Gas Flows Through the Inter-Ring Crevice and Their Influence on UHC Emissions

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ABSTRACT

Influence of the inter-ring crevice, the volume between the top and second piston rings, on unburned hydrocarbon (UHC) emission was experimentally and numerically investigated. The ultimate goal of this study was to estimate the level of UHC emission induced by the blow-up of inter-ring mixture, i.e., unburned gases trapped in the inter-ring crevice.

In the experiments, the inter-ring mixture was extracted to the crankcase during the late period of expansion and the early period of exhaust stroke through the engraved grooves on the lower part of cylinder wall. Extraction of the mixture resulted in the significant reductions of UHC emission in proportion to the increments of blowby flow rate, without any losses in efficiency and power. This experimental study has confirmed the importance of inter-ring crevice on UHC emission in an SI engine and established a relationship between the inter-ring mixture and UHC emission.

A physical model was constructed to predict the gas flows through the piston ring pack and used to interpret the phenomena related to the results of experiments. Amount of the inter-ring mixture returning to the combustion chamber after the exhaust valve open was calculated and converted to the corresponding UHC emission using the relationship between the inter-ring mixture and UHC emission in the experiments. Calculated level of UHC emission caused by the inter-ring crevice was 10% ~ 30% of the entire UHC emission over a range of speeds (1250 rpm ~ 3500 rpm) and loads (185 kPa ~ 556 kPa bmep), which showed maximum at 2500 rpm, 432 kPa bmep. This is a condition most frequently operated by users.

INTRODUCTION

In SI engines, the crevice volumes in the combustion chamber have been shown to be responsible for a significant fraction of unburned hydrocarbon (UHC) emission [1 ~ 5]. There exist several crevice volumes in production engines. As the cylinder pressure rises during compression and combustion, the unburned mixture is forced into the crevice regions. However, once the crevice gas pressure is higher than the cylinder pressure, the trapped gas flows back from the crevices into the combustion chamber. Some of this unburned mixture may exhaust to the atmosphere without in-cylinder oxidation, which could be a major contributor to engine-out UHC emission in SI engines.

The largest and the most important parts of these crevices are volumes between the piston, piston rings and cylinder wall, that is, the piston ring crevices. The piston ring crevices consist of a series of volumes, so that it can be divided into two parts as shown in Fig. 1; the top-land crevice and inter-ring crevice. The former is the crevice above the top ring and the latter is the crevice enclosed by two compression rings. The volume between the second and oil ring rarely contribute to UHC emission because the oil ring does not seal very well. During the compression and expansion stroke, the unburned mixture flows from the combustion chamber, past the piston rings, and into the crankcase as blowby. Some of this mixture is trapped in these two crevice volumes and returns to the combustion chamber in the expansion and exhaust stroke. The mixture trapped in the top-land crevice returns to the combustion chamber after the cylinder pressure starts to decrease (about 15 ~ 20° crank angle after top dead center (TDC)). On the other hand, the mix-
ture trapped in the inter-ring crevice (inter-ring mixture) returns to the combustion chamber (namely, blow-up) late in the expansion stroke through the gap between the ring and cylinder liner, ring gap, and ring side clearance when the inter-ring pressure becomes higher than the cylinder pressure [3]. At this moment, the volume of the inter-ring crevice is usually maximized and is comparable to that of the top-land crevice because the top ring shifts from the lower to the upper groove surface and the second ring is located on the lower groove surface [4].

There is substantial experimental evidence to support the fact that the top-land crevice volume is a major source of UHC emission [1 ~ 5]. However, contribution of the inter-ring crevice volume to UHC emission is not well identified yet, though the influence of the inter-ring crevice was recognized by Namazian [4] and Wentworth [5]. Recently, the importance of inter-ring crevice to UHC emission has been emphasized in the experiments using planar laser induced fluorescence (PLIF) by Green and Cloutman [6]. They visualized the wall jet of UHC near bottom dead center (BDC) driven by the inter-ring pressure that is higher than the cylinder pressure.

