Friction of Sliding Leadscrews under Hydrodynamic Lubrication*

(2nd Report, Verification of Drunkenness-Model Theory and Consideration of Appropriate Driving Condition)

Shigeo FUKADA** and Jiro OTSUKA***

The theory proposed in the previous report is verified experimentally. The proposed theory is based on the clearance shape model with respect to the drunkenness error of an internal thread, and the frictional property of a sliding leadscrew is described under the hydrodynamic condition by some dimensionless numbers derived from the infinitely short bearing theory. To verify the proposed theory, some experiments are performed on several specimens which have different nominal dimensions. The experimental results are arranged according to dimensionless numbers S and M and are compared with the theoretical results. The experimental results agree with the theoretical results in the range of the hydrodynamic condition; thus, the appropriateness of the proposed theory is verified. The appropriate driving condition is considered based on the theory and experimental results, and the selection criterion for oil viscosity is shown.

Key Words: Machine Element, Friction, Lubrication, Sliding Leadscrew, Trapezoidal Screw, Driving Condition, Oil Viscosity

1. Introduction

It has been confirmed experimentally that a screw shaft and a nut are separated by an oil film to meet the hydrodynamic condition when they are driven under oil lubrication and light load at high rotational speed^{(1),(2)}. In the case of sliding bearings used in general, the generation of hydrodynamic pressure to realize fluid lubrication is due to the wedge shape of the oil film, and the frictional properties of the sliding bearings are generally analyzed using Reynolds equations. For example, in the case of journal bearings, the condition of eccentricity between the shaft and the journal is formulated geometrically, so that some dimensionless numbers, such as the Sommerfeld number, are derived from the hydrodynamic theory to

describe the frictional properties based on the relationship among the numbers. On the other hand, in the case of sliding leadscrews, geometrical factors contributing to the generation of the hydrodynamic pressure between the screw shaft and the nut have never been established; thus, the general theory to estimate the frictional properties of leadscrews under hydrodynamic lubrication has not been established.

This study is aimed at clarifying the frictional properties of sliding leadscrews under hydrodynamic lubrication both theoretically and experimentally. In the first report⁽³⁾, the clearance shape model between the screw shaft and the nut is established based on some assumptions with respect to the cyclic drunkenness error of the nut lead. Furthermore, the infinitely short bearing theory is applied to the clearance shape model. Some dimensionless numbers are derived to generally describe the hydrodynamic friction, while their theoretical values are obtained. In this study, friction experiments are performed on some leadscrew specimens with different nominal dimensions in order to verify the lubrication theory based on the drunkenness model derived in the previous report. Furthermore, on the basis of theoretical and

* Received 26th February, 1999. Japanese original: Trans. Jpn. Soc. Mech. Eng., Vol. 64, No. 623, C (1998), p. 2674-2675 (Received 18th July, 1997)

** Faculty of Engineering, Shinshu University, 500 Wakasato, Nagano 380-8553, Japan. E-mail: sfukada@gipwc.shinshu-u.ac.jp

*** Shizuoka Institute of Science and Technology, 2200-2 Toyosawa, Fukuroi 437-8555, Japan

experimental results, the appropriate driving condition for sliding leadscrews is discussed to propose a selection guide for oil viscosity.

2. Theory

In this section, the theoretical results in the previous report are summarized. A leadscrew with nominal pitch diameter d_p and pitch P is considered. The difference in virtual pitch diameter between the screw shaft and the nut is c_p (virtual pitch diameter clearance), while the internal thread of the nut has a drunkenness error with amplitude A_x . When the leadscrew is driven under hydrodynamic lubrication with oil of viscosity η , the frictional property is described by the relation between the following dimensionless numbers S and M .

