

Teardown Analysis for Piston Assembly Friction with Optimization Controls

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ABSTRACT

This research involved creating a method for analyzing the contribution of the skirt and rings to the friction force of the piston assembly, and implementing such analysis. The contribution analysis was carried out by identifying the respective Stribeck curves for the skirt and the rings. The results showed that the skirt is in the fluid-lubricated region and the rings are in the mixed-lubricated region of the curve. The results also made clear that the friction force of the skirt is the dominant component under high-speed, low-load conditions, in particular.

1. Introduction

There is demand for further enhanced fuel efficiency in automobile engines in order to contain global warming. The enhancement of fuel efficiency necessitates a reduction in friction loss together with an enhancement of indicated thermal efficiency. It has been reported that friction loss in the pistons and connecting rods accounts for about one-third of total friction loss⁽¹⁾, and the reduction of piston friction loss, in particular, has been treated as an important issue and subjected to measurement and analysis many times.

A number of methods for measuring piston assembly friction have been implemented by holding a sleeve in a floating mounting and measuring the load that is applied^{(2), (3)}. However, the friction is measured as the resultant forces on the skirt and the rings, so that the contribution of each part is not clearly identified. By contrast, there are examples of individual analysis of the separate parts, such as friction measurement of rings by bench testing⁽⁴⁾, calculation using CAE⁽⁴⁾⁻⁽⁶⁾, and the like, but there have not been any reports of analysis made during actual operation. These issues can be resolved by performing direct teardown analysis of actual measurements made of piston assembly friction.

This research presents a method for identifying Stribeck curves to tear down the measured piston assembly friction into the portion that is generated in the skirt and the portion generated in the rings. This was used to calculate the friction distribution during actual

operation. The curves obtained in this way were then used to examine the influence of engine speed and load on friction loss separately in the skirt and rings and analyze their contribution to piston friction loss. This aids in reducing friction loss.

2. Test Device

In order to measure piston assembly friction, a single-piston engine with a sleeve held in a floating mounting was fabricated. Table 1 and Fig. 1 show the specifications and a sectional view of the fabricated engine.

A triaxial force sensor was used as the sleeve support structure, with reference to the research of Itoh et al.⁽³⁾, because it possesses adequate rigidity and allows measurement up to the high speed range. A total of six triaxial force sensors were disposed [(A) in Fig. 1] on both intake and exhaust sides in order to assure rigidity and prevent the sleeve from tilting. This applies a tensile load on both sides of the sleeve, so that secondary deformation could occur in the bore. In order to prevent this, the sensor holder [(B) in Fig. 1] was made independent of the block.

The combustion chamber seal was formed in the space between the sleeve and the head stuck into the cylinder. This enables prevention of the direct application of cylinder pressure against the upper surface of the sleeve. In this case, however, the use of O rings for the seal has been reported⁽⁷⁾ to result in load transmission to the sleeve

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Table 1 Engine specifications

Engine type	4-stroke
Number of cylinders	1
Bore (mm) × Stroke (mm)	86 × 86
Compression ratio	9.8
Crank offset (mm)	0.0
Cooling system	Water - cooled
Camshaft	SOHC
Valves (IN/EX)	1/1
Lubrication oil	SAE 0W-20 (80±1°C)

due to the head deformation caused by combustion pressure. Therefore, a seal ring with a shape corresponding to the piston rings was used for the seal so as to realize a balance between providing a cylinder pressure seal and preventing load transmission.

The sleeve was given a structure with two layers, inner and outer, to assure the circularity of the bore as well as to reduce weight and enhance maintainability. A flange added on the top surface was used to fasten the sleeve, and the gaps were made into cooling water passages [(C) in Fig. 1].

Figure 2 shows the measuring system.

