# Effect of In-cylinder Liquid Fuel Films on Engine-Out Unburned Hydrocarbon Emissions for SI Engines

by

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Submitted to the Department of Mechanical Engineering in Partial Fulfillment of the Requirements for the Degree of

## DOCTOR OF PHILOSOPHY IN MECHANICAL ENGINEERING AT THE MASSACHUSETTS INSTITUTE OF TECHNOLOGY

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Submitted to the Department of Mechanical Engineering on January 14, 2011 in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy in Mechanical Engineering

### ABSTRACT

Nearly all of the hydrocarbon emissions from a modern gasoline-fueled vehicle occur when the engine is first started. One important contributing factor to this is the fact that, during this time, temperatures throughout the engine are low – below the point at which all of the components of the gasoline can readily vaporize. Consequently, any fuel that enters the combustion chamber in liquid form can escape combustion and subsequently be exhausted as hydrocarbon emissions.

An experimental study was performed in a firing engine in which liquid gasoline films were established at various locations in the combustion chamber and the resulting impact on hydrocarbon emissions was assessed. Unique about this setup was that it combined direct visual observation of the liquid fuel films, measurements of the temperatures these films were subjected to, and the determination from gas analyzers of burned and unburned fuel quantities – all with cycle-level or better resolution.

An increase in the hydrocarbon emissions was observed with liquid gasoline films present in the combustion chamber. This increase depended upon both the location of the film and the temperature of that location, and correlated with estimates of the mass of fuel in the film. The largest impact was observed when the head near the exhaust valve was wetted; the smallest impact was observed when the piston on the intake side of the engine was wetted. In general, as engine temperatures increased the hydrocarbon emissions due to the liquid fuel films decreased. It was also identified when, in the exhaust event, fuel from the films was actually exhausted.

The effect of the location of the liquid fuel film can best be understood in terms of the time before flame arrival at that location, the local flow over the film, and the extent to which the overall flow in the combustion chamber carries fuel from the film to the exhaust valve. The primary effect of wall temperature is to affect the amount of vaporization from the film: as temperature increases more vaporization occurs before flame arrival, resulting in less fuel that can vaporize post-flame as unburned fuel emissions.

Thesis Supervisor: John Heywood Title: Professor of Mechanical Engineering (This page is intentionally blank.)

Thank you, Mom and Dad.

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### **1. INTRODUCTION**

### 1.1. MOTIVATION AND BACKGROUND

#### 1.1.1. Unburned Hydrocarbon Emissions Regulations

Beginning with the 1968 model year, the U.S. government has regulated the emission of unburned hydrocarbons from motor vehicles. These regulations are complex: the intention of what follows is not to be comprehensive but rather to simply give a sense of these regulations over time and at present. Where helpful, further explanation and clarification is given in the endnotes.

The history of U.S. federal unburned hydrocarbon regulations for passenger cars is plotted in Figure 1.1 from 1972 to the present<sup>1</sup> (data from [1] and [2]). The horizontal axis is the model year in which the regulation was first imposed; the vertical axis is the allowable hydrocarbon emissions from a standard test procedure, specified as grams of emissions allowed per vehicle mile driven.<sup>2</sup>

The vertical axis of the plot is logarithmic, and thus the rough trend in these regulations is an exponential drop in allowable hydrocarbon emissions with time. Effective with the 2007 model year, all passenger vehicles sold in the U.S. must satisfy the 0.01 g/mile standard. The need to reduce engines' hydrocarbon emissions to meet these ever-tighter regulations is the underlying motivation of this research.

To put these numbers into perspective, consider that for gasoline a fuel economy of 30 mpg corresponds to roughly 95 grams of fuel consumed per mile. Thus, even at this "good" fuel economy and thus relatively <u>low</u> fuel consumption rate, for today's emissions standards the amount of fuel that can be emitted unburned (0.01 g/mile) is roughly one ten thousandth of the total fuel that is consumed (95 g/mile).

Further, the actual physical quantity of unburned fuel that can be emitted during typical driving can also be estimated. The standards of Figure 1.1 are based upon the FTP75 drive cycle, which is meant to be representative of typical urban driving [3]. It contains a cold start phase, a transient phase, and a hot start phase; a weighted average of the emissions from these three phases is compared to the regulation standards<sup>3</sup>.

A rough estimate of the quantity of unburned fuel that can be emitted can be obtained by assuming the emissions on a g/mile basis are identical in the three phases of the test, in which case the different weighting factors do not matter. Since the entire test is 11.04 miles and 1874 seconds [3], with this assumption the 0.01 g/mile standard corresponds to roughly 110 mg of fuel, which is a spherical droplet of gasoline approximately 1/4" in diameter. Alternatively and perhaps more realistically, if all of the unburned hydrocarbon emissions are assumed to occur in the cold start phase of the FTP75 drive cycle<sup>4</sup>, the corresponding quantity of unburned fuel for the 0.01 g/mile standard is 173 mg – which is a droplet of gasoline roughly 5/16" in diameter. These estimated quantities of unburned fuel can be emitted in about a half hour of typical vehicle operation.

The fact that these estimates are such small fractions and quantities of fuel underscores how tight the current emissions regulations are, and furthermore how good the technology is that is able to meet them. It also suggests that focused research into the details of the engine and its processes is required if one is to understand possible means of reducing the unburned hydrocarbon emissions. This work is one component of that research.

## 1.1.2. Typical Engine-Out and Tailpipe-Out Unburned Hydrocarbon Emissions

In reality, nearly all of the unburned hydrocarbon emissions from a modern vehicle occur when the engine is first started. This can be seen in Figure 1.2, which shows on a fractional basis the cumulative engine-out and tailpipe-out unburned hydrocarbon emissions versus elapsed time in the FTP75 test for a particular vehicle. The engine-out emissions are what exit the engine, prior to any exhaust aftertreatment. The tailpipe-out emissions are what the vehicle ultimately emits after exhaust aftertreatment; it is the tailpipe-out emissions that are regulated.

<u>Aside</u>: the data in Figure 1.2 is somewhat dated, and the emissions certification and model year of the vehicle are unfortunately not known, but the *trend* in both the engine-out and tailpipe-out emissions is nevertheless the same in current engines [6] [7], the only notable difference being that the time to reach ~95% of the tailpipe-out emissions occurs much sooner in modern engines – of the order 10 to 15 seconds [7].

On a fractional basis, nearly all of the tailpipe-out emissions occur during the initial phases of operation. The engine-out emissions, however, do not vary as substantially with time. The reason for this significant difference between the cumulative engine-out and tailpipe-out emissions is due to the effectiveness of modern exhaust aftertreatment systems, primarily the three-way catalyst. When operational, these systems eliminate nearly all of the emissions exiting the engine – resulting in very low tailpipe-out emissions. During the initial phases of engine operation these systems are not yet operational and/or effective, and thus nearly all tailpipe-out emissions occur when the engine is first started.

Moreover, during this critical time when the exhaust aftertreatment system is not effective at reducing emissions, the actual emissions exiting the engine are increased as well. This can be seen in Figure 1.3, which compares the engine-out unburned hydrocarbon emissions

(normalized by the injected fuel mass) for the cold start and hot start phases of the FTP75 test for the same vehicle of Figure 1.2. For the cold start phase (recorded as seconds 0 to 505 in Figure 1.2) the engine starts at ambient temperature. For the hot start phase (recorded as seconds 1369 to 1874 in Figure 1.2), the engine was fully warmed up but shutdown for ten minutes prior to being restarted. Other than the initial temperature of the engine and subsystems, these two phases are identical.

In the initial ten seconds of both starts, the unburned hydrocarbon emissions exiting the engine are many times the level established later – and during this time the exhaust aftertreatment systems are generally not yet hot enough to be operational, resulting in substantial tailpipe-out unburned hydrocarbon emissions. The rise in the cumulative tailpipe emissions for both of these starts is clearly evident in Figure 1.2 at seconds 0 and 1369. A further effect of engine temperature can be seen in that the cold start engine-out emissions are roughly double the hot start engine-out emissions throughout the first two minutes or so of operation.

Thus, during the early phases of engine operation, commonly referred to as the "start" and "warmup", the emissions leaving the engine are higher <u>and</u> the exhaust aftertreatment systems are not yet fully operational. This results in these phases of engine operation constituting nearly all of the unburned hydrocarbon emissions that the vehicle emits.

#### 1.1.3. Strategies to Reduce Unburned Hydrocarbon Emissions

The above suggests two strategies to reduce the unburned hydrocarbon emissions that exit the tailpipe: (1) reduce the unburned hydrocarbon emissions leaving the engine, especially during the critical time when exhaust aftertreatment is ineffective, and (2) improve the exhaust aftertreatment. In particular, strategies to improve the exhaust aftertreatment include improving the three-way catalyst effectiveness and/or getting it operational sooner, encouraging chemical reactions that eliminate the emissions prior to the catalyst (for example with secondary air injection or substantial spark retard), or even somehow storing the emissions in the exhaust system until they can later be processed.

This project focuses on an important aspect of the first strategy, the reduction of the unburned hydrocarbon emissions that leave the engine, and in particular the contribution to those emissions that is due to the presence of liquid fuel in the combustion chamber. The reduction of this "wall wetting" contribution is a key component of the emissions control strategies used to meet current emissions regulations [9].

## 1.1.4. Gasoline: a Fundamental Problem and Challenge

Gasoline is an outstanding fuel for personal transportation because it has a very high energy density per unit volume in the form it is stored "onboard" the vehicle. Figure 1.4 compares the energy density of gasoline to some other fuels suitable for an internal combustion engine. This energy density can be interpreted as the amount of chemical energy carried onboard the vehicle in a given volume. Gasoline and diesel fuel have comparable energy content, whereas ethanol has roughly two-thirds and the gaseous fuels have less than one-third the chemical energy content of gasoline on a per unit volume basis. This high energy density is one of the reasons gasoline is such an attractive fuel for light-duty transportation.

Gasoline has such a high energy density precisely because it is a mixture of liquid hydrocarbons. And in fact, it is <u>engineered</u> to have specific properties that make it an excellent fuel for use in internal combustion engines. Among these properties is the fact that it is a liquid

at ambient temperatures and pressures, but readily vaporizes at typical "warmed up" engine temperatures so that it can be efficiently burned in the engine.

The fundamental problem, however, is that when the engine is "cold" (when it is started) the temperatures in the engine are not sufficient to vaporize all of the gasoline. That is, the very fact that makes gasoline a great fuel for light-duty transportation – that is stored as a liquid but readily vaporizes at typical engine temperatures – causes a challenge when the engine is not at operating temperatures. Any gasoline that does not vaporize can accumulate as liquid films within the combustion chamber. If fuel in those films escapes combustion, it can then be exhausted as unburned hydrocarbon emissions.

And in fact, it is quite likely in a modern engine that liquid gasoline will be present in the combustion chamber under cold engine conditions. The trend in modern engines is from port fuel injection to direct in-cylinder fuel injection. Figure 1.5 shows a schematic of both injection schemes. With port fuel injection the fuel spray is directed at the back of the intake valve; with direct fuel injection the fuel is sprayed directly into the combustion chamber. In both of these scenarios, any liquid fuel that impinges on a "cold" metal wall is unlikely to vaporize and thus is very likely to accumulate within the combustion chamber on its walls.

It is commonly thought that these liquid fuel films in the combustion chamber play a major role in the disproportionately high engine-out unburned hydrocarbon emissions when the engine is first started. The focus of this work is the examination and assessment of that hypothesis.

#### 1.2. PROJECT FOCUS

The focus of this project is to assess the impact on unburned hydrocarbon emissions of having liquid fuel films present at various locations within the combustion chamber. A more practical rephrasing of this question is: if it is inevitable that liquid fuel will be present in the combustion chamber during and after a cold start, how does the location of that liquid fuel impact unburned hydrocarbon emissions? And furthermore, what is the effect of the local wall temperature the liquid films are subjected to at each of the locations?

#### **1.3. PREVIOUS WORK**

Previous studies have examined the relationship between the presence of liquid fuel in the combustion chamber and unburned hydrocarbon emissions. Takeda et al [10] used a specially designed engine to trap and purge the cylinder and intake port contents at a desired phase of operation. This enabled them to determine, on a cycle-by-cycle basis, the quantity of fuel on the intake port walls, on the combustion chamber walls, burned and then exhausted, and exhausted unburned. They observed a correlation between the mass of fuel wetting the combustion chamber walls and the unburned hydrocarbon emissions, and attributed this rise to "vaperization [sic] of the fuel remaining on the cylinder wall, during the expansion stroke". Due to the nature of these experiments, they could not identify where within the combustion chamber the liquid fuel had accumulated.

Landsberg et al [11] deposited liquid fuel at the intake valve seat while the valve was open and observed a significant increase in hydrocarbon emissions. Deposition of the liquid fuel in the intake port on the side closest to the exhaust valve had a substantially larger impact (roughly threefold) than deposition on the side of the intake port furthest from the exhaust valve.

However, with this technique it is not at all clear where within the combustion chamber the liquid fuel goes and/or accumulates.

Kim et al [12] developed a novel technique using special filter paper that absorbed dyed fuel surrogate to identify where within the combustion chamber liquid fuel accumulated, and further the quantity of fuel that had impinged upon a specific location. Experiments using this filter paper were performed under motored conditions in a visualization engine using various fuel injectors. Separately, in a standard engine operating at similar conditions, exhaust hydrocarbon measurements were made and the trends in the unburned hydrocarbon emissions were "consistent with the findings using the [filter paper] footprints techniques" – it is unclear precisely what this statement means and the effect is not obvious from the data, but it appears that the inference is that larger quantities of liquid fuel on the wall correlated with increased hydrocarbon emissions.

Cho et al [13] used laser-induced fluorescence in a visualization rig to measure, for various port injector targetings, the thickness of liquid fuel films on the cylinder liner under steady state intake flow conditions meant to be representative of both part load and wide open throttle conditions. The mass of liquid fuel on the liner was estimated by integrating the liquid film thickness. Separately, in a standard engine, steady state emissions measurements were made and they found those targetings that resulted in larger masses of fuel on the liner also had higher unburned hydrocarbon emissions.

