THE EFFECTS OF CREVICES ON THE ENGINE-OUT HYDROCARBON EMISSIONS IN SPARK IGNITION ENGINES

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Kyoungdoug Min

Submitted to the Department of Mechanical Engineering on December 1, 1993 in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy

ABSTRACT

Combustion chamber crevices into which the flame can not penetrate are identified as the largest contributor to the engine-out hydrocarbon (HC) emissions (~40 percent of the total). Therefore, understanding the crevice hydrocarbon mechanism at steady state and during engine warm-up is of significant importance. The objective of this work is a systematic study of the effects of the in-cylinder crevice on the engine-out HC emissions. There are three elements of this study: the study of the crevice size effects; the study of the crevice location effects; and the study of crevice effects during engine warm-up. The exhaust HC emissions during engine warm-up were measured on a cycle-by-cycle basis with a fast-response Flame Ionization Detector (FID). A methodology for calculating the mass averaged mean HC emissions per cycle from the fast-response FID measurement was developed. Propane fuel was used to minimize the oil layer absorption/desorption effect and eliminate the liquid fuel effect.

To quantify the effects of the crevice volume size and location on the exhaust hydrocarbon emissions, the piston and the head gasket of a single cylinder engine were modified to intentionally change the crevice volume. The experimental results show that hydrocarbon emissions are modestly sensitive to the piston crevice volume size. The trend of HC emissions increase under cold engine conditions with increasing piston crevice volume is similar to that of the HC emissions at steady state even though the HC emission levels are some 40 percent higher. Approximately 20 percent of the decrease in HC emissions during warm-up is due to the piston crevice effect and 80 percent of the decrease is due to change in oxidation level in the cylinder and exhaust port. However, HC emissions are very sensitive to the head gasket crevice size, and the head gasket crevice regions that are closer to the exhaust valve have a larger effect on the HC emissions. Based on the interpretation of the fast-response FID signals, the piston crevice changes have a modest effect on the exhaust hydrocarbon concentration at the end of the exhaust stroke; similarly, the head gasket crevice affects the exhaust hydrocarbon concentration during the blowdown and early exhaust stroke. A model for piston crevice gas transport and oxidation was developed, which is composed of one-dimensional mass, species, energy equation and a one-step oxidation. Based on simulation results, the amount of unburned fuel that survives in-cylinder oxidation modestly increases as the piston crevice volume increases.

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TABLE OF CONTENTS

ABSTRACT ....................................................................................................................... 5

ACKNOWLEDGMENTS ................................................................................................. 6

TABLE OF CONTENTS ................................................................................................... 7

LIST OF TABLES .......................................................................................................... 9

LIST OF FIGURES ....................................................................................................... 10

CHAPTER 1: INTRODUCTION .................................................................................... 17

1-1  Background ........................................................................................................ 17

1-2  Objectives .......................................................................................................... 19

CHAPTER 2: EXPERIMENTS ....................................................................................... 21

2-1  Experimental Apparatus .................................................................................... 21

2-1-1  Engine and Dynamometer .......................................................................... 21

2-1-2  Fuel Supply System .................................................................................... 22

2-1-3  Measurement of Exhaust Hydrocarbon Emissions ......................................... 22

2-1-4  Measurement of Cylinder Pressure ................................................................ 24

2-1-5  Data Acquisition System ........................................................................... 24

2-2  Experiments ....................................................................................................... 24

2-2-1  Procedure .................................................................................................... 24

2-2-2  Experimental Test Conditions ....................................................................... 25

2-2-3  Piston Modification ..................................................................................... 26

2-2-4  Head Gasket Modification .......................................................................... 28

2-2-5  Different Sampling Location of the Fast-Response FID ................................ 29

2-2-6  Liquid Fuel Effect ....................................................................................... 29
CHAPTER 3: DATA ANALYSIS ........................................................................39
  3-1 Calculation of Mass Averaged Mean Hydrocarbon Emissions per
    Cycle ....................................................................................39
  3-2 Prediction of Engine Component Temperatures and Crevice Volumes ....42
  3-3 Calculation of Maximum Hydrocarbons in Crevices .........................43
  3-4 Burn Rate Analysis ..................................................................44
  3-5 Method of Choosing Cold Engine Hydrocarbon Emissions Value .......44

CHAPTER 4: EXPERIMENTAL RESULTS AND DISCUSSION ..................51
  4-1 Introduction ........................................................................51
  4-2 Steady-State Experiments ..........................................................52
  4-2 Cold-Start Experiments .............................................................62
  4-4 Different Fast-Response FID Sampling Locations .........................66
  4-5 Liquid Fuel Effect on the Exhaust Hydrocarbon Emissions ............68

CHAPTER 5: MODELING OF PISTON CREVICE GAS TRANSPORT AND
  OXIDATION................................................................................101
  5-1 Introduction ........................................................................101
  5-2 Piston Crevice Gas Flow ............................................................102
  5-3 Modeling of Transport and Oxidation .........................................103
  5-4 Calculation Results and Discussion ............................................107
  5-5 Construction of Hydrocarbon Emissions Mechanism ....................112

CHAPTER 6: SUMMARY AND CONCLUSIONS ...........................................135
  6-1 Summary of Research Components .............................................135
  6-2 Methodology for Estimating the Mass Averaged Mean Hydrocarbon
    Emissions and Maximum Hydrocarbons in Crevices ......................136
  6-3 Effect of Crevice Volume Size and Location .....................................137
  6-4 Effect of Crevice Volume Size and Location During Warm-up ........137
  6-5 Effect of Liquid Fuel on the Exhaust Hydrocarbon Emissions ........139
LIST OF TABLES

Table 2.1  Ricardo Engine geometry................................................................. 31
Table 2.2  Ricardo Engine Crevice Volumes....................................................... 31
Table 3.1  Predicted warmed-up piston, cylinder wall temperature and percent
           change of total crevice volume during the engine warm-up period.......... 45
Table 4.1  Steady-state exhaust hydrocarbon emissions with different crevice
           volumes at 900 rpm, intake pressure = 1.0 bar.................................... 71
Table 4.2  Steady-state exhaust hydrocarbon emissions with different crevice
           volumes at 1600 rpm, intake pressure = 0.4 bar.................................... 72
Table 4.3  Steady-state exhaust hydrocarbon emissions with different crevice
           volumes at 1600 rpm, intake pressure = 0.7 bar.................................... 73
Table 4.4  Steady-state exhaust hydrocarbon emissions with different crevice
           volumes at 1600 rpm, intake pressure = 1.0 bar.................................... 74
Table 4.5  Burn durations................................................................................. 75
Table 4.6  Crevice outflows and their dispositions ............................................ 75
Table 5.1  List of f_b, f_0,c, f_r, f_0,p, calculated HC_{index}, and measured HC_{index}.... 115

- 9 -
LIST OF FIGURES

Figure 2.1  Schematic of pre-vaporizing gasoline fuel injection system .................. 32
Figure 2.2  Schematic of fast-response FID ......................................................... 33
Figure 2.3  Schematic of fast-response FID sampling set-up and time delay .......... 34
Figure 2.4  Air flow rate during the engine warm-up period ................................. 35
Figure 2.5  Relative air/fuel ratio during the engine warm-up period .................... 35
Figure 2.6  Piston modification procedure ......................................................... 36
Figure 2.7  Head gasket modification procedure .............................................. 37
Figure 2.8  Schematic of the location of the fast-response FID ......................... 38
Figure 3.1  Exhaust gas mass flow rate ............................................................. 46
Figure 3.2  Path lines of each burned gas element ........................................... 46
Figure 3.3  Comparison of the HC emissions of the conventional FID with the mass averaged mean HC concentration of the fast-response FID at steady state ......................................................... 47
Figure 3.4  Predicted engine component temperatures ...................................... 48
Figure 3.5  Predicted piston crevice volume change during warm-up ............... 48
Figure 3.6  A typical exhaust HC emissions during the engine warm-up period ..... 49
Figure 3.7  A typical exhaust HC emissions, $P_{\text{max}}$, and GIMEP as a function of cycle number .............................................................................................................. 50
Figure 4.1.a Effect of piston crevice and head gasket crevice changes on steady-state exhaust HC emissions at 900 rpm, intake pressure = 1.0 bar .......... 76
Figure 4.1.b Effect of piston crevice and head gasket crevice changes on steady-state exhaust HC emissions at 1600 rpm, intake pressure = 0.4 bar .......... 76
Figure 4.2 Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions at 900 rpm, intake pressure = 1.0 bar .......... 77
Figure 4.3  Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions at 1600 rpm, intake pressure = 0.4 bar. .............. 77
Figure 4.4  Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions at 1600 rpm, intake pressure = 0.7 bar. .............. 78
Figure 4.5  Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions at 1600 rpm, intake pressure = 1.0 bar. .............. 78
Figure 4.6  Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions. ............................................................... 79
Figure 4.7  Effect of head gasket crevice volume changes on steady-state exhaust hydrocarbon emissions. ...................................................... 79
Figure 4.8.a Nature of piston crevice flow: out flow velocity with respect to the piston. ............................................................................. 80
Figure 4.8.b Nature of piston crevice flow: out flow velocity with respect to the liner. .............................................................. 80
Figure 4.8.c Nature of piston crevice flow: distribution of crevice flow mass along the liner. ................................................................. 81
Figure 4.8.d Nature of piston crevice flow: core gas temperature. ................................................................................. 81
Figure 4.9  Crevice out flow geometry with respect to the piston. ...................... 82
Figure 4.10 Formation of the scraped-up vortex by the piston (a) and scraped-up vortex partially pushed out of the cylinder. ..................... 82
Figure 4.11 Comparison of the cycle-resolved exhaust hydrocarbon concentration of the original piston with the 2nd grooved piston. ........ 83
Figure 4.12 Comparison of the cycle-resolved exhaust hydrocarbon concentration of the 1st grooved piston with removed the full inner thin head gasket case. ................................................................. 83
Figure 4.13 The cycle-resolved exhaust hydrocarbon concentration for the on-third of the thin inner head gasket crevice case. ...................... 84
Figure 4.14  Extrapolation to zero crevice volume ......................................................... 84

Figure 4.15  Exhaust hydrocarbon emissions during the engine warm-up period
at 1600 rpm, intake pressure = 1.0 bar ................................................................. 85

Figure 4.16  Exhaust hydrocarbon emissions during the engine warm-up period
at 1600 rpm, intake pressure = 0.4 bar ................................................................. 86

Figure 4.17  Exhaust HC emissions index of the cold and warmed-up engine for
the piston top-land modifications at 900 rpm, intake pressure = 1.0
bar ......................................................................................................................... 87

Figure 4.18  Exhaust HC emissions index of the cold and warmed-up engine for
the piston top-land modifications at 1600 rpm, intake pressure =
0.4 bar ............................................................................................................... 87

Figure 4.19  Exhaust HC emissions index of the cold and warmed-up engine for
the piston top-land modifications at 1600 rpm, intake pressure =
0.7 bar ............................................................................................................... 88

Figure 4.20  Exhaust HC emissions index of the cold and warmed-up engine for
the piston top-land modifications at 1600 rpm, intake pressure =
1.0 bar ............................................................................................................... 88

Figure 4.21  Exhaust HC emissions index of the cold and warmed-up engine for
the piston top-land modifications at 2500 rpm, intake pressure =
1.0 bar ............................................................................................................... 89

Figure 4.22  Schematic of the effect of the piston crevice and oxidation level
change on the exhaust hydrocarbon emissions during the engine
warm-up period .................................................................................................. 89

Figure 4.23  Exhaust HC emissions index of the cold and warmed-up engine for
the head gasket modifications at 900 rpm, intake pressure = 1.0 bar ...... 90
Figure 4.24  Exhaust HC emissions index of the cold and warmed-up engine for the head gasket modifications at 1600 rpm, intake pressure = 0.4 bar................................................................. 90

Figure 4.25  Exhaust HC emissions index of the cold and warmed-up engine for the head gasket modifications at 1600 rpm, intake pressure = 1.0 bar................................................................. 91

Figure 4.26  Exhaust HC emissions index of the cold and warmed-up engine for the head gasket modifications at 2500 rpm, intake pressure = 1.0 bar................................................................. 91

Figure 4.27  Decrease in the exhaust HC emissions of the original piston during the engine warm-up period................................. 92

Figure 4.28  Time constant of the exhaust HC emissions, the exhaust gas temperature, and the exhaust valve temperature during the engine warm-up period................................. 92

Figure 4.29  Cycled-resolved exhaust hydrocarbon concentration at different times........................................................................ 93

Figure 4.30  Peak mass fraction burned during the engine warm-up period........... 94

Figure 4.31  Mass fraction burned profile for a cold engine and warmed-up engine........................................................................ 95

Figure 4.32  Cycle-resolved exhaust hydrocarbon concentration profiles inside the jet stream at 1 cm from the exhaust valve................................. 96

Figure 4.33  Cycle-resolved exhaust hydrocarbon concentration profiles on the port wall at 1 cm from the exhaust valve................................. 96

Figure 4.34  Cycle-resolved exhaust hydrocarbon concentration profiles for the different locations of one-third of the head gasket crevice at 1 cm on the port wall................................................................. 97
Figure 4.35  Comparison of the exhaust hydrocarbon emissions of the conventional injector with the propane-assisted conventional injector. ................................................................. 97

Figure 4.36  Steady state exhaust HC emissions of the pre-vaporizing gasoline injector and the conventional injector vs. lambda at 900 rpm, WOT. ................................. 98

Figure 4.37  Steady state exhaust HC emissions of the pre-vaporizing gasoline injector and the conventional injector vs. lambda at 1600 rpm, WOT. ................................................................. 98

Figure 4.38  Exhaust HC emissions for the conventional injector and the pre-vaporizing injector during the engine warm-up period at 900 rpm, WOT. ................................................................. 99

Figure 4.39  Exhaust HC emissions for the conventional injector and the pre-vaporizing injector during the engine warm-up period at 1600 rpm, WOT. ................................................................. 99

Figure 4.40  Exhaust HC emissions of the pre-vaporizing gasoline injector during the engine warm-up period. ................................................................. 100

Figure 5.1  Piston crevice gas distribution along the liner during the expansion process at 1600 rpm, intake pressure = 1.0 bar. ................................................................. 116

Figure 5.2  Piston crevice gas distribution along the liner during the expansion process at 1600 rpm, intake pressure = 0.4 bar. .................................................. 116

Figure 5.3  Percent piston crevice gas flow into the cylinder during expansion. ............... 117

Figure 5.4  Crevice gas velocity with respect to the piston during expansion at 1600 rpm, intake pressure = 1.0 bar. ................................................................. 118

Figure 5.5  Crevice gas velocity with respect to the piston during expansion at 1600 rpm, intake pressure = 0.4 bar. ................................................................. 118

Figure 5.6  Absolute gas velocity of the piston crevice gas during expansion at 1600 rpm, intake pressure = 1.0 bar. ................................................................. 119
Figure 5.7 Absolute gas velocity of the piston crevice gas during expansion at
1600 rpm, intake pressure = 0.4 bar ................................................................. 119

Figure 5.8 The initial thickness of the piston crevice gas at 1600 rpm, intake
pressure = 1.0 bar .................................................................................................. 120

Figure 5.9 The initial thickness of the piston crevice gas at 1600 rpm, intake
pressure = 0.4 bar .................................................................................................. 120

Figure 5.10 The thermal boundary layers growth during the expansion process ...... 121

Figure 5.11 Time of 50 percent propane consumption vs. temperature .................. 122

Figure 5.12 The temperature of core gas during the expansion process .................. 122

Figure 5.13 A typical initial boundary layer temperature and unburned
hydrocarbon profile ............................................................................................... 123

Figure 5.14 The thermal boundary layer profiles and propane concentration
profiles at the location of 1.9 cm from the top of the liner. .............................. 124

Figure 5.15 The thermal boundary layer profiles and propane concentration
profiles at the location of 2.6 cm from the top of the liner. .............................. 125

Figure 5.16 The initial unburned hydrocarbons from the piston crevice during
the expansion process at 1600 rpm, intake pressure = 1.0 bar ......................... 126

Figure 5.17 The remaining unburned hydrocarbons at EVO, 1600 rpm, intake
pressure = 1.0 bar ............................................................................................... 126

Figure 5.18 The initial unburned hydrocarbons from the piston crevice during
the expansion process at 1600 rpm, intake pressure = 0.4 bar ......................... 127

Figure 5.19 The remaining unburned hydrocarbons at EVO, 1600 rpm, intake
pressure = 0.4 bar ............................................................................................... 127

Figure 5.20 The sensitivity of the amount of unburned hydrocarbons left at
EVO to piston crevice volume size .................................................................... 128

Figure 5.21 The initial unburned hydrocarbons distribution along the liner of the
original piston and three different piston crevice reduction cases .................... 129

- 15 -
Figure 5.22  The remaining unburned hydrocarbons at EVO, 1600 rpm, intake pressure = 1.0 bar. ................................................................. 129

Figure 5.23  The sensitivity of the amount of unburned hydrocarbons left at EVO to piston crevice volume size......................................................... 130

Figure 5.24  The core gas temperature of the cold engine. ......................................................... 130

Figure 5.25  The initial piston crevice gas thickness for the cold engine......................... 131

Figure 5.26  The initial piston crevice gas distribution and remaining unburned hydrocarbons after oxidation at EVO, 1600 rpm, intake pressure = 0.4 bar. ................................................................. 132

Figure 5.27  The initial piston crevice gas distribution and remaining unburned hydrocarbons after oxidation at EVO, 1600 rpm, intake pressure = 1.0 bar. ................................................................. 133

Figure 5.28  The sensitivity of oxidation level of the cold engine to crevice volume size................................................................. 134
CHAPTER 1: INTRODUCTION

1-1 Background

Both the Clean Air Act Amendments of 1990 and California's new car emission standards require substantial reduction in exhaust hydrocarbon (HC) emissions. For example, California's standard of the Ultra Low Emission Vehicle (ULEV) in 1995 at 0.04 g/mi is 10 times less than that of the current standard (0.41 g/mi). Thus, many detailed investigations of HC emission sources in spark ignition engines have been conducted to meet these stringent emission requirements. The exhaust hydrocarbon emissions during starting and warm-up are most significant with spark ignition engines since fuel enrichment is necessary to achieve good driveability, and catalytic converters have low conversion efficiency due to their low temperature during starting and warm-up. In addition, poor mixture preparation [1] and the cold combustion chamber surface contribute to high unburned hydrocarbon emissions during the engine warm-up period. Current production vehicles tested in the US Federal Test Procedure (FTP) have HC emissions for the first 5.8 km that are 2.4 times higher than the FTP standard [2]. 70 percent of automotive tailpipe emissions from a 12 km trip occur in the first minute of engine operation [3]. Under city driving conditions, 52 percent of the trips are less than 3 km, and 40 percent of the distance traveled is needed to warm-up the engine [4].