$$\left. \begin{aligned} S &= \left(\frac{d_p}{c_p} \right)^2 \left(\frac{P}{d_p} \right)^2 \left(\frac{n}{m} \right) \frac{\eta N}{p_0} \\ M &= \left(\frac{d_p}{c_p} \right) \left(\frac{P}{d_p} \right)^2 \left(\frac{n}{m} \right) \mu \end{aligned} \right\} \quad (1)$$

where m is the number of threads in the nut, n is the number of drunkenness cycles in the nut (ordinary $m=n$), N is the rotational speed of the screw shaft, p_0 is the mean contact pressure between the screw shaft and the nut, and μ is the coefficient of friction. The theoretical values of S and M in Eq. (1) are obtained by the following equations.

$$\left. \begin{aligned} S &= \frac{\sin^2 \alpha' \cos^2 \alpha'}{C_w} \\ M &= \frac{2 \sin \alpha' \cos^2 \alpha'}{\cos \beta} \cdot \frac{C_f}{C_w} \end{aligned} \right\} \quad (2)$$

where α' is the half angle of the thread in the cross section perpendicular to the thread ridge, and β is the lead angle. The values of C_f and C_w are determined by following equations.

$$\left. \begin{aligned} C_w &= \frac{\epsilon}{(1-\epsilon)^2} - \frac{\epsilon(1+2\lambda)}{\{(1+2\lambda+\epsilon)(1+2\lambda-\epsilon)\}^2} \\ C_f &= \frac{\pi}{\sqrt{(1-\epsilon)(1-\epsilon+2\lambda)}} + \frac{\pi}{\sqrt{(1+\epsilon)(1+\epsilon+2\lambda)}} \end{aligned} \right\} \quad (3)$$

where λ is the ratio of the amplitude of drunkenness error in the internal thread to the clearance between the screw shaft and the nut, as defined by the following equation.

$$\lambda = \frac{2A_x}{c_p \tan \alpha} = \frac{A_x}{c_x} \quad (4)$$

where c_x is one-half of the backlash. ϵ in Eq. (3) means the rate of deviation between the ridge of external thread and the groove of internal thread and is related to the minimum oil film thickness h_{2e} at the pressure flank as follows.

$$(1-\epsilon) = \frac{h_{2e}}{c_p} \cdot \frac{2}{\sin \alpha'} \quad (5)$$

Eliminating C_w from Eq. (2), the following equation is obtained.

$$M = \frac{2}{\sin \alpha' \cos \beta} C_f \cdot S \quad (6)$$

Furthermore, letting $\lambda = \epsilon = 0$, then

$$M = \frac{4\pi}{\sin \alpha' \cos \beta} S. \quad (7)$$

Equation (7) holds under the condition that

- (1) $\lambda=0$, i.e., flanks of external thread and internal thread have a flat planar shape (parallel plane model),
- (2) $\epsilon=0$, i.e., clearances at pressure flank and clearance flank are equivalent to each other (neutral condition).

The situation satisfying both of the above conditions is called the neutral plane condition. Equation (7) corresponds to Petroff's equation for journal bearings.

Rearranging Eqs. (6) and (3) with respect to ϵ under the condition of $\lambda=0$, the following equation is obtained.

$$\epsilon|_{\lambda=0} = \sqrt{1 - \frac{4\pi}{\sin \alpha' \cos \beta} \cdot \frac{S}{M}} \equiv \underline{\epsilon} \quad (8)$$

$\underline{\epsilon}$ determined by Eq. (8) from S and M is called the equivalent ratio of deviation. $\underline{\epsilon}$ means equivalent ratio of deviation in the parallel plane model where only the friction force is equivalent to theoretical estimations or experimental results. The equivalent ratio of deviation, $\underline{\epsilon}$, is related to the thickness of the oil film of the parallel plane model (equivalent oil film thickness⁽¹⁾), h_e , as follows.

$$(1-\underline{\epsilon}) = \frac{h_e}{c_p} \cdot \frac{2}{\sin \alpha'} \quad (9)$$