Finally, a series of studies examining liquid fuel film behavior in a spark-ignition engine were performed at the University of Texas at Austin; those studies pertinent to exhaust emissions are summarized here. The device used for these studies was a specially designed "injection probe" that consisted of a standard port fuel injector connected to a small (~0.2mm ID) tube that

was mounted in an offset electrode spark plug (normally used for pressure transducer access). The end of this tube had an orifice – by rotating or changing the tube a stream of liquid fuel could be directed at the desired location in the combustion chamber. The main advantage of this method to deposit liquid fuel in the combustion chamber over the method used in the current study (a narrow cone angle DI injector) is that with this method the impingement angle is more or less the same for all of the deposition locations. The disadvantages of this method are that head wetting is not possible, the wetted-footprint is not well-controlled, and that periodic "dribbling" from the tube onto the piston occurred that could not be avoided.

These studies used a similar fueling strategy as the current study. Namely, in order to look at the effect of liquid fuel films in a firing engine environment, the bulk of the fuel was provided as a vapor (either LPG or propane depending upon the specific set of experiments) and roughly 10-20% of the fuel was delivered as a liquid via the injection probe. All results in these studies were obtained at steady state with specified coolant temperatures, and further all results (with and without liquid deposition) were obtained at the same exhaust gas relative air-fuel ratio. The specific relative air-fuel ratio used was  $\lambda = 1.1$  (fuel lean). In all cases, they found an increase in unburned hydrocarbon emissions when liquid fuel films were present in the combustion chamber.

The first study in this University of Texas at Austin series was performed by Stanglmaier et al [14]. Gasoline was deposited on the liner and piston. The main findings were that the emissions increase due to the liquid fuel films was relatively insensitive both to deposition timing in the cycle and coolant temperature. From this they concluded that the liquid film vaporization rates were "slow" compared to the engine cycle time. The location that exhibited the most significant increase in unburned hydrocarbon emissions was wetting the liner directly

below the exhaust valves; wetting the center of the piston had the second largest impact and wetting the liner directly below the intake valves had the smallest impact<sup>5</sup>. The emissions level with liquid fuel films present (for an injected liquid quantity of 15% of the overall fueling) was roughly two to three times higher than the unburned hydrocarbon emissions with vapor-only fuel.

Li et al [15] performed a follow-up study that found that the increase in emissions due to the liquid fuel films was roughly linear in the amount of fuel delivered as a liquid. Additionally, injector "shut-off" tests were performed and it was found that the 1/e time constant for the gasoline films was roughly 3 to 5 engine cycles at their test conditions<sup>6</sup>.

Huang et al [16] focused on piston wetting and deposited various single-component hydrocarbons onto the piston. For these experiments the coolant temperature was maintained at 45 °C and the piston temperature was estimated from a cycle simulation to be 150 °C. The fraction of injected liquid that ultimately ended up as unburned hydrocarbon emissions ranged from 5 to 20% based upon their methodology<sup>7</sup>. Interestingly, the largest unburned hydrocarbon emissions impact occurred with the most volatile fuel (normal pentane, boiling point = 36 °C). Its impact was roughly double that of the other normal alkanes used in the study that had boiling points up to 196 °C. This apparent discrepancy was explained by the fact that the normal pentane had transitioned to the film boiling regime, whereas the other fuels had not. And furthermore, the expected film evaporation time for their various fuels correlated very well with their observed trend in the unburned hydrocarbon emissions: longer evaporation times had higher unburned hydrocarbon emissions.

Finally, in a subsequent study Huang et al [17] used a subset of the single-component fuels from [16] and varied the engine speed and load. They found that as speed increased, the

impact of liquid fuel films on unburned hydrocarbon emissions decreased, which they attributed to a combination of decreased time per cycle for vaporization to occur, higher wall temperatures at higher speed operation and thus less fuel mass on the wall, and increased oxidation due to more vigorous mixing in the exhaust port. As load increased, the impact of liquid fuel films on unburned hydrocarbon emissions also decreased. This was attributed to higher wall temperatures<sup>8</sup> (and thus faster film vaporization and smaller wall films) and higher in-cylinder temperatures (and thus a longer period of time for which temperatures are high enough for post-flame oxidation to occur).

\* \* \*

There are a number of shortcomings in these previous studies. First, those that did not involve visualization have no measurement-based information on where the liquid fuel films are located. Those studies involving paired visualization engines and standard engines do not provide simultaneous emissions and visual information, and more importantly many of the visualization studies were not performed in a firing engine environment.

Further, while it is recognized that temperature plays a very important role, no studies actually measured the temperatures the fuel films were subjected to. The results were either correlated to steady state coolant temperatures or to wall temperatures estimated from cycle simulations.

Finally, none of these studies varied the overall relative air-fuel ratio, which provides a means to assess (when near stoichiometric or fuel rich) whether or not fuel vaporized from the liquid films mixes with crevice gases and oxidizes.

This study overcomes these shortcomings: direct visual observation is performed simultaneously with exhaust emissions measurements in a firing engine, special thermocouples

are used to measure the combustion chamber surface temperatures, and different relative air-fuel ratios are examined to assess the possibility of oxidation differences with the various liquid fuel film locations.

:

<sup>2</sup> Prior to 1994 the regulations were on a "total hydrocarbon" basis, while model years 1994 and later use a "nonmethane organic gas" basis. Non-methane organic gases are defined as all compounds containing carbon, excluding methane. Total hydrocarbons are defined as all hydrocarbons, including methane [1]. Note in particular that the non-methane organic gas basis includes compounds containing oxygen and other species, whereas the total hydrocarbon basis does <u>not</u> include such compounds, but does include methane. For practical purposes, these two bases are comparable and generally thought of as the "allowable unburned fuel amount" since they are unreacted or partially reacted fuel molecules. (The fuel is the only significant source of these species exiting the engine.)

<sup>3</sup> For comparison to the regulations standards, the emissions level of the FTP75 test can be expressed as 0.207  $E_{cold}$ start + 0.518  $E_{transient}$  + 0.275  $E_{hot \ start}$ , where the  $E_i$ 's are the emissions levels in g/mile for the respective phases. (from [2] §86.144-94 and used with data from [4]). Note that the explanation in [3] results in a slightly different expression for the composite value, namely 0.215  $E_{cold \ start}$  + 0.500  $E_{transient}$  + 0.285  $E_{hot \ start}$ . This appears to be erroneous as it does not take into consideration the fact that the test phases are <u>slightly</u> different distances in the composite weighting defined in [2], which results in slightly different coefficients.

<sup>4</sup> The cold start phase is 3.59 miles (using data from [4]); see endnote 3 for the weighting of the cold start phase.

<sup>5</sup> Although the deposition locations between the two studies are not completely comparable, the findings in the present study are similar.

<sup>6</sup> Again, although the test conditions were not identical, the findings in this study are consistent with this observation.

<sup>7</sup> For this study, these values were lower: roughly 1 to 10% of the injected liquid ultimately was exhausted as unburned hydrocarbon emissions.

<sup>8</sup> The findings in this study were similar: as wall temperature increased the increase in hydrocarbon emissions due to the liquid fuel films decreased.

<sup>&</sup>lt;sup>1</sup> Prior to 1972 the test procedures were "different enough from the current procedure that the standards are not comparable" [1]; for this reason the regulations are only shown from 1972 onward. Further, except for 1972, the values in this plot are those of the FTP75 drive cycle, which has been used for these regulations since 1975. The 1972 drive cycle was very similar – it simply did not include the "hot start" phase of the FTP75 [3]. More recently, additional drive cycles are used along with the FTP75 drive cycle to certify vehicles. For the sake of comparing historical values, the FTP75 drive cycle standards are examined here.



Figure 1.1: Federal unburned hydrocarbon emissions regulations versus initial year of regulation (For further explanation of the different bases, see note 2 in this chapter.)



Figure 1.2: Cumulative engine-out and tailpipe-out hydrocarbon emission over the FTP-75 drive cycle for a six cylinder SI engine, from [5].



Figure 1.3: Engine-out hydrocarbon (EOHC) emissions for the first (cold start) and third (hot start) phases of the FTP75 drive cycle, for the same engine as Figure 1.2. (Figure from [8], with minor modifications.)



Figure 1.4: Comparison of energy content per unit volume for various fuels suitable for an internal combustion engine



Figure 1.5: Schematics of port fuel injection (left) and direct fuel injection (right)

### 2. SETUP

#### 2.1. ENGINE AND SUBSYSTEMS

#### 2.1.1. Square Piston Visualization Engine

The engine utilized for this study was a special visualization engine with a square crosssection combustion chamber. It is shown schematically in Figure 2.1. The engine was constructed by adding a visualization section to a standard single-cylinder engine assembly [18]. The original circular cross-section piston is connected to the square cross-section piston by a rigid connecting rod that is supported by a crosshead bushing. The original combustion chamber of the engine is not used and is simply vented to the crankcase of the engine (this is not shown in Figure 2.1) so that there is no net change in the volume of the lower original combustion chamber plus crankcase system, and thus there is no work in compressing or expanding gases in the lower unused combustion chamber.

The upper, square cross-section visualization combustion chamber is used. The crosssection is square so as to provide undistorted optical access to the combustion chamber, which would otherwise be distorted with curved windows. One inch thick quartz windows form two opposing walls of the combustion chamber; the other two walls are made of steel.

The piston is sealed by graphite bars as shown in Figure 2.2. These "seal bars" provide a means of sealing the engine that does not require the use of lubrication oil, which could foul the windows and make optical access difficult. The bars are pressed against walls of the combustion chamber by springs, and overlap in lap joints at the corners of the combustion chamber.

Detailed geometry and valve timing information for the engine is given in Table 2.1.

### 2.1.2. Advantages and disadvantages of the square piston engine

The main advantage of this engine was that it afforded excellent optical access. In particular, the location of the liquid fuel films could directly be verified, and other phenomena in the engine could be observed both for troubleshooting and to assist in explaining trends in the quantitative data.

An additional benefit of this engine is that, given its construction, there is a considerable amount of space in the combustion chamber that is easily accessible for the installation of instrumentation and additional hardware. In particular, thermocouples, a pressure transducer, and a fuel injector were mounted in the clearance volume of the combustion chamber for this study.

The main disadvantages in using the square piston visualization engine for this study were that it is not cooled, that it has "nonstandard" geometry, and that it is difficult to seal. Because the engine is not cooled, steady state experiments were not possible. This shortcoming required extra levels of engine control hardware and software, and guided the way the experiments were conducted.

The geometry of this engine is not "standard". While the compression ratio, combustion chamber shape, and of somewhat lesser importance the valve configuration are all quite different from a modern production engine, they are considered an acceptable tradeoff given the other benefits of the engine. The most important factor is that it is a <u>firing</u> engine environment in which to perform the experiments.

Finally, while sealing the engine was difficult, acceptable sealing quality and repeatability were obtained in this study. Engine sealing is discussed in detail in section 2.1.4.
The engine was initially chosen because of its advantages over a more standard engine – any disadvantages that it posed were either able to be overcome or deemed to be an acceptable tradeoff given the advantages.

# 2.1.3. Modifications to the engine for this study

The major modifications made to the engine for this study were the addition of a system of interchangeable wall plates used to target the direct fuel injector, and the addition of special surface thermocouples in the combustion chamber.

The direct injector targeting plates were motivated by the "third window" design of Shelby [19]. The initial design of this system simply adapted Shelby's components. Ultimately, however, these components were redesigned to make them more rigid, to make engine sealing easier and more repeatable, and to make changing of the plates possible without disassembly of any other engine components.

These injector targeting plates are further described in section 2.1.6; the combustion chamber surface thermocouples are described in section 2.2.2.

Additionally, the intake and exhaust systems of the engine were completely redesigned, primarily to obtain more rapid intake pressure stabilization and more accurate intake airflow measurements, and to reduce pressure pulsations in the exhaust to facilitate exhaust gas sampling.

Finally, extensive repairs were made to the engine. The engine was, essentially, completely rebuilt during this study. Most of these repairs were to improve engine sealing, and included the repair of cracked and chipped sections of the combustion chamber components, the replacement of bent or otherwise damaged combustion chamber components, and the

realignment of the entire combustion chamber and piston system. In order to facilitate proper alignment of the components, critical pieces were modified to be pinned in place and custom shims and gage blocks were designed to ensure that accurate spacing of components could be maintained and checked. Because of these modifications and improvements, repeatable engine sealing conditions were realized.

# 2.1.4. Engine Sealing

The most challenging aspect of operating this engine was sealing the combustion chamber. The combustion chamber is formed by two steel plates, a cast iron head, and two quartz windows. Every joint between these components must be sealed and able to withstand combustion pressures and elevated temperatures. Moreover, there are significant blowby paths present in the seal bar design used for the piston sealing that must also be adequately sealed.

The three locations to be sealed in the combustion chamber were the metal-to-metal interfaces, the quartz-metal interfaces, and the piston seal bars. The metal-to-metal interfaces, where it was possible to add glands, were sealed with o-ring sections. In addition, and where o-ring sections were not possible, these joints were sealed from within the combustion chamber with Permatex "Motoseal 1 Ultimate Gasket Maker Grey" (Permatex item #29132). After investigating and trying many different sealants, this particular sealant was found to provide an excellent combination of gap-filling, adhesion, temperature resistance (it is rated to 200 °C), and fuel resistance.

The quartz windows were sealed against the metal of the combustion chamber with flat rubber gaskets. The sealing of these windows was, by far, the most challenging aspect of sealing the engine. The surface each window sealed against was created by four separate components

(the two steel walls, the head, and a "base plate" in the crosshead assembly) – the alignment of these four components was critical. Further, any solution to seal the windows had to be both long-lasting (since the area of the blowby path could change if the gaskets were replaced) as well as easily removable (since the windows did in fact have to removed occasionally and moreover were easily chipped and damaged).

