Technical Paper

# Assessment of Effects of Surface Roughness and Oil Viscosity on Friction Coefficient under Lubricated Rolling-sliding Conditions (Analysis of Lubrication Modes Based on EHL Theory)

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*Recent years have seen remarkable advances in the surface reforming technology for reducing the friction coefficient of sliding parts and enhancing their durability. In addition to conventional macro-level surface reforming approaches such as developing sliding materials and enhancing surface strength by heat treatment, surface reforming techniques focused on lubricity have come to be adopted. These techniques include the development of sliding materials and coating materials focused on surface characteristics at the micro level and the use of surface texturing technology. In conventional research and development efforts, combinations of sliding materials and use conditions (test conditions) were used as parameters for relative assessment of strength and durability. However, this approach involves many assessment parameters if surface characteristics and lubricant properties are included, posing a problem of too many assessment standards and parameters. To address this problem, the author conducted analysis of sliding characteristics using theoretical calculation based on tribology (EHL theory) with a view to unifying various assessment parameters. In addition to using surface roughness and oil viscosity parameters which are the basis of lubricity assessment, heat generated by oil shearing between sliding surfaces were also taken into consideration for analysis of relative oil film thickness. Based on these analyses, this paper discusses techniques of predicting lubrication modes under lubricated rolling-sliding conditions.*

**Key Words:** *Tribology, Rolling-sliding, Lubrication, Oil, Viscosity, Surface roughness, EHL, Coefficient of friction, Micropitting*

## 1. Introduction

To enhance the competitiveness of construction machines, greater added values of main components have been required in recent years. These requirements include higher output, higher strength, smaller size, higher durability, and reduced life cycle cost. Along with this trend, the use conditions, environments, and required quality of individual components have become more harsh and strict. Meanwhile, in sliding parts such as gears, shafts and hydraulic equipment parts, surface forms and textures have come to bear a greater importance on the strength of these parts. Research and development of surface reforming technologies are essential to increase added values of the main components of construction machines.

The conventional approaches to improve the sliding performance and durability of sliding parts have basically looked at the macro characteristics of sliding members themselves by machining the parts to optimal shapes, enhancing surface strength by means of heat treatment, or using copper-based materials. Today, however, with their use conditions and environments becoming more harsh, development efforts focused on lubricity are taking place, including the development of sliding materials and coating materials and the use of surface texturing technology focused on surface characteristics at the micro level. Also, the oil film retention property and lubricant reactivity (tribochemical reaction) in the area between sliding surfaces have come to bear greater importance.

Apart from the development of sliding members, researches have been made on various types of lubricants. However, in terms of their combinations with sliding members, there is not enough data about systematic sliding characteristics. It seems that lubricants have been developed independently for each concerned part.

In this research, the author looked at the surface characteristics of sliding member and lubricant properties with a focus on lubricity, and conducted an analysis of lubrication modes under rolling-sliding conditions. In an experiment using a 4-cylinder micropitting tester, tests were conducted using surface roughness and oil viscosity parameters which are the basis of lubricity assessment. In the analysis of oil film thickness using theoretical calculation based on tribology, surface characteristics and lubricant properties were taken into consideration as well as combinations of sliding members and use conditions (test conditions) which were used as conventional assessment parameters. In addition, heat generated by oil shearing between sliding surfaces were also taken into consideration in this analysis so the effects of slip ratio and part shapes (which affect heat transfer between parts) can be incorporated. This paper discusses the effects of surface roughness and oil viscosity on lubrication modes using lambda value which represents the ratio of the film thickness to the composite surface roughness as an assessment parameter.

## 2. Test Method

### 2.1 Tester and test conditions

**Fig. 1** shows a schematic diagram of the testing section of the 4-cylinder micropitting tester. The central roller test piece is evenly loaded with three discs. The roller slides three times per revolution. **Table 1** shows the test conditions. For assessment of surface roughness, the rotation speeds of the roller and discs were set so that the slip ratio would be 5.2% and the test speed would be 3.150 m/sec. The lubricating oil is supplied to the sliding areas by the two rolling discs at the bottom. A thermocouple is set near the roller to monitor the oil temperature at the macro level.

