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A Study on Engine Oil Consumption

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엔진 오일 소모에 관한 연구

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Abstract – 피스톤-실린더-링 틈새를 통해 일어나는 오일소모와 브로바이가스 증가는 최소화 되어야 하며, 한편으로는 연료저감 및 성능증가 개선 측면에서 피스톤 링 팩의 마찰 손실도 줄일 필요가 있다. 이러한 두 가지 측면에서, 피스톤 링 팩의 최적 설계에 대한 연구가 수행되어야 한다. 따라서 오일소모 및 브로바이가스의 양은 엔진개발과정 및 필드에서의 엔진운전 중에 엔진의 상태가 좋은지 나쁜지를 판단하는 중요한 요인이 된다. 본 연구의 목적은 연소실 내로의 오일 흐름 량과 피스톤 링 팩을 지나 아래로 내려가는 가스흐름을 계산하여 엔진오일 소모 및 브로바이가스를 예측하는 컴퓨터 프로그램을 개발하는 것이다. 향 후 본 프로그램을 이용하여 엔진의 상태를 미리 예측할 수 있을 것으로 본다.

Key words – oil consumption, blow-by, piston-ring pack, top ring gap, inter-ring pressure.

1. Introduction

The status of piston ring-pack lubrication turns out to be fluid film lubrication or boundary lubrication by turns depending on ring profiles, piston motions and fluctuations in inter-ring pressure [1-8]. It is desired that fluid film lubrication is maximized for the reduction of friction and protection against piston ring wear. Meanwhile, the oil consumption and blow-by gas would be better in small amount in terms of saving of resources and engine efficiency. It is important in piston and ring-pack lubrication how these two aspects are combined.

During engine firing, the pressures in the combustion chamber and inter-ring pressures are required at every crank angle. A pressure transducer measures the cylinder pressure and the inter-ring pressures can be calculated by the crevice flow model. Assuming the lubrication between piston ring and cylinder wall is hydrodynamic lubrication, the Reynolds' equation is solved by finite difference method. Then oil film thickness and ring friction at the ring face are obtained. If the film thickness is sufficiently small to indicate boundary lubrication, the friction calculation follows the method for boundary lubrication. Furthermore, the blow-by gas and oil consumption can be calculated.

Oil Consumption in a four stroke reciprocating engine occurs only during about a few degrees when the gas flowing through the top ring gap first reverses, and flows into the combustion chamber from the second land [9].

Suppose that the oil consumed comes from the second land. Then oil Consumption would be expected to correlate with lubricant properties at the temperature of the second land [10].

For modern engines using modern lubricant, where the engine components and lubricant characteristics have been carefully tuned to minimize oil consumption, the crown land was shown to remain dry [11] under various operating conditions with different oils, after oil being forced past the top ring. The observed oil consumption rates and oil film thickness may not be similar in older

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engines, upon which previous mechanisms of oil consumption are based.

The previous theory of oil consumption that oil is forced past the top ring and is later consumed may not be valid, because an accumulation of oil on the crown land is usually not observed. That is, oil evaporation due to the high gas temperature of a combustion chamber based on the oil film thickness distribution on the liner as the piston moves down, and oil throw-off at the top ring due to inertia effect at TDC reversal position are not expected.

A correlation between measured oil consumption rate and oil consumption rates calculated from oil transported by top ring face, based on the oil film thickness measurements under the top ring, is inconclusive.

So, the oil consumption of a modern engine occurs mainly due to the reverse gas flow entraining oil through the top ring gap. If this mechanism is correct, then changing the timing of the top ring reversal and hence the reverse gas flow through the top ring gap would affect the oil consumption of an engine.

Further, the gas-driven flows of viscous lubricants depend on both surface tension, σ and viscosity, μ [12]. The ratio of viscous to surface tension forces is called the Taylor's number, $T_a = \mu U_a / \sigma$ where U_a is the gas flow velocity over oil puddle.

So the present oil consumption concept is related with the oil entrainment in blow-back gases considering together with Taylor's Number.

In this study, three gasoline engines that are same family, and one large diesel engine are observed.