It is worth noting that the temperature of bulk gas in the combustion chamber is too low to oxidize the unburned mixture late in the expansion stroke when the inter-ring mixture begins to return to the combustion chamber. Furthermore, the density of inter-ring mixture is higher than that of top-land mixture, and most of inter-ring mixture consists of unburned mixture. Therefore, it is obvious that the blow-up of inter-ring mixture will exert an important influence upon UHC emission.

In this study, the inter-ring crevice was considered as an important source of UHC emission in SI engines, and its influence was experimentally and numerically investigated. The ultimate goal was to estimate the level of UHC emission induced by the blow-up of inter-ring mixture.

The experiments were performed to reduce the UHC emission by extracting the inter-ring mixture to the crankcase during the late period of expansion and the early period of exhaust stroke through the intentionally engraved grooves on the lower part of cylinder wall. Measurements of UHC emission and blowby flow rate were carried out over a range of speeds (1250 rpm ~ 3500 rpm) and loads (185 kPa ~ 556 kPa bmem). The results of measurements were compared with those of a base engine without grooves, to investigate the influence of extraction of inter-ring mixture on the reduction of UHC emission. The extraction of inter-ring mixture results in the significant reductions of UHC emission in proportion to the increments of blowby flow rate, so that a substantial relationship between the inter-ring mixture and UHC emission could be established from the experimental results.

Because the influence of inter-ring crevice on UHC emission is caused by gas flows into and out of this region, a physical gas flow model coupled with ring motions was also constructed in much the same manner as proposed by Namazian and Heywood [4]. Based on the experimental relationship between the inter-ring mixture and UHC emission, the amount of the inter-ring mixture returning to the combustion chamber was converted to the corresponding UHC emission in the gas flow model. The calculated results will exhibit the portion of this corresponding UHC emission and emphasize the importance of inter-ring crevice volume as a source of UHC emission.

**EXPERIMENTAL SYSTEMS**

**EXPERIMENTAL SETUP** — A 2.0 liter gasoline fueled SI engine was adopted as the test engine. Its specifications are listed in Table 1, while the dimensions of the piston ring pack are listed in Table 2. The volumes of top-land crevice and inter-ring crevice are 744 mm$^3$ and 815 mm$^3$, respectively, which correspond to 1.13 % and 1.24 % of the clearance volume (65,700 mm$^3$) when the top ring is located on the upper groove surface and the second ring is located on the lower groove surface.

<table>
<thead>
<tr>
<th>Table 1. Engine specifications</th>
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<tr>
<td><strong>Type</strong></td>
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<td><strong>Bore × Stroke</strong></td>
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<tr>
<td><strong>Total Disp. Volume</strong></td>
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<tr>
<td><strong>Compression Ratio</strong></td>
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<td><strong>Fuel</strong></td>
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<tr>
<td><strong>Exhaust Valve</strong></td>
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<tr>
<td><strong>Open</strong></td>
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<td><strong>Close</strong></td>
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<tr>
<th>Table 2. Dimensions of piston ring pack (cold condition)</th>
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<tr>
<td><strong>Top land width</strong></td>
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<tr>
<td><strong>Top ring width</strong></td>
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<tr>
<td><strong>Top ring end gap</strong></td>
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<td><strong>2nd land width</strong></td>
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<td><strong>2nd ring width</strong></td>
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<tr>
<td><strong>2nd ring end gap</strong></td>
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<tr>
<td><strong>Ring side clearance</strong></td>
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An overall schematic diagram of the experimental system is shown in Fig. 2. A nondispersive infrared (NDIR) gas analyzer (MEXA-550JK, HORIBA) was used for the measurements of exhaust hydrocarbon emission and excess air ratio ($\lambda$). It has been shown that the values of HC concentrations in engine exhaust gases measured by an NDIR analyzer are about half of the corresponding values measured by a flame ionization detector (FID) on the same carbon number basis [3]. The sampling point of exhaust gas was located at 2.5 m from the exhaust port when the three-way catalyst was removed. Measurement
of the excess air ratio was also obtained with a wide range $O_2$ sensor (TL-7111, NTK). The blowby was measured by an orifice flow meter, which was installed between the intake system and blowby nozzle that is located at the head cover. This orifice flow meter was calibrated with the blowby gases by connecting a bubble meter in series before the measurement and revealed about 1.5% accuracy of measurements. Cylinder pressure and fuel consumption were also measured with a piezoelectric pressure transducer (6051A, KISTLER) and an electronic balance (FA-2000, AND), respectively, for the engine performance diagnoses.

