rolling contact fatigue occurred on the rollers. Finally, the authors succeeded in measuring the traction coefficient of the EHL oil film at higher pressures up to $p_{max} = 2.6$ GPa using test rollers made from Cr-Mo steel which were hardened and tempered care-

fully to a Brinell hardness of about 540 and then ground precisely.

Figures 5 to 7 show changes in the traction coefficients of the oils A and B when the contact pressure and the specific sliding are changed. The contact pressure between the test rollers was varied in the range from $p_{max} = 0.82$ GPa (83.7 kgf/mm²) to 2.6 GPa (265 kgf/mm²). When Fig. 5 is compared with Fig. 7, it is understood that there is only a small difference in the traction coefficients even when the viscosities of oils at atmospheric pressure are different by a factor of six. At a lower speed of 1830 rpm, similar results were obtained (refer to Figs. 6 and 12). These results indicate an interesting relation between the effective viscosity of the EHL oil film (viscosity of lubricating oil under high pressure) and the viscosity at atmospheric pressure.

There was an estimation which indicates that there is a higher limit of traction coefficient μ_{max} when the contact pressure is increased. However, it was found from the present experiments that the value of μ_{max} increases when the contact pressure is increased up to the higher values at which no experiments have been conducted in earlier investigations. For example, $\mu_{max} \approx 0.027$ when $p_{max} = 0.82$ GPa, $\mu_{max} \approx 0.058$ when $p_{max} = 2.4$ GPa, and $\mu_{max} \approx 0.06$ when $p_{max} = 2.6$ GPa (refer to Fig. 6). The value of specific sliding at which the maximum traction coefficient appears is $|\epsilon_f| \approx 5\%$ at $p_{max} = 0.82$ GPa, while it decreases to $|\epsilon_f| \approx 0.5\%$ when p_{max} is increased to 2.6 GPa.

4.1 Shear stress of EHL oil film. The mean shear stress of the EHL oil film of the oil B is shown in Fig. 8, which is calculated using the width of Hertzian contact band $2b$ and the measured traction force between the

Fig. 4 Equivalent friction coefficient for compensating losses at test roller bearings, etc.

Fig. 5 Traction coefficients of test oil A (at 40°C and 3440 rpm)

Fig. 6 Traction coefficients of test oil B (at 40°C and 1830 rpm)

Fig. 7 Traction coefficients of test oil B (at 40°C and 3440 rpm)

rollers. The shear stress is as high as $\bar{\tau} \approx 120$ MPa (12.2 kgf/mm²) at a high pressure of $p_{\max} = 2.6$ GPa ($p_{\text{mean}} = 2.04$ GPa = 208 kgf/mm²). This result indicates that the shear flow stress of EHL oil film at the high pressure (about 20000 times the atmospheric pressure) is nearly equal to the shear flow stress of soft metals under atmospheric pressure.

It is generally accepted that the viscosity of lubricating oil becomes higher when the pressure is increased. The viscosity of oil under high pressure is calculated using the equation $\eta = \eta_0 \exp(\alpha p)$, (indicating the relation between the pressure and viscosity and has been used in the calculation of the theoretical oil film thickness based on the elasto-hydrodynamic lubrication theory). The calculated value of the shear stress of the EHL oil film thickness becomes appreciably greater than that of steel (more than one million times). Therefore, the applicability of the equation must be re-examined. An impossibly high shear stress has been obtained at comparatively low pressures⁽⁸⁾. Many investigators have been working to solve the discrepancy between the theory and experimental results, but no fine theory has been established.

Figure 9 shows the mean shear stress $\bar{\tau}$ of the EHL oil film expressed in terms of the shear strain $(U_1 - U_2)/h_c$. The two lines in the upper part of the figure indicate the

Fig. 8 Relation between sliding speed and shear stress in EHL oil film

Fig. 9 Relation between strain rate and shear stress in EHL oil film

calculated shear stress under the assumption that the distribution of the oil pressures is elliptic in the contact region and also that $\eta = \eta_0 \exp(\alpha p)$ is applicable to the small part of oil film with a pressure of $p = p_{\max} \sqrt{1 - (x/b)^2}$.

$$\bar{\tau} = \bar{\eta} \frac{(U_1 - U_2)}{h_c} = \left[\frac{1}{2b} \int_0^b \eta_0 \exp(\alpha p_{\max} \sqrt{1 - (x/b)^2}) dx \right] \frac{(U_1 - U_2)}{h_c} \quad (2)$$

The effect of temperature upon the shear stress is small as understood from Fig. 9. The mean shear stress estimated using the aforementioned assumption is higher than 10^{18} MPa even at low shear strains for both oils A and B*. In contrast to this, the mean shear stress obtained from experiments is in the range of $\bar{\tau} \approx 50$ to 120 MPa.