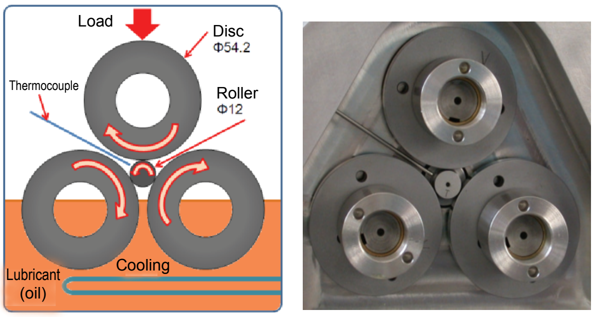


**Fig. 1** 4-cylinder micropitting tester

**Table 1** Test conditions (for surface roughness assessment)

| | |
|---|---|
| Test load | 1,300 N (1.7 GPa) |
| Test speed | 3.150 m/s |
| Slip ratio | 5.2% |
| Testing oil temperature | 70°C |
| Maximum test torque | 20 Nm |

### 2.2 Test pieces

In this research, carburized 16MnCr5 steel (**Table 2**) was used for both the roller (dia. 12 mm × L 12.8 mm) and discs (dia. 54.2 mm × L 8.0 mm). Surface hardness was adjusted to HV 680 for the roller and HV 720 for the discs with a view to causing wear primarily on the roller. The cylindrical portion of the discs was chamfered so that the contact width with the roller would be 3.2 mm. The test consists of a line contact between cylinders.

**Table 2** Chemical composition of 16MnCr5 steel

| Material | | Chemical composition wt (%) | | | | | |
|---|---|---|---|---|---|---|---|
| | | C | Si | Mn | P | S | Cr |
| 16MnCr5 | Min | 0.14 | – | 1.00 | – | – | 0.80 |
| | Max | 0.19 | 0.40 | 1.30 | 0.035 | 0.035 | 1.10 |

Surface roughness was adjusted to provide test standards by grinding the test surfaces after the carburizing process. **Fig. 2** shows the basic surface roughness levels used in this test. Assessment was made using combinations of the same roughness (Rq: root-mean-square roughness) levels for the roller and discs.

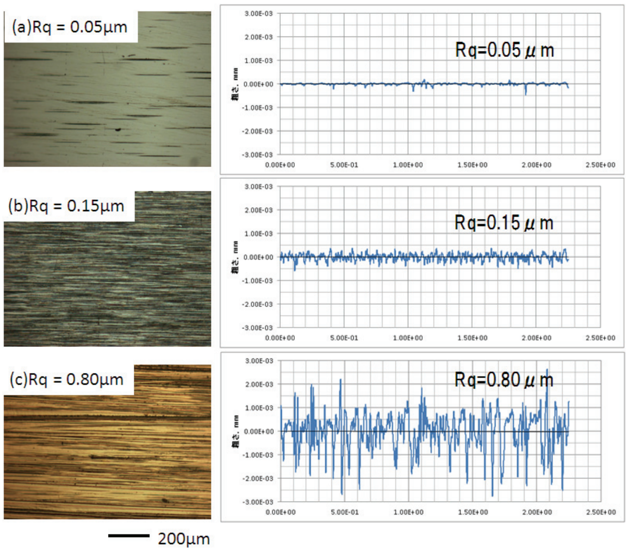


**Fig. 2** Surface roughness levels of test pieces

### 2.3 Lubricant

In this test, three types of oil with different kinematic viscosities (Oil Blends 1 to 3) were used to assess the effect of oil viscosity. **Table 3** shows the lubricant levels, the kinematic viscosity and density at the testing temperature of 70°C is shown in this Table. **Figs. 3** and **4** show the temperature dependent oil viscosity and density, respectively. Here, Blend 1 has the lowest viscosity and Blend 3 has the highest viscosity.

**Table 3** Lubricant levels (kinematic viscosity and density)

| Lubricant level | | | Bland 1 | Bland 2 | Bland 3 |
|---|---|---|---|---|---|
| Kinematic viscosity | cst | 40°C | 48.6 | 125.4 | 294.9 |
| | | 70°C | 15.5 | 32.6 | 64.9 |
| | | 100°C | 7.1 | 12.9 | 22.8 |
| Density | $g/cm^3$ | 40°C | 0.87 | 0.88 | 0.89 |
| | | 70°C | 0.85 | 0.86 | 0.87 |
| | | 100°C | 0.83 | 0.84 | 0.85 |

