

Proposal and Verification of Friction Coefficient Calculation Model by True Contact and Oil Film (Evaluation of Piston Ring and Sleeve)

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ABSTRACT

The investigation of sliding parts takes into account the coefficient of friction of the parts. When the parts are in a relatively high-speed region, such as the piston rings and sleeves in an engine, sliding test performance itself is a challenge and the tribological properties cannot readily be determined. Therefore, surface roughness and waviness after initial familiarity were applied to a stochastic method proposed by Liu, et al., and the oil film thickness reduction coefficient of Gupta, et al., was used in high-speed regions. In addition, a new calculation model was proposed that takes into sliding, which becomes a determining factor in high-speed regions. As conventionally performed, sliding tests could not readily be implemented in high-speed regions. However, a newly developed drum cam test apparatus has made such testing possible to implement. The validity of this model was verified by comparison and the results of sliding tests up to 8 m/s. The results verified that the coefficient of friction in this model match very closely with the results of tests using ordinary sleeve material of cast gray iron (FC250) in high-speed regions up to 8 m/s. In addition, it has become possible to obtain the coefficient of friction in high-speed regions, which had not been possible to determine up to now. It has also become possible to predict the coefficient of friction in high-speed regions where the velocities are equal to or higher than in the verification tests.

1. Introduction

Computer-aided engineering (CAE) technology has now become a necessity on the front lines of engine development. Meanwhile, for the characteristics of sliding parts that are subject to friction and wear, the figures needed to perform CAE analysis are still being taken from test results and entered into the program, but there are cases when the characteristics are not for the speed ranges in actual use, or differ from the true contact pressures, and so on. This is often the case because testing to obtain the tribological properties cannot readily be implemented. Figure 1 shows the results of reciprocating sliding tests implemented in order to evaluate piston rings and sleeves.

The horizontal axis shows the peak values of sliding velocity, which changes sinusoidally as reciprocating movement occurs. The vertical axis shows the coefficient of friction, and this is the friction force, the changes of which form a rectangular wave, divided by the applied load. The coefficient of friction of the cast gray iron (FC250) used in the sleeve is higher than that of Nikasil coating at 1 m/

s, but this relationship changes drastically as the velocity approaches 2 m/s.

There are, however, no reports on tribological properties in high-speed regions up to 8 m/s, which is an engine's normal operating range. The challenges involved in

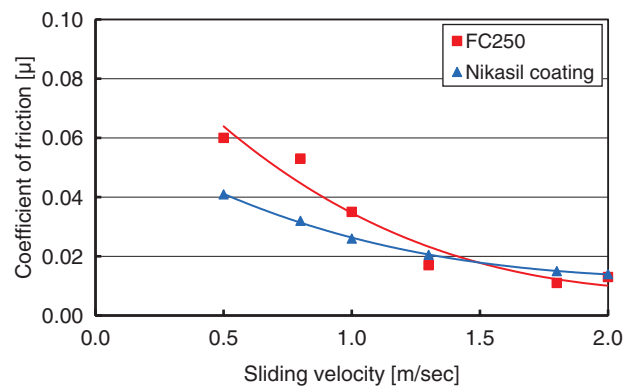


Fig. 1 Relationship between sliding velocity and coefficient of friction

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fabricating evaluation test equipment can be considered a factor in this. For the present research, a new model was created with the capability for calculating tribological properties in speed ranges where actual measurements cannot readily be made using a reciprocating test apparatus. An original high-speed reciprocating test apparatus was developed to conduct verification, and the results are reported herein.

2. Calculation Model for Determining Tribological Properties

2.1. Overview of Calculation Methods

Various researchers have carried out testing and analysis on the question of how the coefficient of friction differs with structures of differing kinds⁽¹⁾ as well as with roughness on machined surfaces⁽²⁾. The distinctive features of the present research lie in the following three points: (1) For speed regions that do not involve heat generation, calculations are performed by the stochastic method of Liu et al.⁽³⁾. (2) For the surface roughness and waviness used in calculation, actual measured values from sliding surfaces after the initial familiarity period following reciprocating sliding tests are used. (3) In high-speed ranges, a new calculation model is used that adds an original calculation taking into account the decrease in oil film thickness due to heat generation caused by sliding. Verification was carried out using these methods. Related calculation models are described below.

