

Ring Pack Crevice Effects on the Hydrocarbon Emissions from an Air-Cooled Utility Engine

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ABSTRACT

The effect of the ring pack storage mechanism on the hydrocarbon (HC) emissions from an air-cooled utility engine has been studied using a simplified ring pack model. Tests were performed for a range of engine load, two engine speeds, varied air-fuel ratio and with a fixed ignition timing using a homogeneous, pre-vaporized fuel mixture system. The integrated mass of HC leaving the crevices from the end of combustion (the crank angle that the cumulative burn fraction reached 90%) to exhaust valve closing was taken to represent the potential contribution of the ring pack to the overall HC emissions; post-oxidation in the cylinder will consume some of this mass. Time-resolved exhaust HC concentration measurements were also performed, and the instantaneous exhaust HC mass flow rate was determined using the measured exhaust and cylinder pressure. At high load the model predicts that the ring pack returns approximately three times as much HC mass to the cylinder as is measured in the exhaust, indicating that the HC emissions are dominated by the ring pack contribution. At the lightest load condition tested, the ring pack model predicts significantly less mass returning to the cylinder from the ring pack than is observed in the exhaust, clearly indicating that another HC mechanism is significantly contributing to the exhaust HC emissions. The integrated exhaust HC mass from the time-resolved HC measurement was found to correlate inversely with the IMEP on a cycle-by-cycle basis for light load condition, which strongly suggest that incomplete combustion is materially contributing to the exhaust HC emissions. The intermediate load condition represents a combination of these two extremes. The ensemble-average ring pack model results indicate that the mass returned to the cylinder from the ring pack is slightly higher than the amount measured in the exhaust. But, a conditional sampling analysis indicates that there are sub-groups, i.e. late-burning cycles, for which this is not true. There is expected to be some in-cylinder post-oxidation of the ring pack HC mass at this condition, and the late burning cycles were not found to excessively contribute to the HC emissions, which both strongly suggests that there are other mechanisms besides the ring pack that are

significantly contributing to the HC emissions at this condition. The most likely mechanism is incomplete combustion.

INTRODUCTION

Small engines are identified as a major source of air pollution by the California Air Resources Board (CARB) and the Environmental Protection Agency (EPA). Every year approximately 100 million small engines are produced worldwide [1], and about 35 million are sold in the US. For comparison, approximately 17 million of cars are sold annually in the US [2]. Small engines usually run for shorter periods of time than automotive engines, however their emissions levels are significantly higher, accounting as much as 13% of the total hydrocarbon emissions in some regions of the US [3]. There is considerable pressure on the manufacturers to reduce emissions levels in order to comply with new emission standards. Since the introduction of the emissions regulations in 1995, small engines have been subjected to increasingly more stringent regulations that focus on the reduction of hydrocarbons (HC) and NO_x levels. The allowable CO emission levels have remained almost constant; for structural reasons small engines need to operate under rich conditions. Small engines are air cooled, and therefore have a high operating temperature. Rich operation keeps the engine temperature at safe levels, which makes the reduction of the CO levels difficult. Additionally, at rich air fuel ratios part of the fuel is not oxidized completely mainly due to the lack of oxygen, generating higher HC emissions.

In small engines HC emissions are a concern at low to medium loads because the HC component dominates the regulated HC+NO_x. This is, in part, due to the slow burn rate, which in some cases leads to incomplete combustion [4]. Therefore it is very important understand the HC sources under these conditions in small engines in order to provide insights for possible reduction of the HC emissions.

Hydrocarbon emissions arise from the storage of the fuel in regions of the chamber that the flame cannot reach. There are several mechanisms of fuel storage to avoid

the complete oxidation of the fuel during the flame propagation. The mechanisms of HC storage described in the literature are: mixture storage in crevice regions, adsorption of fuel by the oil layers and deposits, liquid fuel films, and flame quenching [5-13]. It is believed that at fully warmed up condition the combustion-chamber crevices are the largest contributor to the HC emissions [5-9]. The crevices in the combustion chamber are: the piston ring pack crevices, the head gasket crevice, the spark plug crevice, and the valve seat crevices. The ring pack crevices account for the majority of the engine-out HC emissions [7-10]. However, other mechanisms such as the liquid films and the oil-fuel adsorption-desorption are important as well and their contribution can be significant, especially under cold start and transient conditions [5,6].

