

Impact of Boundary Lubrication Performance of Engine Oils on Friction at Piston Ring-Cylinder Liner Interface

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ABSTRACT

To explore the measures that can be used to improve the fuel economy of internal combustion engines, we investigated how friction at the piston ring-cylinder liner interface is influenced by the boundary lubrication performance of engine oils. We formulated several engine oils with varying boundary lubrication performance and tested them for ring-liner friction by using a floating liner friction tester. We used friction modifiers (FMs) to modify the boundary lubrication performance of engine oils. We found that ring-liner friction is well correlated with the friction coefficients in boundary lubrication regimes when measured by a laboratory friction tester. We also found that the impact of the boundary lubrication performance of engine oils was emphasized in low viscosity engine oils. It makes it possible for improved boundary lubrication performance to inhibit or overcome the viscosity reduction-induced increase of friction energy. These results suggest that FM formulation is as important as viscosity modification for next-generation engine oils to achieve better fuel economy.

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INTRODUCTION

Friction at the piston ring-cylinder liner interface in reciprocating internal combustion engines should be reduced for global energy conservation. Currently, liquid hydrocarbon fuels, such as gasoline, are the main source of energy for human activity, and large amounts of these fuels are consumed in the transportation sector. Reciprocating internal combustion engines—which are widely used to drive transportation vehicles, including passenger cars, heavy-duty trucks, and motorcycles—are essentially equipped with combustion chambers based on pistons and cylinders, which means friction generated between piston ring and cylinder liner is inevitable. Ring-liner friction reportedly constitutes a considerable fraction of energy loss in reciprocating internal combustion engines [1]. Therefore, lubrication of the ring-liner interface is key to improving the fuel economy of transportation vehicles.

For a comprehensive understanding of ring-liner lubrication, we need to study both the physical (viscometric) properties and the chemical (surface-active) properties of the lubricants on ring-liner interfaces, i.e., engine oils. Lubrication regimes at the ring-liner interface dynamically change between hydrodynamic and boundary due to a reciprocating motion. Thus, in addition to the fundamental lubrication parameters (i.e., viscosity of interfacial media, sliding velocity, and normal load), ring-liner friction could also depend on the chemical feature of the ring-liner interface. Our previous study on the relationships

between the viscometric properties of engine oils and ring-liner friction [2] revealed that the addition of viscosity modifiers—the polymeric additives frequently used in fuel economy engine oils—caused strong shear-thinning at the ring-liner interface and that the shear rate was estimated to be approximately 10^7 s^{-1} . This means that in mixed- and boundary-lubrication regimes, viscosity modifiers potentially increase frictional loss and wear probability. In these regimes, boundary lubrication performance is important for reducing friction.

In this study, we investigated the impact of the boundary lubrication performance of engine oils on friction at the ring-liner interface. We used several friction modifiers (FMs), including molybdenum dithiocarbamate (MoDTC) and nonmetallic fatty compounds, to modify the boundary lubrication performance of engine oils. To evaluate the effects of FMs on friction, we used a floating liner friction tester, which is widely used to directly measure ring-liner friction [2, 3, 4, 5, 6]. The floating liner friction measurements show that friction force at dead center and friction energy per cycle highly depend on FM formulation at high temperatures. These ring-liner friction characteristics are well correlated with the friction coefficient measured by laboratory friction testers. Meanwhile, friction characteristics at low temperatures are independent of FM formulation. These results indicate that boundary lubrication performance is a good contributor to ring-liner friction in low viscosity.

METHODS

Engine Oils

Base Formulation

We formulated engine oils for experiments using typical engine oil additives, such as zinc dialkyldithiophosphate (ZnDTP) anti-wear agents, calcium-based detergents, dispersants, and anti-oxidants, to meet the performance equivalent of the Japanese Automobile Standards Organization (JASO) DH-2, a Japanese diesel engine oil standard. The characteristics of base formulations are shown in Table 1. The densities, kinematic viscosities, and viscosity indices of these oils were measured by the procedures determined by ASTM D4052, ASTM D445, and ASTM D2270, respectively. This base formulation corresponds to the oil named “#20” in our previous study [2].