The mean effective viscosity $\bar{\eta}_e$ in the contact region is calculated using the traction force obtained in experiments and shown in Fig. 10.

$$\bar{\eta}_e = \frac{h_c T}{2bB(U_1 - U_2)} \quad (3)$$

The values of the effective viscosity $\bar{\eta}_e$ are shown by the marks $\bullet$, $\blacksquare$, $\blacktriangle$ and $\blacktriangledown$ when a supply oil temperature of 40°C is used in the calculation of the oil film thickness h_c . In the case when the temperature on the roller surface is used as the oil temperature, changes in the viscosity $\bar{\eta}_e$ are shown by the marks $\circ$, $\square$, $\triangle$ and $\triangledown$. It is understood from Fig. 10 that the effective viscosity is mainly decided by the shear strain in the EHL oil film.

It should be noted that the estimated viscosity in the contact region is an impossibly high value of $\bar{\eta}_e = 10^{20}$ Pa·s even at

Fig. 10 Mean effective viscosity in contact region

* It is necessary to introduce a mechanism which can explain the reduction in the shear flow stress of the EHL oil film without decreasing the thickness of the oil film because the theoretical oil film thickness calculated using the relation $\eta = \eta_0 \exp(\alpha p)$ agrees very well with that obtained by experiments. The authors estimate that it is necessary to introduce the inhomogeneity caused by the orientation of molecules of lubricating oil in the contact region.

a low shear strain of $(U_1 - U_2)/hc \neq 10^4$ when the relation $\eta = \eta_0 \exp(\alpha p)$ is assumed in the calculation.

5. Equations for traction coefficient

It is impossible to obtain experimentally the traction coefficient at higher pressure than $p_{\text{mean}} = 1.5 \text{ GPa}$ ($p_{\text{max}} = 1.9 \text{ GPa} = 195 \text{ kgf/mm}^2$). The authors succeeded in estimating the traction coefficients at the high pressures using the experimental results obtained in the low pressure and medium pressure regions, in which the traction coefficients can be easily obtained in experiments.

5.1 Basic equation for traction coefficient. When the many experimental results⁽⁹⁾ are examined, the shape of traction-coefficient curves can be shown schematically by a curve in Fig. 11. In the low sliding speed region, the traction coefficient is proportional to the sliding speed $U_s = (U_1 - U_2)$, and can be shown by Eq. (4).

$$\frac{T}{P} = \frac{\bar{\eta}_s U_s}{p_{\text{mean}} h c} = A \bar{U}_s \quad (4)$$

where, $\bar{U}_s$ is a dimensionless sliding speed expressed by U_s/U_0 . For the sake of simplicity, $U_0 = 1 \text{ m/s}$ is assumed in the following numerical calculation. A is a dimensionless constant.

The authors found that the traction coefficient at higher speeds than $\bar{U}_L$ can be expressed accurately by the following equation.

$$\frac{T}{P} = L[K(\bar{U}_s - \bar{U}_{cr})]^{(1 - [N(\bar{U}_s - \bar{U}_{cr})]^M)} + \mu_0 \quad (5)$$

By analyzing the experimental results obtained by the authors and the earlier investigators⁽⁴⁾⁽⁵⁾, the authors found that the unknown values A , L , K , M , N and μ_0 in Eqs. (4) and (5) can be expressed exactly by the dimensionless Hertzian pressure $\bar{x} = p_{\text{max},x}/p_{\text{max},0}$ [Eq. (6)], where $p_{\text{max},0}$ is the reference Hertzian pressure at which the traction coefficients can be easily obtained in experiments.

$$\left. \begin{aligned} A &= A_1(\bar{x} - A_2)^{A_3} \\ L &= L_1 \bar{x}^{L_2} - L_3 \\ K &= K_1 \bar{x}(K_2 - \bar{x}^{K_3}) \approx K'_1 \bar{x}^{K'_2} \\ N &= N_1 \bar{x}^{N_2} \\ M &= M_1 + M_2 \bar{x}^{-1} \\ \mu_0 &= \mu_1(\bar{x} - \mu_2)^{\mu_3} \end{aligned} \right\} \quad (6)$$

Fig. 11 Schematic drawing of traction coefficient curve and main points on the curve

In the above equation, the symbols with suffixes (A_1, L_1 , etc.) indicate a dimensionless constant which can be decided by experimental results obtained in the lower pressure region in which the traction coefficients can be measured easily.