Figure 2. Overall schematic diagram of experimental system

GROOVED LINER – The key intent of experiments is to prevent the inter-ring mixture from returning to the combustion chamber, by extracting the inter-ring mixture to the crankcase without any losses in performance and efficiency. Sealing with the second ring becomes less important after the exhaust valves open at around 50° BBDC because it does not contribute significantly to power generation any more. Therefore, if the sealing function of the second ring can be artificially deleted at the end of expansion stroke, it is possible to extract the inter-ring mixture without losses in efficiency and power. For this purpose, grooves were engraved on the lower parts of each cylinder liner surface, and the inter-ring mixture was extracted to the crankcase as blowby through the grooves. The extracted inter-ring mixture does not induce UHC emission any more because the crankcase is vented to the engine intake system so that the blowby gases are recycled.

Figure 3 shows the grooved liner of the test engine. There are four grooves 90° apart on the liner in each cylinder, and the width and depth of a groove are 1.5 mm and 1.0 mm, respectively. In this figure, the piston is at the BDC position, and the grooves lie right below the lower surface of the top ring so that the grooves are not exposed to the combustion chamber during the entire engine operating cycle. The exposure of grooves may increase the power and efficiency losses, and oil consumption due to the leakage through the grooves. The role of these grooves is to extract the inter-ring mixture to the crankcase during the late period of expansion and the early period of exhaust stroke. Near BDC, the linear displacement of the piston is very small compared to the angular displacement of the crankshaft, which allows enough residence time for gas extraction. Therefore, the grooves can extract the inter-ring mixture during about 56° crank angles though the top of grooves is only 3.5 mm (4% of stroke) higher than the upper surface of the second ring at BDC.

Figure 3. Configuration of grooved liner for the extraction of inter-ring mixture
When the inter-ring mixture is extracted to the crankcase, two possible effects related to the reduction of UHC emissions are expected. The first is to prevent the inter-ring mixture from returning to the combustion chamber. The second is to pull down the unburned mixture, that leaked out from the piston ring crevice and gathered around the upper corner of the piston, into the piston crevice region again by decreasing the inter-ring pressure under the cylinder pressure.

**EXPERIMENTAL CONDITIONS** – Experiments were carried out under various speeds (1250 rpm ~ 3500 rpm) and loads (185 kPa ~ 556 kPa bmep) conditions with three different types of the cylinder liner: unmodified (BASE), 0.2 mm-deep grooved (MOD1), and 1.0 mm-deep grooved (MOD2) liners. MOD1 was accomplished by filling the 1.0 mm-deep grooves partially with epoxy. Test conditions were selected within the operating ranges of the standard Federal Test Procedure (FTP) driving schedule and the European legislative emissions combined cycles for emission regulation, which cover mainly low and middle engine speed and load conditions. Therefore, the exhaust characteristics of these operating conditions are important to various emission regulations. The excess air ratio of the test engine was set to an almost stoichiometric mixture condition ($\lambda=1.0\pm0.02$). The spark timing was set to the MBT (minimum advance for best torque) timing, and was not influenced by the existence of grooves. Each test condition was repeated 10 ~ 15 times to obtain averaged values of the measurements. Coolant temperature was maintained at 80±2°C throughout the test.

**GAS FLOW MODEL**

**CONFIGURATION OF MODEL** – Figure 4 is the conceptual diagram of the gas flow model used in this study. Considered are the orifice flows through the ring gaps ($m_{13}$, $m_{35}$), the ring motion in the grooves, and the channel gas flows through the ring sides ($m_{12}$, $m_{23}$, $m_{34}$, $m_{45}$) resulting from the ring motion. So are the forces such as pressure force ($F_p$), friction force ($F_f$), ring inertia force ($F_i$), and oil squeeze force ($F_s$). Fundamental assumptions as well as the relationship between parameters followed the model proposed by Namazian and Heywood [4].

