

Study on Lubricating Oil Consumption from Evaporation of Oil-Film on Cylinder Wall for Diesel Engine

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ABSTRACT

It is effective in engine fuel economy to reduce the viscous friction by applying lubricating oil with low viscosity. The lower viscosity such as SAE0W20, however, increase lubricating oil consumption, LOC. In addition, it has become urgent to reduce the LOC because the emission of the sulfide ash, phosphorous and sulfur contents degrades the diesel particulate filter and the de-NOx catalyst, in addition to which the emission of metal oxide contents from oil additives can cause pre-ignition in highly supercharged spark ignition engines.

In order to clarify the LOC mechanism of low viscosity oils, the LOC rates were measured with a supercharged diesel engine under various operating conditions when lubricated with SAE30 or SAE10W30 test oil, and the resulting data have been compared with the rates of the evaporation from the oil-film on the cylinder wall, LOE, as predicted by the devised analysis method for multi-species component oils.

Results obtained show that the change of LOC with the test oil is mostly equal to that of LOE under the same engine operating conditions concerning the gas pressure, the temperature of cylinder wall, the viscosity and the rotational speed. The supercharging operation has the effect of reducing the LOC because the LOE decreases as the gas pressure becomes high. The devised analysis method is more useful to predict the rate of LOE for the multi-species component oil and examine the influence of the oil volatility on the LOC.

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INTRODUCTION

In order to protect the environment and reduce global warming, internal combustion engines have been required to improve fuel economy and reduce exhaust emissions, CO₂ and others. In terms of fuel economy, it is effective for diesel engines to reduce the viscous friction loss by applying piston rings with a narrow width and low tension or the multi-grade lubricating oil with low viscosity such as SAE0W20 [1]. These applications cause serious problems such as the less ability in gas-sealing of rings and the lubricating oil consumption.

Reducing the LOC is an urgent requirement because the emission with sulfide ash, phosphorous and sulfur contents from oil additives causes the soot plugging of diesel particulate filter, DPF, and the deterioration of de-NOx catalyst in the exhaust gas after-treatment device of diesel engines [2, 3, 4]. A further reason

to reduce the LOC is because emissions with metal oxide such as CaCO₃ from oil detergent additives causes the pre-ignition in down-sizing highly supercharged spark ignition engines [5, 6].

For all reciprocating engines, the upward oil transport in the cylinder bore by the piston and piston rings in a pack of ring series, the misty oil-up by the reverse blow-by gas flow through ring gaps and ring-groove clearances, the oil-droplet thrown off the top of piston, the evaporative oil consumption from the oil-film on the surfaces of the cylinder wall and the piston, among others, have been recognized as mechanisms of the LOC. The evaporation of the oil-film on the cylinder wall surface, LOE, is properly supposed to become one of major factors influencing the LOC, because multi-grade engine oils with lower viscosity and higher volatility have to be applied for the gasoline and diesel engines in order to advance in fuel economy.

With regard to oil viscosity and volatility, several studies on the LOC including the LOE have been conducted by many researchers as shown in Table 1 [7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21] so that some-reasonable and proper mechanisms for the LOC have been clarified. However, the detailed mechanism of the LOE has not been clarified sufficiently to design the components of the piston system and the lubricants for various engines.

Through many analytical studies on the LOE various prediction methods have been proposed with respect to aspects such as the oil-film thickness formed by piston rings on the cylinder wall, the pressure and temperature of the gas in the combustion chamber, the heat transfer from gas to wall, the cylinder wall temperature, the vapor pressure, vaporization heat and vapor diffusion rate of mono-species or multi-species component oils and so on but the predicted values of LOE, the estimated oil evaporation rates, are different from each other as shown in Table 1. Particularly in the recent studies, as the LOE is the mass diffusion of oil vapour from oil-film to gas, the phenomenon is similar to the heat transfer from gas to cylinder wall so that the physical laws can be understood to be common with each other [16, 17, 18, 19, 20, 21]. However, an analytical method to predict the LOE for practical multi-species component oils such as many mineral lubricants composed of many hydrocarbons different in molecular weight has yet to be perfectly developed and also it has not been reasonably verified by the experimental method with practically fired and braked engines. This paper attempts to rectify both of these issues.

