Analysis of Ring Pack Lubrication

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This paper describes a method developed for the simulation of ring pack lubrication characteristic in an internal combustion engine. In general, the quantity of oil supply for piston ring lubrication may be insufficient in filling the entire volume formed at the interference between the piston ring and the cylinder liner. Thus the oil starvation condition should be considered in analyzing piston ring lubrication. In order to reasonably estimate the amount of oil left over on the cylinder liner, the flow rate at the posterior portion of the interface should be calculated with an adequate boundary condition that confirms flow continuity condition. In this analysis, oil starvation and open-end boundary conditions are considered at the inlet and outlet of the piston rings. The lubrication characteristic of each piston ring is obtained by an iterative method with sequential steps. It is revealed that piston rings are operated under oil starvation in most operating cycles and the result under these conditions are quite different from that with the fully-flooded assumption.

Key Words : Ring Pack Lubrication, Oil Transport, Open-End Boundary Condition, Minimum Film Thickness

Nomenclature

- h : Height function at the interface
- h_{ex} : Oil film thickness outside of the piston ring interface
- h_{out} : Oil film thickness posterior part of the piston ring interface
- P : Hydrodynamic pressure
- P_{sat} : Saturation pressure
- U : Axial velocity of the piston ring
- x : Axial coordinate system
- μ : Viscosity of the oil

1. Introduction

Generally, an oil scraper ring exists in a ring pack of an internal combustion engine to prevent the consumption of excessive oil that is caused by oil evaporation by controlling the amount of oil transported. In conclusion, the greater frictional force is generated at the interface between the piston ring and the cylinder liner than that expected under the fully-flooded condition, because there is insufficient oil to generate hydrodynamic pressure on the entire surface of the piston ring, and this is due to increasing metal contact (Economou, Dowson, et al., 1982). Because the frictional forces on the piston rings take up a great part of the total mechanical friction in a reciprocating engine, investigation of the theories and experimental designs to minimize these frictional forces has been a critical issue.

To estimate the characteristics of ring pack lubrication under oil starvation, the amount of oil supplied to the interface between the piston ring and the cylinder liner should be accurately calculated. This quantity can be determined from a single ring analysis with proper boundary conditions in considering the oil transport phenome-

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non.

Integrated models of ring pack lubrication and performance were proposed in the work of Dowson et. al. (Dowson, Economou, et. al., 1979) and Keribar et. al. (Keribar Dursunkaya et. al., 1991). However the oil transport model is unclear in many researches. So a more certain procedure is proposed in this paper.

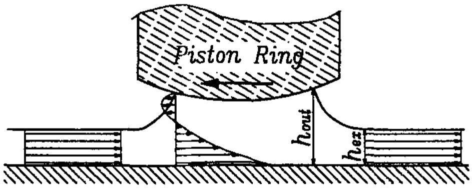
Distribution of inter-ring gas pressure, effective width of the piston ring and the profile of hydrodynamic pressure at the interface are the dominant elements for proper analysis. Exhaust gas pressure distribution in the ring pack is obtained by a simplified method which takes into consideration the gas flow as the flow of an ideal gas, which has a constant discharge coefficient through the square edge orifices (Ting and Mayer, 1973). However, the way to determine the hydrodynamic pressure profile at the interface and an effective width has not been fully verified. Many researchers have proposed many reasonable boundary conditions and analytic methods. And the ring pack lubrication phenomenon may be analyzed on the basis of these methods.

In this analysis, ring pack lubrication analysis considering oil transport is performed by applying open-end boundary condition, which satisfies flow continuity condition.

2. Analysis of Single Piston Ring Lubrication

A one-dimensional Reynolds equation is utilized to analyze the ring pack lubrication. A one-dimensional analysis is adequate for the concentric piston ring and the cylinder liner lubrication, because

1) Almost all the thrust load acting on the piston is supported at the interface between the piston skirt and the cylinder liner, and even the thrust load acting on the piston crown does not affect piston ring motion because there is no interference between the piston groove and piston ring axially. Thus it can be assumed that there is no thrust load acting on the piston rings.