3. Leadscrew Specimens

Figure 1 and Table 1 show the dimensions and specifications of leadscrew specimens. Three kinds of screw shafts (A, B and C) are used. A and B differ in pitch, while B and C differ in major diameter. Furthermore, three nuts with different lengths are engaged to the screw shaft A. Namely, there are five combinations of screw shafts and nuts, as shown in Table 1. The material of the screw shafts is Cr-Mo steel, which is carburized and heat-treated (to achieve a surface hardness of 800 Hv). The external

Fig. 1 Dimensions of specimen

Table 1 Specifications of specimens

Screw thread	Label	A			B	C
	Nominal designation	Tr50x8			Tr50x5	Tr30x5
	d mm	50			50	30
	P mm	8			5	5
	d ₁ mm	41			44.5	24.5
	D mm	51			50.5	30.5
	D ₁ mm	42			45	25
	Nominal pitch diameter d _p mm	46			47.5	27.5
Screw shaft	P/d _p	0.17			0.11	0.18
	H mm	820			820	700
	Actual pitch diameter d _{pr} mm	45.901			47.501	27.490
	Overall lead deviation μm	-12.6			-20.4	-11.9
Nut	Surface roughness R _{max} μm	2.8			1.9	4.3
	Label	A5	A10	A15	B10	C10
	Number of threads	5	10	15	10	10
	I mm	40	80	120	50	50
	K mm	68			68	50
	Actual pitch diameter D _{pr} mm	45.968	45.971	45.983	47.609	27.538
	Drunkeness 2A _x μm	3	4	(4)	11	3
	Surface roughness R _{max} μm	3.3	3.5	4.0	4.0	3.5
Engagement	c _p (=D _{pr} -d _{pr}) mm	0.067	0.070	0.082	0.108	0.048
	Backlash 2c _x mm	0.018	0.019	0.021	0.029	0.013
	λ (=A _x /c _x)	0.17	0.21	(0.19)	0.38	0.23
	d _p /c _p	686	657	561	440	573

threads of the screw shafts are finished by lapping after thread grinding. The nut material is bronze casting, BC6. The internal thread of nut C10 is finished by lapping after tapping, while other nuts are finished by lapping after turning. All nuts are driven under a constant condition (mean contact pressure of 780 kPa) for running-in before the experiments.

Figure 2 shows lead errors of the screw shafts and the nuts. Figure 2(a) shows drunkenness errors of the screw shafts measured by a lead measuring machine with a laser interferometer, where the linear inclination of pitch error is eliminated. The measurement of drunkenness includes some effects of shaft bending in general, so that the measurement results prove that the drunkenness of the screw shafts is less than 2 μm . The nuts are measured after the experiments by a three-dimensional coordinate measuring machine (Zeiss UMM500+RT5). The results of both pressure flanks and clearance flanks are shown. Each lead error of the nuts is composed of a linear inclination over the entire length and significant drunkenness error whose period corresponds to one lead of screw. Nut B10 has especially large drunkenness. The lead errors of the nuts, as shown in the figure, are not intentionally given but are residue remaining in the machining process. Values of λ in Table 1 are

Fig. 2 Lead error of specimens

calculated using Eq. (4) after reading the amplitude of drunkenness from Fig. 2. Nut A15 is too long to be measured because of the restriction in the probe length. Its drunkenness errors are estimated to be equivalent to those of nut A10, because nuts A5 and A10 have a similar error tendency when machined simultaneously with A15.

4. Experimental Procedure

Figure 3 shows the experimental setup. The nuts travel right and left with the forward and reverse rotation of the screw shaft set horizontally. A torque measuring mechanism restraining the nut rotation and an oil bath are devised on the table driven by another feed screw. Figure 4 shows the preloading mechanism and the torque measuring mechanism. Internal loading is used. Two nuts are installed facing each other. A preload is applied between the nuts by coil springs. The left nut in Fig. 4(a) is a specimen for measurement and the right one is a backup nut for loading. A thrust bearing and an insulator are inserted between the nuts, so that the torque of each nut can be measured independently, and both nuts are electrically insulated.