The friction was calculated by subtracting the inertial

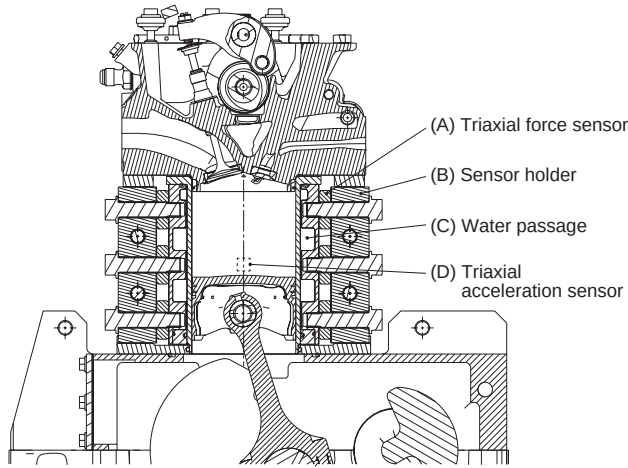


Fig. 1 Engine cross section

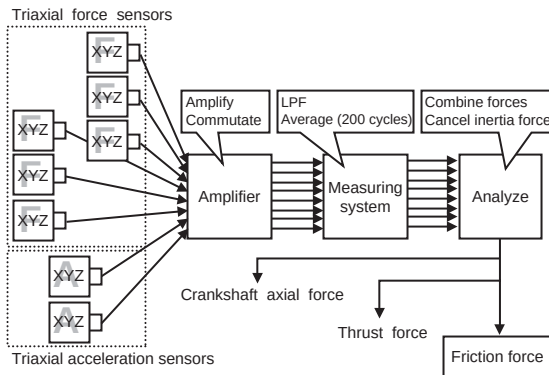


Fig. 2 Friction measuring system

force of the sleeve from the sum of the triaxial force sensor output values. That inertial force was calculated by the values from the two acceleration sensors [(D) in Fig. 1], which were placed so as to pass through the sleeve center of gravity.

3. Identification of the Stribeck Curve

Piston assembly friction F is expanded as shown below, as the sum of the friction of the skirt (hereafter subscript s) and the friction of the rings (hereafter subscript r).

$$F = F_s + \sum F_r \quad (1)$$

The various forces of friction can be represented as products of the coefficient of friction μ and the normal force N . Clarifying these characteristics of μ enables teardown of the friction forces as shown in Eq. (1).

3.1. Stribeck Curve Identification Method

It is known that the coefficient of friction μ on the one hand, together with the viscosity η , speed v , and surface pressure p on the other, can be used to generate a Stribeck curve with characteristics as shown in Fig. 3. In this research, the following three items were used as the control factors for this function.

- (1) The coefficient of friction gradient in the fluid-lubricated region: a
- (2) The minimum coefficient of friction in the mixed-lubricated region: b
- (3) The boundary coefficient of friction: c

The lubricant was taken to be a Newtonian fluid, the diametric clearance during the stroke was taken to be fixed, and the fluid-lubricated region was postulated to be a first-order function that passes through the origin. The mixed-lubricated region was postulated to be a second-order function that passes through minimum coefficient of friction b and boundary coefficient of friction c and links them smoothly.

If the skirt and rings are considered to have their own respective curves, they can be expressed as shown below.

$$(\text{Skirt}) \quad \mu_s(\eta, v, p) = \mu_{a_s, b_s, c_s}(\eta, v, p) \quad (2)$$

$$(\text{Rings}) \quad \mu_r(\eta, v, p) = \mu_{a_r, b_r, c_r}(\eta, v, p) \quad (3)$$

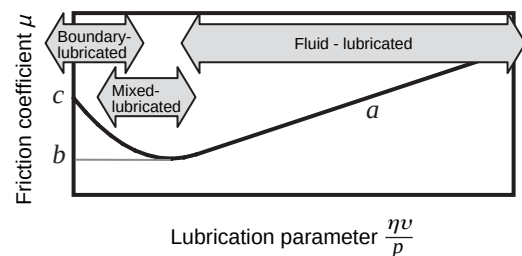


Fig. 3 Stribeck curve

If it is postulated that the normal force that acts on the skirt surface is equal to the thrust load that acts on the piston, and that the normal force that acts on the surface of the outer circumference of the rings is determined by the tensile force and pressure on the reverse surface of the rings, then surface pressure p and the normal force N that are necessary when calculating μ can be expressed as in the following.

$$\text{(Skirt)} \quad p_s = \frac{N_s}{A_s} \quad (4)$$

$$N_s = F_{th} \quad (5)$$

$$\text{(Rings)} \quad p_{r_i} = \frac{N_{r_i}}{\pi D h_{1_i}} \quad (6)$$

$$N_{r_i} = 2\pi F_{t_i} + \pi D h_{1_i} p_{g_i} \quad (7)$$

A_s is the area of one skirt side, F_{th} is the thrust load, D is the bore diameter, h_1 is a ring width, F_t is the ring tension, and p_g is the ring groove pressure, where subscript i (1 to 3) identify the rings. These together with piston speed v and oil viscosity η , are used to express friction F as in the following.