5.2 Example of equations for traction coefficient. The unknown factors in Eqs. (4) and (5) are decided by experiments, and a numerical example of the equations for traction coefficients is shown in the following two cases (a) and (b).

(a) For a comparatively low pressure at which experiments are easy to conduct (Oil A at 40°C , $n_1 = 1830 \text{ rpm}$ and $p_{\text{max}} = 1.3 \text{ GPa}$) For low speed range ($0 < \bar{U}_s \leq 0.0121$),

$$\frac{T}{P} = 1.85 \bar{U}_s \quad (7)$$

For medium and high speed ranges ($\bar{U}_s \geq 0.0121$)

$$\frac{T}{P} = 0.0155[20(\bar{U}_s - 0.0115)]^{(1 - [2.5(\bar{U}_s - 0.0115)]^{0.12})} + 0.0210 \quad (7')$$

(b) For high pressure at which experiments are impossible to conduct (refer to Section 5.5) (Oil A at 40°C , $n_1 = 1830 \text{ rpm}$ and $p_{\text{max}} = 2.4 \text{ GPa}$) For low speed range ($0 < \bar{U}_s \leq 0.0144$),

$$\frac{T}{P} = 2.79 \bar{U}_s \quad (8)$$

For medium and high speed ranges ($\bar{U}_s \geq 0.0144$)

$$\frac{T}{P} = 0.028[73(\bar{U}_s - 0.0118)]^{(1 - [5.9(\bar{U}_s - 0.0118)]^{0.10})} + 0.0244 \quad (8')$$

Similar equations to the above are obtained for Hertzian pressures of 1.0, 2.0 and 2.6 GPa. Numerical results obtained from these equations are indicated in Fig. 12 by full lines. The experimental results are indicated by the marks $\circ$, $\diamond$, $\square$, $\triangle$ and ∇ . The unknown factors A , L , K , M , N and μ_0 in Eq. (6) can be expressed approximately by Eq. (9) in terms of a dimensionless Hertzian pressure $\bar{x} = p_{\text{max}}/1.3$.

$$\left. \begin{aligned} A &\approx 2.60(\bar{x} - 0.62)^{0.35} \\ L &\approx 0.0875 \bar{x}^{0.22} - 0.0720 \\ K &\approx 20.0 \bar{x}(3.6 - \bar{x}^{0.45}) \\ N &\approx 2.5 \bar{x}^{1.4} \\ M &\approx 0.08 + 0.04 \bar{x}^{-1} \\ \mu_0 &\approx 0.023(\bar{x} - 0.40)^{0.17} \end{aligned} \right\} \quad (9)$$

5.3 Effect of oil temperature upon traction coefficient. Changes in unknown factors in the basic equations [Eqs. (4) and (5)] are examined when the temperature of lubricating oil is varied. When the temperature of the oil A is increased from 40° to 70°C , the values A , L , K , N , M and μ_0 change as shown in Fig. 13. It was found that the values A , K and L change with oil temperature, while the values N , M and μ_0 are affected by pressure only.

Figure 14 shows a reduction in the traction coefficients when the temperature of oil

is increased from 40° to 70°C. The reduction is remarkable when the sliding speed and/or the pressure are low. This result suggests that the maximum power transmission capacity of a traction drive is affected easily by the changes in temperature of traction oil when the traction drive is operated at a comparatively low contact pressure.

5.4 Effect of speed upon traction coefficient. When the rolling speed is increased, the traction coefficient becomes smaller under the same contact pressure. Details of the numerical equations for the traction coefficient are omitted. Changes in the unknown factors in Eqs. (4) and (5) are as follows: When the peripheral speed of driving roller is increased from $U_1 = 6.52$ to 12.3 m/s, the values of L , M and N are affected only by the pressure in the case of the oil B. Changes in the other factors such as A , K and μ_0 are shown in Fig. 15. When the rotational speeds are increased, these three factors become smaller, resulting in smaller traction coefficients.