The continuity equations for the regions 2, 3, and 4, and ring motion equations for the two compression rings are as follow;

- **Continuity equations**
  \[
  \dot{p}_2 = \frac{RT}{V_2}(\dot{m}_{12} - \dot{m}_{23}) \\
  \dot{p}_3 = \frac{RT}{V_3}(\dot{m}_{13} + \dot{m}_{23} - \dot{m}_{34} - \dot{m}_{35}) \\
  \dot{p}_4 = \frac{RT}{V_4}(\dot{m}_{34} - \dot{m}_{45})
  \]

where
- $\dot{p}_i$: time derivative of pressure in region i
- $V_i$: volume in region i
- $m_i$: mass in region i
- $R$: gas constant
- $T$: gas temperature

- **Ring motion equation**
  \[
  M_{r,n} \frac{d^2h_n}{dt^2} = (F_{p,n} + F_{f,n} + F_{i,n} + F_{s,n})
  \]

where
- $M_{r,n}$: mass of n-th ring
- $F_{p,n}$: pressure force of n-th ring
- $F_{f,n}$: friction force of n-th ring
- $F_{i,n}$: inertia force of n-th ring
- $F_{s,n}$: oil squeeze force of n-th ring
- $h_n$: ring side clearance of n-th ring

However, in this study, the orifice flows through the ring gap ($m_{13}$ and $m_{35}$) were defined to be caused not through the area of ring gap, $A_{gap}$, but through the effective area, $A_{eff}$, defined as below;

\[
A_{eff} = K \cdot A_{gap}
\]

where
- $K$: effective area factor.

---

![Fig. 4 Conceptual diagram of the gas flow model in the piston ring pack](image-url)
Physical meaning of the effective area factor $K$ is that, if the gas flow through the ring side by ring motion is excluded, the gas flow is made not only in the ring gap but also in other areas. There may exist many other gas flow areas as well as ring gap area \[8\]. The noticeable one originates from the deformation of the cylinder liner \[8 \sim 11\]. This includes the thermal deformation by the non-uniform temperature distribution, and the mechanical deformation by the cylinder pressure, stress induced when mounting the engine head on the block, and so on. When the cylinder liner is deformed, there happens to be a gap between the ring and cylinder liner, and the flow through the ring front surface is induced. In general, if only the ring gap was to be considered as the orifice gas flow path, the blowby flow rate calculated from the gas flow model appears smaller than the measured one. Furuhama \[7\] and Munro \[8\] introduced the effective area factor, $K$ to compensate the discrepancy between the model and reality, and suggested the value of 2 and the value between 1.5 ~ 6, respectively.

**VERIFICATION OF MODEL** – First of all, the effective area factor, $K$ for the modeled engine must be introduced to define the passage area of the orifice flow. Simulations were carried out with several assumed values of $K$ at the condition of 432 kPa bmep, and 2500 rpm, that corresponds to the condition in the middle of experiments. The value of $K=1.85$ was selected at this condition as the proper effective area factor for the modeled engine. With this value of $K$, the blowby flow rate was calculated at all operating conditions and compared with the experimental results for the verification of the gas flow model.

Figure 5 shows the good coincidence between calculated blowby flow rate and measured blowby flow rate regardless of the operating conditions of engine speeds and loads. In contrast, without effective area factor ($K=1.0$), there is a considerable discrepancy between calculated results and measured results. The slope of calculated results is somewhat steeper than that of measured results in Fig. 5(b). The value of $K$ must be lowered with the increase of load to slack away the slope of calculated results. It can be inferred from these comparisons that the cylinder pressure influences the value of effective area factor. The evidence of the influence is that the gap between the ring and cylinder liner created by the deformation of the cylinder liner decreases with pressure. That is, the gap area between the ring and cylinder liner diminishes with the increase of load because the gas pressure is high enough to press the piston ring’s contact surfaces into the distortion recesses \[11\].

Hereafter, all the results from the simulation are based on the value of 1.85 as the effective area factor in the orifice flow.

**EXPERIMENTAL RESULTS AND DISCUSSIONS**

**INFLUENCE OF GROOVE** – Figure 6 shows the results of the tests with BASE and MOD1 (with 0.2 mm-deep grooves). The error bars represent 95% confidence intervals for individual conditions. The UHC emission measured by an NDIR analyzer is displayed on the C₆ basis. The values of UHC emission are somewhat lower than the typical values measured by the FID in other literatures because of lower sensitivity of the NDIR analyzer than that of the FID.