The transport process of the mixture stored in the ring pack crevices back into the cylinder is very complex and plays an important role in the extent to which the mixture will contribute to the engine-out HC emissions. Experimental observations [12, 13] show that as the piston moves downward during the expansion stroke a thin layer of hydrocarbons is formed along the wall due to the outflow of the HC stored in the ring pack. This begins immediately following the time that the cylinder has reached peak pressure. This outgassing process continues until the exhaust valve is closed. It has been shown that the velocity of the outflow gases into the cylinder is slower than the piston speed [8]. Namazian and Heywood [12] used Schlieren photography to study the ring pack crevice outflow in an optical engine during the early stages of the outflow process. They observed two types of flow structures: the outflow of the mass stored in the top land crevice, which extended along the circumference of the piston and initiates just after the peak cylinder pressure; and a jet-type flow that was observed at the end of the expansion process, which happened when the inter-ring pressure was greater than the cylinder pressure. This was verified by a phenomenological flow model of the ring pack.

Green and Cloutman [13], using planar laser-induced fluorescence, studied the crevice gas outflow during the post-combustion period. Two locations were observed, one close to the ring end gap and the other opposite to the top ring end gap. The results revealed different phenomenon occurring at the two locations. At the location opposite the ring gap no crevice outflow was observed up to bottom dead center (BDC), where a wall-jet-type structure was observed. The authors believed that this was due to the release of the mass trapped in the inter-ring volume when the top ring moved from one side of the groove to the other. At the ring gap location a flow structure having features of a plume was observed at 90 CA bBDC. This plume produced a thin layer of HC on the cylinder wall which later, during the exhaust stroke, was scraped by the piston forming a characteristic roll-up vortex structure.

The transport of the HC mass leaving the crevice to the bulk mass of the cylinder is an important process because it influences the post-flame oxidation process. Two main transport mechanisms of the crevice outflow were described schematically by Lavoie *et al* [14]. The first one consists in an entrainment into the bulk gases as they leave the crevices during the blowdown process mainly due to the diffusion process. The second mechanism consists in a roll-up vortex produced by the piston when it scrapes the HC layer of the cylinder wall. Not all of the combustible mixture that escapes the main combustion contributes to the engine-out HC emissions. Depending on the engine operating condition a significant quantity of this mixture can be partially oxidized during post-flame processes, reducing its contribution to HC emissions. The extent of post-flame oxidation depends on the time available, the mixing rate of the unburned mixture into the hot burned gases, the availability of oxygen, and the temperature. It is believed that most of the HC that leave the ring pack are oxidized [15-16]. The extent of the oxidation depends of the type of fuel and the chamber conditions [17]. Several studies have shown that in order to have post oxidation the temperature in the reaction zone must be above a minimum value [15-16, 18-20] between 1250 [18] and 1500 K [16]. From this review, it can be seen that the in-cylinder post oxidation is not in question, however its estimation must be done very carefully since there is significant non-uniformity of the composition and temperature inside the combustion chamber.

As we can see from the review above, there is a good understanding of HC emission mechanisms. However, small engines operate under different conditions than the automotive ones. Small engines still use carburetors as a fuel metering device, use an integral fan as a cooling device, and operate at rich equivalence ratios. Air-cooled engines operate at high temperatures, and can have significant uneven bore distortion. One of the first attempts to understand the ring pack contribution to the HC emissions of a small engine under steady state, fully warmed up conditions was carried out by Salazar and Ghandhi [4]. The tests were carried out by measuring steady-state emissions from a common exhaust tank and cycle-resolved HC emissions from only one cylinder of the two-cylinder air-cooled engine. It was shown that at high loads the ring pack is the major contributor to the HC emissions; however at low loads the ring pack is not the major contributor. Further analysis of the cycle-resolved HC emissions and single-cycle heat release analysis showed that at low loads incomplete combustion can contribute significantly to the HC emissions.

In the present work a detailed study of the estimation of the ring pack contribution to the total HC emission of an air-cooled small engine is presented. This work is an extension of reference [4]; the exhaust system has been separated to isolate individual cylinder effects and conditional sampling has been used to elucidate further the incomplete combustion contribution to the HC emissions.