(a) Inlet (b) Interface part (c) Outlet

Fig. 1 Oil flow model

2) Circumferential oil flow may be ignored except at the region of or near the ring gap.
3) The computing time takes too long to consider two-dimensional effects with oil transport analysis.

A one-dimensional Reynolds equation is as follows.

$$\frac{\partial}{\partial x}\left[\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right]=-6U\frac{\partial h}{\partial x}+12\frac{dh}{dt} \tag{1}$$

For the convenience of defining coordinate system, it can be assumed that the piston ring is stationary and that the cylinder liner reciprocates in the opposite direction of the piston ring. Also, oil flows through the interface, as shown in Fig. 1. When the amount of oil is not sufficient to fill the gap between the ring face and the cylinder liner, the ring operates under oil starvation. The oil in front of the piston ring edge has not formed a laminar flow and moves rigidly at the same velocity as the cylinder wall. The oil which flows into the interface becomes laminar and generates hydrodynamic pressure, and then the flow changes into a rigid motion after it leaves the back edge. From the continuity condition, the thickness of the film left on the cylinder liner behind the piston ring can be calculated by Eq. 2 (Han and Lee, 1999).

$$Uh_{ex}=\frac{-h^3{}_{out}}{12\mu}\frac{\partial P}{\partial x}+\frac{h_{out}U}{2} \tag{2}$$

Oil left on the cylinder liner may be supplied to the following ring, or used to lubricate itself in the case of the top ring. Some of this oil may be evaporated or burnt by combustion heat (Wahiduzzaman and Keribar et. al., 1992). However, oil reduction is not concerned in this analysis.

Variation of the volume formed at the interface as well as flow through the clearance is considered (Gulwadi, 1995).

For a positive load-carrying capacity the thickness of the oil film must decrease in the sliding direction, and hydrodynamic pressure rises in this section. Hydrodynamic pressure drops rapidly in the opposite case(Brown and Hamilton, 1977). To imply this phenomenon and the process of oil transport through the ring pack in the analysis of piston ring lubrication, adequate boundary conditions are necessary to satisfy the continuity of the flow rate. There were many proposed boundary conditions. At the early stage of analysis of piston ring lubrication, full Sommerfeld boundary condition by Sommerfeld (Sommerfeld, 1904) and half Sommerfeld boundary condition by Gümbel (Gümbel, 1921) were proposed. After that Reynolds cavitation boundary condition (Dowson, Economou et. al., 1979), separation condition (Coyne and Elrod, 1970) and open-cavitation condition (Ma, 1996) were applied. Open-end boundary assumption (Han and Lee, 1999) was proposed for reasonable pressure status at the interface and to confirm flow continuity, and thus open-end boundary assumption is used for ring pack lubrication in this analysis.

The hydrodynamic pressure profile with open-end boundary assumption is shown in Fig. 2.

Figure 2 shows the pressure distribution in the oil film formed at the interface between the piston ring and the cylinder liner, and the region of the oil film enveloping the surface of the piston ring partially (from x_1 to x_4) where the entire width of the piston ring is from 0 to x_5. If the amount of oil supplied is not sufficient in filling the entire volume of the anterior portion of the interface, there is no oil film between 0 and x_1, and when the amount of oil seeping through the interface is not enough to fill the open space between the rear side of the piston ring and the cylinder liner, the posterior of the piston ring surface operates without lubricating oil(from x_4 to x_5). The effective rear width of the piston ring can be set with open-end boundary assumption.

The pressure boundary condition is proposed from Eq. 3 to Eq. 5.

$$at\ x=x_1,\ p=p_1 \tag{3}$$

$$at\ x=x_4,\ p=p_2 \tag{4}$$

$$at\ x=x_3,\ p=p_{sat} \tag{5}$$

The saturation pressure is set at atmospheric pressure for this analysis.

This pressure boundary condition is applied only for the case where the hydrodynamic pressure goes below the saturation pressure. This phenomenon occurs during the mid stroke.