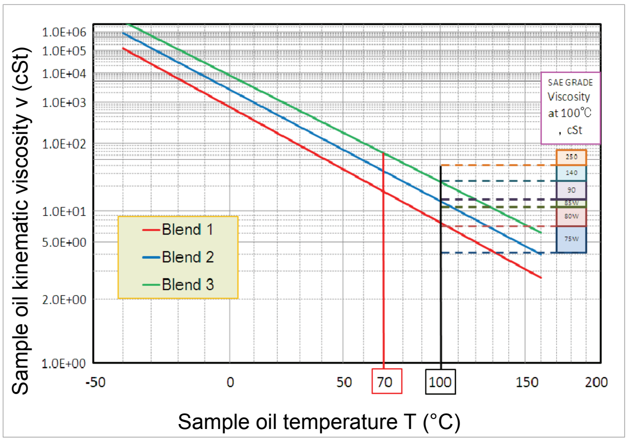


**Fig. 3** Lubricant viscosity-temperature characteristics

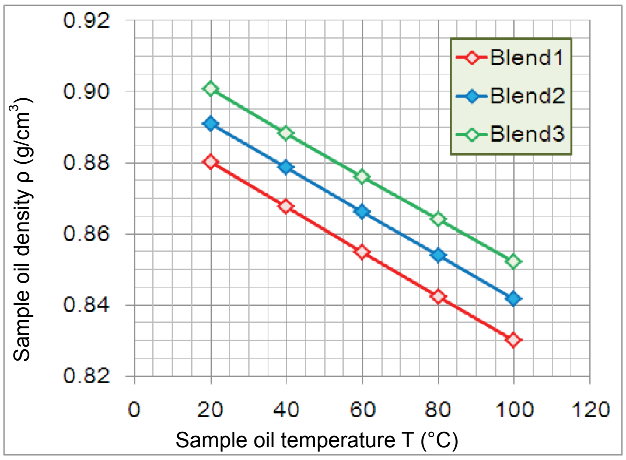


**Fig. 4** Lubricant density-temperature characteristics

## 3. Test Results

### 3.1 Friction coefficient (using bearing characteristics number as assessment parameter)

**Fig. 5** shows the result of friction coefficient measurement in the sliding test where surface roughness and oil viscosity were used as parameters. The horizontal axis indicates bearing characteristic number (speed, contact pressure and oil viscosity taken into account) as conventional assessment parameters, where test conditions are numerically represented, and the vertical axis indicates friction coefficient.

It can be seen that the coefficient of friction increases as the surface roughness increases. Also, the coefficient of friction tends to increase as the viscosity of oil increases at higher surface roughness levels (Rq = 0.15, 0.80 μm). Although the data may be useful for comparison as relative assessment, it is hard to determine the actual lubrication condition from this graph. This is because the bearing characteristics number does not represent the difference in surface roughness. It represents the test conditions as an assessment parameter to show a tendency. The graph also shows that it is difficult to know a tendency based on the oil viscosity without providing many test levels.

In the prediction analysis described in the next section, the effects of surface roughness and oil viscosity were considered based on the result shown in **Fig. 5**, using lambda value as an assessment parameter .

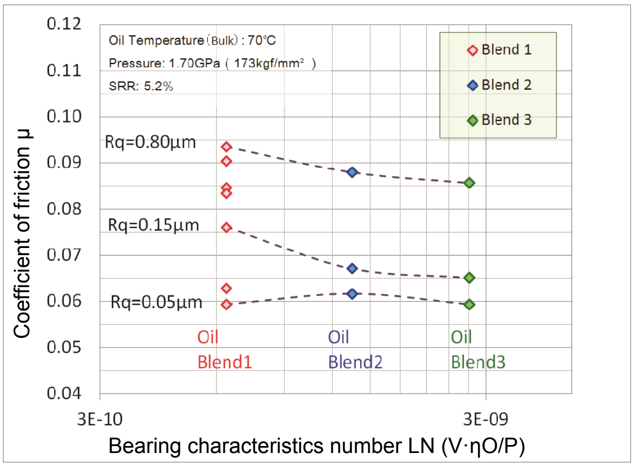


**Fig. 5** Relationship between surface roughness/oil viscosity and friction coefficient (Horizontal axis: Bearing characteristics number)

**Fig. 6** shows the time-series variation of friction coefficient when the Blend 1 oil was used. Here, friction coefficient decreases with time to a greater degree at higher surface roughness levels (Rq = 0.15, 0.80 μm). It is presumed this is due to the fact that the surface becomes smoother and the contact width increases in the wearing process.