Figure 16 shows the effect of peripheral speed of roller upon the traction coefficient. The effect of speed is remarkable even at a very high pressure of $p_{\max} = 2.6$ GPa, at which the authors conducted the measurement of

traction coefficients, and also at a conventional pressure of $p_{\max} = 1.3$ GPa, up to which earlier experiments were conducted.

5.5 Effect of viscosity at atmospheric pressure upon traction coefficient. The viscosity of the oil B at a temperature of 40°C is about six times that of the oil A under atmospheric pressure when the temperature is the same. These two oils manifest almost the same traction characteristics when the pressure is not too high (up to about 1.3 GPa). However, when the pressure is very high (at $p_{\max} = 2.6$ GPa), the traction coefficient seems to tend to a certain value lower than μ_0 in the case of the oil B having a higher viscosity at atmospheric pressure (refer to Fig. 17). Examining the authors' equations derived for calculating the traction coefficients, it is found that the values of μ_0 are the same for the two kinds of oils A and B. The numerical equations for $p_{\max} = 2.6$ GPa are shown in the following two cases (a) and (b).

Fig. 12 Traction coefficient curves calculated from Eqs. (4) and (5)

Fig. 13 Changes in unknown factors in Eqs. (4) and (5); (for test oil A at 1830 rpm)

Fig. 14 Reduction in traction coefficients caused by increase in temperature of test oil

Fig. 15 Effects of peripheral speed on unknown factors in Eqs. (4) and (5) (for test oil B)

Fig. 16 Effects of peripheral speed on traction coefficient (for test oil B)

- (a) For oil A at 40°C and $n_1 = 1830$ rpm
When $(0 < \bar{U}_s \leq 0.0148)$,

$$\frac{T}{P} = 2.91 \bar{U}_s$$

When $(\bar{U}_s \geq 0.0148)$,

$$\frac{T}{P} = 0.030[82(\bar{U}_s - 0.0121)]^{11} - [6.6(\bar{U}_s - 0.0121)]^{10.81} + 0.0249$$

- (b) For oil B at 40°C, $n_1 = 1830$ rpm
When $(0 < \bar{U}_s \leq 0.0177)$,

$$\frac{T}{P} = 2.91 \bar{U}_s$$

When $(\bar{U}_s \geq 0.0177)$,

$$\frac{T}{P} = 0.030[240(\bar{U}_s - 0.0150)]^{11} - [12(\bar{U}_s - 0.0150)]^{10.81} + 0.0249$$

The numerical results obtained from the above equations are shown by the two curves in the upper part of Fig. 17. When the above equations are compared with Eqs. (4) and (5), it is found that the values of A, L and μ_0 of the oil B are the same as those of the oil A.

6. Application of investigated results

6.1 Estimation of traction coefficient under extremely high pressure. It is comparatively easy to measure the traction coefficient of the EHL oil film up to a mean pressure of $p_{\text{mean}} = 1.3 \text{ GPa}$ ($p_{\text{max}} = 1.6 \text{ GPa} = 169 \text{ kgf/mm}^2$). However, it will be impossible in the near future to measure the traction coefficient at higher pressures than $p_{\text{max}} = 2.6 \text{ GPa}$ ($p_{\text{mean}} = 2.04 \text{ GPa}$), at which the traction coefficients were measured in the present investigation.

The authors found that the traction coefficients at an extremely high pressure, which are hardly obtained by experiment, can be estimated from the empirical equations and the experimental results which can easily be obtained by experiments conducted under comparatively low pressure. For example, the unknown factors in the basic equations, Eqs. (4) and (5), can be decided by extrapolating the experimental results with the help of Eq. (9) and shown by broken lines in Fig. 13 for the oil A at 40°C and $n_1 = 1830$ rpm. The numerical values of the unknown factors A, L, K, M, N and μ_0 at higher pressures can be obtained by the extrapolation.

Fig. 17 Effects of kind of test oil upon traction coefficient

When these values are introduced into Eqs. (4) and (5), the estimated traction-coefficient curves are obtained for the higher pressures. The calculated results for higher pressures of $p_{\text{max}} = 3.0 \text{ GPa}$ ($p_{\text{mean}} = 2.4 \text{ GPa} = 245 \text{ kgf/mm}^2$) and 3.5 GPa ($p_{\text{mean}} = 2.7 \text{ GPa} = 280 \text{ kgf/mm}^2$) are indicated in Fig. 12 by broken lines.