$$F = \mu_s(\eta, v, p_s)N_s + \sum_{i=1}^3 (\mu_r(\eta, v, p_{r_i})N_{r_i}) \quad (8)$$

As is apparent from Eqs. (5) and (7), N_s and N_r here have different factors. Using these to make F match the actual measurement value F_{measured} for freely selected conditions and crank angles enables determination of $\mu_s(\cdot)$ and $\mu_r(\cdot)$. The mean square error, ε_f , with respect to the measured value, with the number of actual measurement points as m , will be:

$$\varepsilon_f = \sqrt{\frac{1}{m} \sum_{j=1}^m (F_j - F_{j \text{ measured}})^2} \quad (9)$$

In order to minimize this error ε_f , optimization analysis using variables a_s - c_r was performed and the Stribeck curve functions $\mu_s(\cdot)$ and $\mu_r(\cdot)$ were identified. Figure 4 shows the above system.

3.2. Stribeck Curve Identification and Results

Results under the following two conditions were used to identify the curves.

- (1) Motored at 1500 rpm, intake manifold absolute pressure 25 kPa ($P_{\text{max}} = 0.4$ MPa)
- (2) Motored at 2500 rpm, intake manifold absolute pressure 100 kPa ($P_{\text{max}} = 2.1$ MPa)

In order to inhibit the effects of vibrational components, the components were discretized at every 30 deg of crank angle, as shown in Fig. 5. The data from all 23 points (a total of 46 points) were used. This

excluded the point immediately after compression top dead center, where vibration is great.

The parameters a_s - c_r were subjected to optimization analysis in order to satisfy these discretized empirical values. Simulated annealing⁽⁸⁾ was used for the optimization.

Figure 6 shows the Stribeck curves obtained.

The coefficients of boundary friction were approximately 0.09 for the skirt and approximately 0.04 for the rings. The coefficient of friction for the rings was smaller, about half the figure for the skirt. Gradient a_r in the fluid-lubricated region was smaller, at about one-tenth the figure for the skirt. This is thought to be because the real contact width of the rings was smaller than the nominal width, h_1 , so that the real surface pressure became greater than the nominal value.

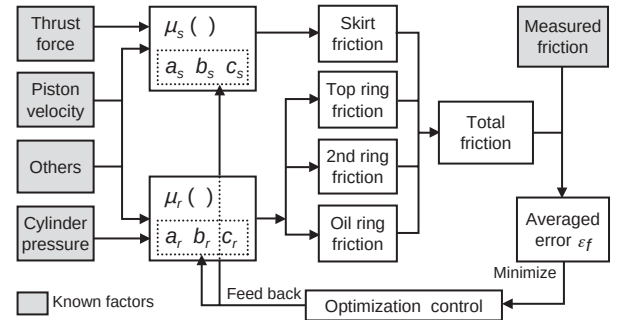


Fig. 4 System for identifying Stribeck curves

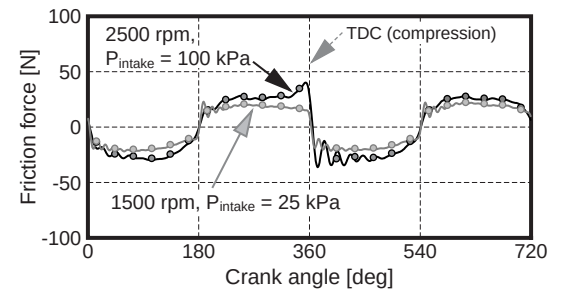


Fig. 5 Discretized empirical data

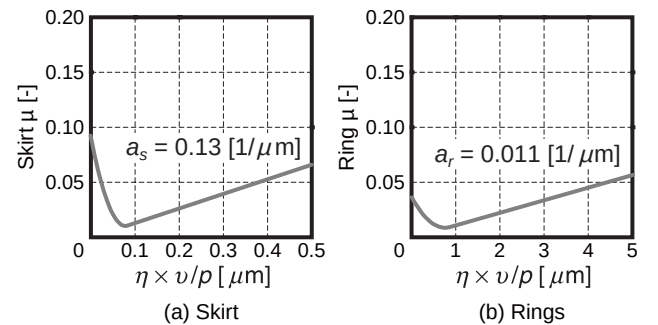
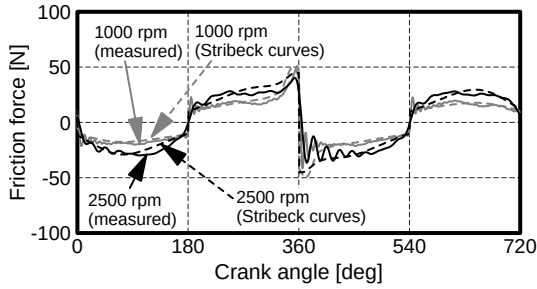