The sources of the HC emissions are 1) crevices, 2) oil layer on the wall, 3) flame quenching (wall quenching and bulk quenching), 4) deposits, and 5) liquid fuel in
the cylinder. Adamczyk et al. [5] observed that oil layers did significantly increase the HC emissions, and this increase was extremely dependent on the specific fuels and oils in their bomb experiment. Carrier [6] and Kaiser showed that there was sufficient time during an engine cycle for an oil absorption/desorption process to occur. Experiments with a lubricant-free piston ring and cylinder [7-8] showed that the oil layer on the cylinder wall contributed 10 to 30 percent of total unburned HC emissions. The solubility of hydrocarbons in the oil layer is very sensitive to oil layer temperature. As a result, the amount of fuel absorbed in the oil layer decreases as the engine warms up.

Flame quenching at the wall was thought to be a major source of unburned HC emissions. According to studies [9-10], the flame-quenched unburned hydrocarbons rapidly diffuse from the wall and are substantially oxidized and the upper limit of the quench layer contribution to exhaust HC emissions was much less than 9 percent. However, bulk quenching, which extinguishes the flame before it arrives at the walls, may significantly contribute to the unburned HC emissions when an engine is operated at its lean limit or with high charge dilution.

A fraction of the fuel that is injected at the back side of the intake valve and port may enter the cylinder without vaporizing or mixing with the mixture. Some of this fuel is stored in crevices, deposits, or on the walls and escapes primary combustion, which contributes to the unburned HC emissions. Two different mixture preparation experiments (a pre-vaporizing gasoline injector and conventional injector) [11] show that the steady-state HC emissions of a conventional injector is about 10 percent higher than the pre-vaporizing gasoline injector case. This liquid fuel effect is especially significant during the engine starting and warm-up. Boam et al. [15] showed that in an engine at 20 °C as little as 20 percent of the metered fuel has evaporated before reaching the inlet valve. A recent study by Fox et al. [1] shows that a substantial amount of the fuel injected during starting does not appear in the combustible charge mixture.
Crevices in the combustion chamber into which the flame can not penetrate are thought to be one of the major sources of HC emissions. The largest crevice volume is the piston ring pack crevice volume. Adamczyk et al. [12] showed that the piston crevice volume contributed 80 percent of the HC emissions from two combustion chambers of production engines. Wentworth's [13] experiments with a sealed ring-orifice system showed that the HC emissions decreased by 20 to 40 percent with an 86 percent decrease in the piston top-land volume (calculated at ambient temperature). Woods et al. [14] observed a 30 percent decrease in the HC emissions with a 95 percent reduction in the piston top land. During engine starting and warm-up, the piston crown expands more than the cylinder liner and the density of crevice gas decreases, therefore the fraction of the total cylinder charge trapped in crevices decreases during engine warm-up.

A recent study [11] suggests that 8.4 percent of the fuel induted into the cylinder escapes primary combustion even though the vehicle-out HC emissions are 0.4 percent. This contributes to significant loss of fuel economy as well as causes high exhaust HC emissions. The major sources of unburned HC emissions and their contribution to engine-out HC emissions at part load in a warmed-up engine are as follows [11]: crevices, about 40 percent; oil layers and deposits, about 20 percent each; flame quenching and in-cylinder liquid fuel effects, about 10 percent each; exhaust valve leakage, less than 5 percent.

1-2 Objectives

At present the knowledge on the HC emission mechanisms during starting and warm-up is not complete. The understanding of the exhaust HC emissions changes during starting and warm-up is very important to reduce the engine-out HC emissions. Crevices are a major contributor to the exhaust HC emissions. One of the objectives of this thesis was to quantify the exhaust engine-out HC emissions during the engine warm-up period and crevices contribution to the HC emissions changes during engine warm-up. Another
objective was to quantify the crevice mechanism contribution to the exhaust HC emissions. Several piston modifications and head gasket modifications were conducted to systematically investigate the effect of the crevice volume size and location on the engine-out HC emissions.

The objectives of thesis can be summarized as follows:

1. Establishment of a methodology for calculating the mass averaged mean hydrocarbon emissions on a per cycle basis from a fast-response FID, which can make it possible to quantify the exhaust hydrocarbon emissions during the engine warm-up period.

2. Quantify the effect of piston crevice volume size on the exhaust hydrocarbon emissions during warm-up and at steady state.

3. Quantify the effect of crevice volume size at the head gasket location on the exhaust hydrocarbon emissions during warm-up and at steady state.

4. Investigate the effect of liquid fuel on the exhaust hydrocarbon emissions during starting and warm-up.

5. Modeling of piston crevice gas transport and oxidation in the cylinder.

6. Describe the crevice hydrocarbon formation and oxidation mechanism in the cylinder and exhaust port during the exhaust process.
CHAPTER 2: EXPERIMENTS

2-1 Experimental Apparatus

2-1-1 Engine and Dynamometer

The engine used in the experiments was a Ricardo Hydra Mark III single cylinder research engine. It has two valves, a naturally aspirated induction system, compression ratio of 8.3, and a hemispherical combustion chamber. The intake port is straight and directed toward the cylinder axis, and swirl or tumble air motion is not expected. The cylinder head of the engine has two 14 mm ports. These ports are parallel to the bore axis and are between the valves 17 mm from the centerline; one is used for the spark plug, and the other for the pressure transducer. Specifications for this engine are summarized in Table 2.1.

The engine was set up with a laminar air flow element (LFE) for measuring the volumetric air flow rate into the engine. The exhaust equivalence ratio was measured with an NTK model MO-100 air/fuel ratio analyzer (lambda sensor). To control the engine coolant temperature while running the engine at light load, two heaters were added to the head and cylinder coolant flow circuits because of the engine's large thermal inertia. The closed coolant flow circuits were cooled down by a heat exchanger that was connected with city water. Also, the engine oil temperature was controlled by a heater or heat exchanger, which were connected with the oil flow circuit.

An Eaton Dynamatic Model AF 6360 dynamometer is coupled to the Ricardo engine. This dynamometer is a variable frequency AC unit that can both drive and absorb,
allowing the engine to be motored at a desired speed. The geometrical configuration is well described by Sztenderowicz [16].

2-1-2 Fuel Supply System

In order to investigate different fuel and fuel vaporization effects on exhaust hydrocarbon emissions, three different fuel systems were employed: (1) solenoid injected propane, (2) port liquid gasoline injection into the intake port, and (3) a pre-vaporizing gasoline injection system. The solenoid injected fuel supply system is composed of a solenoid, a nozzle, and a metering valve is for fine tuning of the fuel flow rate. Most of the experiments were conducted with the solenoid injected propane to minimize engine oil layer, deposits, and liquid fuel effects on the exhaust hydrocarbon emissions. This continuous injection propane injector is located 30 cm from the intake valve seat. For the liquid fuel injection, a standard production Bosch injector was mounted in the inlet port and aimed at the back of the intake valve. The fuel injection control system controls for the injector phasing and duration in terms of engine crank angle degrees.

Figure 2.1 shows a schematic of the pre-vaporizing gasoline injection system. The location of this system is 45 cm from the intake valve seat, which is not critical because the fuel has been vaporized. The pre-vaporizing gasoline injection system consists of a heated tube and pressurized air injection. The heated tube from National Engineering Laboratory in England [17] was maintained at a temperature of about 130 °C. To help fuel vaporization, pressurized air is supplied with two tangential holes and one radial hole. The amount of air supplied by the pressurized air system is about 30 percent of the total air flow rate at 900 rpm WOT.

2-1-3 Measurement of Exhaust Hydrocarbon Emissions

During engine starting and warm-up, the exhaust hydrocarbon emissions change rapidly, and, therefore, a fast response instrument suitable for recording the rapidly
changing concentration is required. A fast-response flame ionization detector (HFR 400 FID) with a time response on the order of a millisecond is able to measure the cycle-by-cycle resolved exhaust hydrocarbon emissions [18-19]. The essential difference from a conventional FID is that the sample gas is mixed with the FID hydrogen fuel gas at the FID nozzle exit and the sample line is very short, which is the reason for the high frequency response [20]. A schematic diagram of a fast-response FID is shown in Figure 2.2.

The FID produces an output which is proportional to the total mass flow rate of hydrocarbons. As the sample pressure fluctuates during the blowdown and displacement process, mass flow rate to the FID sampling head also varies. To minimize the fluctuation in mass flow rate to the FID chamber, a constant pressure (CP) chamber arrangement is used; this arrangement consists of an intermediate chamber between the sampling tube and the FID chamber (Figure 2.2). To keep the mass flow rate through the FID constant in the presence of a fluctuating exhaust port pressure during the engine warm-up period, the FID and CP chamber pressure are controlled with a vacuum control unit. The constant pressure chamber was modified to use a large sample tube diameter in order to reduce the transit time (Figure 2.2). Another vacuum control unit controls the CP chamber pressure.

The overall time delay from the sampling point to the FID detector is dependent on the length of sample line and the chamber pressures. A computer program (SATFLAP3) [20] from Cambustion LTD predicts the transit time and response time of the sample flow. This prediction is based on steady, compressible, and isothermal assumptions. Figure 2.3 shows the FID sampling set-up and time delay for a typical FID operating condition. Three different sampling locations for the fast-response FID were chosen for the experiments. One was located 15 cm downstream from the exhaust valve seat, and another was located 1 cm from the exhaust valve seat. The other locations were inside the exhaust jet stream and on the exhaust port wall. These allowed investigation of the non-uniformity of exhaust hydrocarbon emissions at the exhaust port.
To convert the fast-response FID output voltage to ppm C\textsubscript{1}, the instrument must be calibrated with a gas of known composition (span gas). The fast-response FID was calibrated with span gas before and after each experiment. The fast-response FID data were acquired with a PC-based high-speed data acquisition system. A conventional FID (Rosemount Analytical LTD model 402 hydrocarbon analyzer) was used to measure the steady-state exhaust hydrocarbon emissions.

2-1-4 Measurement of Cylinder Pressure

Cylinder pressure data were taken with a Kistler model 7061 water-cooled piezoelectric transducer which minimized thermal shock. This was connected to a Kistler model 5004 charge amplifier. The data were recorded at every other crank angle degree because approximately 6500 consecutive cylinder pressure and fast-response FID data were collected to monitor exhaust hydrocarbon emissions during the engine warm-up period. The output of the charge amplifier was converted to a digital signal by an A/D data acquisition (DT2828-data translate LTD) card and stored on a PC in compressed form. A shaft encoder supplied the external clock for the data acquisition system.

2-1-5 Data Acquisition System

Two data acquisition systems were employed. A PC(486)-based high-speed data acquisition system recorded in-cylinder pressure and fast-response FID data. To monitor engine coolant temperature, intake air temperature, air flow rate, and exhaust air fuel ratio meter output, another A/D data acquisition system on a PC-XT was used.

2-2 Experiments

2-2-1 Procedure

The test engine starting procedure used was different from actual engine starting due to operational characteristics of the dynamometer and the engine control system. The
simulated starting procedure was as follows: The fast-response FID and conventional FID were ignited. It took 30 minutes for the fast-response FID to reach steady-state operating temperature at approximately 400 to 450 °C. It was checked for leaks with propane. If the fast-response FID was leaking, the output signal of the fast-response FID greatly fluctuated and drifted with time. The engine was then motored at the desired engine speed, the inlet pressure was set with the throttle valve, and the ignition was switched on. Then the two data acquisition systems began collecting data, and the fuel injection control circuit was energized.

During engine starting and warm-up, the volumetric efficiency of the engine decreased due to the increasing engine component temperatures and the hot burned gas back flow into the intake port. Figure 2.4 shows a typical air flow rate during engine starting and warm-up. The initial air flow rate decreased by approximately 10 percent during engine warm-up at WOT. Since exhaust hydrocarbon emissions are sensitive to air/fuel ratio, the exhaust equivalence ratio was controlled and kept constant during engine warm-up. Exhaust lambda (ratio of actual air/fuel ratio to stoichiometric air/fuel ratio) during engine warm-up is shown in Figure 2.5. Data show that exhaust lambda was constant, although the air flow rate changed during engine warm-up. The total number of cycles taken during engine warm-up was 6500 cycles. After an experiment was finished, the engine was cooled down by the heat exchanger that was connected with city water. It took about 1 hour to cool down the engine completely.

2-2-2 Experimental Test Conditions

The combustion chamber crevices into which the flame is unable to penetrate are a major source of unburned hydrocarbon emissions. The piston top-land volume (between the piston and the cylinder wall, above the first ring) and the first ring region (behind the first ring) are the most important crevices. The piston crown expands more than the cylinder liner during engine starting and warm-up. The crevice volume of a cold engine
and the crevice gas density are larger than those of the warmed engine. Therefore, the fraction of the total cylinder charge trapped in the crevices which escapes the primary combustion process will change during engine warm-up. To quantify crevice effects on the exhaust hydrocarbon emissions during engine warm-up, propane was used as the fuel in the engine experiments. The use of a gaseous fuel alleviates the complication of the effects of liquid fuel (such as liquid fuel getting into crevices) and enables one to clearly interpret the results. Additionally, propane is not significantly absorbed and desorbed by the engine oil on the cylinder walls, and a propane/air mixture which is near stoichiometric provides robust combustion so that the exhaust hydrocarbon contribution from the bulk quenching is small.

Engine operating conditions were selected to investigate crevice volume change effects during the engine warm-up period. The experimental conditions were

- 900 rpm, 1.0 bar inlet pressure
- 1600 rpm, 0.4 bar inlet pressure
- 1600 rpm, 0.7 bar inlet pressure
- 1600 rpm, 1.0 bar inlet pressure
- 2500 rpm, 1.0 bar inlet pressure
- Lambda: 1.05 to 1.1

These conditions represent an extensive range of engine component temperatures and warm-up times. In Section 3-2, predicted engine component temperature... and crevice volume changes for each engine operating condition will be discussed.

2-2-3 Piston Modification

The most important crevice is the piston ring pack crevice. During the engine warm-up period the piston crevice volume decreases due to the piston expansion, which depends on engine operating conditions [21]. The crevices of the Ricardo engine at ambient temperature (@ 20°C) are listed in Table 2.2. There were four threaded ports in
the cylinder head and thread crevices were estimated from ISO general purpose metric screw threads basic profile. Regions below the first ring were excluded from total crevice volume. The total mass trapped below the first ring region is less than 1/7th of that trapped above the first ring [22]. The head gasket and valve seat crevices were estimated to be negligible.

To investigate the sensitivity of the exhaust hydrocarbon emissions to the piston crevice size during engine warm-up and at steady state, the original piston of the Ricardo engine was modified in several steps, shown schematically in Figure 2.6. After the cold-start and steady-state experiments with propane and liquid fuel had been finished, the piston top land was grooved to increase the piston crevice volume by 43 percent over the base cold piston crevice volume (top land+first ring region: 0.939 cm³). In order to prevent flame propagation into the piston top-land crevice, the piston was grooved 2.8 mm below the top. At warmed-up conditions, this modified piston had approximately the same crevice volume as the unmodified piston at cold engine conditions. The second step was to groove the piston top land further to increase the top-land crevice volume by 0.4 cm³. These piston modifications may change crevice gas velocity during the expansion stroke, which may change crevice gas mixing with burned gas and, thus, the amount of in-cylinder oxidation. These effects and estimated crevice gas velocity change at steady state due to the piston grooves will be discussed in Chapter 5.