Table 1. Characteristics of Base Formulation

Density @ 15°C	0.876 g/cm ³
Kinematic Viscosity @ 40°C	57.2 mm ² /s
Kinematic Viscosity @ 100°C	8.39 mm ² /s
Viscosity Index	118
Base Oil Composition	Gr.II 150N and Gr.II 500N
Viscosity Modifier	None
Additive System	JASO DH-2

Friction Modifiers

From a wide variety of additives acting as FM, we chose five different additives: three ester-type organic FM, an amine-type organic FM, and a MoDTC-type FM. MoDTC is known as an “extreme pressure additive” that forms low-friction surface films at high pressure through a synergistic tribochemical reaction with ZnDTP [7]. The MoDTC-type FM was mixed with a base formulation at the concentration corresponding to [Mo] = 700 ppm. Organic FM are basically composed of bi-polar molecules that have hydrophilic polar heads with oil-soluble hydrocarbon chains. Hydrophilic heads allow organic FM to be physically adsorbed on the metal surface and modify boundary lubrication performance. The organic FM used in this study are based on ester or amine derivatives of fatty acids. Each organic FM was mixed with a base formulation at a concentration of 1.0 wt%.

Calculation of Viscosity at Given Temperature

We estimated dynamic viscosity at the operational temperature by the procedure described below. Kinematic viscosity at a given temperature was calculated by the following Walther equation:

$$\log \log(\nu + 0.7) = A - B \log T \quad (4)$$

where ν is calculated kinematic viscosity and T is given temperature. A and B were determined from the measured kinematic viscosity at 40°C and 100°C, respectively. Density at

a given temperature was calculated by the following set of equations in accordance with API standard 2540:

$$D = 0.628 / (1000 \cdot \rho_{15^\circ\text{C}}) \quad (5)$$

$$\rho = \rho_{15^\circ\text{C}} \cdot \exp(-D\Delta T(1 + 0.8 \cdot D\Delta T)) \quad (6)$$

where ρ is calculated density, $\rho_{15^\circ\text{C}}$ is the density measured at 15°C and ΔT is the temperature difference from 15°C. Dynamic viscosity η was obtained from the following definition:

$$\eta = \rho \nu \quad (7)$$

Floating Liner Friction Tester

Experimental Device

We evaluated ring-liner friction using a floating liner-type friction tester that was described previously [2]. This tester was developed in-house for research purposes and consisted of a reciprocating four-ring piston driven by a three-phase induction motor through a crank mechanism (Table 2 and Figure 1). We adopted a motoring operation instead of a firing operation to avoid the combustion-derived effects on friction, such as degradation, evaporation, and deposit formation. The rings were coated with chromium nitride (CrN) using the physical vapor deposition (PVD) method. In this tester, the cylinder liner was fixed to a cylinder block through two load cells. These load cells measured the mechanical force applied to the cylinder liner along the direction of the piston stroke. A digital oscilloscope recorded this force with a temporal resolution of about 0.02 milliseconds. The engine oil was heated by an external oil heater and supplied and circulated into the crankcase by an external oil pump. Heat transfer fluid was also heated by another external oil heater and circulated in the oil passage in a cylinder block. The temperatures of the engine oil and heat transfer fluid were measured by thermocouple probes and dynamically modulated through proportional-integral-differential (PID) feedback controllers connected to each oil heater. The temperature at the ring-liner interface was also measured by another thermocouple probe inserted into the liner. We detected TDC using an induction-based crank angle sensor.

Table 2. The Specifications of Our Floating Liner Friction Tester

Stroke	82 mm
Bore	70 mm
Crank Radius	41 mm
Connecting Rod	100 mm
Total Ring Tangential Force	67 N
Crank Offset	None
Ring Material	Steel, CrN-coated
Liner Material	Gray cast iron

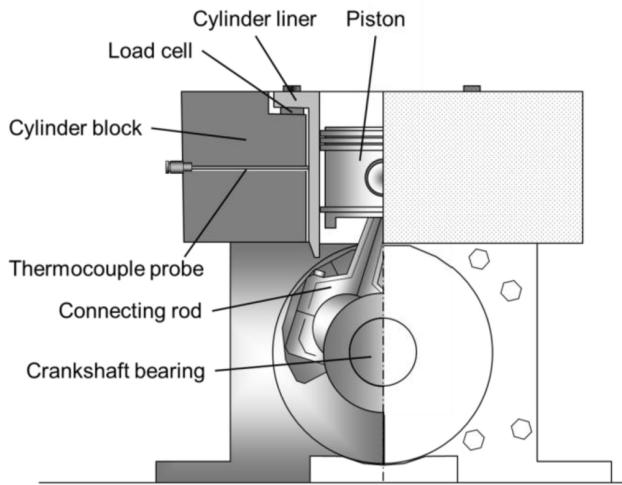


Figure 1. The schematic representation of our floating liner friction tester.