2.2. Division of Load Between Oil Film Portion and Direct Contact Portion

As shown in Fig. 2, the load applied to sliding parts is considered to be divided into two parts, the oil film portion and the convex portion of the sliding surface that has direct contact (true contact portion). When the total load applied between pin and plate is P , the contact load on the convex portion of asperity is P_a , and the load generated on the oil film portion is P_h , and their relationship is expressed by Eq. (1).

$$P = P_h + P_a \quad (1)$$

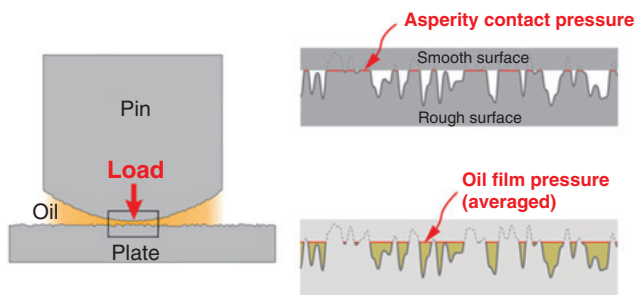


Fig. 2 Division of contact pressure by asperity and oil film

For the ratio of P_h and P_a in P differs according to the sliding conditions. These were, therefore, treated as the two variables γ_h and γ_a , which are the load fraction ratios as defined in Eq. (2), and determined by calculation.

$$\gamma_h = \frac{P_h}{P}, \gamma_a = \frac{P_a}{P} \quad (2)$$

2.3. Method for Deriving Tribological Properties of Oil Film Portion

In order to obtain the coefficient of friction of the oil film portion, the shear stress generated on the oil film is needed. Determining the shear stress requires first that the oil film thickness be determined.

The oil film thickness was determined using the approximation of oil film thickness in elastic fluid lubrication by Nijenhuis et al.⁽⁴⁾. Taking into account that the load P_h , that is applied to the oil film, takes a form in Eq. (2) that includes the variable γ_h , the dimensionless oil film thickness H_c is expressed by Eq. (3) and Eq. (4) below.

$$H_c = \gamma_h^{\frac{1}{2}} \left\{ \left[\gamma_h^{\frac{9}{4}} H_{RI}^{\frac{3}{2}} + \left(\gamma_h^{-\frac{2}{3}} H_{EI}^{-4} + H_{00}^{-4} \right)^{-\frac{3}{8}} \right]^{\frac{2}{3}s} + \left(\gamma_h^4 H_{RP}^{-8} + \gamma_h^4 H_{EP}^{-8} \right)^{-\frac{s}{8}} \right\}^{\frac{1}{s}} \quad (3)$$

This equation is an approximation of the oil film thickness at the central portion under load as obtained from the oil film thickness distribution under different conditions of load and viscosity property.

$$s = 1.5 \left[1 + \exp \left(-1.2 \frac{H_{EI}}{H_{RI}} \gamma_h^{-\frac{7}{3}} \right) \right] \quad (4)$$

Here, H_{RI} , H_{EI} , H_{RP} , and H_{EP} are dimensionless film thicknesses in a region of rigidity and equal viscosity, a region of elasticity and equal viscosity, a region of rigidity and high-pressure viscosity, and a region of elasticity and high-pressure viscosity. H_{00} is a constant.

In general, when sliding velocity increases, heat generation becomes conspicuous, and oil film thickness shows a tendency to decrease due to heat generation. Heat generation of this kind could also conceivably influence oil film thickness at the sliding speed range of pin and sleeve. The calculation model of Liu et al.⁽³⁾, however, is for the low-speed, high-load range, and it does not take this influence into account. For the present research, therefore, the oil film thickness reduction coefficient of Gupta et al.⁽⁵⁾,

shown in Eq. (5), was applied to the calculation of oil film thickness in high-speed regions.

$$\Phi_T = \frac{1 - 13.2 \left(\frac{p_0}{E'} \right) L^{0.42}}{1 + 0.213(1 + 2.23S^{0.83})L^{0.640}} \quad (5)$$

Here, L is a thermal load parameter that is a function of speed, pressure, oil viscosity coefficient, and so on. S is the ratio of sliding and rolling. With plain rolling, heat generation is around zero, but when sliding occurs, then heat generation increases according to that ratio.