The last step of the piston modifications was to move the first ring up into the grooved the piston top land to decrease the piston crevice volume. This is also shown in Figure 2.6. Unfortunately, due to a machining error, the original first ring could not be fitted into the grooved piston top land, and the first ring of another 86 mm piston was used instead. The second ring was removed, and the original first ring was used as the second ring. This piston modification decreased the piston crevice volume by 31 percent of its original value at cold-start conditions. Before conducting the experiments, the engine was run about 20 hours to break-in the new first ring. The cold-start and steady-
state experiments were conducted for each piston modification. These three piston modifications allowed evaluation of the sensitivity of the exhaust hydrocarbon emissions to the piston crevice volume size.

2-2-4 Head Gasket Modification

The amount of unburned fuel/air mixture trapped in crevices in the cylinder head, per unit volume of crevice, is almost the same as that trapped in the piston crevice. However, the situation of crevice gas coming out from the cylinder head crevices differs from that of the piston crevice because the piston moves down during the expansion stroke. The unburned hydrocarbons from the piston crevice are laid along the cylinder wall, and these hydrocarbons may have more time to be mixed with burned gas in the cylinder during their travel to the exhaust port. The purpose of these experiments was to investigate the sensitivity of the exhaust hydrocarbon emissions to crevice location and size. It proved easiest to modify the head gasket crevice. Two different head gasket modifications were made. A schematic of the head gasket arrangement of the Ricardo engine is shown in Figure 2.7. There are two head gaskets, and the thin inner head gasket (thickness: 0.19 mm) was modified to increase the head gasket crevice volume (which was originally zero). This gap size should be small enough to quench flame propagation. At first, 1/2 of the thin inner head gasket (0.283 cm$^3$) was removed, as shown schematically in Figure 2.7. This increased the total cold state crevice volume by 20 percent. Then, the full thin inner head gasket (0.48 cm$^3$) was removed, which increased the cold state crevice volume by 39 percent.

To investigate the effect of retention time of unburned mixture in the cylinder, one-third of the thin inner head gasket was removed at the exhaust valve side as illustrated in Figure 2.7; this increased the cold state crevice volume by 13 percent. Then this crevice volume was rotated to the intake valve side of the combustion chamber. Since the spark plug was equidistant from these locations, the flame propagation toward both crevices
should be identical. From this experiment, the effect of different crevice locations on the 
hydrocarbon emissions could be examined.

2-2-5 Different Sampling Location of the Fast-Response FID

The purpose of this experiment was to investigate non-uniformity of the exhaust 
hydrocarbon emissions near the exhaust valve. The previous experiments were conducted 
at 15 cm from the exhaust valve seat. The exhaust hydrocarbon emissions were quite 
uniform along the radial direction at that sampling location. A diagram of the location of 
the fast-response FID near the exhaust valve is shown in Figure 2.8. One sampling 
location was within the exhaust jet flow about 1 cm from the exhaust valve seat, and the 
other was located on the exhaust port wall.

2-2-6 Liquid Fuel Effect

Preliminary experiments showed that the decrease in the exhaust unburned 
hydrocarbon emissions from start-up to steady state was substantially larger with gasoline 
fuel than with the propane/air mixture, both in absolute and in relative terms. This result 
suggests that liquid fuel in the crevices and on the walls is an additional source mechanism 
for the exhaust hydrocarbon emissions. A major difference between the trapped charge 
mixture and the trapped liquid fuel is that the former is a mixture of air and fuel vapor, but 
the latter has very little oxygen.

To quantify the effects of liquid fuel in the crevices and on the walls on the exhaust 
hydrocarbon emissions during warm-up, a pre-vaporizing gasoline injection system and a 
conventional port injection system were employed. The relative air/fuel ratio according to 
the air flow rate and the amount of fuel injected per cycle was kept constant, and the 
mixture was injected when the intake valve was closed. For the conventional injector, 
additional gaseous propane was used to fix the overall lambda at 0.5 for 4 to 10 seconds
of the operation, because the use of only a stoichiometric amount of liquid indolene required more than 50 cycles before the engine would start firing.
### Table 2.1  Ricardo Engine geometry

<table>
<thead>
<tr>
<th>Model:</th>
<th>Ricardo Hydra MK III</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type:</td>
<td>Single cylinder, iron block and liner (wet), alloy head, two valve, separate overhead cams</td>
</tr>
<tr>
<td>Chamber:</td>
<td>Hemispherical, central ignition</td>
</tr>
<tr>
<td>Compression ratio:</td>
<td>8.3</td>
</tr>
<tr>
<td>Bore x stroke:</td>
<td>85.67 mm x 86.00 mm</td>
</tr>
<tr>
<td>Clearance volume:</td>
<td>68 cm³</td>
</tr>
<tr>
<td>Displacement:</td>
<td>496 cm³</td>
</tr>
<tr>
<td>Valve timings:</td>
<td>IVO: 4⁰ BTC           IVC: 49⁰ ABC</td>
</tr>
<tr>
<td></td>
<td>EVO: 54⁰ BBC           EVC: 16⁰ ATC</td>
</tr>
</tbody>
</table>

### Table 2.2  Ricardo Engine Crevice Volumes*

<table>
<thead>
<tr>
<th>Piston top land:</th>
<th>0.591 cm³</th>
</tr>
</thead>
<tbody>
<tr>
<td>First ring region:</td>
<td>0.348 cm³</td>
</tr>
<tr>
<td>Spark plug** and other threads:</td>
<td>0.133 cm³</td>
</tr>
<tr>
<td>Total estimated crevice volume:</td>
<td>1.072 cm³</td>
</tr>
</tbody>
</table>

(1.6 percent of clearance volume)

* cold engine (@ 20⁰ C)

* neglect valve seat crevice and below the first ring

** only thread crevice counted
Figure 2.1  Schematic of pre-vaporizing gasoline fuel injection system.
Figure 2.2  Schematic of fast-response FID.
Schematic of fast-response FID sampling set-up and time delay.

<table>
<thead>
<tr>
<th></th>
<th>Length (mm)</th>
<th>DIA. (mm)</th>
<th>Transit time (msec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Transfer tube (A)</td>
<td>150</td>
<td>0.688</td>
<td>1.31</td>
</tr>
<tr>
<td>Expansion tube (B)</td>
<td>20</td>
<td>0.150</td>
<td>0.68</td>
</tr>
<tr>
<td>Connecting tube (C)</td>
<td>25</td>
<td>0.30</td>
<td>0.41</td>
</tr>
</tbody>
</table>

- 15 cm

- Exhaust valve

- Calibration tube

- Connecting tube (C)

- Expansion tube (B)

- FID sampling head
Figure 2.4  Air flow rate during the engine warm-up period.

Figure 2.5  Relative air/fuel ratio during the engine warm-up period.
Figure 2.6  Piston modification procedure.
Figure 2.7  Head gasket modification procedure.
Figure 2.8 Schematic of the location of the fast-response FID.
CHAPTER 3: DATA ANALYSIS

3-1 Calculation of Mass Averaged Mean Hydrocarbon Emissions per Cycle

The output of the fast-response FID indicates the time varying hydrocarbon concentration at a fixed location. However, the exhaust gas mass flow rate changes during the blowdown and the exhaust stroke and cycle by cycle. A data analysis technique to calculate mass averaged mean hydrocarbon emissions per cycle from the fast-response FID measurement was established. The exhaust mass flow rate during the blowdown and the exhaust stroke was obtained from an energy balance on the cylinder contents during the blowdown and the exhaust stroke:

\[ dU = \delta Q - \delta W - h_e dm \]

where

\[ U = \text{internal energy} \]
\[ Q = \text{heat transfer to the gas} \]
\[ W = \text{work done by gas} \]
\[ h_e = \text{enthalpy of exhausting gas} \]

The burned gas in the cylinder is regarded as a uniform ideal gas. Then the instantaneous mass flow rate from the engine cylinder is obtained:

\[ \dot{m} = \frac{1}{\gamma} \left( \frac{\dot{p}}{\dot{V}} + \frac{\dot{V}}{\gamma - 1} \frac{\dot{Q}}{pV} \right) \]

where

\[ \dot{m} = \text{mass flow rate} \]
m = total mass in cylinder
p = in-cylinder pressure
V = in-cylinder volume
\( \gamma \) = specific heat ratio

The specific heat ratio (\( \gamma \)) was approximated by a linear function of cylinder temperature [23].

\[
\gamma = 1.346 - 5 \times 10^{-5} T
\]

Woschni's correlation was used to calculate the heat transfer rate [24].

\[
h = c_1 B^{-0.2} p^{0.8} T^{-0.55} W^{0.8} \text{ (w/m}^2\text{k)}
\]

where
B = bore
p = in-cylinder pressure
T = in-cylinder gas temperature
W = the average cylinder gas velocity

and the initial total mass was determined from the measured air flow rate, the exhaust gas equivalence ratio, and the estimated residual fraction:

\[
m_{\text{total}} = \frac{m_{\text{air}} [1 + (F/A)]}{1 - x_r}
\]

where
\( m_{\text{air}} \) = mass of air the cylinder
F/A = fuel/air ratio
\( x_r \) = residual fraction

The residual fraction was estimated using the model developed by Fox [25]. The model is based on the engine speed, inlet pressure, valve overlap, and fuel/air equivalence ratio.

A typical exhaust gas mass flow rate from the calculation is shown in Figure 3.1. The calculation shows that about 40 percent of the burned gas in the cylinder escapes
during the blowdown period and that, early in the exhaust stroke, the flow is reversed. This might be due to the exhaust pressure wave effect. From the calculation of the exhaust gas flow rate, the travel of each burned gas element along the exhaust port was obtained based on no mixing between successive plugs of exhaust gas and no heat transfer to the walls. Figure 3.2 shows the path lines of each burned gas element calculated from Figure 3.1. The dotted line in Figure 3.2 indicates the location of the fast-response FID. The time required for each burned gas element to travel from the exhaust valve to the fast-response FID sampling point was obtained from the path lines. It takes less than 10 CA degrees for burned gas segments to reach the fast-response FID sampling point during the blowdown period and early exhaust stroke and 20 to 40 CA degrees for most of the exhaust process, depending on the mass flow rate. The fast-response FID time delay between the sampling point to FID detector was obtained from the program SATFLAP3 [20]. The fast-response FID signal was matched with the corresponding exhaust gas flow rate. The mass averaged mean exhaust hydrocarbon emissions per cycle were then calculated from:

\[
\overline{[HC]} = \frac{\int_{EVO}^{IVO} [HC]_{FID} \dot{m} \, dt}{\int_{EVO}^{IVO} \dot{m} \, dt}
\]

where

\[
[HC]_{FID} = \text{the fast-response FID signal corrected for time delay}
\]

\[
\dot{m} = \text{exhaust gas mass flow rate}
\]

The comparison of the exhaust hydrocarbon emissions of the conventional FID with the mass averaged mean hydrocarbon concentration of the fast-response FID at steady state is shown in Figure 3.3. Each point of the fast-response FID data was averaged over 400 consecutive cycles. There is good agreement between the two measurement techniques except for lambda = 1.2; our experiments were conducted between lambda = 1.05 and 1.1.
3-2 Prediction of Engine Component Temperatures and Crevice Volumes

During the engine warm-up period, the engine component temperatures increase due to heat transfer from the burned gas. The crevice volume between the piston and the cylinder liner decreases due to the thermal expansion of the piston. The crevice gas density also decreases during the engine warm-up period. The amount of unburned gas which escapes primary combustion thus decreases during the engine warm-up period. To predict the engine component temperatures and the piston crevice volume, a computer model has been developed simulating the thermal processes of the engine [21]. This model was based on a lumped thermal capacitance analysis for each of the five major components: the piston, cylinder block, cylinder head, engine oil, and three separate coolant reservoirs. Thermal expansion rates were calculated for the piston and cylinder liner based on their respective component temperatures and expansion coefficients, so that the piston crevice change was determined during the engine warm-up period.

The input of heat flux from the burned gas to the piston, cylinder head, and cylinder wall was calculated from a cycle simulation program [26]. The calculated coolant temperature of the cylinder head was matched with the measured coolant temperature of the Ricardo engine. Then the engine component temperatures and the piston crevice volume were calculated [21]. The predicted engine component temperatures and the piston crevice volume change of the Ricardo engine during warm-up are shown respectively in Figure 3.4 and Figure 3.5. From the simulation, the time scale for warm-up was on the order of a hundred seconds and changed with the engine speed and load. The piston and the cylinder wall temperature at steady state, and the piston crevice volume (the top land volume and the first ring region) decrease during the engine warm-up period, are listed in Table 3.1. The predicted engine component temperatures and the piston crevice volume were then used to calculate the maximum amount of hydrocarbons in the crevices which escaped burning during the normal combustion process.
3-3 Calculation of Maximum Hydrocarbons in Crevices

The maximum amount of unburned fuel that escaped combustion was calculated as follows. The total crevice volume consists of the piston top land, the first ring region, the spark plug, and the other threads in the cylinder head. The gas temperature in the piston top land and the first ring region was assumed to be the average of the piston and the cylinder wall temperature, and the crevice gas temperature in the threads was assumed to be the same as the cylinder head temperature. The crevices were filled with fuel/air mixture and residual gas and the mixture in the crevice was assumed to be an ideal gas. Maximum in-cylinder pressure was obtained from the experiments. The maximum mass trapped in the crevices is then:

\[
m_{cr} = \frac{p_{max} V_{cr}}{RT_{cr}}
\]

where

\[
p_{max} = \text{maximum in-cylinder pressure}
\]
\[
V_{cr} = \text{crevice volume}
\]
\[
R = \frac{\dot{R}}{M_{cr}}
\]
\[
T_{cr} = \text{crevice gas temperature}
\]

The crevice volume at steady state was estimated by using Kaplan's program [21]. The maximum mass in the crevice was corrected to the cylinder average ppm C, using the total in-cylinder mass, air/fuel ratio, and the residual fraction. The mole ratio of unburned HC in the crevices to the burned gas in the cylinder is:

\[
\frac{n_{HC}}{n_{total}} = \frac{m_{cr}}{m_{total}} \times \frac{M_b}{M_{CH_{2}H_{7}}} \times \frac{(F / A)(1 - x_r)}{1 + F / A}
\]

where

\[
m_{total} = \text{total mass in cylinder}
\]
\[
M_b = \text{molecular weight of burned gas}
\]
The maximum unburned fuel fraction will be compared with measured unburned hydrocarbon emissions at the exhaust port to estimate how much unburned fuel in the crevices is oxidized in the cylinder and the port or left in the cylinder.

3-4 Burn Rate Analysis

The objective is to investigate how the maximum mass fraction burned profile changes during engine warm-up. A one-zone burn rate analysis was applied to calculate the mass fraction burned. The model, proposed by Gatowski [27] and Chun [23], was implemented in FORTRAN on a PC by Cheung [28]. This model was based on the First Law of Thermodynamics and includes sub-models for the effects of residual fraction, heat transfer, and crevices. This burn rate analysis was also used to calculate the burn angle.

3-5 Method of Choosing Cold Engine Hydrocarbon Emissions Value

A typical exhaust hydrocarbon emissions data set taken during the engine warm-up period is shown in Figure 3.6. The exhaust hydrocarbon emissions at early time change rapidly, making it difficult to determine the initial value of the exhaust hydrocarbon emissions during the engine warm-up period. Several cycles were needed to flush out the burned gas of the abnormal cycles in the exhaust port. Data show that the earliest several cycles of combustion after firing were not normal. In addition, the first cycle when the fuel supply system was energized misfired, which contributed to higher exhaust hydrocarbon emissions during the early warm-up period (Figure 3.7).

The method of choosing the initial exhaust hydrocarbon emissions value is as follows: the ratio \( [\text{HC}(t) - \text{HC}(\infty)] / \text{HC}(\infty) \) was plotted on a log scale against cycle number as shown in Figure 3.6. A straight line is a good fit to the data, and the initial exhaust hydrocarbon emissions(\( \text{HC}(0) \)) was calculated from this fitted line.
Table 3.1  Predicted warmed-up piston, cylinder wall temperature and percent change of total crevice volume* during the engine warm-up period

<table>
<thead>
<tr>
<th>condition</th>
<th>piston (K)</th>
<th>cylinder wall (K)</th>
<th>total crevice vol. change (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>900 rpm, 1.0 bar</td>
<td>413</td>
<td>360</td>
<td>22</td>
</tr>
<tr>
<td>1600 rpm, 0.4 bar</td>
<td>400</td>
<td>360</td>
<td>19</td>
</tr>
<tr>
<td>1600 rpm, 0.7 bar</td>
<td>430</td>
<td>360</td>
<td>27</td>
</tr>
<tr>
<td>1600 rpm, 1.0 bar</td>
<td>440</td>
<td>360</td>
<td>31</td>
</tr>
<tr>
<td>2500 rpm, 1.0 bar</td>
<td>480</td>
<td>360</td>
<td>43</td>
</tr>
</tbody>
</table>

* original piston
Figure 3.1 Exhaust gas mass flow rate.