The nut specimen in Fig. 4 alternates its motion, such as movement against the axial load and movement due to pushing back by the load. In this study, the first motion is called an upward operation, and the second motion is called a downward operation. The torque is sampled 200 times (10 times per rotation) while the screw shaft makes 20 rotations in the region near the center of the nut stroke where the rotational speed becomes constant. The average of 200 data is defined as the driving torque under the given condition. The apparent coefficient of friction, μ , is given by following equation.

$$\mu = \frac{\pm 2T \mp Fd_p \tan \beta}{2T \tan \beta \mp Fd_p} \cos \alpha', \quad (10)$$

where F is the axial load, d_p is the pitch diameter, α' is the half angle of the thread of screw, and β is the lead angle of the screw. (top sign represents the upward operation, and bottom sign represents the

downward operation)

In order to electrically measure the contact status between the screw shaft and the nut, a constant voltage is applied between the screw shaft and the nut through a slip ring. The voltage between the screw shaft and the nut when they are completely separated is defined as applied voltage $E_v (=150 \text{ mV})^{(4)}$. The voltage for driving between the screw shaft and the nut is defined as the separation voltage E . The rate of separation is defined by their ratio, $\tau = E/E_v (0 \leq \tau \leq 1)$. After rectifying the separation voltage with a low-pass filter, the voltage is sampled the same number of times in the same region as the driving torque, and the average value is calculated.

Three types of non-additive machine oils, VG9, 68 and 680, as shown in Table 2, are used. An oil bath is installed on the table so that nuts are immersed in the oil for experiments. A thermistor is inserted into the nut flange in order to measure temperature near the frictional interface. The coefficient of oil viscosity at the frictional interface is estimated using Walther's equation⁽⁵⁾.

Fig. 3 Experimental setup

Fig. 4 Details of experimental apparatus

Table 2 Specifications of lubricants

ISO VG	9	68	680
Specific gravity 15/4°C	0.880	0.903	0.962
Kinematic at 40°C	8.47	67.1	705.4
viscosity cst 100°C	2.26	7.16	21.50

5. Experimental Results and Discussion

5.1 Basic property of M - S relation and effect of operational direction

Nut A10 is engaged to screw shaft A under a constant mean contact pressure of $p_0=85$ kPa, and the screw shaft is driven at various rotational speeds between 6 and 500 rpm (0.1–8.3 rps) under lubrication with VG 9, 68 and 680. The S value under each condition is calculated from Eq.(1). The coefficient of friction, μ , is given by the measured driving torque and Eq.(10); then, the M value is calculated using Eq.(1). Figure 5(a) shows the obtained M - S relation. Figure 5(b) shows the rate of separation. In both upward and downward operations, M initially decreases as S increases. Next, M increases along a straight line after taking a minimum value. The property is similar to the Stribeck curve of sliding bearings. Near the minimum point ($S \approx 0.01$), τ starts increasing. The rate of separation approaches 1 in the region where $S > 0.1$, so that the hydrodynamic condition is achieved.

In Fig. 5(a), the theoretical results of M - S relations ($\lambda=0.05-0.4$) and a curve which represents the neutral plane condition of Eq.(7) are also shown. The experimental results agree with the theoretical results in the range where τ increases. In the range where $S < 0.01$, the lubrication is under a mixed condition. In this range, the friction includes solid friction, so that the hydrodynamic theory discussed in this study cannot be applied.