To verify that the analysis with open-end boundary assumption satisfies the flow continuity condition, flow rates at the inlet and at the outlet through the interface supplied with uniformly distributed oil are compared in Fig. 3, and the effective widths of the piston rings are compared in Fig. 4. The piston ring design data and operating conditions come from a passenger car engine. It is assumed that evenly distributed oil with 2μm thickness covers the cylinder liner.

Some discrepancies are found in Fig. 3 between

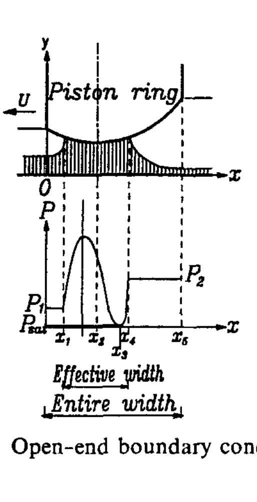
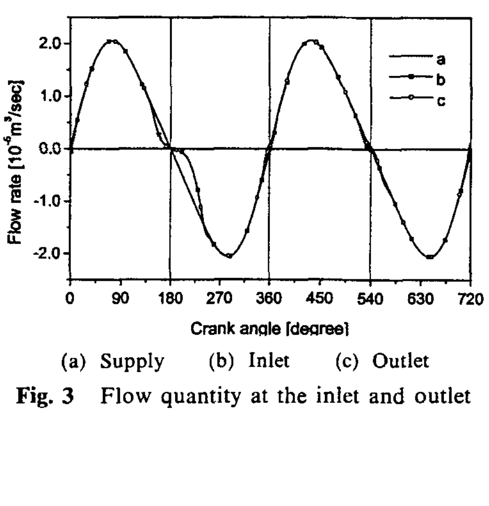


Fig. 2 Open-end boundary condition



(a) Supply (b) Inlet (c) Outlet

Fig. 3 Flow quantity at the inlet and outlet

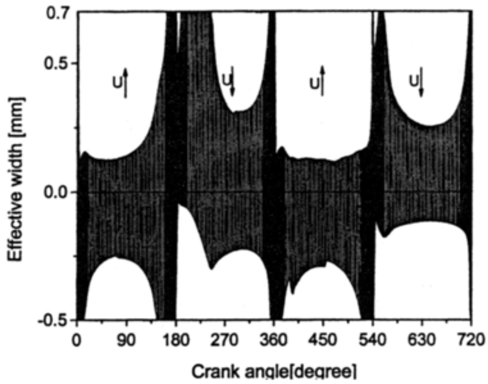
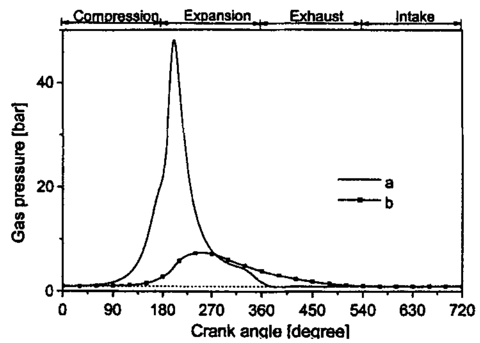
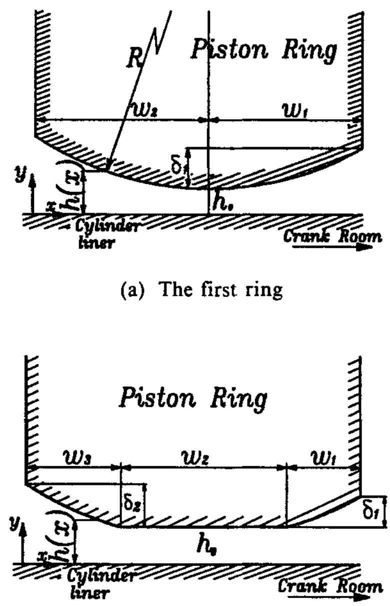


Fig. 4 Effective width of the piston ring



(a) Combustion chamber (b) Between 1st-2nd ring

Fig. 5 Inter-ring gas pressure



(a) The first ring

(b) The second ring

Fig. 6 Surface shape of the compression ring

the flow rate of the supplied oil and the inlet flow rate because the amount of oil is supplied more than sufficient to fill the gap between the piston ring surface and the cylinder liner. The excessive oil accumulates in front of the piston rings.