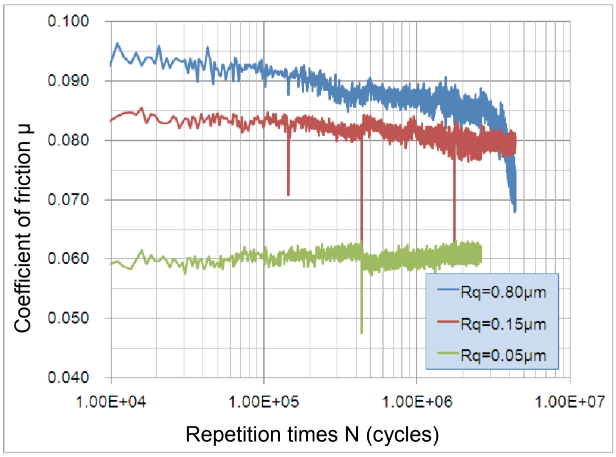


**Fig. 6** Relationship between surface roughness/oil viscosity and friction coefficient (Blend 1)

### 3.2 Surface roughness

**Figs. 7** and **8** show the time-series variations of the surface roughness of the roller and discs, respectively, when the Blend 1 oil was used. A sharp decrease in the surface roughness of the roller was observed on the roller at an early stage. It was found that the surface roughness of the roller decreased early because a greater degree of wear occurred on the roller which has less hardness and was subjected to a greater number of contacts. When the progress of wear reaches a certain level, the surface roughness increases again. On the other hand, the surface roughness of the discs maintained their initial level. For the lowest initial surface roughness level of Rq = 0.05 μm, no change in surface roughness was observed for both on the roller and discs. This may be an indication that the lubrication mode is different from that at the other two roughness levels.

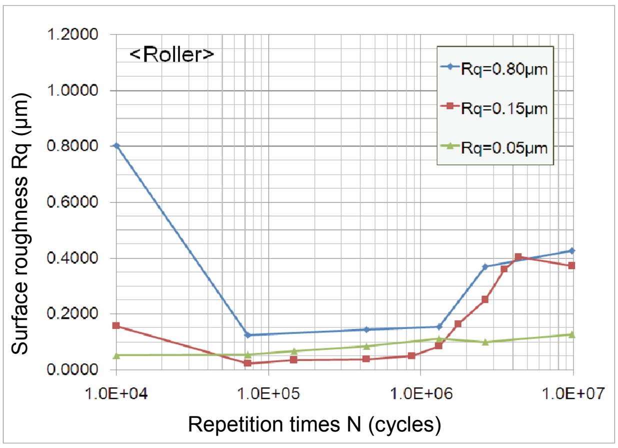


**Fig. 7** Time-series variation of surface roughness of roller test piece (Blend 1)

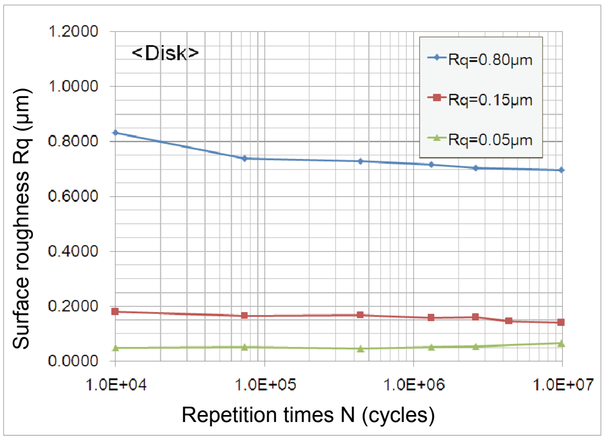


**Fig. 8** Time-series variation of surface roughness of disc test piece (Blend 1)

### 3.3 Wear

**Fig. 9** shows the time-series variation of the wear of the roller test piece for the Blend 1 oil. It was found that the amount of wear increased to a greater degree at higher surface roughness levels, which corresponds to the surface roughness data shown above. As with the surface roughness data, there is no progress of wear at the surface roughness level of Rq = 0.05 μm.