Figure 3.2 Path lines of each burned gas element.
Figure 3.3  Comparison of the HC emissions of the conventional FID with the mass averaged mean HC concentration of the fast-response FID at steady state.
**Figure 3.4**  Predicted engine component temperatures.

**Figure 3.5**  Predicted piston crevice volume change during warm-up.
Figure 3.6  A typical exhaust HC emissions during the engine warm-up period.
Figure 3.7  A typical exhaust HC emissions, $P_{\text{max}}$, and GIMEP as a function of cycle number.
CHAPTER 4: EXPERIMENTAL RESULTS AND DISCUSSION

4-1 Introduction

This chapter presents the experimental results of this study and discusses these results. Four different sets of results are presented: (1) steady-state experiments, (2) cold-start experiments, (3) different fast-response FID sampling location experiments, and (4) liquid fuel effect experiments.

The purpose of the steady-state experiments was to quantify the effects of changes in crevice volume and location on the exhaust hydrocarbon emissions. The object of cold-start experiments, with a propane/air mixture, was to quantify the exhaust hydrocarbon emissions changes in relation to the crevice volume change and the in-cylinder and exhaust port gas temperature change during the engine warm-up period. The purpose of the different fast-response FID sampling location experiments was to investigate the spatial variation of the exhaust hydrocarbon emissions near the exhaust valve during the engine warm-up period and at steady state. Under actual gasoline engine starting conditions, a large amount of overfueling is used in order to reach the first combustion event quickly and to achieve good driveability during engine warm-up. A portion of this gasoline may enter the cylinder as droplets or ligaments. Some of this entering liquid fuel may be stored in crevices, oil layers, and deposits, thereby escaping primary combustion and contributing to the exhaust hydrocarbon emissions during the engine warm-up period. The effects of liquid fuel in the crevices, deposits, and on the oil layer are investigated in Section 4-5.
4-2 Steady-State Experiments

The steady-state exhaust hydrocarbon emissions of the different piston crevice volumes and head gasket crevice volumes are shown in Figures 4.1.a and 4.1.b. The exhaust hydrocarbon emissions data were measured with the conventional FID at the steady-state engine operating conditions and result from the average of several measurements taken on different days. Even though the piston crevice volume is increased from 0.687 cm³ to 1.65 cm³, the steady-state exhaust hydrocarbon emissions only increase by 16 percent (Figure 4.1.a). However, the head gasket crevice volume changes have a larger effect on the exhaust hydrocarbon emissions than the piston crevice volume changes. The steady-state hydrocarbon emissions change from 1400 ppm C₁ to 2080 ppm C₁ when the head gasket crevice volume is increased by 0.483 cm³. The steady-state exhaust hydrocarbon emissions of the 1600 rpm inlet pressure of 0.4 bar case are shown in Figure 4.1.b. The above trends were observed, but the absolute magnitude of the emissions level was different. The steady-state hydrocarbon emissions at the different air/fuel ratios, engine speeds, and loads are summarized in Tables 4.1 through 4.4.

In Figures 4.2 through 4.6 the x-axis contains the crevice volume changes from the piston modification (at steady state to account for the volume decrease from thermal expansion of the piston) normalized with a reference crevice volume. The reference condition was the crevice volume from the first modification of the piston, because the experiments with the head gasket modifications were conducted with this piston with the first groove. Also, the steady-state exhaust hydrocarbon emissions measurement at different lambdas for each case was normalized with corresponding lambda emissions at the reference condition. The symbols indicate the mean exhaust hydrocarbon emissions values and the letters in the figures indicate the corresponding piston modifications, which were described in Section 2-2-3. The crevice volume with the piston modifications was
changed by a factor of 2.6. These experimental results show that the exhaust hydrocarbon emissions at steady state were modestly sensitive to the piston crevice volume change, even though the piston crevice is thought to be a major source of the exhaust hydrocarbon emissions. The results with the different engine speeds and loads are similar. The slope of the normalized exhaust hydrocarbon emissions is about 0.2, and it decreases a little as the engine load and speed increase.

Wentworth's experiments with a sealed ring-orifice system showed that the exhaust hydrocarbon emissions decreased by 20 to 40 percent with an 86 percent decrease in the piston top-land volume (calculated with respect to ambient temperature). The experiments were conducted with iso-octane, which might have other effects on the exhaust hydrocarbon emissions [13]. Similar results were obtained in different studies [14]. Woods et al. [14] showed that the exhaust hydrocarbon emissions decreased by 30 percent with a 95 percent decrease in the piston top-land volume by using a Shamban carbon-impregnated PTFE seal. These experiments were also conducted with gasoline at 2000 rpm and half load. All of these experimental results of the piston modifications are consistent in that the percent decrease in the exhaust hydrocarbon emissions are significantly less than the percent decrease in the piston crevice volume in these piston crevice volume ranges (in our experiments, the normalized slope is 0.2).

The maximum amount of unburned gas trapped in the piston crevice is increased by grooving the piston or decreased by moving up the first ring. As the spark plug is located 17 mm from the center of the combustion chamber, about 85 to 90 percent of the gas in the piston crevice is the unburned fuel, air, and residual gas mixture, based on the flame geometry [26]. However, the hydrocarbon concentration in the exhaust port did not change by the same percentage as the piston crevice volume changes. It is known that significant oxidation of hydrocarbons from the crevices (and from the other HC sources) occurs within the cylinder and in the exhaust port. The level of oxidation in the exhaust port should be the same even though the piston was modified. Thus, a possible reason of
the exhaust hydrocarbon emissions being only modestly sensitive to the piston crevice volume is that the percentage of crevice gas oxidized in the cylinder during the expansion process and the exhaust stroke increases as the piston crevice volume increases. This means that the amount of the crevice gases which survive the expansion and exhaust process increases slightly. Also, the velocity of the gas leaving the piston crevice is proportional to the crevice volume because the clearance between the piston and cylinder wall is the same, which may affect mixing with the burned gas and oxidation in the cylinder. This will be discussed in Chapter 5. The changes in the exhaust hydrocarbon emissions with the head gasket modifications were different from the piston modification case. The experimental results in Figure 4.7 show that the relative sensitivity of the exhaust hydrocarbon emissions to changes in the head gasket crevice volume is about 1.0, and the line goes through the origin.

We now make a distinction between the hydrocarbons that come out of the crevices during the expansion process and those that come out during the blowdown process (i.e. the cylinder/exhaust manifold pressure equilibration process that occurs when the exhaust valve opens). To a good approximation, the crevices are in pressure equilibrium with the combustion chamber. Therefore, the amount of outflow from these crevices is not significant during the exhaust displacement process (after the blowdown process is over), during which the chamber pressure is approximately constant. The crevice outflows and their subsequent dispositions are summarized in Table 4.2. The details are discussed in the following.

We shall first look at the outflow from the piston crevice during the expansion process. In Figure 4.8.a, the outflow velocity relative to the piston, $u_o$ (normalized by the mean piston speed $S_p$) is shown versus the relative distance along the liner. The x-axis is the piston position normalized by the stroke: the left end of the x-axis is the TDC position, and the right end, at a relative distance of 0.81, is the position of the piston at EVO. The
outflow velocity, $u_O$, is obtained based on pressure equilibrium between chamber and crevice:

$$\frac{u_O}{S_p} = \frac{1}{2L} \frac{V_c}{\pi B \delta_c} \frac{1}{p} \frac{dp}{d\theta}$$

Where

- $S_p =$ mean piston speed
- $L =$ stroke
- $V_c =$ crevice volume
- $B =$ bore
- $\delta_c =$ piston top land to liner clearance
- $\theta =$ crank angle
- $p =$ pressure

In Figure 4.8.a, $u_O$ is negative (into the crevice) at first because pressure is rising in the cylinder and gas is being pushed into the crevice. It becomes positive (out of the crevice) after the peak pressure point because the pressure is then decreasing. Because the piston is moving down during the expansion process, the piston velocity must be subtracted from the crevice outflow velocity to obtain the absolute velocity, $u_O'$ (with respect to the liner). This value is shown in Figure 4.8.b. Note that the value of $u_O'$ is always in the negative (downward) direction, i.e. the piston is moving down faster than the crevice outflow velocity $u_O$. The magnitude of $u_O$ or $u_O'$ is of the order of the mean piston speed $S_p$. It can be shown that the normalized velocity is roughly independent of engine load and speed. The piston top land to cylinder liner clearance is on the order of 0.1 mm. The typical Reynolds number associated with the flow is ~ 100. Thus, the flow process associated with the outflow is laminar.

A useful perspective for the crevice outflow is in the reference frame of the piston (Figure 4.9). In this frame of reference the liner is moving upwards at a speed faster than the crevice outflow gas velocity. Thus the outflow of crevice gas is stretched to a thin
layer (of a thickness thinner than the piston to liner clearance) which is laid down in a laminar fashion along the liner. The distribution of the outflow-mass is shown in Figure 4.8.c. This piston crevice gas distribution along the liner was obtained from the ideal gas equation:

\[
\frac{dm}{d\theta} = \frac{V_c}{RT_c} \frac{dp}{d\theta}
\]

Where

\[V_c = \text{crevice volume}\]

\[R = \frac{R}{M}\]

\[\Gamma_c = \text{crevice gas temperature}\]

\[p = \text{cylinder pressure}\]

Note that a majority of the mass comes out in the early part of the expansion process. The burned gas temperature to which the crevice gas is exposed is shown in Figure 4.8.d, which is obtained from the cycle simulation [26] (relative distance here denotes the piston position corresponding to the core burned gas temperature shown). In the expansion process, the layer of the crevice gas on the liner would be stripped by the turbulent motion of the charge to mix with the hot bulk gas. Results from oxidation kinetics calculations [29] have indicated that the crevice gas would oxidize completely if the temperature is above \(\sim 1300K\). Thus, if the crevice gas is mixed with the hot bulk gas at the early expansion process, it would get oxidized. It is therefore plausible that only a very thin layer of the crevice gas adjacent to the cold wall would escape oxidation. Hence a self-regulating mechanism apparently exists: of the hydrocarbons emitted from the piston crevice in the expansion process, only the portion that stays within the cold wall thermal boundary layer would survive oxidation. Thus, the HC emissions from this process are only modestly affected by the piston top-land crevice size. The crevice gas coming out during the blowdown process would not extensively oxidize because the bulk gas temperature quickly decreases during expansion and blowdown. After blowdown, during
the exhaust stroke, the piston rolls up the gas layer adjacent to the liner into a vortex [30]. Therefore, the blowdown crevice exit gas remains within this rolled-up vortex [30]. The amount of blowdown crevice gas in the vortex would be proportional to the piston crevice volume.

We will next look at the fate of the unoxidized piston crevice gas in the exhaust process. Part of this crevice gas would exhaust in the blowdown process. The remaining part, which is located along the liner, would be scraped up into the vortex. At the end of the exhaust process, a part of this vortex would be pushed out of the cylinder by the piston (Figure 4.10). Depending on the extent of valve overlap and the inlet and exhaust pressure, parts of this exhausted crevice gas may be reinjected into the cylinder during the reverse flow process when the intake valve opens [31].

Evidence that the piston crevice gas is exhausted as part of the scraped-up vortex may be found in Figure 4.11. Figure 4.11 shows the time-resolved exhaust hydrocarbon concentration for the original piston and for the second grooved piston obtained from the fast-response FID. As the exhaust valve opens, there is a peak in hydrocarbon concentration; this peak is due to the higher hydrocarbon concentration in the gas from the previous cycle between the exhaust valve and the fast-response FID sampling point, and the unburned mixture retained in the crevices around the exhaust valve seat. Another possibility is that small amounts of unburned mixture may leak through the exhaust valve during compression and combustion. However, the cycle-resolved exhaust hydrocarbon concentration near the exhaust valve shows there was no significant valve leakage. The exhaust hydrocarbon concentration quickly reaches a low level during the main exhaust period. As the piston approaches top dead center (TDC), the exhaust hydrocarbon concentration rises steeply, due to the rolled-up vortex that contains a substantial fraction of unburned hydrocarbons originally located adjacent to the cylinder wall. From Figure 4.11, the exhaust hydrocarbon concentration for both cases is almost the same during the blowdown and early exhaust stroke, but the hydrocarbon concentration of the 2nd
grooved piston is higher than those of the original piston at the end of the exhaust stroke. Based on Figure 4.11, it is possible to conclude that the unburned gas from the piston crevice before the exhaust valve opens, which is proportional to the piston crevice volume size, is quickly oxidized and the amount of the unburned gas left after oxidation is not sensitive to the piston crevice volume size. This may explain why the exhaust hydrocarbon emissions are only modestly sensitive to the piston crevice volume change.

There are two major differences between the head gasket crevice and the piston crevice. First, the head gasket crevice is stationary. Therefore, unlike the piston crevice, the crevice flow during the expansion process is not laid down along the liner wall. As a result, most of this outflow is oxidized in the expansion process. Second, the head gasket crevice is located much closer to the exhaust valve than the piston crevice during blowdown, when the critical crevice outflow occurs. Therefore, almost all the head gasket crevice gas outflow during the blowdown period would escape either in the blowdown flow or the displacement flow produced by the piston. Figure 4.12 shows the fast-response FID output for the 1st grooved piston and the full thin inner head gasket crevice. The basic features of the exhaust hydrocarbon concentration are the same as in Figure 4.11. However, during the blowdown period and early in the exhaust stroke, the exhaust hydrocarbon concentration of the full thin inner head gasket crevice is substantially higher than that with zero head gasket crevice. During the latter part of the exhaust stroke, the two exhaust hydrocarbon concentration profiles are almost the same. The amount of the unburned gas trapped in the head gasket crevice is proportional to the crevice volume size, which is the same as the piston crevice case; however, during the blowdown period the unburned gas that comes out from the head gasket crevice quickly escapes from the combustion chamber, a deduction which is supported by the fast-response FID signals shown in Figure 4.12. The fast-response FID signals in these two figures show that the piston crevice changes affect the exhaust hydrocarbon concentration only at the end of the exhaust stroke, while the head gasket crevice changes affect the exhaust hydrocarbon
concentration during the exhaust blowdown and during the early part of the exhaust stroke.

As discussed above, the crevice component of the engine-out hydrocarbons mainly comprises the outflow from the piston crevice during the expansion process, and the head gasket outflow during the blowdown process. The hydrocarbon emissions contribution of the former changes only modestly with changes in the piston crevice size because only a thin layer (of the order of the thermal boundary layer thickness) of the crevice gas adjacent to the liner survives oxidation. The exhaust hydrocarbon emissions contribution of the latter is proportional to the head gasket crevice volume. As a result, the relative sensitivity of the engine-out hydrocarbon emissions to the piston crevice volume changes is much less than unity (it is about 0.2), while the sensitivity to head gasket crevice volume changes is on the order of unity.

To investigate the effect of the location of crevice volume on the exhaust hydrocarbon emissions, one-third of the thin inner head gasket at the intake valve side was removed to introduce a moveable head gasket crevice. This crevice was then rotated to the exhaust valve side. The spark plug was located on the center line between the intake valve and the exhaust valve. The amount of the unburned gas entering into the crevices should be the same in both experiments. Figure 4.7 shows that the exhaust hydrocarbon emissions with the crevice volume located at the exhaust valve side were on or above the fitted line of the head gasket modifications data. The cycle-resolved exhaust hydrocarbon concentration for the one-third of the inner head gasket crevice cases is shown in Figure 4.13. The fast-response FID data show that the exhaust hydrocarbon concentration for the case with the crevice volume located on the exhaust valve side is higher than that for the crevice volume located the intake valve side only during the blowdown period; this is because all of the unburned mixture trapped in the head gasket crevice comes out during the expansion and blowdown process. The difference in these two cases was only in the geometric location of the head gasket crevice, which affected the crevice gas retention.
time in the cylinder. The crevice gas which comes from further locations to the exhaust valve has more time to mix with the burned gas and to oxidize, or some of it may be trapped in the chamber. These experiments indicate that the head gasket crevice regions that are closer to the exhaust valve have a larger effect on the exhaust hydrocarbon emissions.