Previous studies (e.g. [20]) using this engine used 1/32" silicone rubber sheeting "glued" to the quartz windows with silicone RTV as the gasket material. The purpose of the RTV was to hold the rubber sheet in place for assembly. The author had difficulty in obtaining satisfactory sealing with this approach. The slightest non-uniformity in the overall thickness of the "rubber sheet plus RTV" resulted in either inadequate sealing (i.e. an immediate leak or, more commonly, blowout of the gasketing while operating the engine) or breaking of the window upon clamping due to localized stresses at the non-uniformity. Attempts to improve this method as well as the investigation of alternate means of sealing the windows constituted a substantial effort in the development of the experimental techniques. Ultimately, outstanding window sealing was obtained by using adhesive-backed 1/32" silicone rubber sheeting (40A durometer, McMaster-Carr part number 8991-K711) – the adhesive backing held the gasket in place for assembly; its thickness apparently was uniform enough so that none of the problems described above with the RTV were encountered.

The final aspect of the combustion chamber to be sealed was the blowby paths in the piston seal bar assemblies. The primary blowby path was through the gaps in the lap joints of the seal bars, as shown in Figure 2.2. The approach to sealing these pathways in prior studies was to fill these gaps with high-temperature silicone RTV. The author had difficulty in replicating this technique: the primary failure mode was the "ripping out" of the RTV during the piston motion.

That is, the RTV (apparently) stuck to the combustion chamber walls and was drawn out of the gaps and into the combustion chamber. This may have been due to the particular combination of RTV and graphite compositions used by this author, though every attempt was made to use similar compositions as previous studies. A second, suspected but not verified, failure mode was the "freezing" of the seal bars in place: the RTV bonded very well to the graphite, and appeared to prevent any movement of the bars. This, it is suspected, resulted in additional blowby paths as the bars could not accommodate minor changes in the combustion chamber cross-section throughout the piston stroke.

The technique developed here was to still use RTV to fill the gaps, but to coat all of the surfaces that encountered the silicone RTV with a very thin film of silicone grease prior to injecting the RTV into the seal bar gaps. The RTV did not bond or stick to the grease; rather, after solidifying the RTV slid very well along the grease. As a result, these critical gaps in the seal bars were effectively filled and all components slid easily along one another. The RTV ultimately used by the author was Permatex "Ultra Copper High Temperature RTV Silicone".

As a result of the steps taken here, along with keeping the firing time per experiment to roughly 45 seconds, no combustion chamber seals required replacing throughout <u>all</u> of the experiments presented in this study (and many others that are not). And moreover, the sealing was very repeatable: the motored net mean effective pressure (at a fixed throttle position) for <u>all</u> of the experiments ranged from -94.6 to -92.9 kPa, despite minor variations in ambient conditions (see Table 5.1). The consistency of the engine sealing was essential as the differences in the experimental results were small, and thus sealing variability had to be minimized to avoid artificial variations in the data.

Further, the quality of sealing obtained was fairly good – especially in light of the fact that the engine was designed for optical access, at the expense of sealing quality. The amount of blowby was measured directly, following an approach similar to that of Pischinger [21]. A bronze bushing in the crosshead prevents any significant gas flow from the region below the piston to the lower unused combustion chamber (the location of this bushing is indicated in Figure 2.3). The region below the piston is instead vented to the atmosphere, either via the reed valve assembly discussed below in section 2.1.5 or, to measure the blowby flow, via the rubber diaphragm tank described below. Because the bushing seals the gas flow to the lower combustion chamber, there is a large displacement flow into and out of the region below the piston via this vent as the piston moves up and down.

In order to actually measure the net blowby amount with this substantial displacement flow present, a metal damping tank with a 12 inch diameter, 1/32 inch thick rubber membrane forming one "wall" of the tank was connected to the region beneath the piston. The total volume of the tank with the membrane unstretched was roughly 4.5 gallons. The membrane stretched and absorbed the large displacement flow into and out of the region below the piston and resulted in a fairly steady exit flow from tank. The tank did pressurize somewhat to roughly 0.03 bar above atmospheric pressure, but this is not thought to significantly affect the blowby flow as the driving pressure ratios are quite high.

The exit flow from this tank was measured directly by two flow meters in series. The first flow meter was a Matheson rotameter (type 7640, using a 605 SS tube) and the second was an Omega FMA-2323 (heat transfer based) mass airflow meter. In multiple repeats the two meters agreed very well and the net blowby flow was ~9% of the intake airflow under the fired conditions described in section 3.3.

### 2.1.5. "Volume beneath the piston" ventilator

<u>Aside</u>: In a standard engine, the region beneath the piston is simply the crankcase. However in this engine, since it has a crosshead, the region beneath the piston where blowby gases accumulate is not the crankcase. While somewhat cumbersome, here the term "volume beneath the piston" is used to be explicitly clear about the region that is being referred to.

In order to assess the amount of burned and unburned fuel in the blowby gases each cycle, the species concentrations in this "volume beneath the piston" must be measured. As described above, in the absence of any alterations, there is a substantial displacement flow into and out of this region as the piston moves up and down with a small additional flow due to blowby.

Thus, without alterations to the flows, the net result is a steadily increasing concentration of CO,  $CO_2$ , and unburned hydrocarbons in this region. In order to resolve the effect of an individual cycle, the difference in two steadily increasing numbers must be measured, which is problematic.

A "ventilator" was designed to overcome this challenge and facilitate the determination of the blowby gas composition on a cycle-by-cycle basis. The basic idea behind the ventilator was to purge the contents of the volume beneath the piston when the piston moves down, and to refill the volume with ambient air when the piston moves up. This keeps the concentrations in this region low and at a relatively consistent level, and thus greatly improves measurement accuracy.

The principle of this ventilator is shown in Figure 2.3. Check valves are used so that when the piston moves up, one flow path is used to provide ambient air; when the piston moves

down the other flow path is used to purge the volume's contents. The species concentrations are sampled prior to the flow split.

The actual implementation of this design is shown in Figure 2.4 and Figure 2.5. Four reed valves from a two-stroke motorcycle engine were used as the high-speed check valves. Two reed valves were oriented to allow flow <u>into</u> the volume beneath the piston and two reed valves were oriented to allow flow <u>out of</u> the volume. The concentrations of CO, CO<sub>2</sub>, and unburned hydrocarbons, as well as the absolute pressure, were all measured between the reed valves and the crosshead. The anatomy and interpretation of these species concentration profiles are discussed next.

Figure 2.6 shows the properly phased unburned hydrocarbon concentration at the sampling location in the ventilator device. (The CO and CO<sub>2</sub> concentrations exhibit similar behavior and thus are not plotted here.) Also shown, for reference, are the "ideal" flow rate at the sampling location and the product of this flow rate and the unburned hydrocarbon concentration, which is the mass flow rate of the unburned hydrocarbons at the sample point. The flow rate is referred to as ideal because it was assumed first that the reed valves instantly open and close at top and bottom center and second that the flow was incompressible – neither of these assumptions is completely valid but they are reasonable first approximations to get a sense of the flows at the sample point. Later, an integration of the mass flow rate of each species is used to estimate the net mass of each species in the blowby flow each cycle.

As a check of the phasing of the concentration profile in Figure 2.6, during the times when there is virtually no flow (near top and bottom center), the species concentration is relatively flat. Table 2.2 was constructed to aid in the interpretation of Figure 2.6. It shows the flows between the combustion chamber and the volume beneath the piston, as well as the relative

concentrations of carbon-containing species in those flows and the net mass exchange of each of those species between the combustion chamber and the volume beneath the piston.

Examining the expansion stroke as an example, the relative magnitude of blowby flow (based upon the pressure ratio) is high initially and gradually decreases. The direction of this flow is from the combustion chamber into the volume beneath the piston. The contents of that flow are initially unburned fuel-air mixture (since the flame has not yet arrived at the piston's edges), but later in the stroke it is burned gases. Thus, initially there is a high mass flow of unburned hydrocarbons and a low flow of CO and  $CO_2$  into the volume beneath the piston. Later in the stroke (corresponding to flame arrival at the piston/liner interface) this "switches" and there is a high mass flow of CO and  $CO_2$  and a low flow of unburned hydrocarbons into the volume beneath the piston.

Using similar logic, the net flow of species into and out of the volume beneath the piston can be assessed for each stroke – this is the bottom row of Table 2.2. Virtually all of the mass transfer of carbon containing species to/from the combustion chamber occurs during the compression and expansion strokes. It is important to note that this row is the mass exchange of species between the combustion chamber and the volume beneath the piston, and <u>not</u> what is being sampled at the ventilator device sample point – that is shown in the third row of Table 2.2.

Thus, examining the anatomy of the species concentration in Figure 2.6, starting by looking at the expansion stroke (360 to 540°): the concentration is initially virtually zero because the gas analyzer was sampling fresh air that was drawn in during the compression stroke. Then, as the expansion stroke progresses the gases in the volume beneath the piston are pushed past the sample point. These gases are a mixture of whatever residual gas was in the volume beneath the piston plus the new blowby gases from that engine cycle.

On the exhaust stroke (540 to 720°), the piston is moving upward and fresh air is drawn past the sample point. Then, on the intake stroke (0 to 180°) the piston is moving downward and pushing the gases in the volume beneath the piston past the sample point. Since there was virtually no additional transfer of unburned hydrocarbons to/from this volume, the mixture is well-mixed and the concentration profile is very flat. The actual level of the concentration is lower than during the expansion stroke because the gases in the volume beneath the piston were diluted by fresh air on the exhaust stroke. And finally, on the compression stroke (180 to 360°), fresh air is pulled past the sample point. The gases in the volume beneath the piston are further diluted by this air, resulting in a very low "residual concentration" in the volume beneath the piston with which the blowby gases from the next cycle will mix to produce the concentration that is sampled on the subsequent expansion stroke.

While it cannot be measured directly, the value of this residual concentration will be small: the fractional drop from the expansion stroke to the intake stroke indicates the amount of dilution that occurs. The residual concentration at the start of the expansion stroke will be this same fraction of intake concentration, which is very small relative to the concentrations measured during the expansion stroke.

Thus, this ventilator device both keeps the concentrations in the volume beneath the piston at a consistent level, which facilitates accurate measurements, and more importantly enables cycle-by-cycle tracking of the species that are in the blowby gases.

# 2.1.6. Fuel Systems

Early in the testing of the setup, it was determined that a conventional fuel tank and fuel pump were not the best choice for storing and pressurizing the fuel for these experiments. The approach used here was to store the fuel in a rubber bladder and to pressurize the fuel by pressurizing the exterior of the bladder. Specifically, a 1 quart hydraulic bladder accumulator with a Viton<sup>1</sup> bladder (manufactured by Parker Hannifin, model number BA002B3U28A1) was used to store and pressurize the fuel for each of the fuel injectors in the study. This system was motivated by the design of a similar system by Shelby [19]; the procedures for operating this fuel system were essentially identical to those developed by Shelby (for step-by-step instructions, refer to [19]).

This fuel system is shown schematically in Figure 2.7. Fuel is stored inside a rubber bladder that is completely enclosed in a steel container<sup>2</sup>. The interior of this container is sealed from the atmosphere and pressurized with nitrogen, which then pressurizes the fuel inside the bladder. The main advantages of this system over a conventional fuel tank and fuel pump are first that the system is completely sealed (there is no loss of the more volatile fuel components over time due to the venting of the tank) and second that it is capable of providing near constant fuel pressure (there is no pressure oscillation due to a regulator opening and closing). The disadvantage of this system is that it is difficult to control the fuel temperature. In these experiments the fuel temperature was not controlled or monitored. However, the entire apparatus was soaked at ambient conditions for roughly 1.5 hours between experiments and the engine was only fired for roughly 45 seconds during each experiment.

Minor improvements and modifications were made in this implementation of the fuel system in an attempt to prevent air in the system from reaching the injector tip. In particular, clear tubing was used between the bladder and the fuel injector so that any air in the line could be directly observed. And further, the accumulator was installed well above the height of the

fuel injector and the fuel line monotonically rose from the injector to the bladder so that any air in the line would collect at the top of the bladder and not be trapped anywhere in the fuel line.

Two different fuel injectors were used simultaneously in this study, each connected to a bladder accumulator described above. The first injector was a standard production port injector (Delphi Multec 2) used in the intake port for the delivery of isopentane<sup>3</sup>. (The logic behind the particular fuels used in the study is in section 3.1.)

The second injector was a prototype stratified DI injector designed by Zexel. This injector was originally used by Shelby [19], and was used here both due to difficulties in locating a suitable alternate injector and due to the fact that data on this injector's behavior was available from prior studies [19][20]. It was a pressure-swirl type injector with a nominal cone angle of 20°. Figure 2.8 shows a cross-section of the spray pattern from this injector when operating at an injection pressure difference of roughly 44 psi, which is close to the injection pressure used in this study. This image was obtained in [19] using planar laser-induced fluorescence. For the current study, this injector was operated by a programmable injector driver provided by Siemens.

This DI injector was used to spray liquid gasoline onto the walls of the combustion chamber – the narrow cone angle was desired in order to focus the deposition of the liquid gasoline over a relatively small area of the combustion chamber. In order to deposit the liquid fuel at <u>different</u> locations within the combustion chamber, a system of interchangeable targeting plates was designed. For each desired location, a different plate was used to aim the injector spray at that location. Figure 2.9 shows the injector mounted in a targeting plate, prior to being installed in the engine. Figure 2.10 shows the injector and (a different) targeting plate installed in the engine. These plates were mounted in the upper part of the combustion chamber in one of

the metal walls. The inside surface of the plate actually formed part of the combustion chamber inner wall surface.

The nominally targeted locations in this study were the head, the liner (in the clearance volume), and the piston on both the intake and exhaust sides of the engine. The actual wetted areas for each of these nominal deposition locations were more complex – see section 5.2.