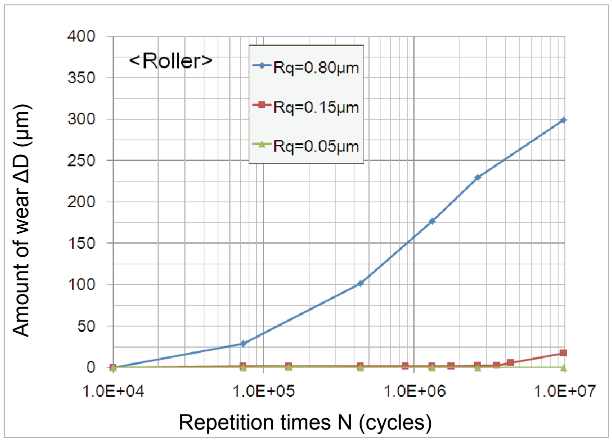


**Fig. 9** Time-series variation of wear of roller test piece (Blend 1)

**Fig. 10** shows the surface conditions after test repetition of $1.3 \times 10^6$ times. As can be seen from this result, the surface conditions after the sliding test vary widely depending on the initial surface roughness. Almost no wear is observed at Rq = 0.05 μm, while micropitting occurred at some places at Rq = 0.15 μm, and the entire sliding surface is seized at Rq = 0.80 μm, indicating the difference in lubrication effect.

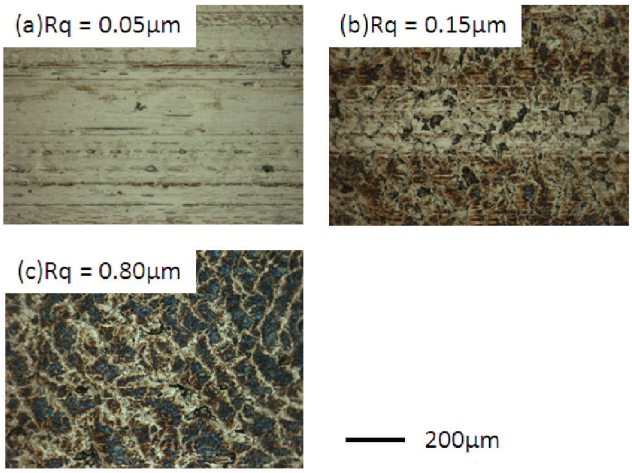


**Fig. 10** Surface conditions of roller test piece (Blend 1 with repetition of $1.3 \times 10^6$ times)

## 4. Prediction Analysis of Lubrication Modes Based on lambda value

### 4.1 Calculation of oil film thickness and lambda value

As shown in **Fig. 5**, it is difficult to view the effects of surface roughness and oil viscosity in terms of a variation in lubrication mode if bearing characteristics numbers represented in terms of test conditions are used as an assessment parameter. For this reason, lambda value was used as an assessment parameter with the effects of surface roughness and oil viscosity taken into account, and the data in **Fig. 5** was represented as follows.

The well-known Dowson and Hamrock's equation (1) was used to calculate oil film thickness. λ (lambda value) is the ratio of the minimum oil film thickness $h_0$ to the composite surface roughness, as shown in equation (2).

<Calculation of minimum oil film thickness>

$$\frac{h_0}{R_x'} = 3.63\overline{U}^{0.68} * \overline{G}^{0.49} * \overline{W}^{-0.073} * \left[1 - e^{-0.70\left(R_y'/R_x'\right)^{0.64}}\right] \quad (1)$$

where

$h_0$: Minimum oil film thickness (µm)

$\overline{U} = \frac{U\eta_0}{E'R_x'}$: Speed parameter (-)

$\overline{G} = \alpha E'$: Material parameter (-)

$\overline{W} = \frac{W}{E'R_x'l}$: Load parameter (-)

$R_x'$: Equivalent radius of curvature in axial direction (mm)

$R_y'$: Equivalent radius of curvature in circumferential direction (mm)

$E'$: Equivalent elasticity coefficient (Pa)

$\eta_0$: Oil viscosity in normal pressure (Pa.s)

$\alpha$: Pressure-viscosity coefficient (1/Pa)

$W$: Test load (Pa.s)

$U$: Average test speed (m/s)

$l$: Contact width (mm)

<Calculation of relative oil film thickness>

$$\lambda = \frac{h_c}{\sqrt{R_{q\_Roller}^2 + R_{q\_Disk}^2}} \quad (2)$$

where

$\lambda$: The ratio of the minimum oil film thickness $h_0$ to the composite surface roughness (-)

$R_{q\_Roller}^2$: Roller RMS roughness (µm)

$R_{q\_Disk}^2$: Disk RMS roughness (µm)

### 4.2 Estimation of temperature between contact surfaces (taking oil shearing heat into consideration)

In this analysis, calculation of heat generation due to oil shearing between sliding surfaces was also conducted. The effect of slip ratio was not considered in the conventional formula for calculating oil film thickness (1). Assessment taking this effect into consideration is possible by replacing the effect of slip ratio with the heat generation due to oil shearing between sliding surfaces. Although the modeling and processes of this calculation are not described here, temperature changes in sample oil were estimated by taking into consideration the heat generation, shapes of sliding members, heat transfer and other factors.