For the shear stress that occurs in the oil film, it was decided to follow the viscosity model of Ree and Eyring⁽⁶⁾, shown in Eq. (6).

$$\tau_h = \tau_0 \operatorname{arcsinh} \left(\frac{\eta \dot{\gamma}}{\tau_0} \right) \quad (6)$$

τ_0 is a constant for stress that depends on the temperature. $\dot{\gamma}$ is a shear rate obtained when sliding velocity is divided by oil film thickness. η is the viscosity of lubricating oil, and the pressure, temperature, and viscosity property of Roelands et al.⁽⁷⁾ were used.

2.4. Method for Deriving Tribological Properties of Convex Portion

The calculation accuracy of the model here is increased by using the surface roughness and waviness obtained by actual measurement made after initial familiarity is achieved by sliding.

Figures 3 and 4 show the surface contour data from measurements of roughness and waviness made when obtaining data for the coefficient of friction in relation to sliding velocity in Fig. 1. The measurements were made using a surface texture and contour measuring instrument (SURFCOM 5000DX manufactured by Tokyo Seimitsu Co., Ltd.).

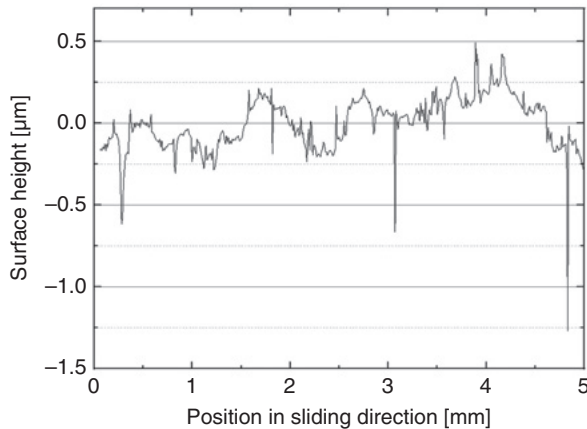


Fig. 3 Surface contour of FC250 after test

Comparison of roughness shows that FC250 is Ra 0.025 while Nikasil coating is Ra 0.007. The latter can be considered to have a smoother surface. As shown in Fig. 5, however, the Nikasil coating surface exhibits waviness that is of low amplitude but has a long period. The influence