Figure 4.14 shows a schematic of how all these crevices affect the exhaust hydrocarbon emissions at steady state. Open symbols in Figure 4.14 indicate the piston modification cases and solid symbols are for the head gasket modification cases. The x-axis is the ratio of the estimated maximum percentage of unburned fuel in the crevices to total fuel in the combustion chamber. For the first grooved piston case (triangular symbol), 13 percent of the maximum amount of unburned fuel in the crevices is due to miscellaneous head crevices (spark plug, pressure transducer, and extra plugs). If the miscellaneous head crevices are removed, a 20 percent reduction in the exhaust hydrocarbon emissions would be expected, based on the slope of the head gasket modification results (solid line in Figure 4.14). To check the background level of the exhaust hydrocarbon emissions, the engine was operated with hydrogen as fuel. The exhaust hydrocarbon emissions level was about 250 ppm C\(_1\), which was attributed to the contribution from the lubrication oil that was vaporized in the cylinder or that leaked through the valve guide of the exhaust valve. This background emissions level, when converted to an emissions index based on the propane fuel flow, amounts to 0.16 percent of the fuel. When this background is taken out, the exhaust hydrocarbon emissions due to the piston crevice only (in configuration B) would lower the exhaust HC emissions index from H to point G in Figure 4.14. From point G, using the sensitivity of the exhaust HC emissions to the piston crevice volume change, the exhaust HC emissions index can be extrapolated (along the dot-dash line in Figure 4.14) to zero piston crevice volume. The intercept value on the exhaust HC emissions index scale is 0.33 percent. It is perplexing that extrapolating the data to zero crevice volume leads to a positive finite exhaust HC
emissions index of 0.33 percent of the fuel under condition where the other hydrocarbon emissions mechanisms have been either eliminated or minimized. For reference, a dotted line through the origin of slope one is also drawn in Figure 4.14. This line represents the relationship between the crevice source and the exhaust hydrocarbon emissions if all the hydrocarbons from the source are transported out of the engine without being oxidized or trapped within the cylinder. Three possible reasons for a non-zero intercept can be proposed: (1) Other sources may be important; e.g. wall quenching, oil layer and carbon deposits absorption/desorption effects. (2) Other crevices in addition to those included may be significant; e.g. the piston crevice volumes below the first ring were excluded from the estimated total crevices, and the only spark plug crevice counted was the threads. (3) The exhaust hydrocarbon emissions behave differently as the piston crevice volume decreases (the most plausible reason). The dotted line and dashed line may be the asymptotes of the exhaust HC emissions index in Figure 4.14.

The previous discussion may be used to explain the nature of the exhaust hydrocarbon emissions behavior as the crevice volume is extrapolated to 'zero'. When the top-land crevice is reduced, the engine-out exhaust HC emissions would decrease modestly, following the dot-dash line of Figure 4.14 because of the self-regulating mechanism discussed above. Although we do not have exhaust hydrocarbon emissions data with small top-land crevices, it is conceivable that when the crevice outflow is small enough so that it is entirely within and protected by the cold wall boundary layer, then it would not oxidize, and the scaling of the engine-out hydrocarbon emissions would follow the dotted line (of slope 1) in Figure 4.14 with a transition region in between. This explanation will be also discussed further in Chapter 5.
4-2 Cold-Start Experiments

In order to quantify changes in the exhaust hydrocarbon emissions during the engine warm-up period, a set of cold-start experiments was conducted. Typical exhaust hydrocarbon emissions during the engine warm-up period are shown in Figures 4.15 and 4.16. Each data point is a mass averaged mean hydrocarbon emissions per cycle, obtained as described in Section 3-1. As the engine starts firing, the exhaust hydrocarbon emissions quickly decrease and reach an approximate steady state in ~ 100 seconds. This result indicates that the engine component temperatures and the oxidation levels in the cylinder and within the exhaust port of the unburned mixture from the crevices come close to the steady-state levels within this hundred second period. The exhaust hydrocarbon emissions for 1600 rpm WOT at steady state fluctuates by approximately 100 ppm C\textsubscript{1}. The cyclic variations of the exhaust hydrocarbon concentration for 1600 rpm with an inlet pressure of 0.4 bar are larger than those for the 1600 rpm WOT case. The steady-state exhaust hydrocarbon emissions were obtained by averaging the steady-state data. To determine the initial exhaust hydrocarbon emissions, the ratio \([\text{HC}(t) - \text{HC}(\infty)]/\text{HC}(\infty)\) was plotted on the log scale in Figures 4.15 and 4.16. The initial hydrocarbon emissions level and the time constant of the change in exhaust hydrocarbon emissions during engine warm-up were calculated from the fitted curve, as explained in Section 3-5. Experimental results from the cold-start experiments show that the exhaust hydrocarbon concentration of several cycles after firing is very high due to misfiring of the first or second injection cycle or poor combustion, as will be discussed later.

Figures 4.17 through 4.21 show the exhaust hydrocarbon emissions index of the cold and warmed-up engine for the piston top-land modifications. The exhaust HC emissions index is the ratio of the HC rate out from the engine to fuel flow into the engine, expressed as a percentage. The maximum HC-in-crevice index (x-axis) was obtained from max. in-cylinder pressure, crevice volume, air/fuel ratio, and residual fraction, again as
explained in Section 3-3. From the calculation of the maximum HC-in-crevice index of the original piston case, 4.5 to 5.5 percent of the fuel in the cylinder escapes primary combustion process at steady state, and 6 to 9.5 percent of fuel escapes for the cold engine. The exhaust hydrocarbon emissions of the cold engine with the piston top-land modifications have the same trend of these of the warmed-up steady-state engine. However, the fitted curve shows that the exhaust hydrocarbon emissions of the cold engine are more sensitive to changes in the piston crevice volume than those of the warmed-up engine. This observation suggests that the self-regulating mechanism discussed in Section 4-2 also applies to the cold engine - i.e. that only a thin layer of the hydrocarbons adjacent to the liner survives oxidation during the expansion process. The environment for oxidation of the crevice gas in the expansion process during warm-up is not that different from the steady-state environment because the in-cylinder core gas temperature should come close to its steady-state value within a few cycles. In Figure 4.17, the maximum HC-in-crevice index of the warmed-up first grooved piston case is the same as that of the cold engine of the original piston case. However, the exhaust HC emissions index of the cold engine of the original piston is substantially higher (by 43 percent) than that of the warmed-up first grooved piston case, which is a consistent result at the different engine speeds and loads tested. The slope of the exhaust HC emissions index from the cold engine to the warmed-up engine changes with the piston modifications because the increase in the maximum unburned mixture trapped in the crevice of the warmed-up engine is less than that of the cold engine, which is due to the crevice gas density difference between the warmed-up engine and the cold engine.

The piston crevice mechanisms contributing to the change of engine-out hydrocarbon emissions during engine warm-up are summarized in Figure 4.22. From the values for the cold engine (the solid round symbol — the values plotted in the figure are for piston configuration A; different values represent repeated experiments at a range of equivalence ratios), the decrease in the exhaust hydrocarbon emissions level to the
warmed-up values (open round symbol) may be broken down into two parts. The first part is due to the change in the piston crevice volume and crevice gas density undergo in the warm-up process. Since the change in volume may be calculated from a model of how engine component temperatures change with time [21], the contribution due to this part may be estimated using the slope of the cold piston crevice volume change data. The remaining part is attributed to differences in the exhaust port oxidation. In this manner, the piston crevice effect contributes to ~ 20 percent of reduction of the exhaust hydrocarbon emissions from cold to warmed-up engine condition, and the change in port oxidation level contributes the remaining 80 percent.

The exhaust hydrocarbon emissions index of the cold and warmed-up engine for the head gasket modifications are shown in Figures 4.23 through 4.26. The sensitivity of the exhaust hydrocarbon emissions of the cold engine to the head gasket crevice size is higher than that of the warmed-up engine. The sensitivities of the exhaust hydrocarbon emissions to changes in the head gasket crevice volume decrease as the engine speed and load increase. From Figure 4.23, the slope for the warmed-up engine is 0.21, which means that only 21 percent of the increase in the maximum unburned mixture in the crevices comes out the tail pipe, and the rest is oxidized in the cylinder or in the exhaust port or is left in the cylinder.

Based on the cold-start experiments, the decrease in the exhaust hydrocarbon emissions, cold to warmed-up engine, and the time constant of this exhaust hydrocarbon emissions decrease as a function of the engine speed and load during the engine warm-up period, before the piston was modified, are quantified in Figures 4.27 and 4.28. Figure 4.27 shows that the amount of decrease in the exhaust hydrocarbons during the engine warm-up period increases as the engine power (thermal loading) increases. When the engine power increases, the piston crevice volume and crevice gas density decrease, and the exhaust port (exhaust valve and port) temperature increases. The former affects the amount of unburned mixture trapped in the crevices, and the latter affects the oxidation

- 64 -
levels of the unburned hydrocarbons in the exhaust port; both effects act to decrease the exhaust hydrocarbon emissions. Based on the steady-state experimental piston modification results, the effects of the piston crevice volume change and the decrease in crevice gas density during warm-up on the exhaust hydrocarbon emissions are modest. Thus, the change in the exhaust hydrocarbon emissions during warm-up is mainly due to the port oxidation level change as the port temperature stabilizes. The region in the immediate vicinity outside the cylinder exit is especially important because this is where the hydrocarbon rich vortex, scraped up by the piston during the exhaust stroke, resides in between exhaust flow events from one cycle to the next (Figure 4.10).

Figure 4.28 shows the exhaust hydrocarbon emissions reduction time constant (1/e time) vs. the engine power. The warm-up time constants are in the range of 10 to 30 seconds and are inversely proportional to the engine power. Also, Figure 4.28 shows the 1/e time constants of the exhaust gas temperature which was measured 1 cm from the exhaust valve, and the exhaust valve surface temperature which was measured in a similar Ricardo Hydra engine [32]. These time constants are in agreement with the exhaust hydrocarbon emissions time constants, which supports the above discussion.

Cycle-resolved exhaust hydrocarbon concentration at different times is shown in Figure 4.29. The characteristics of the fast-response FID HC signal were explained in Section 4-2. As the engine warms up, the magnitude of the first peak is decreased by 55 percent and occurs earlier. The large reduction of the first peak is mainly due to a change in the exhaust port oxidation level, which is dependent on the exhaust valve temperature because the part of vortex that is pushed out into the exhaust port by the piston stays near the exhaust valve until the exhaust valve opens again. The lower level of the exhaust hydrocarbon concentration at the major exhaust stroke is decreased by 20 to 50 percent. The reason for earlier arrival of the fast-response FID signal after the engine warm-up period is that the exhaust valve opening is advanced due to elongation of the exhaust valve stem.
Figure 4.30 shows the peak mass fraction burned during the engine warm-up period. Because of larger heat transfer to the cold walls and characteristic change of back flow from the combustion chamber to the intake port, the peak mass fraction burned several cycles after firing is lower than in the normal combustion cycles. This may explain why the exhaust hydrocarbon emissions of early combustion cycles were relatively higher. Based on the peak mass fraction burned, the combustion becomes normal after about 20 cycles. The variation of peak mass fraction burned for the 1600 rpm inlet pressure of 0.4 bar case (Figure 4.30) is larger than those for wide open throttle cases, which is one of the reasons for the larger cyclic variation of the exhaust hydrocarbon emissions. Typical mass fraction burned profiles for a cold engine and a warmed-up engine are shown in Figure 4.31. The mass fraction burned for the cold engine was averaged over five consecutive cycles after the first firing cycle. Burn angles of 0-2%, 0-10%, 0-50%, 0-90%, and 10-90% for the cold engine and warmed-up engine with speed and load are summarized in Table 4.1. The 0-2% burned angle for the cold engine is substantially higher than that of the warmed-up engine, but for the major combustion period (i.e. 10-90% burn angle) the changes during warm-up are smaller than the changes in 0-2% burn angle changes, which represents the early flame development. Possible reasons for the larger 0-2% burn angle of the cold engine are larger heat transfer from the flame kernel to the spark plug and low mixture temperature at ignition.

4-4 Different Fast-Response FID Sampling Locations

To investigate the non-uniformity of the hydrocarbon concentration at the exhaust port, the fast-response FID sampling location was moved to 1 cm from the exhaust valve, as shown in Figure 2.8. Originally the FID sampling location was 15 cm from the exhaust valve. Two locations at the 1 cm distance from the exhaust valve were used. One was located inside the exhaust valve seat exit jet stream (I), and the other was located on the
exhaust port wall (J). Figure 4.32 shows the exhaust hydrocarbon concentration profiles inside the jet stream. The dashed line indicates the steady-state mean hydrocarbon concentration measured by the conventional FID. As the exhaust valve opens, the exhaust hydrocarbon concentration quickly decreases to zero. The level of the exhaust hydrocarbon concentration before the exhaust valve opening decreases from 7500 to 4500 ppm C1. The exhaust hydrocarbon concentration is almost zero during most of the exhaust stroke. This indicates that large volumes of burned gases do not mix with the unburned hydrocarbons that are laid on the walls from the piston crevice, even though the flow at the exhaust valve is highly turbulent. At the end of the exhaust stroke, the exhaust hydrocarbon concentration steeply increases due to the piston motion during the exhaust stroke which scrapes the unburned gases off the cylinder wall and pushes them toward the top of the cylinder. As the exhaust hydrocarbon profiles are almost flat before the exhaust valve opening, there is no valve leakage.

The exhaust hydrocarbon concentration on the port wall is shown in Figure 4.33. The level of the exhaust hydrocarbon concentration is higher than the mean exhaust hydrocarbon concentration at steady state, which confirms that the unburned hydrocarbons sitting on the walls do not mix with the burned gas at the exhaust valve. The level of the exhaust hydrocarbon concentration is almost constant during the early portion of the exhaust stroke and gradually increases at the end of the exhaust stroke. The exhaust hydrocarbon concentration inside the jet stream and on the port wall is the same before the exhaust valve opening.

Figure 4.34 shows the exhaust hydrocarbon concentration during the steady state for the different locations of one-third of the head gasket crevice. The fast-response FID sampling probe is at the 1 cm location on the port wall. During the blowdown and early exhaust stroke, exhaust hydrocarbon concentration for the head gasket crevice located at the exhaust valve side is higher than those for the head gasket crevice located at the intake valve side. Based on these experimental results, there is non-uniformity of the mixture
near the exhaust valve, and the fast-response FID measurement of the exhaust hydrocarbon emissions which are sampled near the exhaust valve do not represent the mean hydrocarbon concentration.

4-5 Liquid Fuel Effect on the Exhaust Hydrocarbon Emissions

Two different liquid fuel preparation techniques were employed to quantify the effects of the liquid fuel in crevices/on the walls on the exhaust hydrocarbon emissions during the engine warm-up period and at steady state. One is a conventional gasoline port fuel injector which directs fuel at the back of the intake valve and injects when the intake valve is closed. The other is the pre-vaporizing gasoline injector that was described in Section 2-1-2. For the conventional injector case, additional propane was introduced for the first 4 to 10 seconds of the operation. This procedure is necessary because if only the stoichiometric amount of gasoline were introduced into the intake port, it would take more than 50 cycles for the engine to start firing [1]. Figure 4.35 shows the comparison of the exhaust hydrocarbon emissions of the conventional injector with the propane-assisted conventional injector. For the first four seconds of operation, additional propane was used (for an overall value of \( \lambda = 0.5 \)). For the conventional injector case, it takes about 120 cycles to fire. After the engine begins to fire, the exhaust hydrocarbon emissions of the conventional injector are higher than those of the pre-vaporizing gasoline injector by a factor of two. Then the difference of the exhaust hydrocarbon emissions decreases gradually. This experimental result shows that if only liquid gasoline in the stoichiometric amount were used, a large amount of the liquid fuel would accumulate in the intake port due to misfiring and contribute to higher exhaust hydrocarbon emissions during early warm-up.

Figures 4.36 and 4.37 show the steady-state exhaust hydrocarbon emissions of the pre-vaporizing gasoline injector and the conventional injector vs. lambda. The steady-
state exhaust hydrocarbon emissions of the conventional injector case are about 10 percent higher than those of the pre-vaporizing gasoline injector between $\lambda = 0.9$ and $\lambda = 1.1$, for $\lambda = 1.2$, the exhaust hydrocarbon emissions of the conventional injector case are only 5 percent higher than in the pre-vaporizing gasoline injector case. Figure 4.37 for the 1600 rpm case shows the same trend. These results suggest that some of liquid fuel enters the cylinder, and part of this liquid fuel escapes primary combustion, thereby contributing to the exhaust hydrocarbon emissions.

The cycle-by-cycle exhaust hydrocarbon emissions during the engine warm-up period for the conventional injector (propane-assisted) and the pre-vaporizing gasoline injector are shown in Figures 4.38 and 4.39. In these experiments, the fast-response FID data and the cylinder pressure data were recorded with a DSP Technology Transiac 4012A controller and model 2825 12-bit digitizer installed in a Kinetic Systems Camac mini-crate. The maximum local memory of the data acquisition system allowed the collection of only up to 700 consecutive cycles. Thus, the blank periods in the data trace were due to data transfer from the data acquisition system to a VAX 11/750 computer for storage and analysis. For the pre-vaporizing gasoline injector, the exhaust hydrocarbon emissions decrease quickly to the steady-state level and then gradually decrease a little as the engine warms up.