The purpose of this paper is to predict the lubrication status for the piston ring pack and thereby guide as optimum lubrication. Even if all the real firing states cannot be considered and the complicated physical phenomenon cannot be well modeled, it is useful to inform design through parametric study in a short time.

2. Governing Equations

2-1. Piston ring motion equation

$$M_r \frac{d^2h}{dt^2} = F_p + F_f + F_i + F_s + F_{asp} \tag{1}$$

where M_r is the mass of each ring, h the oil film thickness, t time, F_p the pressure force, F_f the friction force, F_i the inertia force, F_s the radial hydrodynamic force and F_{sap} the radial contact force on the ring face.

2-2. Gas flow equations through piston ring pack crevice

$$\frac{m_{02}}{P_{02}}\frac{dP_2}{dt} = \dot{m}_{12} - \dot{m}_{23} \tag{2}$$

$$\frac{m_{03}}{P_{03}}\frac{dP_3}{dt} = \dot{m}_{12} - \dot{m}_{23} - \dot{m}_{34} - \dot{m}_{35} \tag{3}$$

$$\frac{m_{04}}{P_{04}}\frac{dP_4}{dt} = \dot{m}_{34} - \dot{m}_{45} \tag{4}$$

$$\frac{m_{05}}{P_{05}}\frac{dP_5}{dt} = \dot{m}_{35} - \dot{m}_{45} - \dot{m}_{56} - \dot{m}_{57} \tag{5}$$

$$\frac{m_{06}}{P_{06}}\frac{dP_6}{dt} = \dot{m}_{56} - \dot{m}_{76} - \dot{m}_{6c_2} - \dot{m}_{6c_1} \tag{6}$$

$$\frac{m_{07}}{P_{07}}\frac{dP_7}{dt} = \dot{m}_{57} - \dot{m}_{76} - \dot{m}_{7c_1} \tag{7}$$

where m_{oi} is the initial mass of i volume, $\dot{m}_{ij}$ the mass flow rate into j volume from i volume, P_i the pressure of i volume and P_{oi} the initial pressure of i volume. As shown on Fig. 1, 1 indicates the top land clearance, 2 the volume behind the top ring, 3 the volume of second land clearance, 4 the volume behind the second ring, 5 the volume of the third land clearance, 6 the volume behind the oil ring and 7 the volume between the oil ring rails. c_1 and c_2 indicate the crankcase reached through the piston skirt clearance and oil drain hole respectively.

The gas mass flow rates between adjacent lands are calculated using the orifice flow equation such that

$$\dot{m} = C_d \rho A c \eta . \tag{8}$$

where C_d is the discharge flow coefficient, ρ the gas density, A the flow area associated with the orifice, c the speed of sound and η the compressibility factor.

The blow-by gas flow, $\dot{m}$ is

$$\dot{m} = \dot{m}_{7c_1} + \dot{m}_{6c_1} + \dot{m}_{6c_2} . \tag{9}$$

2-3. Reynolds equation for film lubrication

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial P}{\partial x}\right) = -6\mu U \frac{\partial h}{\partial x} + 12\mu \frac{\partial h}{\partial t} \tag{10}$$

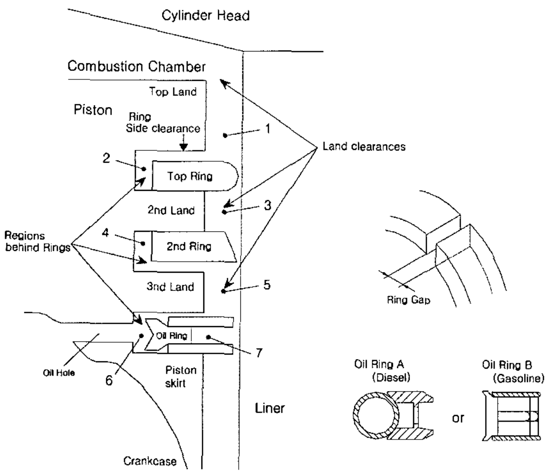

Fig. 1. Schematic diagram of piston-cylinder-ring pack.

where P is the oil film pressure, μ oil dynamic viscosity, U the velocity of piston-ring, h the oil film thickness and x the axial coordinate.