The equivalent rate of deviation, $\underline{\varepsilon}$, is obtained by substituting M and S of each plot in Fig. 5(a) into Eq.(8). Figure 5(c) shows the relationship between $(1-\underline{\varepsilon})$ and S . In the figure, the theoretical results of $(1-\underline{\varepsilon})$ - S relations given by Eqs.(2) and (3) are also shown. $\underline{\varepsilon}$ is given by assuming that the experimental S and M are produced by viscous friction of the parallel plane model ($\lambda=0$). It is noted that, in a strict sense, $\underline{\varepsilon}$ is different from ε under the practical condition of $\lambda \neq 0$. However, as shown in Fig. 5(c), the plots of $(1-\underline{\varepsilon})$ show good agreement with the theoretical curves of $\lambda \neq 0$ in the range where τ increases ($S > 0.01$). Therefore, $\underline{\varepsilon}$ is considered equivalent to ε , so that it is possible to estimate ε by $\underline{\varepsilon}$. This equivalence and the $(1-\underline{\varepsilon})$ - S relation are used for deriving the appropriate driving condition.

(a) M - S relation

(b) Rate of separation

(c) $(1-\varepsilon)$ - S relation

Fig. 5 Fundamental characteristics

The results of upward and downward operations are similar under the hydrodynamic condition, as shown in Fig. 5. The clearance shape model with respect to drunkenness of the internal thread, which is the fundamental model for the theoretical analysis in this study, exhibits a symmetrical configuration (a cosine curve), so that the operational direction does

Fig. 6 Effect of mean contact pressure

not change the theory. The actual lead errors of the nuts have not only the cyclic component but also linear inclination, so that the configuration is not symmetrical. However, the experimental results show no difference between the upward and downward operations. Therefore, the linear inclination of the lead error, as shown in Fig. 2, has no major influence on the hydrodynamic property. These results verify the assumption in the first report that the wedge shape at the cosine curve of drunkenness error rules the hydrodynamic pressure.

The experimental results of three kinds of lubrication oils with different viscosities lie on the same curve because η and N have the same effects on S in Eq. (1). Therefore, in order to avoid complication in the following graphs, the change of N is regarded as being equivalent with the change of η , so that the plots are not distinguished among the kinds of oils.

5.2 Effect of mean contact pressure

Similar experiments are performed under several mean contact pressures, and the results are shown in Fig. 6. The theoretical curves in Fig. 6 are the same as

Fig. 7 Effect of dimensions of specimens

in Fig. 5. S in Eq. (1) includes p_0 , so that p_0 is normalized in the relationships among dimensionless numbers ε , M and S . The experimental results under different contact pressures in Fig. 6 converge into the theoretical curve in the region of hydrodynamic lubrication where the theory is applicable (where the Stribeck curve rises). In the region of mixed lubrication, where the theory is not applicable, the plots of different contact pressures branch off as in the case of sliding bearings. M does not change under $p_0 = 780$ kPa. Under those conditions, hydrodynamic lubrication is not achieved, while τ does not increase.

5.3 In the case of leadscrews with different nominal dimensions

Experiments similar to those described in section 5.1 are performed on all specimens in Table 1 under $p_0 = 85$ kPa. Figure 7 shows the results. S and M in Eq. (1) include dimensional parameters d_p , P and c_p . Because the specimens have different dimensions, their S values are different even if their p_0 , N and η values are the same. Therefore, their minimum points of M are different. However, those results converge

into the same curve in the region where the Stribeck curves rise. The relationship among M , S and ε depend on λ . The specimens in Table 1 have λ of 0.17–0.38, so that the experimental M - S relations agree with the theoretical curves in Fig. 7 of $\lambda=0.05$ –0.4. These results prove that the lubrication theory on sliding leadscrews proposed in this study based on the infinitely short bearing theory is appropriate under these conditions.

By investigating the M - S relation of Fig. 7 in detail, it is found that the experimental values of M are larger than the theoretical curve in the range of $S=0.01$ –0.1. The reasons considered are as follows.

(1) The effect of higher order modes in the surface profile (such as surface roughness or higher order modes of lead error in Fig. 2)

(2) Though this theory assumes that the bodies are rigid, there is elastic deformation by means of axial load in practice. λ under the loaded condition is less than that without load.