6.2 Analysis of experimental results obtained by other investigators. K.L. Johnson, et al. measured the traction coefficients at contact pressures up to $p_{\text{max}} = 1.54 \text{ GPa}$ ($p_{\text{mean}} = 1.21 \text{ GPa}$)⁽⁴⁾. Their experimental results are very famous and have been quoted in many articles because they are accurate and very valuable.

The authors analyzed the experimental results obtained by K.L. Johnson, et al. and decided the unknown factors in Eqs. (4) and (5). The traction coefficient curves obtained by the authors' analysis are shown in Fig. 18 using full lines. The experimental results are indicated by the marks O, □, △, ◇, ● and x. The broken lines in Fig. 18 indicate the values estimated from the authors' equations for higher pressures at which no experiments were conducted by them.

Figure 19 shows the numerical values of

Fig. 18 Traction coefficient curves calculated from the authors' equations and experimental results by K.L. Johnson, et al.

Fig. 19 Numerical values of unknown factors obtained by analyzing experimental results of K.L. Johnson, et al.

the unknown factors which were obtained by analyzing the experimental results obtained by K.L. Johnson, et al. The broken lines in the figure indicate the extrapolated values for the unknown factors.

Trachman, et al. measured the traction coefficients of synthetic oil at pressures up to $p_{\max} = 1.72$ GPa, and they also calculated the traction coefficients at pressures of $p_{\max} = 0.80$ and 1.38 GPa⁽⁵⁾. The authors analyzed the results of Trachman, et al. with the help of Eqs. (4) and (5), and obtained equations which can express the traction coefficients. The calculated results are indicated in Fig. 20 by full lines. It is found that the experimental results obtained under four different pressures (indicated by the marks Δ , $\square$, $\circ$ and $+$) can be expressed accurately by the authors' equations. The results calculated theoretically by Trachman, et al. can not be expressed so accurately as the results calculated using the authors' empirical equations. They did not present the equations for calculating the traction coefficients. For reference, the authors' equations for $p_{\max} = 1.72$ GPa are as follows.

$$\text{For low speed range } (0 < \bar{U}_s \leq 0.0232) \\ \frac{T}{P} = 1.30 \bar{U}_s \dots\dots\dots (10)$$

For medium and high speed ranges
($\bar{U}_s \geq 0.0232$)

$$\frac{T}{P} = 0.016[30(\bar{U}_s - 0.0210)]^{11} - [2.0(\bar{U}_s - 0.0210)]^{14} \\ + 0.0255 \dots\dots\dots (10')$$

Traction curves estimated from the authors' equations for $p_{\max} = 2.1$ and 2.5 GPa, at which Trachman, et al. could not measure the traction coefficients, are given in Fig. 20 as broken lines.

7. Discussion

The recent investigations about the estimation of the traction coefficients are in such a tendency that the experimental results about the basic characteristics of test oils such as the viscosity at high pressures, the limiting shear stress, etc. are used in the equations for estimating the traction coefficients⁽⁵⁾⁽⁹⁾⁽¹⁰⁾. However, the results of these investigations can not predict the traction coefficients so accurately as the

authors' equations when the contact pressure is as high as in the case of present investigation. This may be ascribed to the fact that the basic characteristics of the oils are hardly obtained at a high pressure and also at a high rate of pressure increase such as those observed in rolling contact. In contrast to this, the authors used the results of experiments conducted at a comparatively low pressure at which the measurement of traction coefficients is easy, and derived empirical equations which make it possible to estimate accurately the traction coefficients at high pressures at which experimental values of the traction coefficients are hardly obtained.

8. Conclusions

(1) Using a rolling contact testing machine designed and made by them, the authors measured the traction coefficients of lubricating oils at high pressures in the range of $p_{\text{mean}} = 1.6$ to 2.0 GPa ($p_{\max} = 2.0 - 2.6$ GPa = $204 - 265$ kgf/mm²), at which no experimental results had been presented in earlier investigations.

(2) Empirical equations were derived, which can express accurately the traction coefficients obtained by experiments. These equations make it possible to estimate numerically the traction coefficients at high pressures up to $p_{\text{mean}} = 2.7$ GPa ($p_{\max} = 3.5$ GPa = 375 kgf/mm²).

(3) Experimental results about the traction coefficients at pressures up to $p_{\max} = 1.7$ GPa, which were presented in America and Britain, could be expressed accurately by the authors' equations.

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Fig. 20 Traction coefficient curves calculated from the authors' equations and experimental results by Trachman, et al.