Data Analysis

We first expressed friction force as a function of the crank angle with the help of a time-course TDC signal and obtained the peak values of the friction forces just before TDC (F_{TDC}) and BDC (F_{BDC}). We calculated the dead center friction force (F_{DC}) as follows:

$$F_{DC} = (F_{TDC} + F_{BDC})/2 \quad (1)$$

We then calculated the friction energy per cycle by determining the piston position, Y , as a function of crank angle θ using a standard piston motion equation, as follows:

$$Y(\theta) = R \cos \theta + \sqrt{L^2 - R^2 \sin^2 \theta} \quad (2)$$

where R is the crank radius and L is the connecting rod length. TDC and BDC corresponds to $\theta = 0^\circ$ and $\theta = 180^\circ$, respectively. Note that θ is discrete and expressed as θ_i , where i is discrete time step 0, 1, 2... n . Friction energy, E , was calculated from friction force, F , as follows:

$$E = \sum_{i=a}^b F_i \cdot \{Y(\theta_{i+1}) - Y(\theta_i)\} \quad (3)$$

where a and b are the time steps chosen to cover the crank angle range of 360° .

Reference Data

Figure 2 shows reference data measured in our previous study [2]. The experimental procedure and the additive package of the tested oils were exactly same as this study. The relationships between dead center friction force and the product of viscosity, η , and velocity, v , (Figure 2B) and the relationships between friction energy and ηv (Figure 2C) were strikingly consistent with the parts of Stribeck curve. We identified that the transition between the boundary and mixed

lubrication regimes on the dead center friction force occurs at $\eta v = 300 \text{ mPa}\cdot\text{s}\cdot\text{rps}$. We also identified that the mixed lubrication regime was dominant in the friction energy at $\eta v < 600 \text{ mPa}\cdot\text{s}\cdot\text{rps}$, whereas the hydrodynamic lubrication regime was dominant at $\eta v > 600 \text{ mPa}\cdot\text{s}\cdot\text{rps}$.

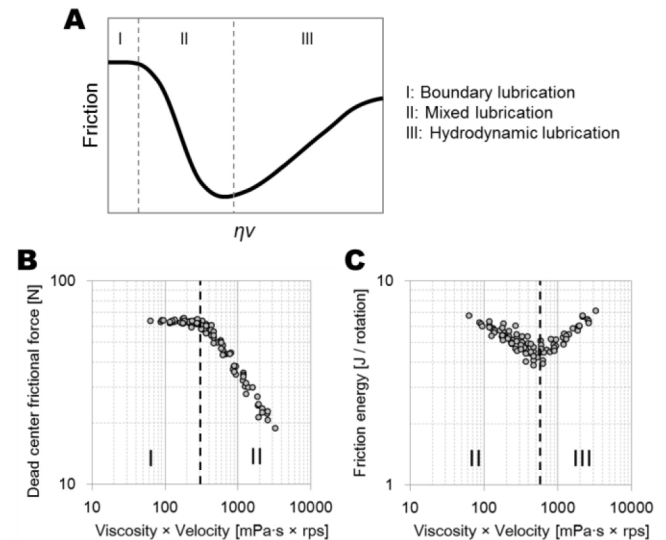


Figure 2. Reference data taken from our previous study [2]. (A) Schematic drawing of typical Stribeck curve. Regime I, II, and III correspond to boundary, mixed, and hydrodynamic lubrication regime, respectively. (B) Dead center friction force plotted against ηv . (C) Friction energy plotted against ηv .

Cameron Plint TE77 Friction Tester

We used a Cameron Plint TE77 friction tester [8] to evaluate the boundary lubrication performance of the engine oils used in this study. The test condition is described in Table 3. We used SUJ-2 cylinders coated with CrN using the PVD method and a FC250 steel plate to mimic the surface of the piston ring of our floating liner friction tester. We measured the average friction coefficient with elevating oil temperature from 60°C to 75°C and 90°C in a stepwise manner.

Table 3. The Experimental Condition of the TE77 Friction Tester

Upper Specimen	SUJ-2 cylinder, CrN-coated
Lower Specimen	FC250 steel plate
Load	100 N
Frequency	10 Hz
Stroke	8.0 mm

RESULTS AND DISCUSSION

Evaluation of Boundary Lubrication Performance

First, we examined the boundary lubrication performance of the oils using a TE77 friction tester (Figure 3), which is widely used to characterize boundary lubrication [8]. If this tester operated in the boundary lubrication regime, the friction coefficient would have been independent of the viscosity of the tested oils. The