Finally, because knowledge of the injected fuel quantity was critical for this study, the calibrations for each of these injectors with their respective accumulators and fuels<sup>4</sup> were directly measured by injecting a fixed number of times into a flask immersed in an ice bath.

# 2.1.7. Engine Control

The engine was controlled by a custom controller created by Prof. Wai Cheng; the author made some modifications and improvements to the software used to operate the controller. This controller used a Kontron ADIO 1600 multifunction analog and digital I/O card and a Cyber Research CDI024C digital I/O card, along with some custom circuitry to enable and disable the controller outputs. The control software was custom written C code.

The combination of this hardware and software was capable of controlling, <u>on a cycle-by-</u> <u>cycle basis</u>, the timing and duration of an ignition signal used to operate the spark plug as well as two injection signals, one for each of the fuel injectors used in the study. Additionally, the controller had two analog inputs: one of these inputs was used for feedback control of the isopentane fueling to obtain a desired exhaust relative air-fuel ratio during portions of each experiment.

The feedback control scheme used in this study was a modified discrete incremental PI control scheme. Pseudocode for it is shown below – the variable inj\_dur is controlled to obtain the desired target\_lambda.

last\_error = current\_error current\_error = sensor\_lambda - target\_lambda delta\_inj\_dur = [k\_i \* current\_error + k\_p \* (current\_error last\_error)] \* inj\_dur inj\_dur = inj\_dur + delta\_inj\_dur

Here, k\_i is the integral gain and k\_p is the proportional gain. The values found to work quite well for this setup were k\_i = k\_p = 0.3. The difference between this scheme and a "traditional" discrete incremental PI controller is that the change in the control variable (here, that change is delta\_inj\_dur) is expressed as a fraction of its current value, and not as a fraction of some fixed quantity. This was found to greatly improve controller performance as, due to hardware limitations, the resolution of the exhaust relative air-fuel ratio ( $\lambda$ ) as read by the controller was roughly 0.005 units of  $\lambda$ . In the absence of the above modification, for a case in which the scaling factor of delta\_inj\_dur was not "close enough" to the actual required injection duration, quantization error could result in oscillation and the inability to reach the desired target\_lambda.

For the conditions of section 3.3 that these experiments were conducted at and with fueling on isopentane only (which essentially completely vaporizes), the engine was able to stabilize at a newly commanded target  $\lambda$  in roughly 8-10 cycles, and was stable within  $\pm 0.01$  units of  $\lambda$ . This time was fairly short, despite the "slow" response of the UEGO used as input for the feedback control and the displacement of some unburned mixture into the intake port prior to intake valve closing.

An improvement made by the author to the control software was the checking of the exhaust relative air-fuel ratio and all controller timings and durations each cycle, and the immediate termination of the code if an out-of-range value was encountered. This was both to ensure that an error in the code could not cause engine damage as well as to ensure that the feedback scheme for the fueling did not "run away".

Additionally, all engine timing parameters, along with the number of crank angles detected, were streamed to disk for each cycle. This data was used for debugging as well as later post-processing of the results. For example, each cycle's injection durations were used to calculate the mass of fuel injected in that cycle.

Finally, some minor cosmetic improvements were made to the software to enable and facilitate interactive changing of the control parameters.

## 2.1.8. Engine Position Sensing

In order to synchronize the engine controller to the engine, an incremental shaft encoder (BEI, model XH25E-F1-SS-720-ABZC-28V/V-SM18) was connected to the camshaft of the engine. This encoder provided 720 pulses per revolution and (separately) one pulse per revolution. Because it was connected to the camshaft, it provided one pulse every crank angle degree and one pulse per cycle. The one pulse per cycle was positioned at bottom center of the compression stroke.

Because the integrity of the encoder signals was absolutely essential to controlling the engine, several measures were taken to reduce or eliminate noise in them. First, 24 volt digital logic was used to transmit the signals from the encoder. This 24 volt logic is more noise-tolerant than the TTL logic that was required by all of the devices that used the encoder signals:

immediately prior to the devices the signals were converted to TTL logic by the broadcast module described below. Second, special shielded cabling (also by BEI) was used to reduce electromagnetic interference as the signals were transmitted. And finally, because these signals were not only used to time the engine controller but also to time the data acquisition and the high-speed camera, the encoder output was optically isolated from all of these devices – and all of the signals to these devices were optically isolated from each other. This optical isolation was achieved with a "broadcast module" (BEI, model number BX-24-AD/V-IC/V-IC/V-IC/V). As stated above, the broadcast module also converted the 24-volt logic of the shaft encoder to a TTL basis which the devices required.

#### 2.2. INSTRUMENTATION

### 2.2.1. Pressure

The pressure at various locations throughout the engine was monitored. The combustion chamber pressure was monitored by a Kistler type 7061 piezoelectric transducer, along with a Kistler model 5004 dual mode charge amplifier. This particular transducer type has a very rapid time response, but exhibits "drift" and thus its value must be referenced (or "pegged") to absolute pressure measurements made elsewhere in the engine. Although this transducer had the option of water-cooling, it was not water-cooled because doing so can actually decrease measurement accuracy, primarily due to increased heat transfer to the sensor and vibration of the water within the sensor body [23].

The absolute pressure was monitored in the intake port, exhaust port, the volume beneath the piston, directly downstream of the intake throttle, and at a location in the exhaust far downstream of the exhaust port. The transducers used were either by Omega (type PX176) or by

Honeywell / Data Instruments (type SA). (Despite the different brandings, these transducers all have identical construction and specifications). In practice, these transducers had excellent linearity but all exhibited some degree of offset – which was verified by a deadweight tester and compensated for in the post-processing.

The cylinder pressure was "pegged" each cycle by matching its average value over a 10 degree window at bottom center of the compression stroke to the average value of the intake manifold pressure over this same window. Typical pressure data for a fired cycle at the conditions of section 3.3 is shown in Figure 2.11 and Figure 2.12. Figure 2.12 is a detail view that only shows the 0 to 150 kPa range. As a check of the cylinder pressure pegging, the cylinder pressure during the later part of the exhaust stroke is slightly above the exhaust port pressure, as expected. The "noise" in the cylinder pressure near 200° and 0° (and 720°) is due to vibration of the transducer when the intake and exhaust valves close.

## 2.2.2. Temperature

Gas temperatures in the intake port, exhaust port, ventilator exit flow, and muffler exit were monitored with exposed junction thermocouples (Omega, type GKMQSS-062E-6). Additionally, the intake port wall temperature directly above the valve seat was monitored at two locations with "cement-on" surface thermocouples (Omega, type CO1-K). The first location was on the injector side of the port, and the second was directly opposite this location, shadowed from the injected spray by the valve stem. This same cement-on surface thermocouple type was also used on the exterior wall of the exhaust runner at the gas sampling location to ensure that each experiment was performed at similar exhaust system wall temperatures.

While the temperature data in the above paragraph was not used explicitly in the processing of other results, it was monitored to ensure that the thermal conditions throughout the engine did not change between experiments. Typical data from these thermocouples is shown in Figure 2.13. The intake port wall temperatures start very close to one another, but at the end of firing the side of the port that is shadowed from the injected fuel spray (and closest to the exhaust valve) is roughly 40 °C hotter – this is consistent with and a check of the temperatures measured within the combustion chamber described below. The intake port gas temperature was essentially atmospheric temperature, while the exit flow from the ventilator was roughly 15 °C above atmospheric temperature, consistent with the fact that it is exposed to the higher metal temperatures of the swept volume beneath the piston. The variation in the exhaust gas temperature is due to the fact that the relative air fuel ratio is changing as the experiment is conducted.

In order to obtain actual data on the surface temperatures the liquid fuel films in the combustion chamber were subjected to, special surface thermocouples were installed in the combustion chamber. These thermocouples were custom made by Nanmac Corporation (with their "eroding thermocouple" E10 series design). The body of the thermocouple assembly was made of the material the assembly was mounted in so that the temperature field was minimally changed by the installation of the thermocouple. In this body, a sandwich consisting of thin ribbons of insulation / thermocouple metal A / insulation / thermocouple metal B / insulation is enclosed. The total thickness of these layers is roughly 0.005". The very end of this assembly is mounted flush to the surface it is mounted in – by simply sanding across the thermocouple plus insulation sandwich, a friction weld creates a thermocouple junction directly at the surface.

Further, because this junction is so small, its time response is very fast (of the order milliseconds to microseconds).

These thermocouples were installed in the head, liner, and piston on both the intake and exhaust sides of the engine. They were positioned as close as possible to the locations wetted by the liquid depositions. Figure 2.14 shows these thermocouples installed in their respective engine components. The thermocouple bodies are <sup>1</sup>/<sub>4</sub>" in diameter; the sandwich of thermocouple metal and insulation is so thin it is not even visible in the photographs.

Typical data from these thermocouples is shown in Figure 2.15. The fast time response of these thermocouples is evident both when firing begins as well as when it ends.

The most challenging location in which to install these thermocouples was the piston because it is moving up and down so rapidly. The piston intake-side thermocouple data is not shown in Figure 2.15 because (apparently) fatigue fracture occurred in one of its lead wires shortly after the data presented subsequently began to be acquired. This thermocouple could only be read with the piston close to bottom center: apparently contact was made between the two broken pieces and the signal could be read with the piston in this position. However, limitations of and built-in filtering in the data acquisition system prevented this signal from being usable while the engine was rotating. Thus, the piston intake-side temperature data used subsequently in the study was estimated from previous data and also verified by performing a "rapid shutdown" of the engine in which the piston was at rest at bottom center roughly 5 seconds after the last fired cycle. The variation in the combustion chamber temperature profiles between experiments was small and moreover the agreement between the expected and actually observed final piston intake-side temperature from the rapid shutdown test was very good, so this is not thought to be a major limitation on the experimental results.

### 2.2.3. Mass airflow

The mass airflow was monitored in the intake system by an EPI Master-Touch flow meter (model number 8689MPNH-SSS-133-AC115-AIR). The typical response time of the meter was 1 second (which is roughly 8 cycles at 1000 RPM) and thus it did not provide accurate cycle-resolved measurements, but was usable once the engine had stabilized.

## 2.2.4. Gas Composition

### <u>UEGO</u>

The exhaust gas oxygen content when fuel lean, or oxygen deficiency when fuel rich, was monitored in the exhaust runner roughly 7 inches from the exhaust valve with a universal exhaust gas oxygen sensor (UEGO). The particular model used was a Horiba MEXA-110 $\lambda$ . Based upon the fuel composition, the oxygen content or deficiency in the exhaust gases is used to infer the exhaust gas relative air-fuel ratio.

This particular UEGO did not have a fast enough response to provide cycle-level resolution: its 1/e response time is of the order 100 ms [24], which is 600 CAD at 1000 RPM. It was, however, used for feedback control of the vaporous fueling level to obtain a desired exhaust relative air-fuel ratio as the engine was running and to provide a means of checking the cycle-resolved exhaust gas relative air-fuel ratio calculated from the fast CO and  $CO_2$  data.

### Fast FID

The hydrocarbon concentration was monitored in the exhaust runner roughly 3 inches from the exhaust valve and in the "volume beneath the piston" ventilator with a fast flame ionization detector (Cambustion, HFR400). The term "fast" is used colloquially to refer to an analyzer that is capable of providing cycle-level resolution. The 10-90% response time of this particular analyzer is roughly 2 ms, which is 12 CAD at 1000 RPM.

The important thing about this analyzer, for the purposes of processing and interpreting the results of this study, is that it counts carbon atoms that were in hydrocarbon molecules. And furthermore, its relative sensitivity is essentially the same for all hydrocarbons.

### Fast NDIR

The carbon monoxide and carbon dioxide concentrations were also monitored at the exact same sampling locations as the fast FID with a fast nondispersive infrared analyzer (Cambustion, NDIR500). Its typical 10-90% response time is roughly 8 ms, which is 50 CAD at 1000 RPM. While this response time is much slower than the fast FID, it is still usable for cycle-resolved measurements.

A problem was encountered when sampling hot  $(500+ ^{\circ}C)$  exhaust gases with this analyzer. The reported concentrations were too high; the exact cause of this problem was never determined. However, a means of using the data from the analyzer when sampling these hot exhaust gases was developed – it is discussed in detail in section 4.3.

# 2.2.5. Imaging

A high speed camera was used in order to determine where the liquid fuel films were present, as well as more generally to observe phenomena in the engine to aid in both troubleshooting and the interpretation and support of the quantitative results. The particular camera used was a monochrome Vision Research Phantom version 4.2. Its sensor is 512 by 512 pixels and has 8-bit image depth (256 shades of grey) and an ISO rating of 4800. At its full

resolution, the maximum frame rate of the camera is 2100 frames per second; as the resolution is decreased the maximum frame rate increases. A Nikon Zoom-Nikkor 35-105mm lens was used for all images captured in this study.

The camera was actually used in two distinct ways in the study: for either high-speed image capture or for stroboscopic once-per-cycle image capture with the camera synced to an external strobe light. The reason these two different methods were used was that with the high speed capture the maximum acquisition time was roughly 2 seconds, which is about 15 engine cycles at 1000 RPM. As a result, high speed images could not be captured for an entire experiment: in order to obtain images of the wetted fuel footprint for an entire experiment the stroboscopic technique was developed.

The high speed image capture was typically done at a 3000 to 7000 frames per second, and either utilized no external lighting or a 700 watt halogen Lowel Tota-Light positioned next to the combustion chamber (but not in the line of sight of the camera).

For the stroboscopic image capture, a Perkin Elmer VIGI-Lux MVS-5002 machine vision strobe was used to illuminate the combustion chamber. In order to synchronize this strobe light and the camera shutter to each other, and to synchronize both of them to the engine, an additional, modified version of the engine controller hardware described in section 2.1.7 was used. This controller would open the camera shutter, fire the strobe light, and then close the camera shutter at the desired point in the engine cycle. Two separate control codes were written: one that acquired an image at the same crank angle every cycle and one that acquired an image at a successive crank angle each cycle, using a user-defined step size and starting and ending crank angle.