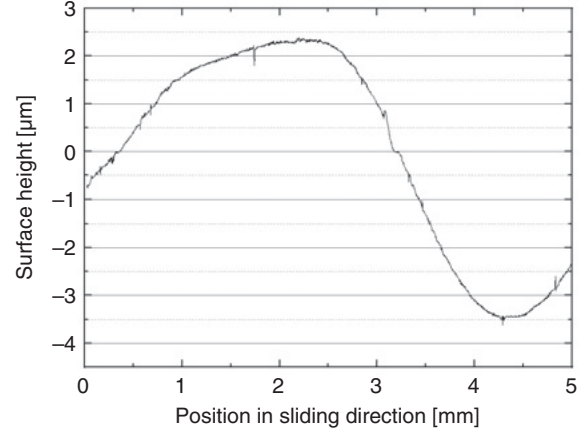


Fig. 4 Surface contour of Nikasil coating after test

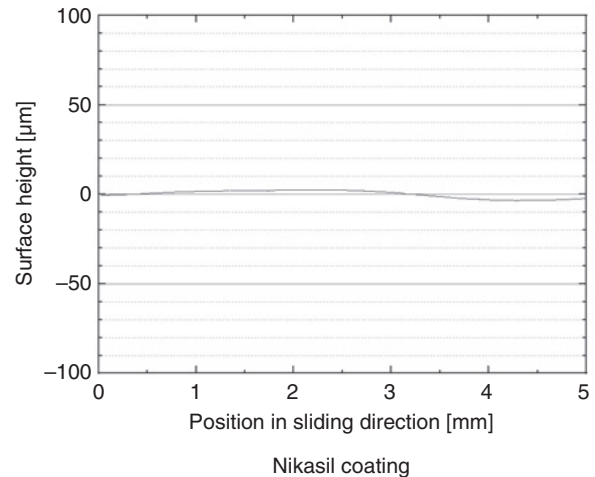
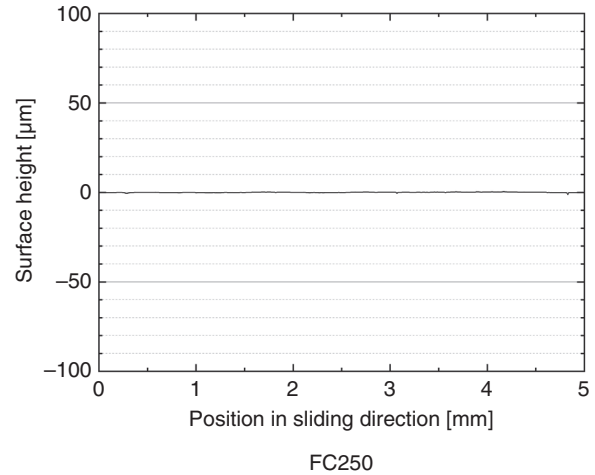


Fig. 5 Comparison of surface contours

of this kind of surface characteristics on the tribological properties was taken into consideration.

Table 1 shows the material parameters for sliding surfaces as extracted from the sliding surface contour curves. The calculation for the coefficient of friction was derived using these conditions, and verification of the model calculations was carried out as a comparison of the velocity dependence of the coefficient of friction with the test figures.

The probability density distribution of this waviness and roughness after sliding appeared generally as a Gaussian distribution. It was assumed, therefore, that the convex portion with Gaussian distribution $\phi_s(z)$ as in Eq. (7) would come in contact with a completely smooth surface.

$$\phi_s(z) = \frac{1}{\sigma_s \sqrt{2\pi}} e^{-\frac{z^2}{2\sigma_s^2}} \quad (7)$$

z is the protrusion height from the mean position, and σ_s is the standard deviation for the protrusion height. An equivalent σ_s was calculated from the contour measurement results for the pin and plate, which show a Gaussian distribution, and this was inserted into the formula.

2.5. Method for Calculating Contact Load of Convex Portion

The convex portion that is in contact bears the load P_a only on the peaks and in some cases may undergo elastic deformation in which the height is lowered or convex portions may be raised higher than h_{ss} . Equation (8) from Johnson et al.⁽⁸⁾ was therefore used for the mean contact pressure applied to the area of true contact, which is the convex portion.

$$p_{a,av} = \frac{2}{3} n \beta \sigma_s E' \sqrt{\frac{\sigma_s}{\beta}} F_{\frac{3}{2}} \left(\frac{h_{ss}}{\sigma_s} \right) \quad (8)$$

h_{ss} is the height of the contact surface shown in Fig. 6. n is the density of the convex portion. β is the radius of the peaks. E' is the equivalent Young's modulus of the material expressed as in Eq. (9).

$$\frac{2}{E'} = \left(\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right)^{-1} \quad (9)$$

Here, E_1 , E_2 , ν_1 , and ν_2 are the Young's modulus and Poisson's ratio for the pin and plate. Function F in Eq. (8) is expressed as follows:

$$F_{\frac{3}{2}}(u) = (2\pi)^{-\frac{1}{2}} \int_u^\infty (t-u)^{\frac{3}{2}} e^{-\frac{t^2}{2}} dt \quad (10)$$

Ordinarily, model calculations do not consider changes

Table 1 Calculation conditions for surface irregularities of sliding parts

	σ_s [μm]	n [m^{-2}]	β [μm]	E' [GPa]
Nikasil coating	0.064	2.25×10^{10}	27.0	247
FC250	0.170	3.40×10^{10}	42.5	263

in the surface state due to initial familiarity. The calculation model here, however, has taken waviness with a long period and asperity after testing into consideration. In order to include their influence in the calculations, the pressure concentration factor (P.C.F.) of Seabra et al.⁽⁹⁾ was used. This is expressed as shown in Eq. (11).