The effective width of the piston ring is about 30 to 80 percent of the entire width, and this portion drastically increases near the dead centers to up to 100 percent.

To consider the effect of oil transport, more than 70 nodes are needed because the oil flow rate is greatly affected by the gradient of hydrodynamic pressure distribution.

3. Application to the Ring Pack Lubrication

To generate the pressure boundary condition for the piston ring, blow-by and dynamic analysis should be performed first. The gas flow through the ring gap is assumed to be of an ideal gas with a constant discharge coefficient in one-dimensional flow passing through the square edge orifices (Ting and Mayer, 1973). The resultant pressure distribution in the ring pack and inertia of the ring are considered to determine ring position. Blow-by and the dynamics of the piston rings interact simultaneously. The engine considered here has an oil scraper ring and three compression rings, and the combustion pressure and inter-ring gas pressure with respect to the crank angle are shown in Fig. 5.

The top ring has a circular-shaped face (Fig. 6 (a)) and the second ring has a complex surface shape (Fig. 6(b)). The operating condition and design parameters are shown in Table 1.

When there is no additional oil supplied to each piston ring, the amount of oil that flows into the interface is the summation of the amount of oil that the foregoing piston ring leaves on the cylinder liner and oil that cohered the cylinder liner due to surface tension. This can be schematized in Fig. 7.

Because the oil scraper ring is of a twin-landed type, sufficient oil is supplied through holes and thus the lubrication condition is considered to be fully-flooded. The analysis sequence for ring

Table 1 Engine specification

Rotational speed (rpm)		2000			
Piston bore (mm) × Stroke (mm)		82×93.5			
Fuel		Gasoline			
Connecting rod length (mm)		146.5			
RING WIDTH	Top ring (Circular shape)	W_1	0.5	δ_1	0.01
		W_2	0.7		
	Second ring (Taper-Flat-Taper)	W_1	0.2	δ_1	0.005
		W_2	0.1		
		W_3	0.9	δ_2	0.05
	Oil scraper ring (Circular shape)	W_1	0.2	δ_1	0.002
		W_2	0.2		