It can be found from Fig. 6 that the reduction of UHC emission by MOD1 covers the entire range of test conditions. Particularly at 1500 rpm and 2500 rpm of engine speed, MOD1 resulted in the reductions of UHC emission by 15% ~ 21% compared to the BASE case. The reductions of UHC emission are related to the increments of blowby. The lower engine speeds led the increase in both
the increment of blowby and the reduction of UHC emission. This trend is shown clearly when the results of blowby and UHC emission are displayed with respect to the engine speeds under the constant load conditions (Fig. 7). At 1250 rpm, the lowest engine speed condition, the longest available time to extract the inter-ring mixture into the crankcase led the largest increment of blowby, and eventually the largest reduction of UHC emission. As engine speed increased, on the contrary, both the increment of blowby and the reduction of UHC emission decreased. These decreases are also possibly due to the fact that the 0.2 mm-deep grooves are too shallow to extract the inter-ring mixture under higher engine speed conditions. The same explanation could be applied to the case of relatively high load conditions.

Deeper grooves can extract an additional amount of the inter-ring mixture. Figure 8 shows the results of the tests with the deeper grooves of MOD2 (1.0 mm-deep grooved) which has 0.8 mm deeper grooves than MOD1. However, deterioration of UHC emission was found at low loads and low speeds, even though MOD2 could extract more inter-ring mixture than MOD1. At 185 kPa bmep and below 2500 rpm, the UHC emission of MOD2 was higher than that of BASE. This increment of UHC emission may be caused by the suction of oil, which is originally filled in the grooves, into the combustion chamber during the late period of intake and the early period of compression stroke. It is certain that, as the engine speed and load decrease, more oil spurts into the combustion chamber because of the longer residence time and lower intake pressure. In the case of MOD2, therefore, the increments of blowby were no longer related to the reductions of UHC emission at low load (185 kPa bmep) and/or low engine speed (1250 rpm) conditions as shown in Fig. 9. This trend is contradictory to the case of MOD1 shown in Fig. 7. The characteristics of UHC emission became worse as engine speed and load were lowered. In the idling condition represented by the lowest load and speed condition, spark plugs were found stained with unburned oil, which was not seen in the case of BASE and MOD1.

However, it is noticeable that the blowby of MOD2 was higher than that of MOD1 under the entire test conditions, and the sizeable reduction of UHC emission was accomplished with MOD2 at higher load condition; by 23 % under 556 kPa bmep, for example. As engine load increases, the amount of spurted oil will decrease because the difference between the intake and crankcase pressure decreases. Therefore, the increased blowby of MOD2 could result in additional reductions of UHC emission under higher load conditions.

Fuel consumption under entire test conditions and power under wide open throttle conditions were also measured though not displayed here. These were almost unchanged regardless of the existence of the grooves on the cylinder liners.
Green and Cloutman [6] visualized unburned mixture ejected from the inter-ring crevice late in the expansion stroke using PLIF. They concluded that this unburned mixture can survive oxidation and it is mixed with combustion products in the clearance volume. These statements imply that the large portion of the unburned mixture ejected from the inter-ring crevice exhaust from the combustion chamber without in-cylinder oxidation and directly affect the engine-out UHC emission. The results in this study could be explained by these findings in the work of Green and Cloutman. The increased blowby of the grooved liner is essentially the inter-ring mixture that, if the grooves do not exist, must return to the combustion chamber during the late period of expansion and the entire period of exhaust stroke. In this period, the temperature of the combustion chamber is too low to oxidize the unburned mixture. Therefore, the reduction of UHC emission could be accomplished by extracting the inter-ring mixture to the crankcase.

The optimization process on the configurations of grooves, such as depth and number of grooves, were not carried out because the purpose of this study was to test, first of all, whether the influence of inter-ring crevice on UHC emission is indeed considerable. However, it would be suggested to find the optimum configuration of grooves for the further decrease of UHC emission and for the further understanding of the influence of inter-ring crevice.