**Figs. 11** and **12** show the calculation results of the sample oil temperature between contact surfaces and the minimum oil film thickness, respectively, in terms of the effects of slip ratio (SRR, %) and test speed. The sample oil temperature between contact surfaces displayed a tendency to increase with the increase in the slip ratio and test speed. It was also found that, supposing that it is under EHL lubrication, the oil film thickness tends to decrease because the oil viscosity decreases with the increase in temperature. By recalculating oil viscosity and density based on the sample oil temperature obtained here, it became possible to make predictions of lubrication modes that better reflect the actual conditions.

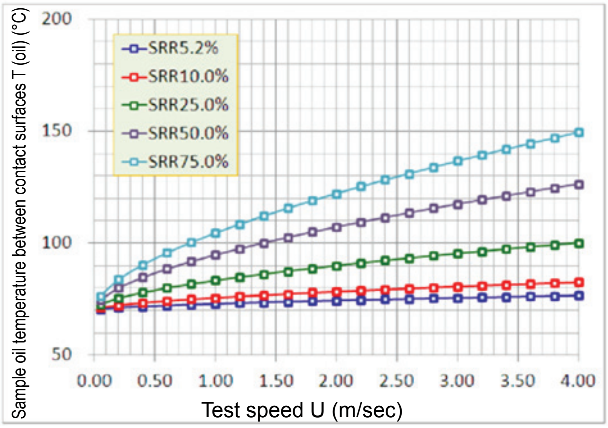


**Fig. 11** Example of calculation of sample oil temperature between contact surfaces (Blend 1 with bulk sample oil temperature of 70°C)

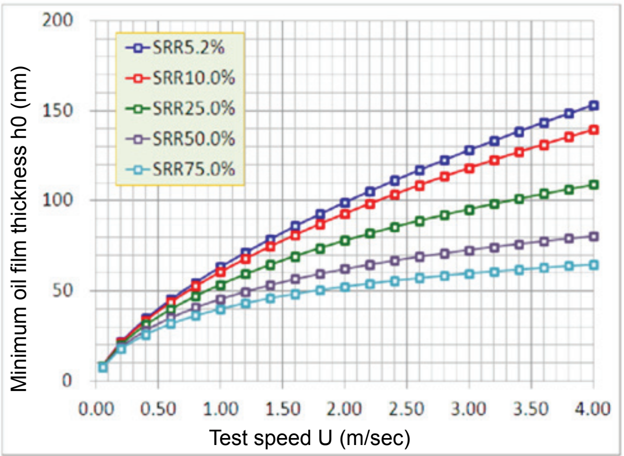


**Fig. 12** Example of calculation of minimum oil film thickness (Blend 1 with bulk sample oil temperature of 70°C)

### 4.3 Analysis of lubrication modes based on lambda value

Using the above analysis method, the author conducted analysis of the test results with reference to lambda value. Based on the test result shown in **Fig. 5**, calculations were made with the horizontal axis being the lambda value (taking into consideration the heat generation due to oil shearing between sliding surfaces). Here, comparison of results was made by overlapping with the stribeck curves which were obtained using the test conditions as parameters.

**Fig. 13** shows the analysis result. With the result of the analysis using bearing characteristics numbers shown in **Fig. 5**, it was difficult to predict the lubrication modes using the surface roughness and oil viscosity levels. On the other hand, it was possible to represent the effects of surface roughness and oil viscosity on stribeck curves, as shown in **Fig. 13**. The author believes that the result of this test shows the surface roughness and oil viscosity levels represent the transition of lubrication mode from the EHL lubrication region to mixed lubrication region. It is also presumed that the time-series variation of surface roughness or amount of wear corresponds well with the analysis tendency of lubrication mode, from the level of rapid progress of wear (Rq = 0.80 μm) to the level of no progress of wear (Rq = 0.05 μm).