Fig. 6 Stribeck curves for skirt and rings

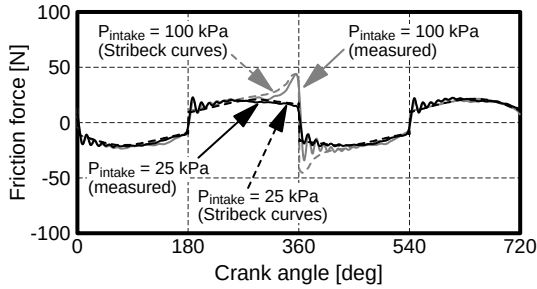
3.3. Verification of the Validity of Stribeck Curves

The friction forces calculated from Stribeck curves for multiple conditions were compared with the empirical results to verify the validity of the curves. Figure 7 shows the results.

This clearly illustrates the way in which friction force in the piston high speed region changes with engine speed [Fig. 7(a)] and the distinctive phenomena whereby friction force increases near compression top dead center due to differences in cylinder pressure [Fig. 7(b)]. The validity of the Stribeck curves obtained could therefore be confirmed.



(a) Influence of speed (motored, intake pressure = 100 kPa)



(b) Influence of intake pressure (1500 rpm motored)

Fig. 7 Validation of Stribeck curves

4. Contribution Analysis of Piston Assembly Friction

Stribeck curves $\mu_s(\cdot)$ and $\mu_r(\cdot)$, obtained using Eq. (8), were applied to perform contribution analysis of the friction forces.

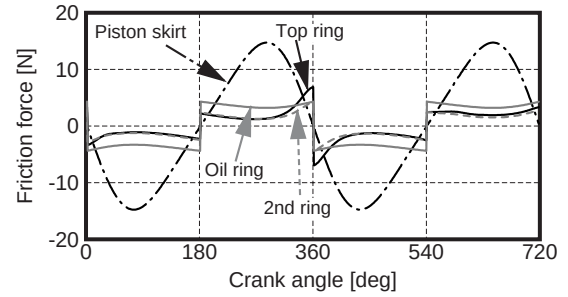
4.1. Contribution Analysis of Actual Friction Measurements

An example of the results obtained through teardown analysis of actual friction measurements together with the application areas of the Stribeck curves at that time are shown in Figs. 8 and 9. The friction loss distribution is shown in Fig. 10.

It is apparent from Fig. 8 that friction in the skirt occurs in proportion to speed, and from the application

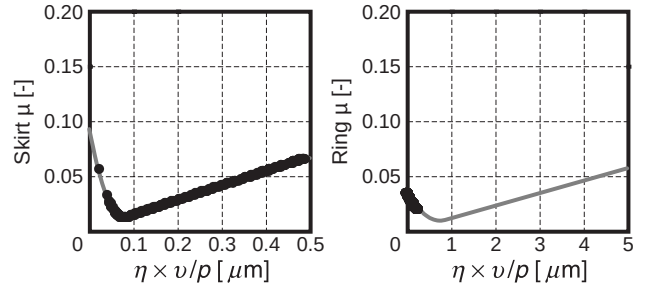
area of the curve in Fig. 9(a) that the friction is occurring under adequately fluid-lubricated conditions. In Fig. 8, by contrast, the friction waveforms for the rings show largely fixed values, so that it can be predicted that the boundary-lubricated condition will be dominant. This can also be confirmed from the results in Fig. 9(b).

It was also apparent from the friction loss distribution in Fig. 10 that the skirt accounts for approximately half of the piston friction loss under these conditions. This was followed by the large influence from the top ring, which is susceptible to the influence of cylinder pressure, and from the oil ring, which experiences high tensile force.