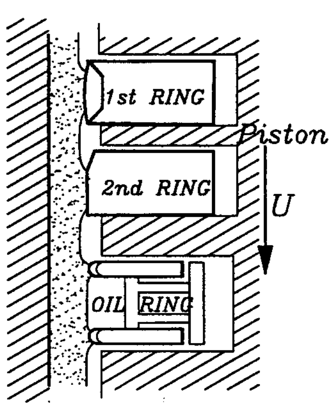


Fig. 7 Oil flows in the ring pack

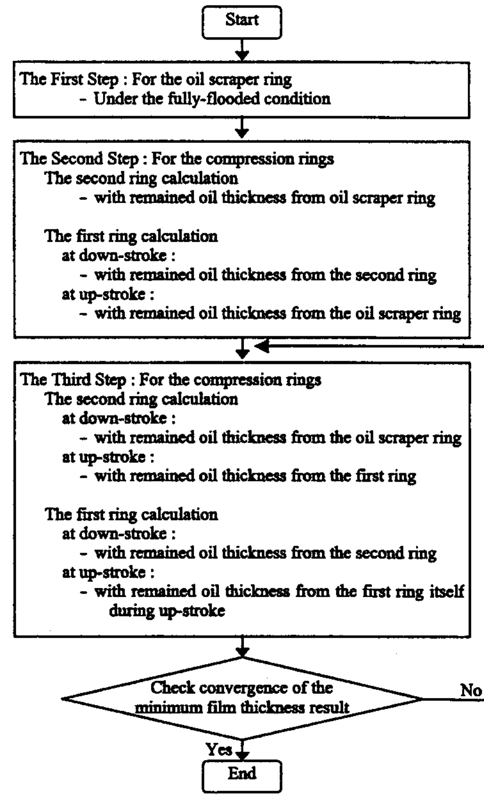


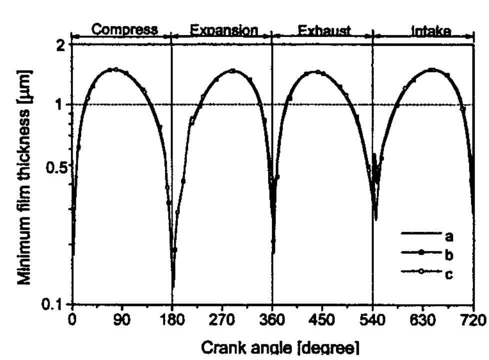
Fig. 8 Flow chart for ring pack analysis

pack lubrication is described in Fig. 8. When the second ring moves from the bottom to the top in the second step of analysis, it is assumed that the oil left by the oil scraper ring was used.

Analysis results of the top ring with respect to the calculation step are shown in Fig. 9. The effect of oil cohered on the cylinder liner is ignored and only the oil transported from the previous piston ring is considered.

The minimum film thickness and friction force acting on the piston ring is converged at reasonable tolerance as the calculation steps proceed, and final results on the minimum film thickness can be obtained by three iterative calculations.

The total frictional force acting on the piston ring is the summation of the forces due to the viscosity of oil and asperity contact (Greenwood and Tripp, 1970-71). This is also converged in



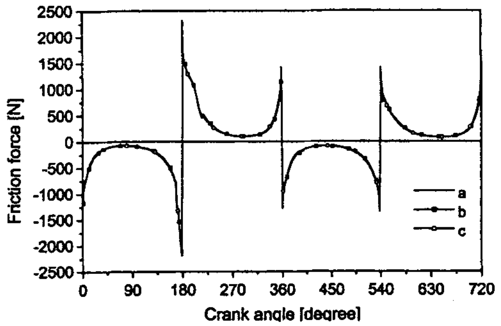
(a) First (b) Second (c) Third calculation

Fig. 9 Minimum film thickness between the top ring and the cylinder liner according to analysis sequence

sequential analyses as shown in Fig. 10.

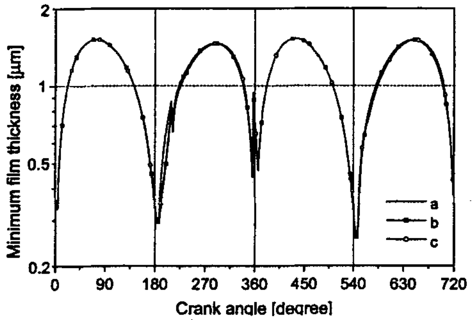
The results of the second ring show the same tendency as that of the top ring. These results are shown in Fig. 11.

Friction is increased near the dead centers mainly because of the increase in the asperity



(a) First (b) Second (c) Third calculation

Fig. 10 Friction force on the top ring according to analysis sequence



(a) First (b) Second (c) Third calculation

Fig. 11 Minimum film thickness between the second ring and the cylinder liner according to analysis sequence

contact force, and the same tendency is seen in the results with a single piston ring.

4. Conclusions

(1) The minimum film thickness and the frictional force of the piston rings in the ring pack are calculated while taking into consideration oil transport by the iterative method.

(2) The compression rings in the ring pack operate under oil starvation in most strokes because the oil scraper ring supplies only a little oil for the compression rings to avoid overconsumption of oil. This makes the mixed lubrication region broader in the reciprocating stroke.

(3) The effective width of the piston ring is reduced to about 20 to 30 percent in most strokes because of oil starvation and the pressure drop phenomenon at the posterior portion. However, it extends over the whole surface as the piston ring approaches the dead center because the squeeze effect is dominant in this region.

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