Figure 4.40 shows the ratio $[\text{HC}(t) - \text{HC}(\infty)]/\text{HC}(\infty)$ of the pre-vaporizing gasoline injector case on a log scale against time. The data show two different slopes. The quick decrease in the exhaust hydrocarbon emissions during 100 seconds after firing is due to the effects of port/in-cylinder oxidation level change and the crevice volume and crevice gas density change. The other gradual decrease is most likely due to the oil layer effect on the exhaust hydrocarbon emissions. The amount of fuel absorbed into the engine oil layer on the cylinder wall is strongly dependent on the oil temperature, which depends on the cylinder liner temperature. Even though the piston and the exhaust port temperatures quickly increase, the cylinder liner temperature slowly increases during the
engine warm-up period. Thus, the amount of fuel absorbed into the oil layer decreases as the cylinder liner temperature increases; this fuel is desorbed into the combustion chamber during the expansion and the exhaust stroke and contributes to the exhaust hydrocarbon emissions.

For the propane-assisted injector case, the exhaust hydrocarbon emissions are higher than those in the pre-vaporizing gasoline injector case by a factor of two. As the engine warms up, the difference of the exhaust hydrocarbon emissions between the propane-assisted case and pre-vaporizing gasoline injector case decreases gradually. This indicates that the engine coolant temperature plays a major role in the liquid fuel effects on the exhaust hydrocarbon emissions, because the intake valve and port temperature quickly increase and reach steady state. During the engine warm-up period, the decrease in the exhaust hydrocarbon emissions of the conventional injector are due to the liquid fuel effects. These are thought to be liquid fuel in the piston crevice or on the combustion chamber walls, and a change in the oil layer absorption/desorption phenomenon (which strongly depends on the oil layer temperature).

The cyclic variations of the exhaust hydrocarbon emissions of the conventional injector case are much higher than those of the pre-vaporizing injector case, and decrease as the engine warms up. Based on these experiments, the liquid fuel effects on the exhaust hydrocarbon emissions are larger during the early warm-up period and decrease as the engine warms up.
Table 4.1 Steady-state exhaust hydrocarbon emissions with different crevice volumes at 900 rpm, intake pressure = 1.0 bar

<table>
<thead>
<tr>
<th>λ</th>
<th>first ring moved-up</th>
<th>original piston</th>
<th>first grooved piston</th>
<th>second grooved piston</th>
<th>1/3 head crevice (intake valve side)</th>
<th>1/3 head crevice (exhaust valve side)</th>
<th>1/2 head gasket removed</th>
<th>full head gasket removed</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.9</td>
<td>1700 **</td>
<td>1780</td>
<td>1790</td>
<td>1830</td>
<td>1810</td>
<td>1990</td>
<td>2140</td>
<td>2420</td>
</tr>
<tr>
<td>1.0</td>
<td>1300</td>
<td>1400</td>
<td>1410</td>
<td>1510</td>
<td>1530</td>
<td>1645</td>
<td>1840</td>
<td>2080</td>
</tr>
<tr>
<td>1.1</td>
<td>1070</td>
<td>1120</td>
<td>1180</td>
<td>1280</td>
<td>1280</td>
<td>1450</td>
<td>1670</td>
<td>1760</td>
</tr>
<tr>
<td>1.2</td>
<td>1020</td>
<td>1080</td>
<td>1170</td>
<td>1330</td>
<td>1330</td>
<td>1345</td>
<td>1600</td>
<td>1720</td>
</tr>
<tr>
<td>total crevice*</td>
<td>0.69***</td>
<td>0.85</td>
<td>1.25</td>
<td>1.65</td>
<td>1.41</td>
<td>1.41</td>
<td>1.51</td>
<td>1.73</td>
</tr>
</tbody>
</table>

*: steady-state condition  **: ppm C<sub>1</sub>  ***: cm<sup>3</sup>
Table 4.2  Steady-state exhaust hydrocarbon emissions with different crevice volumes at 1600 rpm, intake pressure = 0.4 bar

<table>
<thead>
<tr>
<th>( \lambda )</th>
<th>piston modification</th>
<th>head gasket modification</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>first ring moved-up</td>
<td>original piston</td>
</tr>
<tr>
<td>0.9</td>
<td>1650**</td>
<td>1720</td>
</tr>
<tr>
<td>1.0</td>
<td>1110</td>
<td>1140</td>
</tr>
<tr>
<td>1.1</td>
<td>870</td>
<td>860</td>
</tr>
<tr>
<td>1.2</td>
<td>910</td>
<td>960</td>
</tr>
<tr>
<td>total crevice*</td>
<td>0.68***</td>
<td>0.88</td>
</tr>
</tbody>
</table>

*: steady-state condition  **: ppm C\(_1\)  ***: cm\(^3\)
Table 4.3 Steady-state exhaust hydrocarbon emissions with different crevice volumes at 1600 rpm, intake pressure = 0.7 bar

<table>
<thead>
<tr>
<th>λ</th>
<th>first ring moved-up</th>
<th>original piston</th>
<th>first grooved piston</th>
<th>second grooved piston</th>
<th>1/2 head gasket removed</th>
<th>full head gasket removed</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.9</td>
<td>1380**</td>
<td>1400</td>
<td>1430</td>
<td>1450</td>
<td>1860</td>
<td>2040</td>
</tr>
<tr>
<td>1.0</td>
<td>1000</td>
<td>1070</td>
<td>1170</td>
<td>1160</td>
<td>1500</td>
<td>1650</td>
</tr>
<tr>
<td>1.1</td>
<td>800</td>
<td>860</td>
<td>870</td>
<td>920</td>
<td>1290</td>
<td>1400</td>
</tr>
<tr>
<td>1.2</td>
<td>750</td>
<td>900</td>
<td>910</td>
<td>940</td>
<td>1250</td>
<td>1350</td>
</tr>
<tr>
<td>total crevice*</td>
<td>0.67***</td>
<td>0.83</td>
<td>1.23</td>
<td>1.63</td>
<td>1.52</td>
<td>1.72</td>
</tr>
</tbody>
</table>

*: steady-state condition  **: ppm C<sub>1</sub>  ***: cm<sup>3</sup>
Table 4.4  Steady-state exhaust hydrocarbon emissions with different crevice volumes at 1600 rpm, intake pressure = 1.0 bar

<table>
<thead>
<tr>
<th>$\lambda$</th>
<th>piston modification</th>
<th>head gasket modification</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>first ring moved-up</td>
<td>original piston</td>
</tr>
<tr>
<td>0.9</td>
<td>1360 **</td>
<td>1300</td>
</tr>
<tr>
<td>1.0</td>
<td>1010</td>
<td>960</td>
</tr>
<tr>
<td>1.1</td>
<td>760</td>
<td>780</td>
</tr>
<tr>
<td>1.2</td>
<td>700</td>
<td>750</td>
</tr>
<tr>
<td>total crevice*</td>
<td>0.65***</td>
<td>0.78</td>
</tr>
</tbody>
</table>

*: steady-state condition  **: ppm C$_1$  ***: cm$^3$
### Table 4.5  Burn durations

<table>
<thead>
<tr>
<th>Burn duration</th>
<th>900 rpm, WOT</th>
<th>1600 rpm, 0.4 bar</th>
<th>1600 rpm, WOT</th>
<th>2500 rpm, WOT</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>cold/warmed-up</td>
<td>cold/warmed-up</td>
<td>cold/warmed-up</td>
<td>cold/warmed-up</td>
</tr>
<tr>
<td>0 - 2 %</td>
<td>15.6 / 10.4</td>
<td>24.8 / 21.2</td>
<td>15.6 / 11.2</td>
<td>20.4 / 15.6</td>
</tr>
<tr>
<td>0 - 10 %</td>
<td>24.0 / 17.6</td>
<td>33.2 / 30.4</td>
<td>22.0 / 18.0</td>
<td>27.2 / 22.4</td>
</tr>
<tr>
<td>0 - 50 %</td>
<td>38.4 / 29.6</td>
<td>44.0 / 40.8</td>
<td>33.6 / 28.4</td>
<td>39.2 / 32.8</td>
</tr>
<tr>
<td>0 - 90 %</td>
<td>51.6 / 43.2</td>
<td>56.7 / 52.8</td>
<td>44.8 / 39.2</td>
<td>52.0 / 43.2</td>
</tr>
<tr>
<td>10 - 90 %</td>
<td>27.6 / 25.6</td>
<td>23.6 / 22.4</td>
<td>22.8 / 21.2</td>
<td>24.8 / 20.8</td>
</tr>
</tbody>
</table>

### Table 4.6  Crevice outflows and their dispositions

<table>
<thead>
<tr>
<th>Piston Crevice</th>
<th>Head Gasket Crevice</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outflow in Expansion Process</td>
<td>Mostly oxidized, except for the layer adjacent to the combustion chamber wall; Exhaust partially as portion of the scraped-up vortex</td>
</tr>
<tr>
<td></td>
<td>Mostly oxidized</td>
</tr>
<tr>
<td>Outflow in Blowdown Process</td>
<td>Not extensively oxidized; trapped at the center of scraped-up vortex; not exhausted</td>
</tr>
<tr>
<td></td>
<td>Not extensively oxidized, mostly exhausted</td>
</tr>
</tbody>
</table>
Figure 4.1.a Effect of piston crevice and head gasket crevice changes on steady-state exhaust HC emissions at 900 rpm, intake pressure = 1.0 bar.

Figure 4.1.b Effect of piston crevice and head gasket crevice changes on steady-state exhaust HC emissions at 1600 rpm, intake pressure = 0.4 bar.
Figure 4.2 Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions at 900 rpm, intake pressure = 1.0 bar.

Figure 4.3 Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions at 1600 rpm, intake pressure = 0.4 bar.
Figure 4.4 Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions at 1600 rpm, intake pressure = 0.7 bar.

Figure 4.5 Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions at 1600 rpm, intake pressure = 1.0 bar.
Figure 4.6  Effect of piston crevice volume changes on steady-state exhaust hydrocarbon emissions.

Figure 4.7  Effect of head gasket crevice volume changes on steady-state exhaust hydrocarbon emissions.
Figure 4.8.a  Nature of piston crevice flow: out flow velocity with respect to the piston.

Figure 4.8.b  Nature of piston crevice flow: out flow velocity with respect to the liner.
Figure 4.8.c  Nature of piston crevice flow: distribution of crevice flow mass along the liner.

Figure 4.8.d  Nature of piston crevice flow: core gas temperature.
Figure 4.9  Crevice out flow geometry with respect to the piston.

Figure 4.10  Formation of the scraped-up vortex by the piston (a) and scraped-up vortex partially pushed out of the cylinder.
Figure 4.11  Comparison of the cycle-resolved exhaust hydrocarbon concentration of the original piston with the 2nd grooved piston.

Figure 4.12  Comparison of the cycle-resolved exhaust hydrocarbon concentration of the 1st grooved piston with removed the full inner thin head gasket case.
Figure 4.13  The cycle-resolved exhaust hydrocarbon concentration for the one-third of the thin inner head gasket crevice case.

Figure 4.14  Extrapolation to zero crevice volume.
Figure 4.15 Exhaust hydrocarbon emissions during the engine warm-up period at 1600 rpm, intake pressure = 1.0 bar.
Figure 4.16 Exhaust hydrocarbon emissions during the engine warm-up period at 1600 rpm, intake pressure = 0.4 bar.
Figure 4.17 Exhaust HC emissions index of the cold and warmed-up engine for the piston top-land modifications at 900 rpm, intake pressure = 1.0 bar.

Figure 4.18 Exhaust HC emissions index of the cold and warmed-up engine for the piston top-land modifications at 1600 rpm, intake pressure = 0.4 bar.
Figure 4.19  Exhaust HC emissions index of the cold and warmed-up engine for the piston top-land modifications at 1600 rpm, intake pressure = 0.7 bar.

Figure 4.20  Exhaust HC emissions index of the cold and warmed-up engine for the piston top-land modifications at >600 rpm, intake pressure = 1.0 bar.
Figure 4.21  Exhaust HC emissions index of the cold and warmed-up engine for the piston top-land modifications at 2500 rpm, intake pressure = 1.0 bar.

Figure 4.22  Schematic of the effect of the piston crevice and oxidation level change on the exhaust hydrocarbon emissions during the engine warm-up period.
Figure 4.23  Exhaust HC emissions index of the cold and warmed-up engine for the head gasket modifications at 900 rpm, intake pressure = 1.0 bar.

Figure 4.24  Exhaust HC emissions index of the cold and warmed-up engine for the head gasket modifications at 1600 rpm, intake pressure = 0.4 bar.
Figure 4.25  Exhaust HC emissions index of the cold and warmed-up engine for the head gasket modifications at 1600 rpm, intake pressure = 1.0 bar.

Figure 4.26  Exhaust HC emissions index of the cold and warmed-up engine for the head gasket modifications at 2500 rpm, intake pressure = 1.0 bar.
Figure 4.27  Decrease in the exhaust HC emissions of the original piston during the engine warm-up period.

Figure 4.28  Time constant of the exhaust HC emissions, the exhaust gas temperature, and the exhaust valve temperature during the engine warm-up period.
Figure 4.29  Cycled-resolved exhaust hydrocarbon concentration at different times.
Figure 4.30  Peak mass fraction burned during the engine warm-up period.
Figure 4.31  Mass fraction burned profile for a cold engine and warmed-up engine.
Figure 4.32  Cycle-resolved exhaust hydrocarbon concentration profiles inside the jet stream at 1 cm from the exhaust valve.

Figure 4.33  Cycle-resolved exhaust hydrocarbon concentration profiles on the port wall at 1 cm from the exhaust valve.
Figure 4.34  Cycle-resolved exhaust hydrocarbon concentration profiles for the different locations of one-third of the head gasket crevice at 1 c·n on the port wall.

Figure 4.35  Comparison of the exhaust hydrocarbon emissions of the conventional injector with the propane-assisted conventional injector.
Figure 4.36  Steady state exhaust HC emissions of the pre-vaporizing gasoline injector and the conventional injector vs. lambda at 900 rpm, WOT.

Figure 4.37  Steady state exhaust HC emissions of the pre-vaporizing gasoline injector and the conventional injector vs. lambda at 1600 rpm, WOT.
Figure 4.38  Exhaust HC emissions for the conventional injector and the pre-vaporizing injector during the engine warm-up period at 900 rpm, WOT.

Figure 4.39  Exhaust HC emissions for the conventional injector and the pre-vaporizing injector during the engine warm-up period at 1600 rpm, WOT.
Figure 4.40  Exhaust HC emissions of the pre-vaporizing gasoline injector during the engine warm-up period.
CHAPTER 5: MODELING OF PISTON CREVICE GAS TRANSPORT AND OXIDATION

5-1 Introduction

Piston crevices are one of the major sources of exhaust hydrocarbon emissions. During compression and combustion, while in-cylinder pressure continues to rise, in-cylinder gas continues to flow into the crevice volumes. The fraction of the total cylinder charge trapped in the piston crevice volume is about 5 to 9 percent at the time of peak cylinder pressure, and the fuel in this mixture escapes the primary combustion process (see Chapter 4). As the cylinder pressure decreases, most of the crevice gases flow back into the cylinder. Some of the piston crevice gases coming out early in the expansion stroke mix with burned gas and then oxidize. When the exhaust valve opens, some of the unburned and partially oxidized hydrocarbons come out with the blowdown exhaust gas. The piston then scrapes up the unburned hydrocarbons near the cylinder wall and pushes them up to the top of the cylinder during the exhaust stroke. Overall, about half of the unburned hydrocarbons formed by the source mechanisms will oxidize within the engine cylinder [24].

The experimental results developed in this thesis showed that when the piston crevice volume is increased substantially by grooving the piston top land, the exhaust hydrocarbon emissions proportionately increase by a much smaller amount. However, this does not necessarily suggest that the piston crevice volume is not a major source of the exhaust hydrocarbon emissions. As discussed already, a likely explanation is that the
extent of in-cylinder oxidation of the unburned hydrocarbons from the piston crevice increases as the piston crevice volume is increased.

To quantify the sensitivity of the amount of in-cylinder oxidation of the unburned hydrocarbons before the exhaust valve opening to the piston crevice volume size, a piston crevice gas flow model [22] was used to predict piston crevice gas flow into the combustion chamber during the expansion and blowdown period. A model with one-dimensional unsteady piston crevice gas transport into the burned gas and a one step kinetic mechanism for the oxidation of propane was developed.