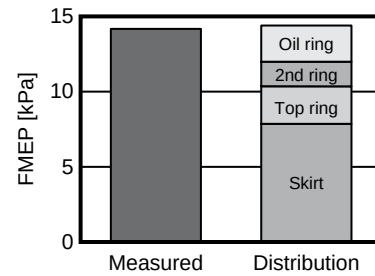
(1500 rpm motored, intake pressure = 25 kPa)

Fig. 8 Friction distribution



(a) Skirt
(b) Rings
(1500 rpm motored, intake pressure = 25 kPa)

Fig. 9 Application area of Stribeck curves



(1500 rpm motored, intake pressure = 25 kPa)

Fig. 10 FMEP distribution

4.2. Changes in Friction Loss and Contribution due to Engine Speed and Load

In order to confirm how the contribution to friction loss changes due to engine speed and load, calculations were performed to determine friction loss when engine speed and intake pressure were changed. Figure 11 shows the results.

Examination of Fig. 11 (a) shows that, because the skirt is in the fluid-lubricated region, friction loss increases in proportion to engine speed. It is also apparent, from Fig. 11(b), that there is less friction loss in the rings than in the skirt, and that it diminishes still further as engine speed increases because it more closely approaches the fluid-lubricated region. Friction loss increases when the load increases because of the increase in the load on the reverse surface of the rings due to cylinder pressure.

Figure 12 shows friction loss in the piston as a whole and its percentages in the skirt.

Friction loss in the whole increases as engine speed becomes higher and load becomes greater. Moreover, it was ascertained that the percentage of friction loss occurring in the skirt is at least 50% more or less across the entire range. Under conditions of high engine speed and low load, in particular, it accounts for 90%.

In this research, the actual measurement results were used to examine the piston assembly friction and ring friction separately. The following are considered necessary in order to further enhance accuracy:

- (1) Identification of curves using results from fired conditions
- (2) Methods to make curve shapes more closely resemble actual operating conditions

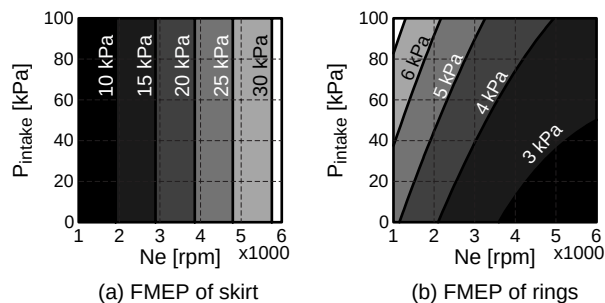


Fig. 11 FMEP maps of skirt and rings

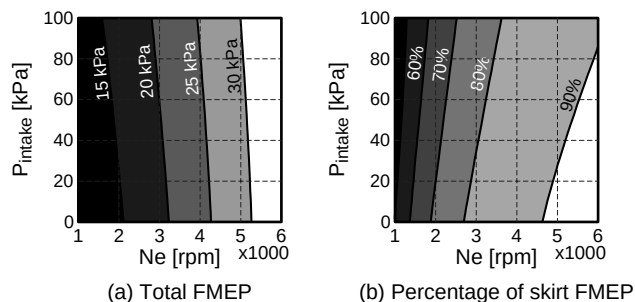


Fig. 12 Total FMEP and skirt percentage

- (3) Introduction of independent curves for each ring

It is also thought that this method could be applied, for example, so that friction measurement results for an entire engine could be used to analyze the contribution of each individual component without making use of motored teardown methods.

5. Conclusion

Analysis of friction was performed using empirical results with the aim of reducing piston friction loss, and the following findings were obtained:

- (1) A method of identifying Stribeck curves on the basis of empirical results was devised, and it yielded good matching results with empirical data under a variety of conditions.
- (2) Use of the above method succeeded in tearing down the actual measured friction into the portion contributed by the skirt and the portion contributed by the rings. As a result, it was confirmed that in this engine, the fluid-lubricated condition was dominant in the skirt while the rings were mainly under boundary-lubricated or mixed-lubricated conditions.
- (3) Analysis of the influence on skirt friction and ring friction related to engine speed and load confirmed that skirt friction is strongly influenced by engine speed and that ring friction is strongly influenced by engine speed and load. The percentage of friction loss in the skirt was found to be about 50-90%, and the percentage of skirt friction loss was found to increase as the engine speed increased and the load diminished.

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