$$\text{P.C.F.} = C1 \left(\frac{\lambda}{\text{Amp}} \right)^\alpha \left(\frac{a}{R} \right)^\beta \left(\frac{\lambda}{a} \right)^\gamma \quad (11)$$

Here, $C1=4.3884$, $\alpha=-0.4234$, $\beta=-0.4204$, and $\gamma=-0.0155$. λ and Amp are the wavelength and amplitude, respectively, of surface waviness. a is Hertzian contact half length and R corresponds to the equivalent radius of curvature of the pin. At this time, the influence of waviness causes the area of Hertzian contact to be reduced by the amount of the reduction of area (R.A.) in the coefficient below. In other words, in the Hertzian contact region, which is reduced in area by the amount of this coefficient, pressure on the convex portion is exerted by the amount of the P.C.F. multiplied by the mean contact pressure ($p_{a,av}$).

$$\text{R.A.} = \ln \left[C2 \left(\frac{\lambda}{\text{Amp}} \right)^\theta \left(\frac{a}{R} \right)^\eta \left(\frac{\lambda}{a} \right)^\Psi \right] \quad (12)$$

Here, the constants are $C2=0.9404$, $\theta=0.3802$, and $\Psi=0.0395$. The waviness assumed in this model is somewhat smaller than in the Hertzian contact region, and it is larger than the period of asperity.

2.6. Method for Calculation of Shear Force of Oil Film and Surface Roughness

The shear stress τ_{ci} that is generated in the contact region i of an asperity surface is determined as shown in Eq. (13) using the value for the coefficient of friction f_{ci} (Fig. 1) obtained in a boundary lubrication area, or in other words, low-speed-region sliding tests.

$$\tau_{ci} = f_{ci} p_{ci} \quad (13)$$

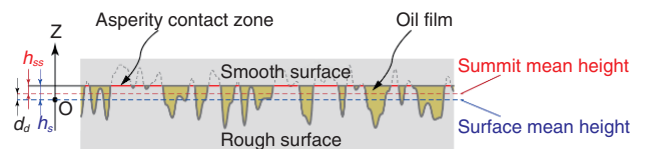


Fig. 6 Relationship of surface parameters

The shear stress in the oil film and the friction force in the true contact portion caused by surface asperity are divided and determined as the respective quantities in this way, and consequently, the net coefficient of friction f is taken to be the sum of these quantities divided by the load P as expressed in Eq. (14).

$$f = \frac{\sum_{i=1}^N \iint_{A_{ci}} \tau_{ci} dA_{ci} + \iint_{A_h} \tau_h dA_h}{P} \quad (14)$$

Here, A_c indicates the load-bearing area of the protrusion contact portion, and A_h that of the oil film-forming region. Figure 7 shows the calculation flow.

First, the load fraction ratios γ_h and γ_a are assumed. This determines the relative distances of the uneven surface and the smooth surface, the pressure on the convex portion is determined, and the oil film thickness can be determined from this pressure.

The pressure applied to the convex portion can also be derived from the load P_a divided by the area A_{nom} of the pressure-generating region. When the convex contact portion pressures obtained by the two methods are identical, the assumed values for γ_h and γ_a are considered to be

correct, and the coefficient of friction was determined from Eq. (14).

Since the present research involves calculations for high-speed ranges, the original step was taken of adding Eq. (5), Eq. (11), and Eq. (12).

3. Comparison of Test Results and Calculation Results for Coefficient of Friction

3.1. High-Speed Reciprocating Test Apparatus

In the case of a test stroke of 50 mm, the reciprocating movement here started where the edge velocity was zero. On advancing 25 mm, the highest velocity is reached, and upon arrival at the edge 50 mm ahead, the velocity becomes zero. This is treated as one cycle, and the reciprocating action is continued while applying a load.

With the piston in an actual engine, there is an air spring effect from the combustion gases and air-fuel mixture at top dead center and bottom dead center, the positions where the velocity is zero. No vibration is generated due to the friction caused by the sealing of combustion gas or crank deflection.