5-2 Piston Crevice Gas Flow

To predict the piston crevice out flow into the cylinder during the expansion process, a model for piston-cylinder crevice flow [22] was employed. The model was coupled with a ring motion model, and flow between ring regions was assumed to be isothermal [22]. In-cylinder pressure data for input to the model were obtained from the experiments. Figures 5.1 and 5.2 show the crevice gas distribution along the liner during the expansion process calculated by the model for piston-cylinder crevice flow. The x-axis contains piston location from the top of the liner normalized by the stroke (86 mm). The spark plug is located 17 mm from the center of the cylinder head; thus, about 15 percent of the gas trapped in the piston crevice is burned gas, based on the flame geometry. This gas comes out first as the cylinder pressure decreases. The majority of the piston crevice gas flows out during the early expansion process. The piston crevice gas flow rates scale with the piston crevice size (piston top land+first ring region). Figure 5.3 shows the percent piston crevice gas out flow during the expansion process. Before the exhaust valve opens, more than 86 percent of the total crevice gas trapped in the piston crevice flows into the combustion chamber. Less than 14 percent of the maximum piston crevice gas comes into the cylinder during the blowdown period.
The crevice gas velocity with respect to the piston, as it comes out from the piston crevice, is shown in Figures 5.4 and 5.5, which were obtained from Figures 5.1 and 5.2. The relative gas velocity for the original piston case for 1600 rpm WOT is smaller than the mean piston speed (4.58 m/s), and for the 1600 rpm 0.4 bar inlet pressure case it is larger than the mean piston speed; this is mainly due to the difference of in-cylinder pressure between the WOT and intake pressure 0.4 bar cases. As the piston top-land crevice is increased, the relative gas velocity increases proportionally. Figures 5.6 and 5.7 show the absolute gas velocity of the piston crevice gases during the expansion process. The absolute gas velocity of the original piston case for 1600 rpm WOT is negative; thus the crevice gases move downward when they come out of the piston crevice. After coming out of the crevice, The piston crevice gases probably stay on the cylinder wall even though they flow out like a jet, because the piston gap size is on the order of 0.1 mm. If the crevice gases are adjacent to the cylinder wall without mixing with burned gas, the initial crevice gas thickness can be calculated from the crevice gas flow rate. The changes in the density of the crevice gases are on the of order of one during the expansion process because the cylinder pressure decreases by the same order of magnitude. Figures 5.8 and 5.9 show the initial crevice gas thickness based on the crevice gas flow (Figures 5.1 and 5.2) and in-cylinder pressure from the experiments. The horizontal dotted line in Figures 5.8 and 5.9 indicates the average gap between the piston and cylinder wall. The initial crevice gas thickness of the original piston case in Figures 5.8 and 5.9 is smaller than the piston crevice gap. Also, the initial crevice gas thickness is proportional to the piston crevice volume (piston top land + first ring region).

5-3 Modeling of Transport and Oxidation

The initial thicknesses of the crevice gas layers are calculated from the crevice gas distribution along the liner. The crevice gas which flows out the piston crevice is divided
into crank angle intervals for the modeling of transport. The number of segments of the crevice gas for 1600 rpm WOT is 51. Basic assumptions of the piston crevice gas transport are as follows:

1) There is no transport between each segment in the axial direction.

2) Each segment of the piston crevice gases is stationary in the axial direction and only moves toward the center of the chamber.

3) There is no turbulent transport.

4) Temperature of the cylinder wall is constant during the expansion process.

5) The core gas temperature during expansion is obtained from a thermodynamic cycle simulation [26].

6) A one-step kinetic mechanism for the oxidation of propane is employed.

The thicknesses of the thermal boundary layers are changing during the expansion stroke and affect the mixing and oxidation of the unburned hydrocarbons in the thermal boundary layers. The thicknesses of the thermal boundary layers on the cylinder liner of a spark-ignition engine have been measured throughout the complete operating cycle [33]. It was shown that the growth of the liner thermal boundary layer was correlated with an expression. The simple correlation is

\[ \delta_t = 0.6 \sqrt{\alpha \cdot t} \cdot \text{Re}^{0.2} \]

Where

- \( \text{Re} = \rho v x_o / \mu \), \( v = v_p (x_o / x) \)
- \( \alpha = \text{thermal diffusivity} \)
- \( t = \text{time} \)
- \( v_p = \text{piston velocity} \)
- \( x = \text{distance of top of piston from cylinder head} \)
- \( x_o = \text{distance of measurement location from cylinder head} \)

The thermal diffusivity is obtained from the calculated Prandtl number data [34].

\[ \text{Pr} = 0.05 + 4.2(\gamma - 1) - 6.7(\gamma - 1)^2 \]
The ratio of specific heat ($\gamma$) is approximated by a function of temperature [23].

$$\gamma = 1.3655 - 6.105 \times 10^{-5} T$$

A fitted correlation for the viscosity of hydrocarbon-air combustion products is employed [34]:

$$\mu = 3.3 \times 10^{-7} x T^{0.7} / 1.0256$$

$$\alpha = \frac{\mu}{(\rho Pr)}$$

This correlation is used to predict the growth of the thermal boundary layer for each segment of the piston crevice gases. All gas properties were evaluated at the mean of the core gas temperature and the cylinder wall temperature. Figure 5.10 shows the thermal boundary layer grows during the expansion process. The initial thermal boundary layer thickness is fixed at 1 mm in Figure 5.10, based on the measurement of the thickness of the thermal boundary layer by Lyford-Pike and Heywood [33]. The thermal boundary layers grow about 1 to 1.7 mm during the expansion process; and the thermal boundary layer for the intake pressure of 0.4 bar case is thicker than that of the WOT case.

A one-step overall kinetic mechanism for the oxidation of propane is used:

$$\frac{d[C_{3}H_{8}]}{dt} = -1.0 \times 10^{23} x[C_{3}H_{8}] [O_{2}] \exp\left(-\frac{-25000}{T}\right)$$

Where [ ] denotes concentration in moles per cubic centimeter, t is in seconds, T in Kelvins. This one-step oxidation model was obtained from the Chemkin calculation, and the hydrocarbon oxidation mechanism was developed by Dagaut [38].

Figure 5.11 shows how the time of 50 percent of propane consumption varies with temperature. If the gas temperature is above 1500 K, it takes less than 0.1 msec to burn
up 50 percent of the propane. Thus, gas temperature plays an important role of the hydrocarbon oxidation.

The one-dimensional unsteady mass conservation, species, and energy equation are:

\[
\frac{\partial p}{\partial t} + \frac{\partial (\rho u)}{\partial x} = 0
\]

\[
\frac{\partial Y_i}{\partial t} + u \frac{\partial Y_i}{\partial x} = \frac{1}{\rho} \frac{\partial}{\partial x} \left( D \frac{\partial Y_i}{\partial x} \right) + \frac{\omega_i}{\rho} \quad (i = 1, 2)
\]

\[
\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} - \frac{1}{\rho C_p} \frac{Dp}{Dt} = \frac{1}{\rho C_p} \frac{\partial}{\partial x} \left( k \frac{\partial T}{\partial x} \right) - \frac{1}{\rho C_p} \Delta h_r \omega
\]

Gilliland has proposed a semiempirical equation for the diffusion coefficient in gases:

\[
D_{12} = 0.04357x\left( \frac{1}{M_1} + \frac{1}{M_2} \right)^{1/2} \left( V_1^{1/3} + V_2^{1/3} \right)^2 \frac{T^{3/2}}{p}
\]

Where

- \( M \) = molecular weight
- \( V \) = molecular volume
- \( T \) = temperature (K)
- \( p \) = pressure (N/m²)

and specific heat \( (c_p) \) is given by

\[
c_p = 947.8 + 0.2493T \quad (J/kg \ K)
\]

Thermal diffusion coefficient \( (k) \) is obtained from the thermal diffusivity, gas density, and specific heat.

\[
k = \alpha \rho c_p
\]
One-dimensional unsteady equations are solved by the Crank-Nicolson method [35]. Cell size ($\Delta x$) is 0.01 mm and the number of cells per segment is 300. The time step ($\Delta t$) is $1.3 \times 10^{-6}$ sec. Figure 5.12 shows the core gas temperature during the expansion process for 1600 rpm WOT and for an inlet pressure of 0.4 bar. The core gas temperature for an inlet pressure of 0.4 bar is about 100 K lower than that of inlet pressure of 1.0 bar case, and the difference decreases a little during expansion. Figure 5.13 shows a typical initial temperature and unburned hydrocarbon concentration (propane) profile. From the experimental results of Lyford-Pike and Heywood [33], the initial thickness of the thermal boundary layers is about 1 mm. In this calculation, 1 mm of thermal boundary layer is added outside the piston crevice gas. The temperature profile of the added gas is obtained from the fitted curve of the experimental measurements of thermal boundary layer [36]. The temperature profile is given by

$$T(x) = T_{core} + \frac{(T_{core} - T_{wall})}{2.6} \log\left(\frac{x}{\delta_t}\right) \quad 10^{-2.6} < x < \delta_t$$

Where
- $T_{core}$ = core gas temperature (K)
- $T_{wall}$ = wall gas temperature (K)
- $\delta_t$ = thickness of thermal boundary layer
- $x$ = distance from wall

Concentration of unburned fuel in the piston crevice gas is calculated with a relative air/fuel ratio of 1.05 and the predicted residual fraction [25]. The unburned hydrocarbon concentration in the burned gas is assumed to be zero.

5-4 Calculation Results and Discussion

Figures 5.14 and 5.15 show the thermal boundary layer profiles and propane concentration profiles near the cylinder wall every four crank angle degrees. The initial
thickness of the crevice gas for the location of 1.9 cm from the top of the liner is 0.16 mm, and the piston crevice gas initially contacts with the core gas temperature of 2250 K. As the unburned hydrocarbons diffuse into the hot burned gas and heat diffuses into the piston crevice gas, the unburned hydrocarbons that mix with the hot burned gas start to oxidize and increase the temperature of the thermal boundary layer, which accelerates the oxidation of the unburned hydrocarbons. Due to the oxidation of propane, the temperature close to the wall reaches close to 2000 K in Figure 5.14, and only a small amount of propane survives due to the cold wall. The amount of remaining unburned hydrocarbons at the exhaust valve opening (EVO) point is a few percent of the initial unburned hydrocarbons. The unburned hydrocarbons exist between the wall and 0.4 mm from the wall. The unburned hydrocarbons more than 0.4 mm from the wall are oxidized. From Figure 5.11, the rate of oxidation is strongly dependent on gas temperature. In Figure 5.15, the oxidation of unburned hydrocarbons does not increase the thermal boundary layer temperature enough to accelerate the oxidation of propane because the rate of oxidation of the unburned hydrocarbons is small, and heat generation from the oxidation quickly diffuses to the wall. Therefore, about 50 percent of unburned hydrocarbons survives oxidation at the EVO point. As the core gas temperature decreases quickly during the expansion process, the unburned hydrocarbons reach 0.8 mm from the wall. Based on calculation results, the core gas temperature plays a major role in the unburned hydrocarbon oxidation, and the unburned hydrocarbons only exist very close to the wall.

Figures 5.16 and 5.18 show the initial hydrocarbons from the piston crevice during the expansion process; the initial amount of hydrocarbons is proportional to the piston crevice volume (piston top land+first ring region). The amount of gases trapped in the piston crevice is proportional to maximum in-cylinder pressure, crevice volume size, and crevice gas density. The fraction of the charge trapped in the crevices is almost the same for intake pressure of 0.4 bar and WOT case, which was shown in Chapter 4. Based on
the flame geometry and location of the spark plug, 15 percent of the gas trapped in piston crevice is burned gas. If the burned gas in the piston top land does not mix with the mixture of air, fuel, and residual gas, it will come out first as soon as the cylinder pressure decreases. The dotted line in Figures 5.16 and 5.18 indicates that 15 percent of the crevice gas that flows into the combustion chamber. The majority of the gas in the piston crevice flows into the cylinder during the early expansion process.

The calculations of piston crevice gas transport and oxidation were conducted until the exhaust valve opening because the gas motion in the cylinder dramatically changes after EVO. Figures 5.17 and 5.19 show the remaining unburned hydrocarbons at EVO after the calculation. In Figure 5.19, the unburned hydrocarbons that come out from the relative distance of 0.04 to 0.2 are almost completely oxidized, and the amount of unburned hydrocarbons left is not dependent on the initial unburned hydrocarbons. Between the relative distance of 0.2 and 0.4, the remaining unburned hydrocarbons for the first grooved piston case and the second grooved piston case are quite small compared with the original piston case. The initial thickness of piston crevice gas for the grooved cases are bigger than that of the original piston, and the temperature of the crevice gas for the thicker crevice gas layers is quickly increased; in addition, the diffusion coefficient strongly depends on temperature. Thus, the thicker initial crevice gas will start to oxidize early. After the relative distance of 0.4, the crevice gases mix with relatively lower temperature core gas, and the thermal boundary layer growth is faster (see Figure 5.10). The amount of oxidation is thus quite small, and the amount of unburned hydrocarbons left at EVO is almost proportional to the initial unburned hydrocarbons. For the intake pressure of 1.0 bar case, the unburned hydrocarbons that come out from the relative distance 0.04 to 0.36 are almost all oxidized. The amount of oxidation for intake pressure of 1.0 bar is larger than that for an intake pressure of 0.4 bar, mainly due to hotter core gas and thinner thermal boundary layers during the expansion process.
30 percent of the initial unburned hydrocarbons coming out from the piston crevice for the original piston case at 1600 rpm and intake pressure of 0.4 bar remain at the exhaust valve opening and 20 percent of the initial piston crevice gas for 1600 rpm WOT case survives at EVO. As the piston top-land crevice volume is increased, the amount of unburned hydrocarbons that survived the in-cylinder oxidation at EVO increases modestly. Figure 5.20 shows the sensitivity of the amount of unburned hydrocarbons left at EVO to the piston crevice volume size. The x-axis contains the steady-state piston crevice volume (piston top land+first ring region) normalized with the reference crevice volume. The reference crevice volume used is the steady-state piston crevice volume (piston top land+first ring region) of the first grooved piston. The y-axis contains the remaining unburned piston crevice hydrocarbons at EVO normalized with the remaining unburned hydrocarbons of the first grooved piston. The slope of the 1600 rpm WOT case is 0.29, and the slope of the 1600 rpm intake pressure of 0.4 bar case is 0.42.

Based on these experimental and calculation results, the amount of unburned hydrocarbons surviving oxidation within the cylinder modestly increases as the piston crevice volume increases in this piston crevice volume range. The sensitivity of the amount of piston crevice gas left at EVO to the piston crevice volume size increases as the intake pressure decreases.

To quantify the behavior of the oxidation level for smaller piston crevice volumes, artificially reduced piston crevice volume (piston top land+first ring region) cases were investigated. Figure 5.21 shows the initial piston crevice gas distributions along the liner for the original piston and three different piston crevice reduction cases. The crevice volume size of case 3 is only 10 percent of the crevice volume of the first grooved piston. Thermal boundary layer conditions are the same with the previous calculations. Figure 5.22 shows the remaining piston crevice hydrocarbons at EVO. As the piston crevice is decreased, a substantial amount of the unburned hydrocarbons coming out early from the crevice survives oxidation. The likely reason is that the thickness of the crevice gas layer
as it comes out from the piston crevice is very thin, and the temperature of this crevice gas increases slowly due to large heat transfer to the wall. Thus, the majority of these unburned hydrocarbons can survive the expansion process.

Figure 5.23 shows the sensitivity of the unburned hydrocarbons left at EVO to the piston crevice volume reductions. The solid symbol indicates the artificially reduced piston crevice volume case; the slope is 1.75, which is substantially different from the normal operating piston crevice volume size slope (0.29). The intercept on the y-axis that corresponds to no piston top land and first ring region is 0.175. This non-zero intercept could be due to the crevice volume beneath the first ring. Crevice gas from beneath the first ring comes out late, and only a small amount is oxidized. From calculation results, some 20 percent of unburned crevice gas left at EVO comes from beneath the first ring. As the piston crevice volume is decreased, two different trends are seen, as shown in Figure 5.23. This may explain why the piston crevice is one of the major sources of the exhaust hydrocarbon emissions, even though the exhaust hydrocarbon emissions modestly increase as the piston crevice volume increases in a certain crevice volume range.

To quantify the oxidation level in the cold engine and the sensitivity of the amount of remaining unburned hydrocarbons at EVO to crevice volume size, calculations for cold engine cases were conducted. Figure 5.24 shows the core gas temperatures of the cold engine during expansion; they are about 50 K lower than for the warmed-up engine. The difference of the core gas temperature between intake pressure of 1.0 bar and 0.4 bar is similar to the steady-state case. Initial piston crevice gas thicknesses for the cold engine are shown in Figure 5.25 and are thicker than in the warmed-up engine case. The dotted line indicates the gap between the piston and the cylinder wall at ambient temperature. The maximum amount of crevice gas for original piston, cold engine case is about 7 to 9 percent of the total charge gas (Chapter 4). 7 to 9 percent of fuel escapes the main combustion process, leading to higher exhaust emissions and a substantial loss of fuel conversion efficiency.
Figures 5.26 and 5.27 show the initial piston crevice gas distribution and remaining piston crevice gas after oxidation at EVO. 32 percent of the unburned hydrocarbons from the piston crevice for the original piston at intake pressure of 0.4 bar survive from the in-cylinder oxidation at EVO and 18 percent survive for the intake pressure of 1.0 bar case. The amount of remaining unburned hydrocarbons at EVO for the cold engine is 58 percent larger than for the warmed-up engine case for intake pressure of 0.4 bar, and 83 percent for intake pressure of 1.0 bar. Figure 5.28 shows the sensitivity of the oxidation level of the cold engine to crevice volume size. Calculation results show that the oxidation level of the cold engine is more sensitive than that of the warmed-up engine.