RELATIONSHIP BETWEEN INTER-RING MIXTURE AND UHC EMISSION – The reduction of UHC emission by MOD1 was strongly related to the increment of blowby as shown in Fig. 7. As engine speed decreased, both the increment of blowby and the reduction of UHC emission increased. The other tested conditions showed the same trends. It is necessary to establish the relationship between the increments of blowby and the reduction of UHC emission for the further understanding of the influence of inter-ring mixture on UHC emission with respect to the variation of operating conditions.

The relationship between these two parameters was assessed and given in Fig. 10. The increment of blowby flow rate was expressed as the percentage of the intake air-fuel flow rate ($\frac{m_{\text{in-blowby}}}{m_{\text{intake}}} \times 100$%). The vertical axis in Fig. 10(b) means the reduction of UHC emission per unit percent of the increased blowby flow rate to the intake air-fuel flow rate. The increased blowby of the grooved liner is essentially the inter-ring mixture which, if there were no grooves, must return to the combustion chamber, as mentioned previously. Therefore, the relationship between the increment of blowby and the reduction of UHC emission with the grooved liner in Fig. 10(b) is coincident with the relationship between amount of inter-ring mixture and its corresponding UHC emission which actually influences the exhausted UHC emission in the base engine.
Interpreting the results in Fig. 10(b) from a viewpoint of inter-ring mixture as a function of engine speed, it can be said that the blow-up of inter-ring mixture at high engine speeds has a relatively small influence on UHC emission. In other words, considerable amounts of the blow-up of inter-ring mixture would not be exhausted to the atmosphere at high engine speeds. It is supposed to be due to the less available time to exhaust the inter-ring mixture through the exhaust valve under the higher engine speed conditions.

Figure 11 shows the relationship between the inter-ring mixture and UHC emission at constant speed condition (2500 rpm). UHC emission is more affected when the inter-ring mixture returns to the combustion chamber under the load condition at around 432 kPa bmep than the lower or higher load conditions (Fig. 11(b)). A reason for the relatively small influence of inter-ring mixture at lower load is supposedly that the residual gas fraction increases as load decreases [12]. The higher residual gas fraction increases the portion of the burned gas composition in the inter-ring mixture, so that the influence of inter-ring mixture is relatively lowered. At the higher load (556 kPa bmep), the possibility of oxidization of inter-ring mixture in the combustion chamber is higher than that at lower load because of the higher exhaust gas temperature.

These experimental and substantial relationships between the inter-ring mixture and UHC emission, Fig. 10(b) and Fig. 11(b) were used when the influence of inter-ring mixture on UHC emission was estimated in the gas flow model.

Figure 11. Relationship between the inter-ring mixture and UHC emission with respect to engine loads

**NUMERICAL RESULTS AND DISCUSSIONS**

**INFLUENCE OF GROOVE** – The grooves of 0.2 mm depth on the cylinder liner (MOD1) same as the experiments were considered in the gas flow model. The increment of blowby flow rate as the simulated result was described in Fig. 12. Good accordance of simulated results with measured ones provides the indications of validation of the gas flow model. Extraction of the inter-ring mixture to the crankcase brings about the inter-ring pressure to fall down as shown in Fig. 13, where the traces of inter-ring pressure for the base engine without grooves were also shown as dotted line for the comparison. The inter-ring pressure starts to decrease at $28^\circ$
BBDC and does not lower down to the cylinder pressure at most operating conditions. However, as it falls down under the cylinder pressure due to the influence of grooves at below 1500 rpm, the unburned mixture in top-land crevice flows into the crankcase via the inter-ring crevice. It indicates that a part (about 25% at 1250 rpm and 5% at 1500 rpm) of the calculated increment of blowby flow rate was, strictly saying, originated not from the inter-ring crevice but from the top-land crevice. Although it may be possible to reduce UHC emissions by extracting unburned mixture in top-land crevice to the crankcase, it is not the influence of inter-ring crevice. Therefore it might well be that the influence of top-land crevice was included in the influence of inter-ring crevice under low speed conditions in Fig. 10. It can also be found from Fig. 13 that the grooves can extract the inter-ring mixture sufficiently under the low speed conditions but not under the high speed conditions.