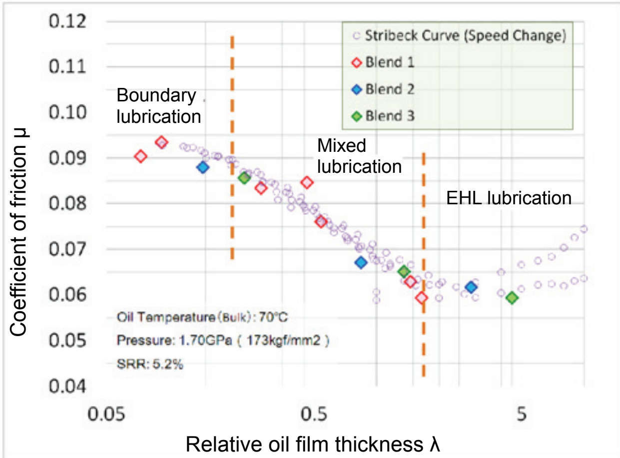


**Fig. 13** Relationship between surface roughness/oil viscosity and friction coefficient (Horizontal axis: Relative oil film thickness)

### 4.4 Analysis of wear based on lambda value

The amount of wear was analyzed using the lambda value considered above as an assessment parameter. **Fig. 14** shows the result of the analysis of wear in each level after test repetition of $1.3 \times 10^6$ times with the horizontal axis representing the lambda value. Here, we can see a sharp increase in the amount of wear at levels below λ = 0.5 (i.e. higher levels of surface roughness Rq). Though further discussion is required with regard to the threshold of λ values taking the testing time into consideration, this data indicates the existence of a certain relationship between λ values and the amount of wear, just like the relationship with friction coefficient. The author believes this result makes an example that shows that the lambda value can be used as an assessment parameter.

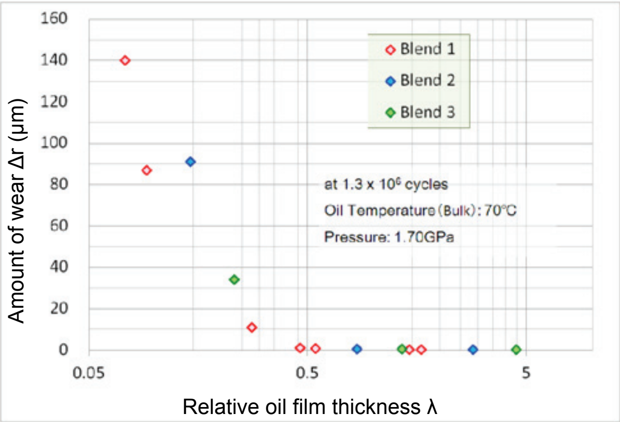


**Fig. 14** Relationship between roller wear and relative oil film thickness (Blend 1 with repetition of $1.3 \times 10^6$ times)

## 5. Conclusion

In this research titled "Assessment of Effects of Surface Roughness and Oil Viscosity on Friction Coefficient under Lubricated Rolling-sliding Conditions", the author conducted a prediction analysis of lubrication modes using calculations based on the EHL theory. The results are summarized below.

(1) With regard to the effects of surface roughness and oil viscosity on sliding characteristics, the transition of friction coefficient was represented by analysis using the lambda value as an assessment parameter. This approach was also assumed to be effective for analysis of lubrication modes (which was difficult using a conventional assessment parameter of bearing characteristics number).

(2) In analyzing the lambda value, the effect of slip ratio was incorporated by taking into consideration the heat generation in oil between sliding surfaces. Furthermore, it was indicated that lubrication modes could be analyzed in a way that better reflects the actual conditions by recalculating the oil viscosity and density based on the sample oil temperature between sliding surfaces.

(3) A certain relationship between the lambda value and the amount of wear was found by using the analysis method.

(4) Although further discussion is required to clarified the relation between the absolute values of λ and the trends of the transitions of friction coefficient and the amount of wear, the author would like to further investigate into this issue in future.

**Introduction of the writers**

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Joined Komatsu Ltd. in 2007
Currently a member of the Manufacturing Engineering Development Center, Production Division

**[A few words from writers]**

I hope the analysis method achieved in this research will help develop quality enhancing techniques in the efforts to improve sliding performance and durability of sliding parts. I would also like to study the effects of various assessment parameters on sliding characteristics while improving my skills in analysis for more efficient research.