# 2.3. DATA ACQUISITION

Data from all of the instruments was logged as each experiment was conducted, either once per crank angle or once per cycle. The facility was capable of simultaneously logging 16 channels of differential analog input on a once per crank angle basis, along with 16 differential analog inputs and 16 thermocouple inputs on a once per cycle basis. All of the hardware used for this acquisition was manufactured by National Instruments. The key components of this hardware and their specifications are summarized in Table 2.3.

All of the above-mentioned pressures and fast gas analyzer readings, along with the ignition and fuel injection signals from the engine controller (primarily for debugging purposes in the event problems were encountered), were logged on a once per crank angle basis for each experiment. All of the above-mentioned temperatures, the exhaust UEGO, and the intake airflow were all logged on a once per cycle basis.

Custom "virtual instruments" (VI's) were created and programmed by the author using National Instrument's Labview 7.1 programming environment to log and monitor the data from the various instruments. The VI that was written to perform the data logging simultaneously recorded the once per crank angle and once per cycle data to a text file for subsequent post-processing. Figure 2.16, Figure 2.17, and Figure 2.18 show the front panel, or user interface, of this program. Figure 2.19 shows the block diagram of the program.

Additional VI's were written to monitor the engine between and prior to experiments. As an example, the front panel of the VI used to monitor the temperatures throughout the engine is shown in Figure 2.20.

### 2.4. EXPERIMENTAL PROCEDURE

Prior to any experiments being conducted, all of the instrumentation in the cell was installed in the engine and powered on for a minimum of 90 minutes. This period of time was necessary in order for the fast gas analyzers to thermally stabilize (they are very temperature sensitive and require a minimum of one hour to "warm up"). As a consequence of this, the exhaust system wall temperatures did increase slightly due to heating from the fast gas analyzers and the UEGO: the <u>exterior</u> wall temperature at the gas sampling location stabilized at roughly 35 °C for ambient temperatures ranging from 20 to 22 °C. (For reference, most of the experiments conducted for this study were performed with the initial combustion chamber metal temperatures at 30 °C.)

A summary of the procedure for a typical experiment is:

0. check all settings, valve and switch positions, and that all analyzers and hardware are installed

When the combustion chamber wall temperatures <u>are close to</u> the desired initial temperature

- 1. calibrate the NDIR
- 2. start:
- a. camera capture
- b. strobe & camera sync code (for stroboscopic experiments)
- c. engine control code
- d. data acquisition code

When the combustion chamber wall temperatures <u>reach</u> the desired initial temperature (typically 30 °C) 3. turn the motor on

- after motoring some initial cycles the engine controller starts firing the engine using the prescribed fuel schedule
- the data acquisition and control codes log all of the data as the engine runs

#### when firing completes

- a. open the coarse throttle valve to purge residual hydrocarbons (as monitored on the fast FID display)
- b. turn off the motor
- c. copy and save all data
- d. calibrate the fast FID
- e. prepare for the next experiment (change injector plates, turn on monitoring VI's, etc).

A minor nuance is that, because it must be manually calibrated (and the calibration process is

time-consuming), the fast FID was calibrated immediately after a successful experiment.

However, the fast NDIR calibration was both automated and included built-in linearization of the analyzer output, and further could be completed very quickly – and thus it was calibrated before each experiment.

The typical time between experiments was 75 to 90 minutes. During this time, all of the instrumentation and analyzers remained powered on and installed in the engine in order to ensure consistent exhaust system wall temperatures between experiments.

<sup>1</sup> The manufacturer of the accumulator recommended a Hydrin bladder for use with "fuels". Numerous problems were encountered by the author and other investigators in the lab in using a Hydrin bladder. These problems ranged from tearing of the bladder to (apparent) slow diffusion of the small quantity of hydraulic oil used to lubricate the exterior of the bladder into the fuel (most certainly there was hydraulic oil in the fuel, with no evidence of tears or leaks in the bladder). This is believed to be due to a material incompatibility between the Hydrin and gasoline / isopentane. No such problems were encountered when using a Viton bladder.

<sup>2</sup> The "standard" use of these hydraulic accumulators is to put dry nitrogen inside the bladder with hydraulic fluid inside the container. Thus, the usage here is the reverse of how the accumulator is typically used and some minor modifications are necessary to adapt the fluid connections. In particular, the bladder exit is typically sealed by a Schrader-like valve that must be removed and the "cap" to this valve must be modified to permit a fluid connector. Further, in ordering the accumulator it should be requested with a minimal amount of hydraulic fluid inside the accumulator to lubricate the bladder only (or no hydraulic fluid at all).

<sup>3</sup> These injectors did fail occasionally. It is believed that this was due to the poor lubricity of isopentane and the resulting "severe" adhesive wear it causes [22]. It is not known whether these particular injectors were designed for lower lubricity fuels or not.

<sup>4</sup> Originally, both injectors were calibrated using gasoline since it was less volatile and thus easier to work with as well as being cheaper. However, even after accounting for differences in fuel density, a roughly 15% difference was found in the calibration for the port injector using isopentane versus using gasoline. This was somewhat unexpected and is believed to be due either to slight differences in the bladder thicknesses affecting the actual pressure the fuel experiences or due to other fluid properties such as viscosity affecting the calibration.

These notes are primarily for anyone attempting to replicate this setup.

Combustion	Square cross-section,
Chamber Shape	flat head and flat piston
Displaced volume	785 cm <sup>3</sup>
Square bore	82.55 mm
Stroke	114.3 mm
Connecting rod length	254.0 mm
Compression ratio	6.0
Number of valves	2
IVO	4° aTC
IVC	26° aBC
EVO	33° bBC
EVC	1° bTC

Table 2.1: Geometry and valve information for square piston visualization engine. Valve timings are lash-adjusted zero-lift values.

Stroke	Intake	Compression	Expansion	Exhaust
Piston Motion	$\downarrow$	$\uparrow$	$\checkmark$	$\uparrow$
What's being sampled?	Below piston gases	Fresh incoming air	Below piston gases	Fresh incoming air
Magnitude of blowby flow (based on pressure ratio)	<b>Moderate</b> (~ 2:1 pressure ratio for most of stroke)	<u>1<sup>st</sup> half:</u> Moderate to Low (~2:1 pressure ratio, falling off) <u>2<sup>nd</sup> half</u> : Moderate to High (reaches ~2:1 pressure ratio 120 CAD into stroke, increasing to ~ 6:1 at end of stroke)	High (peaks at ~10:1 pressure ratio, falls off to roughly 1.5:1 toward end of stroke)	<b>~ Zero</b> (pressures "same") <u>note</u> : blowby ends 30° into stroke
Direction of blowby flow	From below piston Into combustion chamber	1 <sup>st</sup> half: From below piston Into combustion chamber 2 <sup>nd</sup> half: From combustion chamber Into below piston	From combustion chamber Into below piston	No significant flow
[CO] & [CO2] in flow	<b>Low</b> – at below piston concentration	<u>1<sup>st</sup> half</u> : <b>Low</b> – at below piston concentration <u>2<sup>nd</sup> half</u> : <b>Low</b> – only unburned mixture in combustion chamber	Low initially, but high toward end of stroke	No significant flow
[HC] in flow	<b>Low</b> – at below piston concentration	<u>1<sup>st</sup> half</u> : <b>Low</b> – at below piston concentration <u>2<sup>nd</sup> half</u> : <b>High</b> – unburned mixture	High initially, but low toward end of stroke	No significant flow
Mass of each species into/out of "below piston" (i.e. the net effect)	Very low amount of all species OUT of below piston into combustion chamber	1 <sup>st</sup> half: very low amount   of all species OUT of below   piston   2 <sup>nd</sup> half: very low amount   of CO & CO2 and high   amount of HC INTO below   piston	<u>Initially</u> , <b>very high HC</b> and <b>very low CO &amp; CO2</b> <i>INTO</i> below piston. <u>Late in stroke</u> , <b>very low HC</b> and <b>high CO &amp; CO2</b> <i>INTO</i> below piston.	No significant flow

Table 2.2: Flows, concentrations, and mass exchange between combustion chamber and volume beneath the piston.

Description	Model Number	Specifications / Comments			
High speed, once per crank angle acquisition					
Data Acquisition Card	PCI-6224	16-bit, 250 kS/s, 16 differential inputs			
BNC Terminal Block	BNC-2090	(2 of these used)			
Low speed, once per cycle acquisition					
Data Acquisition Card	PCI-6023E	12-bit, 200 kS/s, 8 differential inputs			
Signal Conditioning and	SCXI-1000	Enables more & different types of			
Multiplexing Chassis		channels to be logged simultaneously on			
		one card			
Thermocouple Input Module	SCXI-1112	(2 of these used)			
Analog Input Module	SCXI-1100				
BNC Terminal Block	BNC-2095	Used with SXCI-1100			

Table 2.3: Summary of National Instruments data acquisition hardware



Figure 2.1: Basic construction of the square piston visualization engine



Figure 2.2: Isometric view (left) and top view (right) schematics of piston and seal bars showing leakage paths, from [21].





Figure 2.4: Front view of ventilator showing reed valves



Figure 2.5: Installed ventilator: flow paths and gas sampling location



0 degrees is TC-intake.



Figure 2.7: Schematic of fuel system, from [19].



Figure 2.8: Cross-section of fuel spray for the DI injector used in this study, operating at 44 psi injector rail pressure. Image taken 18° after start of injection, for fired operation with an intake pressure of 0.5 bar and the start of injection at 90° aTC-intake. From [19]. Note that the injector targeting used in [19] was not the same as the targeting used in this study, but the injector and engine are otherwise the same.





Figure 2.10: DI injector mounted in a targeting plate (installed in the engine)



Figure 2.11: Pressure data for a typical fired cycle (full view)



Figure 2.12: Pressure data for a typical fired cycle (detail view showing 0-150 kPa)



Figure 2.13: Typical gas and non-combustion chamber surface thermocouple measurements. Firing occurs from cycles 150 to 491. Note that the exhaust port gas temperature is divided by 10.



Figure 2.14: Photograph of the surface thermocouples installed in the piston (left) and head and liner (right)



Figure 2.15: Typical combustion chamber surface thermocouple measurements


Figure 2.16: Tab 1 of the data acquisition VI front panel

Tasks	# cycle	File saving							
	Number	of cycles to log*: 500							
	* including motored cycles at beginning of acquistion								
<u>At 1</u>	000 RPM:								
15	sec = 125 c	cles 90 sec = 750 cycles							
30	sec = 250 c	cles							
45	sec = 375 c	cles 120 sec = 1000 cycles							
60	sec = 500 c	cles							
		(8 1/3 cycles per second)							

Figure 2.17: Tab 2 of the data acquisition VI front panel



Figure 2.18: Tab 3 of the data acquisition VI front panel



Figure 2.19: Block diagram for the data acquisition VI



Figure 2.20: Front panel of the temperature monitoring VI

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## **3. EXPERIMENTAL DESIGN**

#### 3.1. FUELING STRATEGY

The goal of this study was to assess the impact of liquid fuel films on engine-out unburned hydrocarbon emissions in a <u>firing</u> engine environment. In order to maintain <u>controlled</u> firing conditions in which to examine this impact, the bulk of the fuel was delivered to the combustion chamber as a vapor. To this, a controlled amount of additional fuel was deposited as a liquid on the combustion chamber surfaces.

The "vapor" fuel used in the implementation of this strategy was isopentane delivered in the intake port. Isopentane is a very volatile liquid (it boils at 28 °C). Thus, it can be delivered with a standard port fuel injector but will readily vaporize after injection. Data in section 3.3.2 shows that in these experiments the isopentane is essentially completely vaporized.

The liquid fuel chosen to be deposited on the combustion chamber surfaces was gasoline. The particular gasoline used was Haltermann EPA Tier II EEE US Federal Emission Certification Gasoline (HF 00437). The specification sheet for the actual batch of gasoline used in this study is shown as Figure 3.1; its distillation curve is plotted in Figure 3.2.

Although its multi-component nature may make interpreting the data more difficult, gasoline was chosen precisely because it is the fuel used in "real" engines. That is, gasoline was deposited on the combustion chamber surfaces in this study because in a "real-world" engine the fuel on the combustion chamber surfaces is gasoline.

Deposition of the gasoline onto the combustion chamber surfaces was achieved with a narrow cone angle DI injector mounted in one of the combustion chamber walls. Details of this injector and the method of deposition are described in section 2.1.6.

In this study, unless noted, the mass of gasoline sprayed onto the combustion chamber walls was roughly 17% of the stoichiometric fuel mass. The overall delivered relative air-fuel ratio for the mixture of isopentane and the gasoline varied from roughly 0.83 to 0.87 for the data obtained here with liquid deposition. Further information on the fueling conditions is described later in section 3.4.1.

## 3.2. FUEL TRACKING APPROACH

In order to make inferences about the amount of liquid fuel on the combustion chamber walls, and ultimately how much of that fuel results in unburned hydrocarbon emissions, a means of tracking what happens to the liquid fuel on the walls was required. The approach used here was to track <u>all</u> of the fuel molecules entering and leaving the combustion chamber – by knowing the amount of fuel delivered as a vapor inferences can be made about the liquid fuel deposited on the combustion chamber walls.

This tracking is achieved simply by tracking carbon. Figure 3.3 shows this approach schematically. Thinking of the combustion chamber as a black box, the only carbon-containing species entering the combustion chamber are the fuel molecules. (The contribution of atmospheric  $CO_2$  is negligible). Fuel that is burned will exit the combustion chamber either in the exhaust gases or the blowby gases as CO or  $CO_2$ ; fuel that is not burned (completely) will exit as unburned hydrocarbon emissions.