EXPERIMENTAL SETUP AND TEST CONDITIONS

The engine for this study was a two cylinder (V-twin) air-cooled engine, that originally comes equipped with a carburetor and has fixed spark timing. The specifications of the engine are given in Table 1, where bTDC is before top dead center, aTDC is after top dead center, IVO is intake valve opening, IVC is intake valve closing, EVO is exhaust valve opening, and EVC is exhaust valve closing. A pre-vaporized homogeneous mixture system was used as the fuel system instead of the carburetor. It was previously shown that under steady, fully warmed up conditions the carburetor does not affect the HC emissions [21]. The engine was modified to be controlled by the MotoTron suite of products which allows the use of electronic fuel injection system, electronic spark timing control, closed-loop air-fuel ratio control, and electronic throttle control. The experimental set up was almost the same as the one used in [4, 21]. However, in the present work some important modifications to the exhaust system were made. A preliminary study [4] showed that the engine has a significant cylinder-to-cylinder variability. The use of a common exhaust tank to measure the average emissions was, therefore, masking the behavior of each cylinder. Thus, a split exhaust system was built, consisting of a separate exhaust tank for each cylinder. With this new arrangement the exhaust gases were sampled from the exhaust surge tank of each cylinder and their averaged emissions were analyzed using a Horiba 5-gas (CO, CO₂, NO_x, O₂ and HC) emissions bench. Cycle-resolved hydrocarbon concentrations were measured using a Cambustion HFR500 Fast FID analyzer, with the sample probe located 1 cm downstream the exhaust valve stem of each cylinder. This new configuration allowed the calculation individual cylinder air-fuel ratio and the estimation of the HC mass at exhaust by carrying out at mass balance at the exhaust port [4]. Pressure data for each cylinder were measured using water-cooled Kistler 7063A piezoelectric pressure transducers. Additionally, crank-angle resolved intake and exhaust manifold pressures were also recorded. The engine was tested over a wide range of load, air-fuel ratio, speed and spark timing (Table 2). The fuel used for this research was a US EPA TIER II EEE certification fuel provided by Haltermann Products.

A single-zone heat release analysis was performed to characterize the combustion event. This analysis was carried out for averaged and single-cycle pressure traces. The heat release analysis requires the knowledge of the mass trapped in each cylinder. This was estimated by considering that the charging of each cylinder was proportional to the IMEP in each cylinder. To avoid the influence of incomplete combustion and other factors, spark sweep tests at constant throttle position were carried out. The mass trapped in each cylinder was considered to be proportional to the maximum IMEP in each cylinder as follows:

$$\frac{m_{cyl-1}}{m_{cyl-2}} = \frac{IMEP_{max,cyl-1}}{IMEP_{mac,cyl-2}} \quad (1)$$

$$m_{cyl-1} + m_{cyl-2} = m_{total} \quad (2)$$

where m_{cyl-1} , m_{cyl-2} are the mass per cycle trapped in cylinder 1 and cylinder 2, respectively; m_{total} is the mass of fuel and air that is delivered to the engine per cycle (both cylinders), and is measured experimentally; and $IMEP_{max,cyl-1}$ and $IMEP_{mac,cyl-2}$ are the maximum IMEP obtained from the spark sweep tests.

Table 1 Engine specifications

Type	Two cylinder 90° V
Bore	80 mm
Stroke	67 mm
Displacement	674 cc
Compression ratio	8.15
Spark timing	22 bTDC
Intake valve timing	IVO: 22 bTDC/IVC:134 bTDC
Exhaust valve timing	EVO:130 aTDC/EVC:26 aTDC

Table 2 Test matrix, highlighted conditions are discussed in detail in the text

	LOAD										
SPEED (RPM)	10%			25%				50%		100%	
	AF			AF				AF		AF	
	12	13	14	11	12	13	14	11	12	11	12
1750	✗	x	x		x	x	x				
3060				x	✗			x	x	x	✗

RESULTS AND DISCUSSION

RING PACK ESTIMATION

RING PACK MODEL

The estimation of the ring pack contribution to the HC emissions was carried out by implementing and using a simplified ring pack model [4]. The model has been successfully used by Eng *et al.* [15, 16], and was tested in previous work by the same authors [4]. The model considers the ring pack crevices as a system of interconnected volumes. Each volume communicates with adjacent volumes through the ring end gap. A model of the ring dynamics was not considered, instead when the net pressure force on the ring changes direction the ring instantaneously changes position, going from one side of the groove to the other. The pressure in each volume is assumed uniform due to the small pressure drop across the volume. Both the measured crankcase and the cylinder pressure were

used to calculate the mass flow rates. In addition, following the results of [12,15,16], the gas in all volumes was assumed to be isothermal and equal to the cylinder wall temperature. Mass conservation was applied to each volume, and the gas flow through the ring end gap was approximated as compressible flow through an orifice. A discharge coefficient of 0.85 was used, which is the same as used in previous similar studies [12,15,16]. The model was tested against the measured blowby, and found to be in good agreement, see [4] for further details.

The ring pack crevice contribution to engine-out HC emissions was evaluated using two metrics. In the first metric the mass stored in the top piston land at the peak cylinder pressure was considered to escape the main combustion. This value, $m_{crev,max}$, is given as

$$m_{crev,max} = \frac{P_{cyl,max} V_{crev}}{RT_{wall}} \quad (3)$$

where V_{crev} is the volume above the top ring, $P_{cyl,max}$ is the maximum cylinder pressure, and T_{wall} is the cylinder wall temperature which was considered constant and equal to 385 K. The second metric considered that only mass that leaves the ring pack crevices after the flame propagation period contributes to engine-out HC emissions, *i.e.* the integration of the HC mass flow rate from the time that the cumulative mass fraction burn reached 90% (CA90) to the time when the exhaust valve closes (EVC),