These effects cannot readily be reproduced in the reciprocating sliding tests, however, so vibration is generated in the test piece, centered on the point where

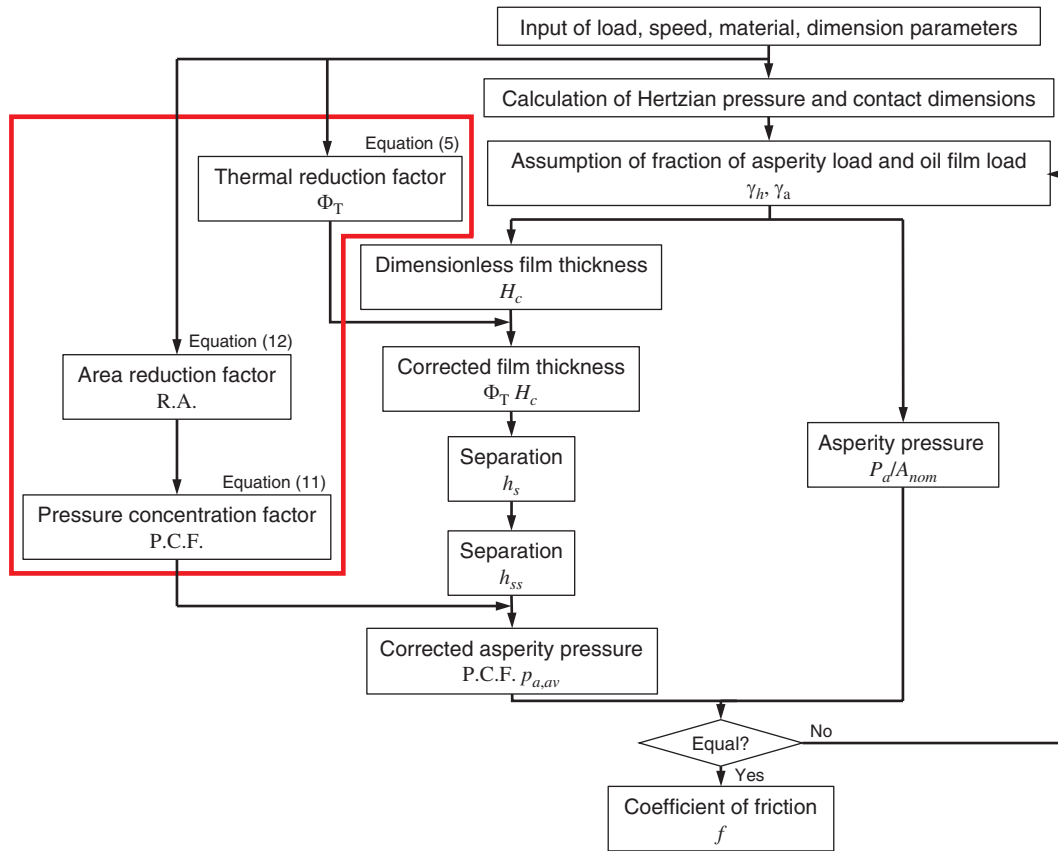


Fig. 7 Calculation flow

the load is applied. In some cases the test apparatus itself shakes back and forth in a large motion. In order to avoid this, most tests⁽¹⁰⁾⁻⁽¹³⁾ are implemented at speed ranges up to 2 m/s where the mean velocity provides for reproducibility of results and the reciprocating sliding test apparatus is stable in screening tests. These velocities are one order of magnitude lower than normal maximum mean piston velocity (about 18 m/s).

This was addressed by reducing the weight of the holder that supports the test piece, which is the plate, and also by adopting a drum cam system to drive the sliders in parallel via ball bearings placed in a helical trench made in the rotating drum. This made it possible to achieve high-speed reciprocating movement. Figure 8 shows the test apparatus mechanism. Table 2 shows the test apparatus specifications.

3.2. Sliding Tests

The sliding tests are implemented as virtual engine piston rings and sleeves. The test conditions are shown in Table 3.