Figure 3.4 shows the implementation of this idea in the engine. The amount of fuel entering the combustion chamber each cycle can be determined from the injector calibrations. Then, by measuring the unburned hydrocarbons and CO and  $CO_2$  in the exhaust and blowby gases the amount of fuel exiting the combustion chamber both burned and unburned can be

determined. Using fast gas analyzers for these measurements enable this determination to be done on a cycle-by-cycle basis. (Recall that an apparatus was designed for this engine to facilitate the cycle-by-cycle determination of the blowby gas composition – see section 2.1.5).

Restating this approach a final time, looking only at the liquid fuel on the combustion chamber walls, Figure 3.5 shows the various pathways it can follow in a given cycle and how the aforementioned fuel tracking pertains to it. First, a particular parcel of liquid fuel on the combustion chamber wall can simply remain on the wall throughout the entire cycle. Second, it can vaporize and be exhausted as unburned hydrocarbon emissions – which is directly detected in the exhaust unburned hydrocarbon measurement. Third, it can vaporize and burn, which is detected in both the exhaust and blowby gas CO and  $CO_2$  measurements. And finally, it can vaporize but be lost to the crankcase, which is detected in the blowby gas unburned hydrocarbon measurement.

## **3.3. OPERATING CONDITIONS**

The operating conditions used for all of the experiments in this study (unless explicitly noted otherwise) are summarized in Table 3.1. Some of these conditions were chosen so as to be representative of the warmup phase of operation, some had essentially no effect on engine operation, and others involved delicate balances that required tradeoffs to be made in order to produce repeatable experiments. The logic used for the selection of each of these conditions, along with important facts and implications regarding them that are relevant to the overall experimental conditions and the interpretation of the results, are discussed in turn below.

## 3.3.1. Engine speed and intake pressure

The engine speed and intake pressure were chosen to be representative of the "warmup" phase of engine operation, the period of time after the engine is started but before the temperatures throughout the engine have stabilized. As described in chapter 1, it is this phase of engine operation that contributes most to the unburned hydrocarbon emissions that a modern vehicle emits.

The engine speed was nominally chosen as 1000 RPM, but with this valued commanded to the motor controller the actual speed obtained was later determined to be 976 RPM. This value was very consistent and was verified by both a digital tachometer and post-processing of the shaft encoder signal. Limitations of the setup precluded engine speeds faster than 1000 RPM (the motor drive would overheat after roughly 30 seconds of operation at ~1100 RPM).

Engine speed was very stable while operating fuel rich, and only slightly less stable when operating fuel lean, suspected to be due to the engine output being more sensitive to fueling level when lean. Figure 3.6 shows engine speed versus cycle number and time for operation in which the engine started at rest, was motored for 150 cycles, fired for 150 cycles, and then motored and shutdown. The fueling level is fuel rich for all but the last 30 fired cycles as shown in the figure. This data was obtained from a truncated version of the experimental scheme described in section 3.4 (only batches 1-5 were performed).

There is an initial transient in the engine speed when firing starts (due to the sudden change in engine output); after this time the engine speed is stable to within  $\pm 0.2\%$  when fuel rich and  $\pm 1\%$  when fuel lean.

The intake pressure was originally chosen as 0.45 bar. The reason for this choice was so as to ensure choked flow at the throttle, and thus a fixed mass of air inducted per cycle at fixed

engine speed irrespective of any volumetric efficiency change as the engine warmed up. However, it was discovered during the experimentation that there was an error in the digital display used to report the intake manifold pressure, and that a display value of 0.45 bar corresponded to an actual average intake pressure of roughly 0.55 bar. Because a significant knowledge base had been built using this condition by the time the discrepancy was found, an average intake pressure of 0.55 bar (or 55 kPa) was maintained for all experiments.

The intake pressure was not actively controlled; a fixed throttle position was used throughout a given experiment. Moreover, this same throttle position was used for all experiments. (The ambient pressure did not vary significantly – see Table 5.1. The total range of the ambient pressures was 99.7 kPa to 100.5 kPa.)

Figure 3.7 shows intake manifold pressure at intake valve closing versus cycle number for a typical experiment. Firing begins at cycle 152 and ends at cycle 491. Note that due to pressure waves in the intake system, the pressure at IVC is greater than the average manifold pressure. This is the reason that, in the stable fired region, the values plotted are greater than the above mentioned value of the average intake pressure.

Although there was not choked flow at the throttle, the airflow was fairly constant once the engine speed and intake manifold pressure had stabilized. Figure 3.8 shows the airflow meter reading versus cycle number for the same dataset as Figure 3.7. For cycles 200-491 (the fired cycles after which the MAP has stabilized), the maximum deviation in the airflow from its average value is  $\pm 0.6\%$ , which includes any noise in the signal.

The conditions and scheme used to produce Figure 3.6, Figure 3.7, and Figure 3.8 are those of Table 3.1 and described in section 3.4. Thus, for cycles 200 and greater in these "quasi-steady fuel deposition" experiments (note that all of the data in chapter 5 comes from cycles 200

or greater in this scheme), the engine is operating at conditions representative of the warmup phase of operation with essentially constant average intake manifold pressure, engine speed, and intake airflow. This fact is later used in the processing and interpretation of the data.

## 3.3.2. Port (isopentane) injection pressure and timing

Isopentane was injected in the intake port using a production port fuel injector. Neither the injection pressure nor the timing of this injection were critical because isopentane is so volatile.

The injection pressure chosen was the injector's nominal design pressure of 55 psig, or roughly 70 psia. Because investigations of the injection timing showed very good vaporization of the isopentane, there was no reason to consider alternate injection pressures.

Closed-valve injection of the isopentane was investigated, the idea being to have all of the isopentane vaporize in the intake port and thus avoid any cylinder wall-wetting that might occur with open-valve injection. A sweep of closed-valve injection timings showed virtually no effect of injection timing on either engine output (the net imep) or the exhaust unburned fuel (i.e. hydrocarbon) emissions.

The experiment performed was as follows: feedback control was used to maintain the relative air-fuel ratio ( $\lambda$ ) at 0.95 ± 0.01 and every 20 cycles the port injection timing was changed. The engine started at ambient temperature and warmed up as the experiment was conducted. Average values were computed for each of the 20-cycle "batches". Figure 3.9 shows the net imep and exhaust HC concentration from this experiment scattered versus injection timing. There is no clear trend in either the net imep or exhaust HC concentration. Figure 3.10 shows these same quantities, but plotted versus time or order in the experiment. Batch 1 is the

first 20 fired cycles, batch 2 is the second 20 fired cycles, and so on. The trend here is clear – whatever variation there is in Figure 3.9 is due to a time variation, and not to changes in the port injection timing.

<u>Aside</u>: since there is little to no effect of port injection timing of the isopentane on the engine output or exhaust emissions, one conclusion useful in later interpretations of the data is thus that under these conditions the isopentane is completely vaporizing.

A particular value for the start-of-injection of the isopentane was chosen as 20° aTCexpansion; as described above, for these closed-valve timings under consideration, because the isopentane is so volatile the timing of this injection showed no impact on the emissions or performance of the engine.

#### 3.3.3. Spark Timing

The choice of spark timing required tradeoffs to be made. Spark timing directly affects combustion phasing; as combustion is phased later in the cycle peak pressures are lower (which is good for engine sealing) but engine output is reduced and there is the possibility of increased cycle-to-cycle variability.

A sweep of spark timings, similar to the port injection timing sweep described above, was performed. The engine was operated on isopentane with feedback control used to maintain  $\lambda =$  $0.95 \pm 0.01$ . The spark timing was changed every 20 cycles, and average values of multiple engine parameters were then computed for each of these 20-cycle "batches". Figure 3.11 is a scatter plot of those parameters versus spark timing. In order to show all of them at once, some were scaled up by the factors indicated in the legend.

While there are many things plotted in Figure 3.11, the underlying driving factor is that as spark is retarded combustion is phased later. This can directly be seen in the location of peak

pressure data. Moreover, as combustion is phased later all pressures in the cylinder are lower. This is evident in the value of peak pressure and the net imep data, and also in the exhaust HC emissions data since less unburned mixture is forced into the crevices with lower cylinder pressures. There is no clear trend in the COV of net imep data; one would expect the COV to increase with spark retard and the lack of a clear trend here is likely due to insufficient sample size.

Because engine sealing was so challenging and so critical to obtaining repeatable results, it dictated the spark timing chosen for the study. As the sealing techniques described in section 2.1.4 were developed, it was determined that at conditions with average peak pressures of roughly 800 kPa or lower all of the seals could be maintained for well over twenty (cumulative) minutes of engine operation. In contrast, for conditions when the average peak pressures approached roughly 1000 kPa, "blowout" of the window gaskets could occur. Having to replace these gaskets was highly undesirable because the blowby areas could change, which could introduce additional variability into the data. With the spark timing chosen below, through all of the experiments reported here (and more that are not), none of the engine seals required replacing and the sealing conditions remained essentially the same (see section 5.1).

At a spark timing of  $20^{\circ}$  bTC, the average peak pressure in Figure 3.11 is very nearly 800 kPa, and thus this spark timing was chosen. The small drop in net imep is accepted as the nature of this study does not require operation close to the maximum brake (or, really, indicated) torque spark timing, and furthermore during warmup operation of modern engines the typical strategy is to retard combustion – as is done here. The possibility of increased combustion variability is a tradeoff that is also accepted and later accounted for as best as possible (see Section 4.7).

## 3.3.4. Liquid deposition injection pressure and timing

A prototype direct injection (DI) fuel injector was selected as the method to deposit liquid fuel films in the combustion chamber. Because the goal was to deposit liquid fuel films on the walls, poor spray atomization was desired and the injector was operated at conditions very different than it was designed for and typically operated at. This, along with the fact that the desired injected fuel quantities were a fraction of the overall fueling, created challenges not anticipated when this method was selected.

In particular, it was found that for the lower injection pressures necessary both for poor atomization as well as to obtain the relatively low desired fuel quantities, the momentum of the fuel spray was so low that it could be heavily deflected by the intake flow. This deflection was problematic because the fuel was not deposited at its intended location, nor was the deposition location completely repeatable cycle-to-cycle as observed in high-speed videos.

Because the focus of this study was to look at the effect of liquid fuel films in the combustion chamber, the best possible deposition quality was desired. It was not obvious a priori what conditions produced the most suitable deposition as there were several competing effects. Consequently, the underlying physics of the injector spray behavior was examined in terms of the desired deposition characteristics. This analysis is presented below. Ultimately, it was determined that the particular DI injector chosen could be used for this study, and that with some tradeoffs acceptable and repeatable depositions could be obtained.

#### Injector Pressure Differential

Two constraints set the range of injector pressure differentials that were considered. First, the minimum pressure differential was determined from the injector functionality. In

bench tests, a pressure difference less than roughly 30 psi did not produce a conical spray – a narrow stream instead exited the injector. (In colloquial terms, it looked like the flow out of a garden hose that did not have a nozzle). A conical spray pattern was desired so as to deposit the film over a given area, and thus the minimum injector pressure differential considered was 30 psi.

Hardware limitations of the driver used to operate the injector set the minimum possible injection duration that could be used. With this constraint on the injection duration, the minimum desired injected fuel quantity determined the maximum injector pressure differential that could be used to obtain that particular mass of fuel per injection. For this specific hardware and a desired minimum injected mass of 10% of the stoichiometric fuel amount at the conditions described above, this maximum pressure difference was roughly 110 psi.

In order to select an injector pressure differential, the goals of the deposition and how they scale with the injector pressure differential were considered<sup>1</sup>. These goals are:

- 1. able to fit the injection into a portion of the engine cycle such that it is not affected by the in-cylinder flow
- 2. minimum vaporization from the spray while it is in transit to the wall
- 3. minimum rebounding from the wall (i.e. good sticking to the wall).

In order to fit the injection into a period of time in the cycle that has low flow, a small spray residence time in the combustion chamber was desired<sup>2</sup>. This residence time can be thought of in two components: the spray tip transit time, which is the time for tip of the spray to leave the injector and travel to the wall it is being deposited onto, and the injection duration, which is the total time the spray is exiting the injector. Their sum, assuming all parts of the spray travel at the

tip velocity, is the total time the spray is present in the combustion chamber before impinging upon the wall.

The spray tip transit time as a function of injector pressure differential was determined directly from high speed video of the injection, and corresponded well to values estimated from an extrapolation of previously reported [19] spray velocities using this injector at much higher injection pressures. The spray tip transit time as a function of injector pressure differential is plotted as the dotted line in Figure 3.12. The units of this time are crank angle degrees at 1000 RPM, the nominal engine speed for these experiments. (For reference, at 1000 RPM, 6 CAD = 1 ms).

Also plotted in Figure 3.12 is the sum of this transit time and the required injection durations (determined from the measured injector calibration) for various injected fuel quantities. As described above, this sum can be thought of as the total spray residence time in the combustion chamber. For reference, 3 mg is roughly 10% of the stoichiometric fuel mass for the conditions under consideration here.

The transit time dominates the residence time for all of the injection pressures, and the total residence time shows diminishing returns as injector pressure differential increases. Both the transit time and the injection duration scale inversely with the square root of the pressure difference.

The scaling of the other two deposition goals is considered next before returning to this plot in deciding upon an injector pressure differential to be used for the study.

First, however, a brief aside is necessary: for these subsequent scalings a sense of how the spray droplet diameters scale with the pressure drop across the injector is required. For this particular injector type (which is a pressure-swirl injector), as well as for simple pressure-jet

atomizer (i.e. a plain orifice) injectors, all characteristic spray diameters scale with spray exit velocity to some power, where that exponent ranges from -1.0 to -0.5 depending upon the particular correlation [25]. That is,

$$d_{characteristic} \sim v^{\alpha} \tag{3.1}$$

where  $\alpha = -1.0$  to -0.5 and  $d_{characteristic}$  is any of the characteristic spray diameters such as the SMD, DV90, etc, and *v* is the spray exit velocity. (For further reference on these characteristic diameters, see [25]). From basic fluid mechanics, the fluid velocity exiting the injector scales as the square root of the pressure drop across the injector, and thus

$$d_{characteristic} \sim \left(\Delta p_{injector}\right)^{\beta} \tag{3.2}$$

where  $\beta = -0.5$  to -0.25. This scaling is used below to look at how the desired deposition characteristics scale with injector pressure differential.