Furthermore, this theory regards only the drunkenness error of internal thread, not the lead error of the screw shaft. Further consideration would be needed for the case in which the drunkenness of screw shaft is not negligible, and for the influences of the above (1) or (2), which are not assumed in this study.

The experimental M - S relation near $S=1$ of Fig. 7 levels off, so that the experimental values of M are smaller than the theoretical values. In the region where S is larger than 1, ε reaches the value of 0. The increase in the thickness of the oil film is not expected, and the oil viscosity decreases due to excessive heat generation, so that it is difficult to increase S even if N is increased further. These regions should be avoided in the practical use of sliding leadscrews.

6. Discussion on Appropriate Driving Condition

In the case of sliding leadscrews, the coefficient of friction under hydrodynamic lubrication is larger than 0.1. This value is ten or more times larger than those of sliding bearings. Therefore, it is not advisable to use sliding leadscrews under exceedingly large N or η for pursuing the hydrodynamic condition because this results in excess heat generation. The desirable condition for sliding leadscrews lies in the region where the coefficient of friction is as small as possible under the hydrodynamic condition, i.e., the minimum of the Stribeck curve.

The region near the minimum of the Stribeck curve represents the transitional state from mixed to hydrodynamic lubrication. The transition is ruled by the limit of the hydrodynamic condition, i.e., the lower limit of oil film thickness. Figure 8 shows the relation-

Fig. 8 μ as a function of h_e

ship between the coefficient of friction and the equivalent oil film thickness which is calculated by substituting the equivalent rate of deviation in Fig. 7 obtained from the experimental results into Eq. (9). It is considered that the minimum of μ in Fig. 8 corresponds to the limit of the hydrodynamic lubrication, so that h_e at the minimum of μ corresponds approximately to the lower limit of oil film thickness. Taking note of the minimum point of each specimen, the value of h_e where μ takes a minimum value is about $1 \mu\text{m}$ ^{*1}. Namely, the appropriate driving condition for these sliding leadscrews is the state where h_e takes a value of 1–2 μm , if the chosen condition is slightly more hydrodynamic than the minimum point.

If h_e is given, ε is determined by Eq. (9). Moreover, S and M are determined by ε which is considered equivalent to ε as described in section 5.1. The relationship among these parameters is shown schematically in Fig. 9. For example, if desired h_e is 2 μm , the value of h_e is substituted into Eq. (9) to obtain ε , which is considered as ε . Using the value of ε , S and M are determined by Eqs. (2) and (3), so that the values of $\eta N/p_0$ and μ are determined by substituting P , d_p , and c_p of the leadscrew into S and M . The appropriate coefficient of viscosity for the lubricant is determined by the value of $\eta N/p_0$ under a given rotational speed N and mean contact pressure p_0 .

In order to more simply determine the value of S at the desired driving condition, the relationship

^{*1} It is generally considered that the critical oil film thickness is ruled by surface roughness. The value of $1 \mu\text{m}$ is smaller than the surface roughness of the specimens in Table 1. It should be noted that h_e obtained here is the equivalent oil film thickness, which is different from the minimum oil film thickness in a strict sense.

Fig. 9 Relationship among parameters

Fig. 10 Approximation of $(1-\epsilon)$ - S relation

between $(1-\epsilon)$ and S is described by approximation. Figure 10 is obtained by drawing the $(1-\epsilon)$ - S relation of Fig. 7 in the logarithmic scale. The dot-dashed line in the figure represents the approximation of both plots under the hydrodynamic condition and theoretical curves in the range of $\lambda=0.05-0.4$. The approximation line is given by the following equation.

$$\log(1-\epsilon) = 0.425 \log S + 0.125 \quad (11)$$

From Eqs. (11), (1) and (9), the following equation is obtained.

$$\left. \begin{aligned} \frac{\eta N}{p_0} &= \left(\frac{c_p}{P} \right)^2 10^{G_1} \\ G_1 &= 2.352 \log \frac{2h_e}{c_p \sin \alpha'} - 0.2941 \end{aligned} \right\} \quad (12)$$