Harigaya *et al.* have examined the influences of some factors on the LOE by predicting the LOE for mono-species component oils analytically [18]. Also, they have investigated the mechanism of the LOC by examining the influences of the temperature and oil-film thickness distributions of the cylinder surface under the cylinder bore distortion on the LOC or the LOE by measuring the LOC with a practical gasoline engine and predicting the LOE [22].

Further, they have proposed the analytical method to predict the LOE of multi-species component mineral oils by using the saturated vapor pressure and the vaporization heat of oil species, and investigated the LOC mechanism comparing the LOE with the LOC measured under the whole operating conditions with a practical diesel engine. The findings from these studies included that the changes of the LOE with the load rate and the rotational speed qualitatively coincide with those of the LOE predicted under the analysis model for the multi-species component oils. The major species of the evaporated oil are the hydrocarbons with smaller molecular weights in the oil because the median number of carbon atoms in the distribution of the mass fraction of evaporation for the evaporated oil is smaller than that of the mole fraction for the liquid phase of oil. The LOE markedly increases as the viscosity of oil becomes low since the mean molecular weights become lower [20, 21].

In the present study the mechanism of the LOC including the LOE from the oil-film on the cylinder wall surface is investigated and clarified experimentally and analytically with an operating supercharged diesel engine.

Table 1. Experimental and analytical study results on LOC and LOE concerning practical engines and oils

Researchers (year)	Oil	API grade	Engine	LOC	LOE
				(max)	(max)
Kawamoto, <i>et al.</i> , (1977)	mono	SAE30	G-4cyl-85x70	20 g/h	15 g/h
Wahiduzzaman, <i>et al.</i> , (1992)	multi	SAE15W40 SAE30	D-1cyl-125x140	200 g/h	8.51 g/h 3.35 g/h
Petris, <i>et al.</i> , (1997)	mono	C ₂₄ H ₅₀ C ₂₇ H ₅₆ C ₃₀ H ₆₂	D-1cyl-75x86		10 g/h 1.4 g/h 0.4 g/h
Audette, <i>et al.</i> , (1999)	multi	SAE15W40	D-1cyl-114x135		1.1 g/h/cyl.
Herbst, <i>et al.</i> , (2000)	mono		D-6cyl-128x155	50 g/h	
Yilmaz, <i>et al.</i> , (2002)	multi	SAE10W30 SAE0W30	G-4cyl-86x86	24 g/h 18 g/h	26 g/h 5 g/h
Yilmaz, <i>et al.</i> , (2004)	multi	SAE10W30 SAE0W30	G-4cyl-86x86	28 g/h 16 g/h	15 g/h 8 g/h
Inagaki, <i>et al.</i> , (2004)	mono		G-1cyl-81x77	16 g/h	17 g/h
Soejima, <i>et al.</i> , (2007)	mono	SAE10W30 SAE30	D-4cyl-91.1x90	100 g/h 25 g/h	160 g/h 20 g/h
Harigaya, <i>et al.</i> , (2009, 2014)	mono	SAE10W30 SAE30	G-4cyl-89x80.3	8 g/h/cyl.	2.43 g/h/cyl. 0.14 g/h/cyl.
Harigaya, <i>et al.</i> , (2016)	multi	C _{22.5} H ₄₇	D-4cyl-86x96	10.9 g/h/cyl.	7.90 g/h/cyl.

The validity of the analytical method in the evaporation model for the multi-species component oils is verified concerning various factors influencing the LOE such as the operating conditions with or without supercharging, the gas pressure in the combustion chamber, the rotational speed, the temperature of cylinder wall surface, the oil-film thickness formed by piston rings under the mixed lubrication in the ring-pack and the viscosity and volatility of the oil. The verifying method is measuring the LOC under the various operating conditions of the test engine by using two kinds of test oils, the mono-grade oil SAE30 and the multi-grade oil SAE10W30, having volatilities different from each other and the same viscosity properties at a temperature of around 140 degrees centigrade. And the difference of measured LOC rates between them is compared with that of predicted LOE rates between them, or the influences of some factors on the measured LOC are compared with those on the estimated LOE to study how their characteristics are related with each other.

Furthermore, in order to examine the applying availability of the super-low viscosity oils indispensable for the development of innovatively advanced engines, the detailed LOC contents for the multi-species component mineral oil should be clarified so that the measures to prevent the increase of LOC can be more effectively investigated according to the engine design.