Vaporization from the spray while in transit to the walls can occur both by heat transfer and by mass transfer. For both of these modes, the mass vaporized from the spray  $m_{vaporized}$ while in transit to the walls can be cast as:

$$m_{vaporized} = h A_{spray} \begin{pmatrix} \Delta T \text{ or} \\ \Delta concentration \end{pmatrix} t_{transit}$$
(3.3)

where *h* is either the heat or mass transfer coefficient,  $A_{spray}$  is the total surface area of the spray,  $\Delta T$  and  $\Delta concentration$  are the temperature and concentration differences between the droplets and the surroundings, and t<sub>transit</sub> is the time the spray is in transit to the wall.

Both the Nusselt and Sherwood numbers for a sphere scale with the square root of the Reynolds number. Thus, both the heat and mass transfer coefficients scale with the square root of the velocity, or the fourth root of the pressure difference across the injector. That is,

$$h \sim v^{\frac{1}{2}} \sim \left(\Delta p_{injector}\right)^{\frac{1}{4}}$$
 (3.4)

The Sauter Mean Diameter, or SMD, is a useful characteristic spray diameter for examining spray vaporization. It is the diameter of a droplet whose volume to surface area ratio is simply the total volume of the spray divided by the total surface area of the spray. Using this definition and some basic algebra,

$$\frac{SMD}{6} = \frac{V_{spray}}{A_{spray}}.$$
(3.5)

For a given injected mass, the total spray volume is fixed since gasoline is incompressible. Thus, since the SMD is a characteristic spray diameter, for a fixed quantity of fuel injected:

$$A_{spray} \sim \frac{1}{d_{characteristic}} \sim (\Delta p_{injector})^{0.25 \text{ to } 0.5}.$$
(3.6)

For a spray traveling a fixed distance, the transit time scales inversely with velocity. Thus,

$$t_{transit} \sim \frac{1}{v} \sim \left(\Delta p_{injector}\right)^{-0.5}.$$
(3.7)

Combining all of these scalings we get that

$$m_{vaporized} \sim (\Delta p_{injector})^{0 \ to \ 0.25}$$
 (3.8)

This relatively weak dependence of the mass of fuel vaporized from the spray while in transit to the walls as a function of injection pressure for those values under consideration is plotted in Figure 3.13. The values plotted are normalized by the value at 30 psi. The two curves shown bracket the expected behavior given the exponent range in the characteristic diameter - injector pressure differential scaling. Despite the fact that at higher injection pressures there is less time for spray vaporization to occur, both the spray area and the heat and mass transfer coefficients increase as the pressure difference increases, resulting in the weak trend shown.

The two most critical dimensionless numbers used to study spray-wall interactions and spray sticking are the Weber number and the Ohnesorge number, with the former being the most critical [25][26][27][28]. The Weber number can be thought of as the ratio of the kinetic energy of a drop to its surface energy, and is defined as:

$$We = \frac{v^2 \rho_{liquid} d_{characteristic}}{\sigma}$$
(3.9)

where *We* is the Weber number, *v* is the fluid velocity (typically at the nozzle exit),  $\rho_{liquid}$  is the liquid density,  $d_{characteristic}$  is a characteristic diameter for the spray droplets, and  $\sigma$  is the liquid's surface tension coefficient. The Ohnesorge number can be thought of as the ratio of the internal viscous force in the droplet to its surface tension force, and is defined as:

$$Oh = \frac{\mu_{liquid}}{\left(\rho_{liquid} \sigma d_{characteristic}\right)^{0.5}}$$
(3.10)

where *Oh* is the Ohnesorge number,  $\mu_{liquid}$  is the liquid's dynamic viscosity, and the other parameters are as defined above.

For a given fluid, putting in the above scalings results in:

$$We \sim (\Delta p_{injector})^{0.5 \text{ to } 0.75}$$
$$Oh \sim (\Delta p_{injector})^{0.125 \text{ to } 0.25}$$
(3.11)

Both of these dimensionless numbers are plotted relative to their value at 30 psi injection pressure in Figure 3.14. As both dimensionless numbers increase, the amount of the spray sticking to the wall it impinges upon decreases [25][26].

In light of Figure 3.12, Figure 3.13, and Figure 3.14, there is no obvious choice for an injector pressure differential. As the pressure difference increases the residence time of the spray

in the combustion chamber decreases, but the quantity of fuel expected to stick to the wall decreases, and to a lesser extent the amount of fuel vaporized while the spray is in transit to the wall also increases.

Because the main goal of the injection was to deposit a liquid fuel film on the combustion chamber surfaces, a lower injector pressure differential was chosen so as to reduce spray rebounding and spray vaporization. The resulting tradeoff in terms of spray residence time was able to be accommodated – and as will be shown in the next section even with the highest injector pressure differentials under consideration the spray residence time in the combustion chamber was so long as to require deposition at one of two times in the cycle, at which the increased residence time required for the lower injector pressure differential could be accommodated.

A particular value of the injector pressure differential of 40 psi was chosen. It offered a slight reduction in residence time relative to the minimum value of 30 psi with a slight increase in rebounding and spray vaporization. Moreover, existing data was available for the spray behavior of this injector at roughly this injector pressure differential [19], which confirmed the spray was "well-behaved" at this condition (see Figure 2.8). And finally, this injector pressure differential is typical of port fuel injection, for which wall wetting is most certainly obtained and for which some literature existed for reference (e.g. [27] and [28]).

#### Injection Timing

From Figure 3.12, for the 40 psi injector pressure differential selected above and an injected fuel quantity of 6 mg (which is roughly 20% of the stoichiometric fuel mass at the conditions of Table 3.1 – slightly more than the 17% value ultimately used in the study), the total

time the spray is present in the combustion chamber before impinging upon a wall is roughly 50 crank angle degrees. (Even for the maximum possible injector pressure differential of 110 psi, this time is roughly 30 crank angle degrees – which, as will be evident below, does not offer a substantial benefit over 50 crank angle degrees.) The challenge in choosing the timing for the liquid deposition was to not have the intake flow "deflect" the spray while it was in transit to the walls so that it would actually be deposited at the intended location.

Figure 3.15 shows examples of sprays that are and are not deflected by the intake flow. The images in this table were extracted from high speed videos of the spray. The image at the top center of the table labels the various components of the combustion chamber. A mirror was used to provide an additional view of the spray: the left-hand portion of each image is the view through the mirror – in it the spray travels from left to right. The right-hand portion of each image is a direct view looking from the right side and up into the combustion chamber – in this view the spray travels from right to left. For the 19.9° and 39.8° after start of injection (SOI) images, the spray is visible in both views. The deflection of the spray is clearly evident when comparing the two sets of images.

Deflection of the spray is the result of momentum exchange between the in-cylinder flow and the spray. The details of this exchange depend upon the entire flow field in the combustion chamber. However, the momentum flow rate through the intake valve gap gives a sense of the momentum of the flow the fuel spray will encounter. (All of the targeting locations are in the clearance volume; the piston targeting is achieved with the piston near top center).

The momentum flow rate through the intake valve gap was estimated from the results of a WAVE cycle simulation. It is plotted in Figure 3.16. Momentum flow into the combustion chamber is positive, and momentum flow out of the combustion chamber is negative. This

momentum flow rate is the product of the mass flow rate through the valve and the valve gap flow velocity. The valve gap flow velocity was calculated from the mass flow rate through the valve, the valve open area, and an estimated density in the valve gap. The first two quantities were directly obtained from the simulation code; the latter was estimated from a compressible flow calculation assuming that the "upstream" location (either the intake port or the combustion chamber, depending upon the direction of the flow) was stagnant. That is, it was assumed that the upstream velocity was low relative to the valve gap velocity.

The typical spray residence time of roughly 50 crank angle degrees is shown to scale in Figure 3.16 for reference. If no part of the spray is to be deflected by the intake flow, the injection must be timed so that this entire 50 crank angle degree window occurs during a period in which the momentum flow rate is low. Thus, the deposition must either occur late in the exhaust stroke (with the injection starting prior to intake valve opening) or late in the intake event near intake valve closing.

There are drawbacks with both of these injection timing options. For the early injection timing, the injection is into hot exhaust gases and thus increased spray vaporization would be expected. For the late injection timing, "sooty" combustion was observed in the high-speed videos: secondary droplets that rebounded off the wall did not completely vaporize prior to combustion and created locally fuel rich inhomogeneities, which then resulted in soot formation upon flame passage. Figure 3.17 shows images taken from high speed videos 36° after spark for both the "early" and "late" deposition timings. A halogen lamp was used to illuminate the combustion chamber for this particular experiment: any chemiluminescence from the propagating flame is overpowered by the halogen lamp. As a result, for the early timing the

flame is not visible. However, for the late injection timing, soot particles are formed as the flame propagates and their radiation is so powerful that they over-exposured the image.

This soot formation was undesirable as it could possibly affect the unburned hydrocarbon emissions, which are the ultimate interest of this study. Further, any effect that these inhomogeneities had on the unburned hydrocarbon emissions could depend on local mixing and in-cylinder flow and thus be non-repeatable cycle-to-cycle. For later analysis, it was desired to have the behavior of the fuel that is not on the wall to be repeatable, well-behaved, and if possible completely vaporized.

Furthermore, due to geometric constraints of the targeting plate system, deposition of the liquid fuel on the piston required "early" (near top center) injection. It was also desirable to have the same time between deposition and combustion for all of the targeted locations.

For these two reasons, "early" liquid deposition late in the exhaust stroke was used for this study. The actual injection timing used was a middle-of-injection of 25° bTC-intake (the typical injection duration was 12°). This timing was chosen from examination of high-speed videos of the spray to ensure that no part of the spray was deflected by the intake flow: this was the latest possible timing at the end of the exhaust stroke for which the spray was not deflected. Referring to Figure 3.16, the spray is present in the combustion throughout the entire backflow period – the sink flow from the combustion chamber to the intake port apparently does not disrupt the fuel spray substantially.

This injection is with the exhaust valve open and the intake valve closed (see Table 2.1), and is into hot exhaust gases. The possibility of increased spray vaporization resulting from injection into the hot exhaust gas is an accepted tradeoff given the drawbacks of the alternative deposition timing. The injection did occur with the exhaust valve (slightly) open: a sweep of the

injection timing was performed with the liquid deposition targeted directly at the exhaust valve. No short-circuiting was observed with the chosen injection timing. Furthermore, as mentioned above, with this injection timing all of the secondary drops that rebounded from the wall appeared to have ample time to vaporize prior to combustion, and thus in the subsequent analysis the fuel that did not stick to the wall could be assumed to be completely vaporized.

Ultimately, despite some of the shortcomings and tradeoffs mentioned above, with this injection timing films of liquid fuel were able to be established on the combustion chamber walls – and more importantly those films were observed to have a direct impact on the unburned hydrocarbon emissions.

## 3.3.5. Fueling level, overall

The overall fueling level for data used in this study was fuel rich: specifically, the relative air-fuel ratio was less than 0.96 for any data points that were used. This restriction was primarily due to problems with the fast NDIR, from which the cycle-resolved burned gas relative air-fuel ratio ( $\lambda$ ) was calculated. The data from the analyzer could be used to determine  $\lambda$  only when it was sufficiently fuel rich (here, that limit was determined to be a relative air-fuel ratio less than 0.96). The methodology used for this determination, and the establishment of this particular limit on  $\lambda$ , is described in section 4.3.

An additional reason for and benefit of operating fuel rich was that the engine output is relatively flat when fuel rich. Thus, as discussed in section 3.3.1, any speed variations due to load variations that the motor controller could not compensate for were minimized.

A final benefit of operating fuel rich is that the possibility of post-flame oxidation of any fuel vaporized from the liquid films is reduced since there is negligible oxygen in the combustion

products. As a result, any increase in unburned hydrocarbon emissions due to fuel vaporized from liquid films would presumably be larger, and thus more easily measured and detected, than if it were reduced by post-flame oxidation.

#### 3.3.6. Initial combustion chamber metal temperatures

The baseline initial combustion chamber wall temperature used for this study was 30 °C. Ambient conditions could not be controlled for this particular setup: they changed day-to-day and within a particular day. (The ambient temperature for these experiments ranged from 20 to 24°C – see Table 5.1). Foremost, with this initial combustion chamber temperature choice, effects of liquid fuel films on hydrocarbon emissions were observed. That is, this initial combustion chamber wall temperature was not so high that injected liquid fuel had no impact on the hydrocarbon emissions. The choice of initial combustion chamber wall temperature was a compromise between the time required for the engine to cool between experiments and the size of the impact the liquid fuel had on the hydrocarbon emissions, but as mentioned above for these conditions an impact was observed. For this particular initial combustion chamber wall temperature and range of ambient conditions, roughly 75 to 90 minutes were required for the engine to cool between experiments. With this time between experiments, a full set of 8 to 10 experiments could be completed in a single (long) day, reducing the effect of ambient conditions on the data.

## 3.3.7. Number of cycles motored before firing

For these experiments, 151 cycles were motored prior to firing the engine. (The intention was for this value to be 150 cycles, but due to how the engine controller was setup it was actually

151 motored cycles.) As described earlier in section 3.3.1, this number of motored cycles allowed the engine speed, intake manifold pressure, and intake airflow to stabilize prior to firing. This motoring did result in the temperatures throughout the combustion chamber increasing somewhat, but was necessary in order to have the aforementioned quantities stable prior to firing the engine.

## 3.4. QUASI-STEADY EXPERIMENT SCHEME

Because the engine was not cooled, steady state experiments were not possible. The engine was warming up, i.e. temperatures throughout the engine were increasing, as an experiment was conducted. This natural warmup was accepted and used to assess the impact of combustion chamber wall temperature on the liquid fuel film behavior and the unburned hydrocarbon emissions resulting from the liquid fuel films.