Equation (12) gives the bearing constant, $\eta N/p_0$, which is necessary to drive the leadscrew with pitch P and clearance at pitch diameter, c_p , under the condition that the oil film thickness is h_e . The unit of all dimensions is given in mm. By determining the value

Fig. 11 Selection of oil viscosity

of $h_e=0.002$ mm as the desired driving condition from the critical oil film thickness obtained above, the following equation is obtained from Eq. (12).

$$\frac{\eta N}{p_0} = \left(\frac{c_p}{P} \right)^2 10^{(-4.553 - 2.352 \log c_p)} \quad (13)$$

Taking nut A10 in Table 1 for example, the values of $c_p=0.07$ mm and $P=8$ mm are substituted into Eq. (13) to calculate the value of $\eta N/p_0=1 \times 10^{-6}$. The appropriate viscosity of the lubricant is determined by the following equation under the driving conditions of N and p_0 .

$$\eta = 1 \times 10^{-6} \cdot \frac{p_0}{N} \quad (14)$$

Figure 11 is obtained by calculating Eq. (14) to derive the oil viscosity at the rotational speed of the screw shaft. The specific gravity of the lubricant is assumed to be 0.9 in the figure. Figure 11 gives a selection of oil viscosities that are appropriate for the leadscrews used in the experiments.

This study shows the theory of the frictional properties under hydrodynamic lubrication and verifies it experimentally. This theory gives the frictional properties under any condition as long as the interface between the screw shaft and the nut is under hydrodynamic lubrication. If the desired oil film thickness is given, the driving condition to realize the desired oil film can be calculated as mentioned above. However, in the final consideration for the appropriate oil viscosity, the desired driving condition is determined on the basis of estimation that the critical oil film thickness is 1–2 μ m in the experimental results (Fig. 8). The applicability of Fig. 11 to other leadscrews has not been verified. In particular, in the case that the surface roughness is much different from these experiments, other considerations are necessary to determine the critical oil film thickness.

7. Conclusions

Some experiments are performed on leadscrew

specimens which have different nominal dimensions to verify the hydrodynamic theory of sliding leadscrews on the basis of the drunkenness model proposed in the first report. The following conclusions were reached.

(1) The experimental results are arranged according to dimensionless numbers S and M , and the equivalent rate of deviation is calculated. The experimental values are compared with the theoretical values. The experimental results under the hydrodynamic condition correspond to the theory, thus verifying the appropriateness of the theory.

(2) The region near the minimum of the friction coefficient is chosen to determine the desirable driving condition, and the equivalent oil film thickness is estimated where the coefficient of friction takes a minimum value. Moreover, the calculation process is shown to be able to derive the driving condition necessary to realize the desirable oil film thickness from this theory.

(3) For example, a selection of oil viscosities is

shown for the leadscrews in these experiments under a given rotational speed and mean contact pressure.

The authors would like to thank Mr. Toshiaki Ito at Tokyo Institute of Technology (at the time) for his cooperation in the experiments.

References

- (1) Fukada, S., Otsuka, J., Izuhara, A. and Tanigawa, H., Journal of Japanese Society of Tribologists, (in Japanese), Vol. 36, No. 12 (1991), p. 990-997.
- (2) Fukada, S., Otsuka, J. and Izuhara, A., Journal of Japanese Society of Tribologists, (in Japanese), Vol. 38, No. 6 (1993), p. 545-552.
- (3) Fukada, S., Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 64, No. 653, C (1998), p. 2666-2673.
- (4) Takeuchi, A., Sato, M. and Aoki, H., Journal of Japan Society of Lubrication Engineers, (in Japanese), Vol. 33, No. 1 (1988), p. 62-68.
- (5) Japan Society of Lubrication Engineers, Ed., Lubrication Handbook (Revised Edition), (in Japanese), (1987), p. 210, Yoken-do.