EVAPORATION OF MULTI-SPECIES COMPONENT OIL

The evaporation of the lubricating oil is analyzed under the prediction model based on Fick's law of diffusion where the oil vapor diffuses from the oil-film adhering onto the cylinder wall to the combustion gas as shown in Figure 1. The thickness of the oil-film is changed on the cylinder wall surface where the oil-film is formed between the wall surface and the top compression piston ring during the down-stroke of the piston as described in the previous section.

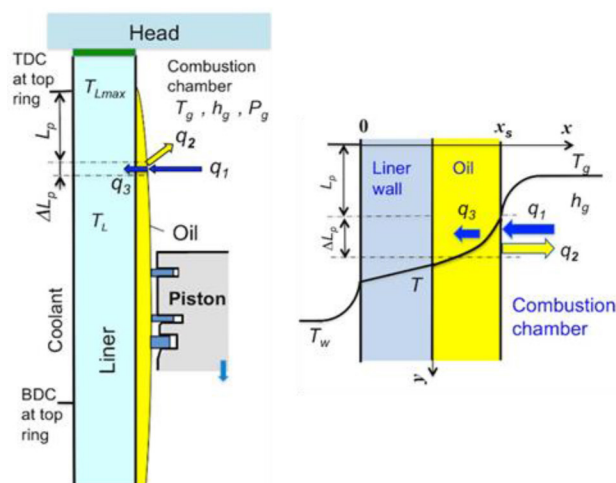


Figure 1. Evaporation model for oil-film on cylinder wall

The evaporation rate is predicted by summing the estimated values of local diffusive mass flux varying at the piston position. They are summed during the individual time from the top dead center to the next top dead center when the concerned oil-film is deposited to the charged and fired gas, the exhaust gas, in the cycle of engine. As

described in the previously, the prediction model has been developed for multi-species component mineral oils, where the evaporation depends on several factors such as the pressure, temperature and convectional velocity of the combustion gas, the heat transfer between gas and oil-film, the surface temperatures of cylinder wall and oil-film, the thickness of the oil-film and the physical properties of the oil including oil species as below.

The evaporative consumption of the multi-species component oils on the cylinder wall is predicted under the following physical and analytical principles. The surface temperature of oil-film varies under the energy balance among the heat flowing from or to the gas or the cylinder wall surface and the heat dispersing with the evaporation. The oil species consist of hydrocarbons different in molecular weight. Each species evaporates as the diffusion governed by the saturated vapor pressure characteristics changing with the temperature. The instantaneous evaporation rate changes with the piston stroke under the engine operation condition. The pressure and the temperature of the gas in the combustion chamber are measured and estimated respectively according to the practical data from the engine tests.

Evaporation Analysis

The evaporation of oil is analyzed under the diffusion based on Fick's law and the energy conservation balancing at the cylinder wall under the engine operation as well as the analysis model previously proposed for mono-species component oils by the authors [18, 19]. The following assumptions are made to predict the evaporation for the practical lubricating oil like the multi-species component mineral oil.

1. Lubricating oil is a mixture of n-Paraffin composed of hydrocarbons with various carbon numbers, and the influence of oil additives is not taken into account.
2. The oil is an ideal solution governed by Dalton's law and Raoult's law.
3. The direction of heat conduction in the lubricating oil and the cylinder wall is perpendicular to the wall surface.
4. The temperature of cylinder wall varies along the axial direction but the mean temperature during the cycle is constant while the temperature varies with time in the oil-film and the cylinder wall.

Under unsteady energy conservation the equation for the evaporative heat flux q_2 at the oil surface is expressed with the convection heat flux q_1 and the conduction heat flux q_3 by considering the heat transfer from the combustion gas through the oil-film and the cylinder wall to the cooling water as below.

$$q_2 = q_1 - q_3 \quad (1)$$

The temperature of oil-film and cylinder wall T is governed by the unsteady one-dimensional heat conduction equation.

$$\rho C \frac{\partial T}{\partial t} = \kappa \frac{\partial^2 T}{\partial x^2}$$

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As expressed in Equation 3 the convection heat flux q_1 is obtained by Newton's law of cooling.

$$q_1 = h_g (T_g - T_s) \quad (3)$$

where T_g and T_s are the temperature of the combustion gas and the one at the oil-film surface, respectively. The heat transfer coefficient h_g is estimated by Woschni's formula on the heat transfer in the cylinder of diesel engines and applied to the present study.