2-4. Friction Calculation

Friction for film lubrication, F_f

$$F_f = \int\left[\frac{h}{2}\frac{\partial P}{\partial x} + \mu\frac{U}{h}\right](\pi D_b)dx \quad (11)$$

where D_b is the diameter of cylinder bore.

Friction for boundary lubrication, $F_{f,asp}$

$$F_{f,asp} = c_f F_{asp} \quad (12)$$

where c_f is the friction coefficient and F_{asp} the radial contact force on the ring face.

2-5. Oil consumption

For modern engines using modern lubricant, where the engine components and lubricant characteristics have been carefully turned to minimize oil consumption, the observed oil consumption rates and oil film thickness may not be similar in older engines. Changing the timing of the top ring reversal, the reverse gas flow through the top ring gap would mainly affect the oil consumption of the modern engine [11]. The gas-driven flows of viscous lubricants depend on both surface tension and viscosity. The ratio of viscous to surface tension forces is called the Taylor's number. The related theory of oil consumption is called puddle theory [11]. The side view of piston showing

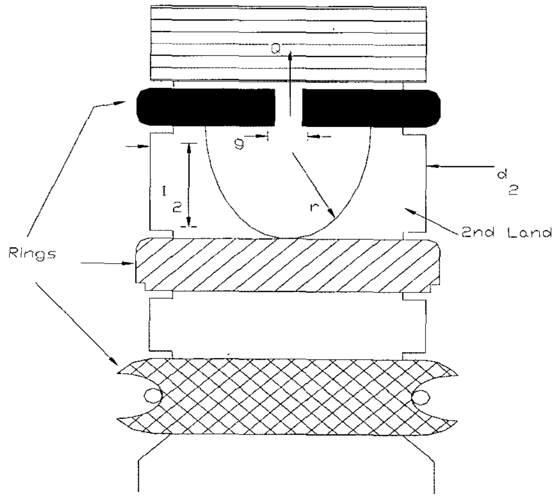

Fig. 2. Side view of piston showing dimensions relevant to oil consumption.

dimensions relevant to oil consumption is shown as Fig. 2. Finally, the oil consumption can be expressed as follow.

$$Oil\_Consumption = 3\times10^4 RPM\rho h_i h^* A_{ref} A^*\left(\frac{g}{hr}\right) \quad (13)$$

where ρ the density of oil, A_{ref} the area of second land oil puddle beneath top ring gap, $A_{ref} = \frac{\pi l_2^2}{2}$, l_2 the length of the second land, A^* the non-dimensionalized second land puddle area, $A^* = 0.03056 (T_a)^{-2/3}$ for diesel engine, $A^* = 0.3056(T_a)^{-2/3}$ for gasoline engine, h_i the second land oil film thickness before top ring reversal, $h^* = (h_i - h_f)/h_i$ the non-dimensionalized change in second land oil film thickness correlated with $h^* = 1.30(U_a t_{max})(T_a)^{1/3} \left(\frac{\mu_a}{\mu}\right) + 0.61$, t_{max} time for Q to reach Q_{MAX} after top ring reversal, μ_a air dynamic viscosity, μ oil dynamic viscosity, Q mass flow rate of blowby gas through top ring gap, Q_{MAX} maximum value of Q, h_f the final height of the oil film, and RPM the speed of engine.

3. Results

In the Table 1, the specifications of four test engines are listed.

Table 1. Engine specifications

Items	Specifications			
	TBI-2 V	MPI-2 V	DOHC (4 V)	4 V
Fuel		Gasoline		Diesel
Engine Type		L-4		L-6
Displacement (L)		1.5		12
Bore Diameter (mm)		76.5		130
Stroke(mm)		81.5		155
Connecting Rod Length (mm)		130		260
Compression Ratio	9.0	9.5	9.5	17.2