The material used for the piston rings was SUS440C with the usual material properties and with nitriding treatment applied. The material used for the sleeve was FC250 fabricated by spin casting and with Nikasil coating, which has also come into frequent use on mass-production vehicles in recent years, applied on the aluminum parent material.

The lubricant used in testing was the ordinary, genuine Honda brand oil (Ultra Mild) supplied at room temperature at the rate of 1 ml per minute from the point of a needle next to the pin. Figure 9 shows testing underway with high-speed reciprocating movement.

As to the test time, testing was continued until the pin and plate had undergone wear, and after initial familiarity had been achieved when a stable coefficient of friction could be obtained.

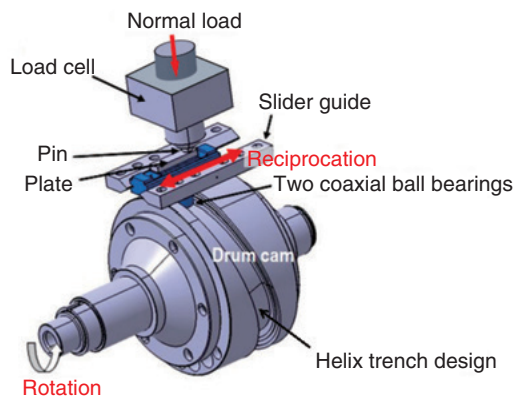


Fig. 8 Mechanism to generate reciprocating movement

Table 2 Specifications of apparatus

Sliding speed	0.1 m/s to 8.0 m/s
Reciprocation stroke	50 mm
Normal load	30 to 200 N
Pin size	φ 15 mm × 25 mm
Plate size	17 mm × 70 mm × 9 mm

Table 3 Test conditions

Sliding speed	0.5 to 8.0 m/s (Peak in reciprocating cycle)
Pin material	Nitriding Treatment on SUS440C
Plate material	FC250 (Cast gray iron) Nikasil coating on Cast aluminum
Normal load	100 N
Lubricant	Ultra Mild 10W-30

3.3. Comparison of Test Results and Calculation Results

The calculation results for the coefficient of friction with respect to sliding velocity using FC250 and Nikasil coating are shown in Figs. 10 and 11. The horizontal axis is the sliding velocity. The vertical axis is the mean coefficient of friction. The test results are also overlaid on the figures for the purpose of comparison. With regard to the FC250, the calculated value for the coefficient of friction largely

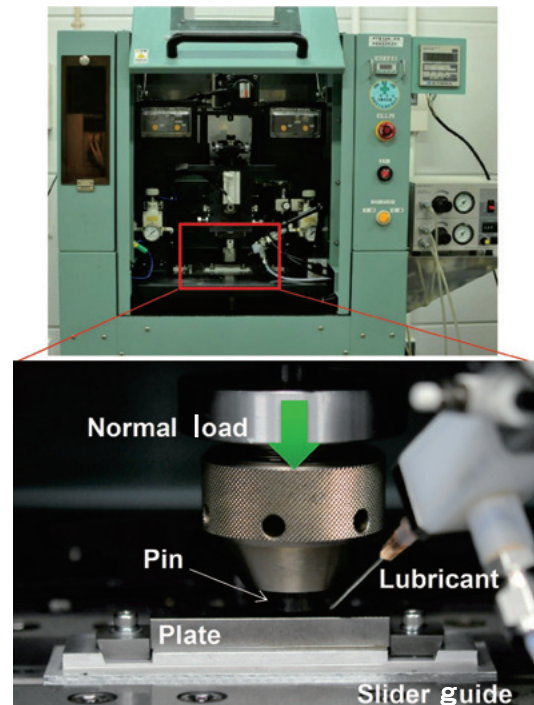


Fig. 9 Sliding part during reciprocating frictional test (4 m/s)

matched the experiment value throughout in the lubricated state.

As shown in Fig. 11, on the other hand, the coefficient of friction calculated for the Nikasil coating showed marked differences in the region up to 2 m/s. This is conceivably because the Nikasil coating surface layer and the base surface have differing hardness. It is also apparent that virtually no difference is seen when the influence of waviness shown in Eq. (6) and Eq. (7) is taken into account. As to methods for analyzing the coating surface structure by this kind of stochastic friction analysis, further investigation is probably needed.