$$m_{CA90-EVC} = \int_{t_{CA90}}^{t_{EVC}} \dot{m}_{out-crev} dt \quad (4)$$

where t_{CA90} and t_{EVC} are the times of CA90 and exhaust valve closing. The second method was considered to be more realistic since it assumes that the outflow of HC mass from the ring pack crevices before the flame

reaches the cylinder walls is consumed by the flame. The mass stored in the crevices was considered to be fresh mixture, with temperature equal to the cylinder wall. No penetration of burned mixture in the sections of the crevices behind the flame front was considered because the nature of the flame propagation inside the combustion chamber is random and generally unknown. Thus, the assumption of any geometric shape could lead to results that are not realistic. This results in over prediction of the model to the total ring-pack crevice HC contribution by the model.

In the following discussion, the results will be reported as the fuel-specific emission index of HC (EIHC). The HC *measured* in the exhaust using the steady state emission bench will be denoted as $EIHC_{M,EXH}$. The *calculated* mass of HC returned to the combustion chamber using the ring pack model will be denoted as $EIHC_{C,RP}$. Later, another *calculated* HC emission index will be developed from the cycle-resolved HC measurements and an estimate of the instantaneous exhaust mass flow rate, and this will be denoted $EIHC_{C,MB}$. All of these measures have been performed separately for each cylinder of the engine, referred to hereafter as cylinder 1 and cylinder 2.

ENSEMBLE-AVERAGE RESULTS

Analysis of the results using the ensemble-average pressure traces gives a global sense of the ring pack effect on the HC emissions. Figure 1, shows the estimation of the ring pack contribution to the HC emissions using the average pressure traces for air-fuel ratios shown in Table 2. Three conditions from Table 2 will be used for comparison going forward: 1750 RPM, 10% load, 3060 RPM, 25% load, and 3060 RPM, 100% load, at a constant air-fuel ratio of 12:1 (the stoichiometric air-fuel ratio for the fuel used was 14.6:1). These conditions span the operating range of the engine. The data in Fig. 1 are presented with the steady-state measured emissions ($EIHC_{M,EXH}$) on the horizontal axis, and the estimated ring-pack contribution using the model on the vertical axis, $EIHC_{C,RP}$. Both the maximum possible ring pack mass and the integrated CA90-EVC mass values are shown.

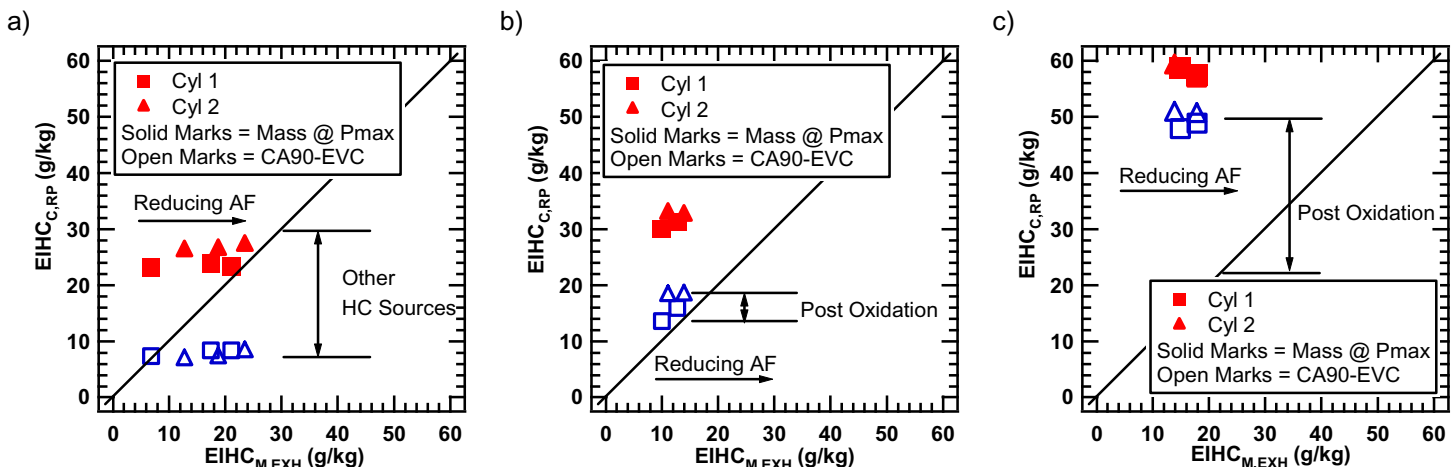


Figure 1 Ring pack model results for three different conditions, (a) 1750 RPM, 10% load, (b) 3060 RPM, 25% load, (c) 3060 RPM, 100% load