The heat flux due to the oil evaporation q_2 is given by summing the product q_{2i} of the evaporation mass flux $\dot{m}_i$ and the vaporization heat Lv_i for each oil species i as below.

$$q_2 = \sum_{i=1}^n q_{2i} \quad (4)$$

$$q_{2i} = Lv_i \dot{m}_i \quad (5)$$

The vaporization heat Lv_i is estimated by Watson's equation. The flow of gas inside the combustion chamber is assumed to be turbulent and the mass fraction of the oil species vapor Y_i in the combustion chamber at the infinite distance from the oil/gas interface is assumed to be $Y_{\infty i} = 0$. Therefore, the evaporation mass flux $\dot{m}_i$ is expressed by Equations 6 and 7.

$$\dot{m}_i = \frac{h_g}{C_p} \ln \left(1 + \frac{Y_{Si}}{1 - Y_{Si}} \right) \quad (6)$$

$$Y_{Si} = \frac{Pv_i X_i W_i}{\sum_{i=1}^n Pv_i X_i W_i + P_g W_g} \quad (7)$$

where C_p is the specific heat at constant pressure of the gas and Y_{Si} is the mass fraction of oil species i at the oil/gas interface, and P_g is the gas pressure, and W_g is the molecular weight of gas and W_i is the one of oil species i , and Pv_i is the saturated vapor pressure of oil species i , and X_i is the mole fraction under the liquid phase of oil species i . The overall saturated vapor pressure for the multi-species component oil Pv is estimated by multiplying the products of Pv_i and X_i as below.

$$Pv = \sum_{i=1}^n Pv_i X_i \quad (8)$$

The evaporation mass flux $\dot{m}$ at the position with a distance L_p from the top rig reversal position is obtained by summing $\dot{m}_i$ of each oil species as below.

$$\dot{m} = \sum_{i=1}^n \dot{m}_i \quad (9)$$

The mass fraction of the oil species vapor Y_i is defined as the ratio of the evaporation mass fluxes $\dot{m}_i$ and $\dot{m}$ as below.

$$Y_i = \frac{\dot{m}_i}{\dot{m}} \quad (10)$$

The evaporation rate m at an arbitrary crank angle is obtained by multiplying the products of $\dot{m}$ and the surface area exposed to gas $A_{L_p} = \pi D A_{L_p}$, and the total evaporation rate during the cycle m_{all} is obtained as below.

$$m = \sum_{L_p=0}^{L_p=St} \dot{m}_{L_p} A_{L_p} \quad (11)$$

$$m_{all} = \int_0^{\tau} m dt \quad (12)$$

where D is the cylinder bore and St is the piston stroke, and τ is the cyclic period.

Then the oil consumption rate per one hour operation of engine Qoe is obtained by using the number of cylinder Z and the cyclic period τ as below.

$$Qoe = \frac{m_{all} Z}{\tau} \quad (13)$$

As described above, the total evaporation rate during the cycle m_{all} is estimated by using the evaporation mass flux $\dot{m}_i$ of oil species i predicted with the mole fraction X_i , the saturated vapor pressure Pv_i , the vaporization heat Lv_i , the gas pressure, the temperatures of gas, oil-film and cylinder wall, and the heat transfer from gas to oil-film varying cyclically at the axial position of the cylinder wall.

Vapor Pressure

In the present study the evaporation characteristics are examined for the ideal mixture of oil species composed of n-Paraffin hydrocarbons $C_n H_{2n+2}$ with various numbers of carbon atoms n and saturated vapor pressures Pv as the multi-species component oils. So it is necessary to determine the values of Pv varying with the carbon number and the temperature. Figure 2 shows the relation of Pv with the temperature as functions of the n-Paraffin hydrocarbons $C_n H_{2n+2}$. These values of vapor pressure Pv are estimated with the equations recommended by API and Chickos where they are applicable for n of 10 to 60 [23, 24, 25, 26]. As shown, the values of Pv_n decrease with the number of carbon atoms n , and the Pv_n for n of 20 is around 1000 times greater than that for n of 30 at the temperature of 100 degrees centigrade.