The oil layer absorption/desorption effect is not considered in the piston crevice gas transport model because propane is little absorbed and desorbed by the engine oil on the cylinder wall. However, if gasoline is used as fuel, the oil layer on the cylinder wall does absorb fuel during compression and combustion, and these hydrocarbons are desorbed out of the oil layer during the expansion and exhaust process. As the piston moves down during the expansion process, the piston crevice gas flows into the cylinder and stays adjacent to the cylinder wall. Thus the piston crevice gas covers the oil layer and may delay the desorption of the hydrocarbons in the oil layer. The unburned hydrocarbons from the oil layer have a higher possibility of avoiding oxidation than those from the piston crevice.

5-5 Construction of Hydrocarbon Emissions Mechanism

Some of the unburned hydrocarbons which escape the primary engine combustion process in the crevices may oxidize in cylinder and remain in the cylinder with the residual gas. A fraction of the unburned hydrocarbons which leave the cylinder through the exhaust valve will oxidize within the port, and the rest of the unburned hydrocarbons are emitted as engine-out emissions. From Chapter 4, a total of 5 to 9 percent of the fuel
inducted into the cylinder escapes the main combustion process, and engine-out emissions are 0.7 to 1.4 percent of the fuel, depending on the engine operating conditions. The above process may be represented as follows:

\[ HC_{\text{index}} = HC_{\text{crevice}} \cdot (1-f_b) \cdot (1-f_{o,c}) \cdot (1-f_r) \cdot (1-f_{o,p}) \]

Where

- \( HC_{\text{crevice}} = \int \text{dm}_{\text{fuel in crevices}} / \text{m}_{\text{fuel in cylinder}} \)
- \( HC_{\text{index}} = \text{m}_{\text{fuel in exhaust}} / \text{m}_{\text{fuel in cylinder}} \)
- \( f_b = \) the fraction of burned gas
- \( f_{o,c} = \) the fraction of oxidized in cylinder
- \( f_r = \) the fraction of remained in cylinder
- \( f_{o,p} = \) the fraction of oxidized in port

\( HC_{\text{crevice}} \) is obtained from estimated crevice volume, maximum cylinder pressure, and residual gas fraction, as shown in Chapter 4. The fraction of burned gas in the crevice gases is obtained from the flame geometry and in-cylinder pressure trace; it is about 15 percent of the maximum gases trapped in piston ring pack crevices for the spark plug location used here. For simplicity, the other crevices are treated as the piston crevice. The fraction of \( HC \) oxidized in the cylinder \( (f_{o,c}) \) is calculated from the modeling of crevice gas transport and oxidation. The fraction of the crevice hydrocarbons that survives the expansion process and exits the cylinder is 20 to 30 percent, depending on the operating conditions. 10 to 14 percent of the maximum crevice gases flow into the cylinder during the blowdown period and are assumed to remain without oxidation. Based on this assumption, \( f_{o,c} \) can be quantified.

The \( HC_{\text{index}} \) of the warmed-up engine of the original piston at 1600 rpm and intake pressure of 0.4 bar is 0.9, based on the experimental result. The burned gas fraction in the crevice \( (f_b) \) is 0.15. The fraction of oxidation in the cylinder is 0.61, which
is based on 70 percent oxidation at EVO and 10 percent of the crevice gas coming out during the blowdown period which is assumed not to be oxidized. It was estimated that about 30 percent of the unburned hydrocarbons remain in the cylinder with the residual gas. Previous studies [37] show that about 30 percent of the exhaust hydrocarbons exiting the cylinder are oxidized in the exhaust port. Thus, the right-hand side of equation 5.1 can be calculated as

\[ 5.5(1-0.15)(1-0.61)(1-0.3)(1-0.3) = 0.91 \]

This value agrees well with the HC\text{index} obtained from the experiments. For other operating conditions, the value of \( f_b, f_{o,c}, f_r, f_{o,p} \) are listed in Table 5.1.
**Table 5.1** List of $f_b$, $f_{o,c}$, $f_r$, $f_{o,p}$, calculated $HC_{\text{index}}$, and measured $HC_{\text{index}}$

<table>
<thead>
<tr>
<th></th>
<th>$HC_{\text{crevice}}$</th>
<th>$f_b$</th>
<th>$f_{o,c}$</th>
<th>$f_r$</th>
<th>$f_{o,n}$</th>
<th>calculated $HC_{\text{index}}$</th>
<th>measured $HC_{\text{index}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>warmed-up engine</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1600 rpm, 0.4 bar</td>
<td>5.5</td>
<td>0.15</td>
<td>0.61</td>
<td>0.30</td>
<td>0.30</td>
<td>0.89</td>
<td>0.91</td>
</tr>
<tr>
<td>1600 rpm, WOT</td>
<td>5.3</td>
<td>0.15</td>
<td>0.67</td>
<td>0.20</td>
<td>0.40</td>
<td>0.71</td>
<td>0.74</td>
</tr>
<tr>
<td><strong>cold engine</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1600 rpm, 0.4 bar</td>
<td>7.0</td>
<td>0.15</td>
<td>0.60</td>
<td>0.30</td>
<td>0.20</td>
<td>1.33</td>
<td>1.33</td>
</tr>
<tr>
<td>1600 rpm, WOT</td>
<td>8.7</td>
<td>0.15</td>
<td>0.68</td>
<td>0.20</td>
<td>0.30</td>
<td>1.32</td>
<td>1.27</td>
</tr>
</tbody>
</table>
Figure 5.1  Piston crevice gas distribution along the liner during the expansion process at 1600 rpm, intake pressure = 1.0 bar.

Figure 5.2  Piston crevice gas distribution along the liner during the expansion process at 1600 rpm, intake pressure = 0.4 bar.
Figure 5.3  Percent piston crevice gas flow into the cylinder during expansion.
Figure 5.4  Crevice gas velocity with respect to the piston during expansion at 1600 rpm, intake pressure = 1.0 bar.

Figure 5.5  Crevice gas velocity with respect to the piston during expansion at 1600 rpm, intake pressure = 0.4 bar.
Figure 5.6  Absolute gas velocity of the piston crevice gas during expansion at 1600 rpm, intake pressure = 1.0 bar.

Figure 5.7  Absolute gas velocity of the piston crevice gas during expansion at 1600 rpm, intake pressure = 0.4 bar.
Figure 5.8  The initial thickness of the piston crevice gas at 1600 rpm, intake pressure = 1.0 bar.

Figure 5.9  The initial thickness of the piston crevice gas at 1600 rpm, intake pressure = 0.4 bar.
Figure 5.10  The thermal boundary layers growth during the expansion process.
Figure 5.11  Time of 50 percent propane consumption vs. temperature.

Figure 5.12  The temperature of core gas during the expansion process.
Figure 5.13  A typical initial boundary layer temperature and unburned hydrocarbon profile.
Figure 5.14 The thermal boundary layer profiles and propane concentration profiles at the location of 1.9 cm from the top of the liner.
Figure 5.15  The thermal boundary layer profiles and propane concentration profiles at the location of 2.6 cm from the top of the liner.
Figure 5.16 The initial unburned hydrocarbons from the piston crevice during the expansion process at 1600 rpm, intake pressure = 1.0 bar.

Figure 5.17 The remaining unburned hydrocarbons at EVO, 1600 rpm, intake pressure = 1.0 bar.
Figure 5.18  The initial unburned hydrocarbons from the piston crevice during the expansion process at 1600 rpm, intake pressure = 0.4 bar.

Figure 5.19  The remaining unburned hydrocarbons at EVO, 1600 rpm, intake pressure = 0.4 bar.
Figure 5.20  The sensitivity of the amount of unburned hydrocarbons left at EVO to piston crevice volume size.
Figure 5.21  The initial unburned hydrocarbons distribution along the liner of the original piston and three different piston crevice reduction cases.

Figure 5.22  The remaining unburned hydrocarbons at EVO, 1600 rpm, intake pressure = 1.0 bar.
Figure 5.23  The sensitivity of the amount of unburned hydrocarbons left at EVO to piston crevice volume size.

Figure 5.24  The core gas temperature of the cold engine.
Figure 5.25  The initial piston crevice gas thickness for the cold engine.
**Figure 5.26** The initial piston crevice gas distribution and remaining unburned hydrocarbons after oxidation at EVO, 1600 rpm, intake pressure = 0.4 bar.
Figure 5.27  The initial piston crevice gas distribution and remaining unburned hydrocarbons after oxidation at EVO, 1600 rpm, intake pressure = 1.0 bar.
Figure 5.28  The sensitivity of oxidation level of the cold engine to crevice volume size.
CHAPTER 6: SUMMARY AND CONCLUSIONS

6-1 Summary of Research Components

The exhaust hydrocarbon emissions of a single cylinder spark-ignition engine during the engine warm-up period were measured on a cycle-by-cycle basis with a fast-response FID. A methodology for estimating the mass averaged mean hydrocarbon emissions per cycle from the fast-response FID measurement was developed. Experiments with two different locations of the fast-response FID sampling were used to investigate the non-uniformity of the exhaust hydrocarbon emissions near the exhaust valve during the engine warm-up period.

In order to investigate the sensitivity of exhaust hydrocarbon emissions to crevice volume size and location, the piston top land and head gasket were modified to change the crevice volume and location. Propane fuel was used to minimize the oil layer absorption/desorption HC emissions mechanism and eliminate liquid fuel effects (such as liquid fuel getting into crevices, oil layers, or deposits). Engine operating conditions were selected to investigate the effect of different crevice volume changes on the exhaust hydrocarbon emissions. The exhaust hydrocarbon emissions at cold and warmed-up conditions with different crevice volume sizes and locations were thereby quantified.

The maximum amount of unburned fuel which escapes primary combustion because it is within the crevices was calculated during engine warm-up. The predicted engine component temperatures and piston crevice volumes were obtained from a
computer model that simulates the thermal processes of the engine piston, block, and head [21].

A fraction of the fuel that is injected onto the back side of the intake valve and port areas will not be evaporated due to the cold wall temperatures. Some of this liquid fuel in the port enter the cylinder without vaporizing and may be stored in piston ring pack crevices, oil layers, and the deposits, thereby escaping primary combustion. If it is not oxidized, it may contribute to the exhaust hydrocarbon emissions. To quantify the effect of the liquid fuel in crevices, oil layers, and deposits on the exhaust hydrocarbon emissions during the engine warm-up period, two different liquid fuel preparation techniques were employed: a conventional gasoline port fuel injector and a pre-vaporizing gasoline injector.

A model of piston crevice gas transport and oxidation was developed. This model is composed of one-dimensional unsteady mass, species, and energy equation. A model of piston-cylinder crevice flow [22] was employed to predict the piston crevice out flow into the cylinder during the expansion process. The model of crevice gas transport and oxidation was then used to estimate how much of the piston crevice gas is oxidized and the sensitivity of the in-cylinder oxidation level of the unburned hydrocarbon emissions to the piston crevice volume size.

6-2 Methodology for Estimating the Mass Averaged Mean Hydrocarbon Emissions and Maximum Hydrocarbons in Crevices

1 Based on the in-cylinder pressure data and the fast-response FID signals, the cycle-resolved mass averaged mean hydrocarbon emissions of the fast-response FID are well matched with data of a conventional hydrocarbon emission analyzer at steady state.
2. The crevice volume of the original piston changes about 19 to 43 percent during the engine warm-up period over the experimental conditions examined.

3. Between 4.5 to 6 percent of the fuel inducted into the cylinder (original piston case at warmed-up condition) escapes the primary combustion process, and 6.5 to 9 percent of fuel are trapped in crevices for the cold engine conditions.

6-3 Effect of Crevice Volume Size and Location

The results of crevice volume size and location study at steady state can be summarized as follows:

1. At steady state, the exhaust hydrocarbon emissions modestly increase as the piston crevice volume is increased. The slope of the normalized exhaust hydrocarbon emissions plot against normalized crevice volume size is about 0.2.

2. The change in the exhaust hydrocarbon emissions at steady state is proportional to the change in head gasket crevice volume (the sensitivity is 1.0).

3. Based on the fast-response FID signals, the piston crevice volume changes affect the exhaust hydrocarbon concentration at the end of the exhaust process, and the head gasket crevice size changes affect the exhaust hydrocarbon concentration during the blowdown period and at the early exhaust process.

4. The head gasket crevice regions that are closer to the exhaust valve have a larger effect on the exhaust hydrocarbon emissions due to the shorter retention time in the cylinder.

5. The head gasket crevice has more effect on the exhaust hydrocarbon emissions than the piston crevice per unit crevice volume.

6-4 Effect of Crevice Volume Size and Location During Warm-up

The results of crevice volume size and location study during starting and warm-up
are summarized as follows:

1. The exhaust hydrocarbon emissions quickly decrease during warm-up and reach approximate steady state in ~ 100 seconds.

2. The HC emissions index at steady state for the original piston case is about 0.7 to 1.0 percent and is 1.3 to 1.5 percent for cold conditions.

3. The HC emissions index of the cold engine is substantially higher than that of the warmed-up engine at the same crevice size, mainly due to changes in the in-cylinder and exhaust port oxidation levels.

4. The trend of the exhaust hydrocarbon emissions increase under cold engine conditions with increasing crevice volume is similar to that of the exhaust hydrocarbon emissions at steady state.

5. 80 to 90 percent of the decrease in the exhaust hydrocarbon emissions during warm-up is due to the change in oxidation level in the cylinder and the exhaust port; the rest (10 to 20 percent) of the decrease is due to the piston crevice volume and crevice gas density decrease.

6. The sensitivity of the exhaust hydrocarbon emissions to the head gasket crevice volume decreases as the engine speed and load increase, presumably due to an increasing fraction that are oxidized.

7. The decrease in the exhaust hydrocarbon emissions during warm-up is about 25 to 60 percent, depending on engine operating conditions. The decrease in the exhaust hydrocarbon emissions is larger at conditions with higher thermal loading.

8. Even though the flow at the exhaust valve is highly turbulent during the blowdown and the exhaust process, large volumes of burned gas do not mix with the unburned hydrocarbons which come from the piston crevice.
9. There is non-uniformity of the mixture near the exhaust valve, and the fast-response FID measurements of the exhaust hydrocarbon emissions near the exhaust valve do not represent the mean hydrocarbon concentration.

6-5 Effect of Liquid Fuel on the Exhaust Hydrocarbon Emissions

The results of liquid fuel on the exhaust hydrocarbon emissions during warm-up can be summarized as follows:

1. The steady state exhaust hydrocarbon emissions of the conventional gasoline port fuel injector are 10 percent higher than the pre-vaporizing injector case between $\lambda = 0.8$ and $\lambda = 1.1$, and about 5 percent higher for $\lambda = 1.2$. This indicates that liquid fuel may enter the crevices, oil layers, and deposits and escape primary combustion, thereby contributing to the exhaust hydrocarbon emissions.

2. The liquid fuel effect on the exhaust hydrocarbon emissions is much larger during the early stage of the engine warm-up period (2 times higher) and decreases as the engine warms up.

6-6 Results of Modeling of Piston Crevise Gas Transport and Oxidation

The modeling study of piston crevice gas transport and oxidation close to the cylinder liner showed the following:

1. About 15 percent of the crevice gas trapped in the piston ring pack crevices is burned gas, depending on the flame geometry and location of the spark plug.

2. The initial thickness of the layer of piston crevice gas (of the original piston) along the liner is less than the gap between the piston and cylinder wall.

3. The piston crevice gases remain between the cylinder wall and 1 mm from the wall during the expansion process.
4. The core gas temperature plays a major role in the unburned hydrocarbon oxidation process.

5. 30 percent of the initial unburned hydrocarbons from the piston crevice of the original piston case at 1600 rpm and intake pressure of 0.4 bar remains unburned at the exhaust valve opening and 20 percent for 1600 rpm and intake pressure of 1.0 bar.

6. The slope of the normalized amount of unburned hydrocarbons left at EVO to the normalized piston crevice volume at 1600 rpm and intake pressure of 0.4 bar is 0.42, and the slope of 1600 rpm WOT case is 0.29. The amount of unburned hydrocarbons surviving in-cylinder oxidation modestly increases as the piston crevice volume increases.

7. The calculations of artificially reduced piston crevice volumes show that, with much smaller crevices, a substantial amount of the unburned hydrocarbons from the piston crevice survives oxidation due to very small thickness of the crevice gas layer and large heat transfer to the wall. Calculation shows two different asymptotes of the unburned hydrocarbons as the piston crevice volume decreases.

8. The percent of unburned hydrocarbons left at EVO in the cold engine experiments is almost the same in the warmed-up engine case. However, the amount of remaining unburned hydrocarbons in the cold engine at EVO is about 60 to 80 percent larger than in the warmed-up engine cases.

9. Based on the modeling of piston crevice gas transport and oxidation and experimental results, the hydrocarbon emissions mechanism was constructed.
REFERENCES