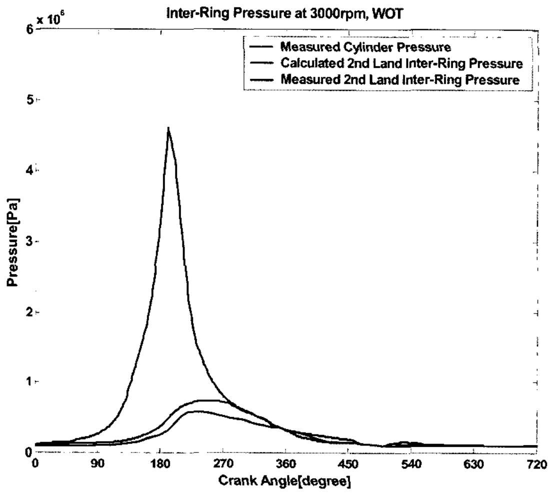

Fig. 3. Comparison between measured 2nd land pressure [13] and calculated 2nd land pressure at 3000 rpm, WOT(2V-MPI).

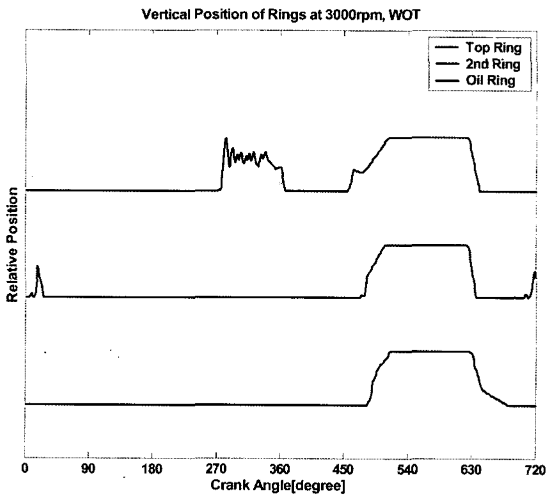

Fig. 4. Ring axial motion at 3000 rpm, WOT(2V-MPI).

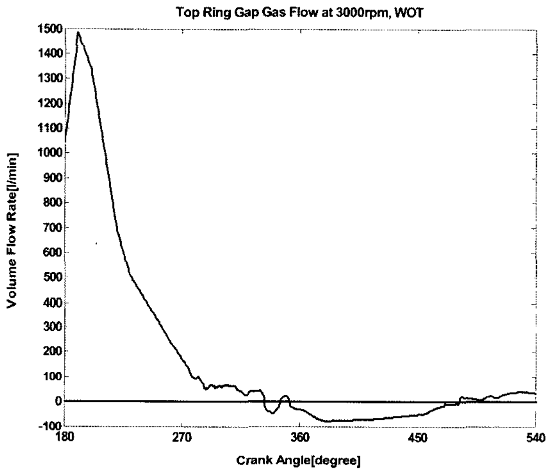

Fig. 5. Volume flow rate through top ring gap at 3000 rpm, WOT(2V-MPI)

In order to examine the reliability of this developed computer program, the measured 2nd land pressure [13] was compared with the calculated value. For example, in a 1.5 L 2 V-MPI-gasoline engine operating at 3000 rpm and full load, the result showed quite a good correlation, Fig. 3. The axial motions of the piston rings are shown on Fig. 4 and the gas flow rate through the top ring gap on Fig. 5. Those explain the mechanism of oil consumption.

For a 12 L 4 V-diesel engine operating at 1800 rpm and full load, the same results as the gasoline engine are shown on Fig. 6, Fig. 7 and Fig. 8.

Also, for several firing engines, the calculated values of blow-by and oil consumption are compared to the ranges of measured values in Fig. 9 and Fig, 10. The calculated blow-by is scattered more widely compared to the measured range. Meanwhile, the calculated oil

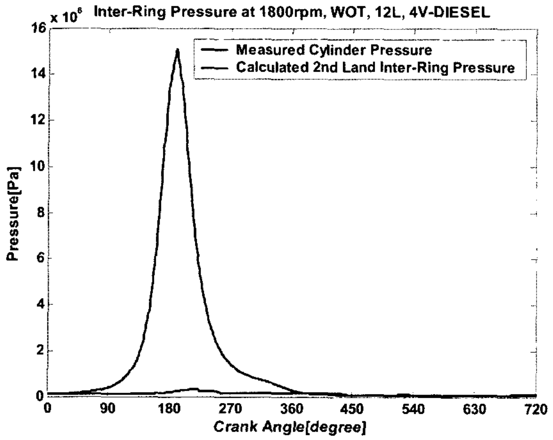

Fig. 6. Calculated 2nd land pressure at 1800 rpm, WOT (4 V-Diesel).