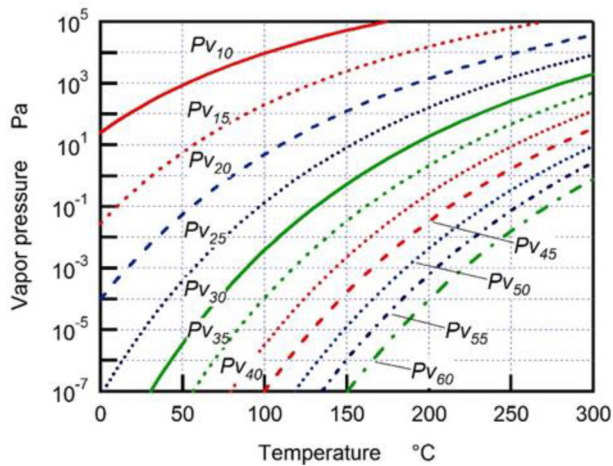


Figure 2. Vapor pressure for n-Paraffins from C_{10} to C_{60}

MEASUREMENT OF LOC WITH TEST ENGINE

Test Equipment and Method

The test engine is a water-cooled, in-line-four cylinder, four-stroke cycle diesel engine with specifications shown in Table 2. It can be operated under the naturally aspirated condition denoted as the normal operation and the turbo-charged condition due to an exhaust gas turbine denoted as the turbo-charged operation.

Test lubricating oils are the multi-grade mineral oil SAE10W30 and the mono-grade mineral oil SAE30 for light duty diesel engines and have the physical properties and the evaporation properties shown in Table 3 and Figures 3 and 4, respectively. The viscosities at 140 degrees centigrade were adjusted to the same for these test oils used in the present experimental and analytical study.

As shown in Figure 3 the evaporation properties were measured by the thermo-gravimetric method. The volatility of SAE10W30 is higher than that of SAE30 at the same temperature and the difference between the evaporation rates of these oils becomes large as the temperature rises higher than around 130 degrees centigrade. Also as shown in Figure 4 they have respective characteristics on the change of the vapor pressure with the temperature where the vapor pressure is predicted by Equation 8 as the mixture of the oil species, n-Paraffins of hydrocarbons, corresponding to the estimated distribution of mole fraction shown in Figure 5.

As shown in Figure 5 each of these oils has the median number of carbon atoms average in the mole fraction that reduces to become low in viscosity and high in vapor pressure as the oil viscosity grade is changed from SAE30 to SAE10W30.

Table 2. Specifications of test diesel engine

Description	Normal	Turbo-charged
Bore × Stroke, mm	91.1 × 90.0	
Piston displacement, cc	2347	
Compression ratio	21	
Combustion chamber	Swirl chamber type	
Valve system	OHV – SOHC	
Cylinder liner	Dry liner type	
Rated output, kW/rpm	55/4200	70/4200
Max torque, Nm/rpm	147/2500	181/3000

Table 3. Physical properties of test oils

Specification	SAE10W30	SAE30
Specific gravity (14 °C)	0.882	0.902
Kinematic viscosity, cSt		
40 °C	74.9	108.1
100 °C	11.3	12.7
140 °C	5.4	5.4
HTHS viscosity, mPas		
150 °C	3.25	
Average molecular weight	394	522
Specific heat, kJ/kgK		
20 °C	1.89	1.87
200 °C	2.55	2.52
Thermal conductivity, W/mK		
20 °C	0.134	0.131
200 °C	0.122	0.119
Vapor pressure, Pa		
100 °C	2.7×10^0	1.3×10^{-1}
200 °C	1.2×10^2	2.0×10^1
300 °C	1.2×10^4	1.2×10^3
Critical pressure, Pa	9.8×10^5	7.8×10^5
Critical temperature, °C	588	657

The rate of oil consumption was measured by the weight method where the weight decreasing rate of the oil in the oil-pan with time is measured. In order to measure the weight of the oil as precisely and quickly as possible, the measurement was conducted by detecting the weight of the oil drained in the oil-tank entirely isolated from the engine body and suspended with stabilized rods mounted to the isolated outer structure [16].

The signal of the load transducer to detect the weight was also stabilized with a low-pass filter so that the decreasing rate of oil weight with time can be indicated as a regression line of recorded values. The measurement became effective after both the temperature of the oil and that of the cooling water became stable. The decreasing rate of oil weight per one hour is defined as the oil consumption rate Q_{oc} .