The inter-ring pressure is higher than the cylinder pressure until the latter half of exhaust stroke under the high speed (over 2500 rpm) conditions. The higher the engine speed, the lower the possibility for the inter-ring mixture, which returned to the combustion chamber during the latter half of exhaust stroke, to exhaust out to the atmosphere. Therefore, the extraction of such inter-ring mixture can hardly decrease UHC emission although it can raise blowby flow rate at high engine speeds. This phenomenon can be regarded as one of the reasons for the reduction of the values (UHC_{reduction}/(\dot{m}_{in-blown}/\dot{m}_{intake} \times 100)) under high speed conditions in Fig. 10(b). In the same manner, the simulation results can explain one of the reasons for the reduction of the value (UHC_{reduction}/(\dot{m}_{in-blown}/\dot{m}_{intake} \times 100)) at high load in Fig. 11(b). The period when the inter-ring pressure is higher than the cylinder pressure extend over the whole exhaust stroke at 556 kPa bmep and 2500 rpm, though it is not presented in this paper. Therefore, the large portion of the blow-up of inter-ring mixture may not exhaust to the atmosphere under the high load conditions.

Figure 12. Simulated results of increased blowby flow rate induced by the grooves

Figure 13. Traces of inter-ring pressure and cylinder pressure at 432 kPa bmep (simulated)
The amount of the blow-up of inter-ring mixture was calculated from the gas flow model and converted to the corresponding UHC emission using the experimental relationship in Fig. 10(b) and Fig. 11(b). In this calculation, the inter-ring mixture, that returns to the combustion chamber before the exhaust valve open (47° BBDC), was excluded because of the possibility of oxidization in the cylinder. The calculated results expressed by percentage were displayed in Fig. 14.

It was found from the simulated results that the influence of inter-ring crevice on UHC emission reaches to 10 % ~ 30 % over a range of speeds (1250 rpm ~ 3500 rpm) and loads (185 kPa ~ 556 kPa bmeP), biggest at the condition of 432 kPa bmeP and 2500 rpm. The reasons for the relatively small influence of inter-ring crevice at lower/higher load, and higher speed than 432 kPa bmeP and 2500 rpm were already mentioned. At low engine speeds, the blow-up of inter-ring mixture is relatively small because the pressure difference is smaller, and its effective duration is shorter than those under the high speed conditions as shown in Fig. 13.

It is worth noting the fact that the influence of inter-ring crevice on UHC emission is highest at 432 kPa bmeP and 2500 rpm. This operating condition is very important to UHC emission because the users operate engines most frequently around this condition. Therefore, the influence of inter-ring crevice is very important in a viewpoint of the emission regulations.

**CONCLUSIONS**

The influence of inter-ring crevice volume on UHC emissions was experimentally and numerically investigated. Several conclusions were obtained as follows:

From the experimental results with the grooved liner

1. Extraction of inter-ring mixture resulted in the significant reductions of UHC emission without any losses in efficiency and power.

2. The reduction of UHC emission closely related to the increment of blow-by flow rate, which implied the direct influence of the blow-up of inter-ring mixture on UHC emissions.

3. It could be known from the relationship between the inter-ring mixture and UHC emission that the considerable amount of the inter-ring mixture returning to the combustion chamber does not affect UHC emission at high engine speeds.

4. The influence of inter-ring mixture was highest at 432 kPa bmeP condition and was relatively small at lower or higher load conditions.

From the simulated results with the gas flow model

5. The increased blow-by flow rate induced by the grooves originated not only from the inter-ring crevice but also from the top-land crevice under the low engine speed conditions.

6. The period when the inter-ring pressure was higher than the cylinder pressure lasted even after the middle of exhaust stroke under the high engine speed and load conditions. It indicates some part of the blow-up of inter-ring mixture may not exhaust to the atmosphere under these conditions.

7. The influence of inter-ring crevice on UHC emission that estimated from gas flow model reached to 10 ~ 30 % over a range of speeds (1250 rpm ~ 3500 rpm) and loads (185 kPa ~ 556 kPa bmeP). The most influential operating condition was 432 kPa bmeP and 2500 rpm around which the engines are most frequently operated by users.
REFERENCES