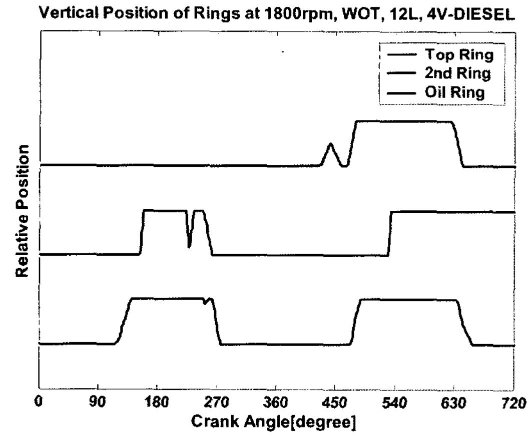

Fig. 7. Ring axial motion at 1800 rpm, WOT(4V-Diesel).

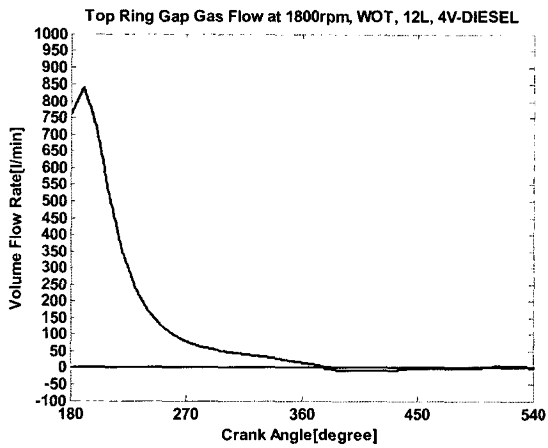

Fig. 8. Volume flow rate through top ring gap at 1800 rpm, WOT(4 V-Diesel).

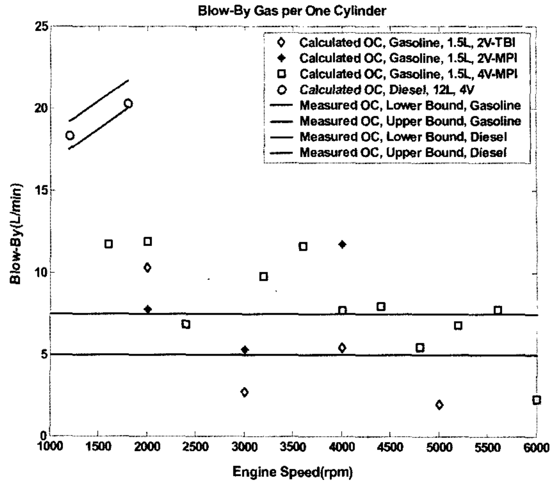

Fig. 9. Blow-by calculation results at various speeds.

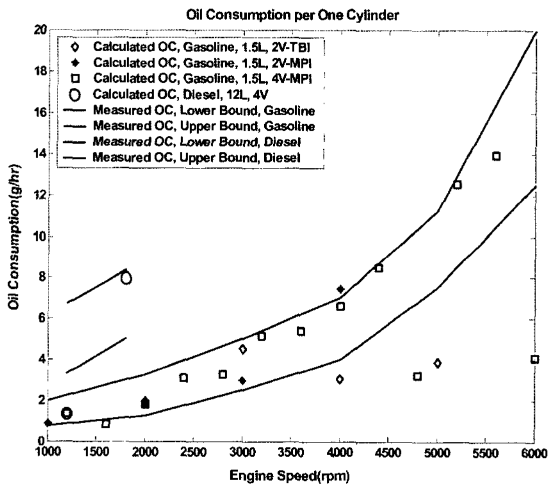

Fig. 10. Oil consumption calculation results at various speeds.

consumption was well correlated with the measured range, except in a few cases.

4. Conclusions

The calculated values of oil consumption are well correlated with the ranges of measured values for several firing engines. Using the developed computer program, the real time condition of an engine in terms of oil consumption can be predicted with good reliance.

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