Figure 12 shows the Stribeck curves for the results of testing and calculation performed here.

The calculation results obtained using the calculation model created here are shown by lines, while the test results are shown by points. The horizontal axis shows the velocity (m/s) just as it did in Figs. 10 and 11.

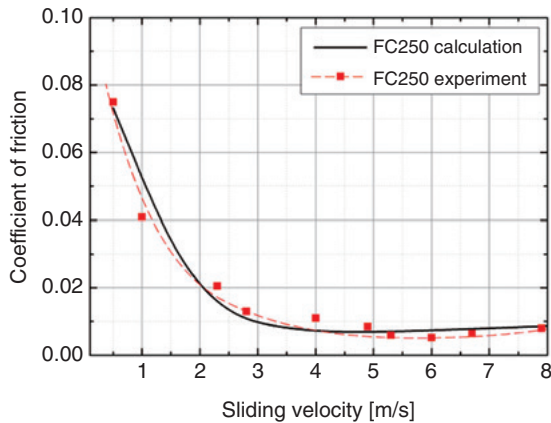


Fig. 10 Comparison of calculated and measured coefficient of friction to sliding velocity, FC250

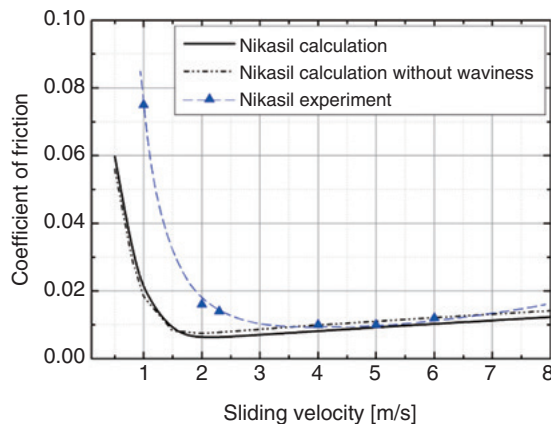


Fig. 11 Comparison of calculated and measured coefficient of friction to sliding velocity, Nikasil coating

There are also regions where the calculation results and test results do not match at velocities of 2 m/s or less. It is apparent, however, that results generally match at subsequent speed ranges. Moreover, this model made it possible to derive values at speed ranges that could not be tested experimentally.

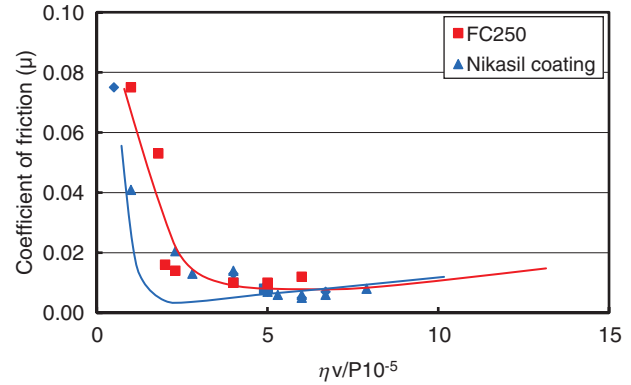


Fig. 12 Comparison of calculated and measured coefficient of friction to sliding velocity

4. Conclusion

Surface roughness and waviness after initial familiarity has been achieved were applied to the stochastic method proposed by Liu et al. A new calculation model was created to take into account slippage, which becomes a determining factor in higher-speed regions.

An apparatus was developed that makes it possible to implement stand-alone sliding at 8 m/s, which is the normal operating range for actual engines.

In this way it has become possible to obtain the coefficient of friction in high-speed regions where it has been unable to derive up to now. It has also become possible to predict the coefficient of friction in high-speed regions where the velocities are equal to or higher than in the verification tests.

With coatings, the hardness of the surface layer differs from that of the base surface below it. For this reason, although the present model was able to make the calculations, they did not match with the test results. Going forward, the model will be rebuilt so that it also takes these material properties into consideration, and the matter of enhancing the accuracy of calculation of the coefficient of friction at low speeds will be investigated.

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