The basic idea behind the experimental scheme was that, while the experiments were not conducted at steady state wall temperatures, over a small number of cycles the wall temperatures did not change significantly. Thus, fairly stable conditions could be established over a small window of cycles. It is for this reason the term "quasi-steady" is used to describe the scheme. By comparing windows of cycles that were spaced sufficiently far apart, the effect of wall temperature could be examined.

## 3.4.1. Desired Fueling Conditions

Table 3.2 shows the desired "stable" fueling conditions, and their sequence, in the scheme developed for this study. (Figure 3.18 shows the anticipated unburned hydrocarbon emissions behavior for each of these conditions; it will be discussed subsequently but can be

helpful to refer to in reading the description of each of these conditions.) In this scheme, the mass of liquid fuel sprayed at the combustion chamber walls in a given experiment was fixed – that is, it did not change as the "vapor" fueling level changed. A consequence of this is that the relative fractions of fuel delivered as a vapor and as a liquid did change slightly between the two liquid deposition conditions. However, maintaining the same injected liquid mass irrespective of the vaporous fuel quantity facilitated the examination of the liquid film behavior and (possible) oxidation differences between different deposition locations, as well as ensured that any small errors in the injector calibrations did not introduce artifacts into the data. Thus, to restate for emphasis, these experiments are conducted at "constant mass of liquid sprayed at the walls" and <u>not</u> at "constant fraction of fuel delivered as a liquid".

For the experiments in this study, unless noted otherwise, the mass of liquid fuel injected each cycle was roughly 17% of the stoichiometric fuel amount at the conditions under consideration (see Table 3.1; the actual injected mass of liquid gasoline was 5.00 mg).

The values used for the various fueling levels in Table 3.2 were not at all arbitrary. Condition 1 was vaporous only fueling at relative air-fuel ratio ( $\lambda$ ) of 0.95. For  $\lambda$  greater than 0.96 the data obtained from the fast CO and CO2 analyzer could not be used to accurately determine the cycle-resolved burned gas  $\lambda$  (see section 4.3), and the engine controller was able to maintain a desired fueling level to within ±0.01 units of  $\lambda$  (see section 2.1.7). Thus, condition 1 is the least rich condition for which usable cycle-resolved data could be obtained.

Condition 2 is generated simply by maintaining the vaporous fueling level of condition 1 and additionally injecting the liquid fuel targeted at the combustion chamber walls. The quantity of liquid fuel was chosen so as to be as large as possible while not resulting in an overall delivered  $\lambda$  for this condition that was "too rich" so as to be unrealistic for a gasoline engine. In

other words, the mass of liquid sprayed on the combustion chamber walls each cycle was chosen to be as large as possible given the aforementioned constraints on the "least rich" and "most rich" conditions to be considered.

Condition 3 is a vaporous only fueling condition that brackets condition 2 on the "more rich" side.

Conditions 4 and 5 are similar to conditions 1 and 2 in that condition 5 is produced by maintaining the vaporous fueling of condition 4 and additionally injecting the liquid fuel. The specific fueling levels for conditions 4 and 5 were chosen so as to provide separation between the overall delivered  $\lambda$ 's for the two liquid deposition conditions – so that if there was an effect of  $\lambda$  (for example on postflame oxidation) it could be identified. The engine speed was slightly less stable when operating lean (see section 3.3.1). Thus, for condition 4, a vaporous-only  $\lambda$  of 1.03 was determined to be a reasonable compromise for which the engine speed was relatively stable, but that provided some separation between the overall delivered  $\lambda$  values for the two conditions with liquid deposition (conditions 2 and 5).

Because it was fuel lean, cycle-resolved burned gas  $\lambda$  could not be determined for condition 4 (see above and section 4.3). For this reason, data from condition 4 was not used in order to examine the emissions impact of liquid fuel films. Data during the transition between condition 4 and condition 5 was, however, usable: during the transition the burned gas  $\lambda$  was less than 0.96, and thus cycle-resolved data could be obtained.

While the ultimate interest was in the conditions with liquid fuel deposition, the vaporous-only conditions served several purposes. First, they provided an opportunity within the experiment to adjust the vaporous fueling (via feedback control) in order to obtain the actual desired vaporous fueling levels. Second, they provided a means of verifying the emissions

behavior of the engine when all of the fuel was vaporized. And finally, the transitions between the vaporous-only conditions and the conditions with liquid deposition provided useful information about how much fuel from the depositions was actually ending up on the combustion chamber walls.

The logic behind this sequence and the specific fueling levels is perhaps more apparent when viewed graphically. Figure 3.18 shows the anticipated unburned hydrocarbon emissions versus the burned gas relative air-fuel ratio ( $\lambda$ ) for each of the conditions described above. The lower gray line indicates an expected linear trend in the unburned hydrocarbon emissions with vaporous-only fueling (e.g., see Fig 11.2 in [29]). The upper gray line is a hypothetical trend for the unburned hydrocarbon emissions with liquid fuel films present in the chamber: the unburned hydrocarbon emissions are expected to increase with liquid fuel films present, and this is shown schematically as a direct upward shift of the vaporous-only trend.

Thus, to recap and referring to Figure 3.18: condition 1 involves vaporous-only fueling at the "least rich" conditions for which a cycle-resolved burned gas relative air-fuel ratio can be determined given the limitations of the instrumentation. To this fueling, a mass of liquid fuel that is roughly 17% of the stoichiometric fuel mass is added, creating condition 2 – which is not so rich as to be unrealistic or unfeasible for a gasoline engine. This mass of liquid was chosen so as to be as large as possible given these two constraints on the "least rich" and "most rich" conditions to be considered. Condition 3 then operates with vaporous-only fueling at a level that is slightly richer than condition 2, bracketing it. The data from condition 4 is not usable on a cycle-by-cycle basis, but by adding to it the same mass of liquid fuel as above, condition 5 is created. The positioning of conditions 4 and 5 were determined so that condition 4 was not so lean as to have the engine speed unstable while still having some separation between conditions

2 and 5 so that any effects of different relative air-fuel ratios on the emissions contribution of the liquid fuel films could be identified. It is the data from conditions 1, 2, 3, and 5 that are used subsequently to examine the impact of liquid fuel films on unburned hydrocarbon emissions; condition 4 is necessary but not useable to look at on a cycle-by-cycle basis.

#### 3.4.2. Implementation

In practice, for this engine at the operating conditions and deposited liquid fuel quantities under consideration, after 20 cycles of liquid deposition fairly stable conditions were reached. Thus, each liquid deposition was performed for 30 cycles: the last 10 cycles were considered "stable" and used to examine the impact of liquid fuel films on the unburned hydrocarbon emissions.

Vaporous-only fueling was likewise performed for 30 cycles. A minor nuance is that, since the vaporous only fueling stabilized very rapidly to the desired  $\lambda$  value, when immediately following a liquid deposition the first 10 of these 30 cycles were performed at the previous vaporous fueling level. This facilitated the examination of the "clean out" of the liquid fuel film: since the vaporous fuel amount was fixed, any excess amount of fuel observed as leaving the combustion chamber could be attributed to the liquid fuel films. Then, in the next ten cycles (cycles 11-20) the fueling was transitioned to the desired  $\lambda$  value. The last ten cycles (cycles 21-30) were "stable" and used to look at the unburned hydrocarbon emissions without liquid fuel films present.

An additional nuance was that, since during the transition to lean operation the exhaust relative air-fuel ratio as measured by the UEGO stabilized more slowly (suspected to be due to engine speed instabilities and UEGO mode switching), operation at condition 4 was given more

time to stabilize. This was made possible by positioning it in the sequence so that it did not occur after a liquid deposition and thus it could transition to the desired  $\lambda$  value over the first 20 cycles: the last 10 cycles were stable.

In the testing, it was determined that with a maximum of roughly 45 seconds, or about 350 cycles at this engine speed, of firing the engine sealing could be maintained – see section 2.1.4. Therefore in the actual experiments, each of the conditions in Table 3.2 could be performed twice.

Finally, in order to get the engine stabilized under firing conditions, 30 cycles of vaporonly fueling were performed before commencing the fuel schedule described above. These 30 cycles are labeled as "Batch 0" in the subsequent plots; data from these 30 cycles were not used in the processing of the results as the engine was not yet stabilized.

The net result of this fueling scheme, then, is ten stable cycles at each of the desired fueling conditions – with some spacing between them, which implicitly builds in wall temperature variation. And furthermore, the vaporous fueling level is "locked" from 10 cycles before to 10 cycles after each liquid deposition which facilitates examination of the liquid fuel film buildup and decay.

## 3.4.3. Typical data

Typical data for the stable cycles of an experiment using this fueling scheme is shown in Figure 3.19 to Figure 3.21. The "batch" labels at the top of Figure 3.19 and Figure 3.20 identify the appropriate condition in Table 3.2: Batches 1-5 correspond to conditions 1-5, and Batches 6-10 are then a repeat of conditions 1-5. Condition 4 data is not shown because, as discussed above, its burned gas relative air-fuel ratio cannot be determined on a cycle-by-cycle basis.

The symbol color in these figures indicates whether the data is with (shown in red) or without (shown in black) liquid deposition. The shape of the polygon indicates the sequence of the data in time: the greater the number of sides, the later in the experiment and thus the higher the wall temperatures.

Figure 3.19 shows the cycle-resolved exhaust burned gas relative air-fuel ratio versus cycle number for a typical experiment; Figure 3.20 shows the cycle-resolved mass-averaged exhaust unburned hydrocarbon concentration versus cycle number for the same experiment. These plots are neither useful nor interesting on their own other than to show the degree of scatter in the so-called "stable" conditions that were established.

However, when the exhaust unburned hydrocarbon concentration is scattered versus the relative air-fuel ratio as in Figure 3.21, a collapse is seen. At a given relative air-fuel ratio, the exhaust hydrocarbon emissions are higher when liquid films are present in the combustion chamber (section 4.7.2 establishes that it is valid to interpolate between the vaporous only data). Furthermore, as wall temperatures increase (i.e. as time in the experiment increases) the size of that impact appears to decrease.

The production of this scatter plot was the motivation behind the development of this experimental scheme. The interpretation of plots such as this one, and the extraction of the impact of the liquid fuel films from them, is addressed subsequently in section 4.7.

The goal of this study is to examine these impacts on unburned hydrocarbon emissions both as a function of the liquid fuel film location and the wall temperature the liquid films are subjected to. First, however, the methodology used for this analysis is discussed in the next section before presenting those results.

 $^{2}$  Even at the highest injection pressures considered, the spray was still substantially deflected by the intake flow. For this reason, the challenge of not having the spray deflected by the in-cylinder flow became one of fitting the injection into a period of time in the cycle when the flow in the combustion chamber is low.

<sup>&</sup>lt;sup>1</sup> Attempts to explicitly calculate the spray behavior were challenging due to unknown spray characteristics and uncertainty in the gasoline properties. Using bounds for both of them resulted in the behavior bridging multiple spray type and wall interaction regimes; for this reason the scaling of relevant parameters was instead investigated.

Engine speed	976 RPM			
Intake pressure	55 kPa			
Port (isopentane) injection pressure and timing	70 psia rail pressure			
	$SOI = 20^{\circ} aTC$ -expansion			
Spark timing	20° bTC			
Liquid deposition (via direct injector) injection	55 psia rail pressure			
pressure and timing	$MOI = 25^{\circ} bTC$ -intake			
Initial metal temperatures (at start of motoring)	30° C			
Motored cycles prior to initiating firing	151			
Fueling level, overall	Fuel rich, $\lambda \leq 0.96$ for all data used			

Table 3.1: Summary of operating conditions used (unless otherwise noted)

Batch numbers in	Condition	Vanan 2	Liquid	Overall	
implementation		vapor $\lambda$	Deposition	Delivered $\lambda$	
1,6	1	0.95		0.95	
2, 7	2	0.95	YES	0.83	
3, 8	3	0.80		0.80	
4,9	4	1.03		1.03	
5, 10	5	1.03	YES	0.87	

Table 3.2: Fueling sequence for the quasi-steady liquid deposition experimental scheme

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## **Product Information**

#### FAX: (281) 457-1469

Johann Haltermann, Ltd.

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PRODUCT: EPA TIER II EEE FEDERAL REGISTER		Batch No.: YC2621LT10							
PRODUCT CODE:	FEDERAL REGISTER			т	ank No ·	107			
PRODUCT CODE:	<u>nr437</u>			Tank No.:		2/27/2010			
Analysis Date: 3/2//20									
TERT	METHOD	UNITS			Snore	DESULTS			
11231	METHOD	UNITS	MIN	TADOLT	MAY	RESULTS			
Distillation IPD	ACTH DOG	• <u>r</u>	75	TARGET	05	07			
	ASTMIDOO	- -	13		90	0/			
5%			120		125	111			
10%			120		135	125			
20%						147			
30%						1/1			
40%			000		000	199			
50%		P P	200		230	221			
60%		F				233			
70%		-F				244			
80%		-F				265			
90%		0F	305		325	316			
95%		°F				339			
Distillation - EP		°F			415	398			
Recovery		vol %		Report		97.2			
Residue		vol %		Report		1.1			
Loss		vol %		Report		1.7			
Gravity	ASTM D4052	°API	58.7		61.2	59.5			
Density	ASTM D4052	kg/l	0.734		0.744	0.742			
Reid Vapor Pressure	ASTM D5191	psi	8.7		9.2	9.1			
Carbon	ASTM D3343	wt fraction		Report		0.8647			
Carbon	ASTM E191	wt fraction		Report		0.8604			
Hydrogen	ASTM E191	wt fraction		Report		0.1357			
Hydrogen/Carbon ratio	ASTM E191	mole/mole		Report		1.879			
Stoichiometric Air/Fuel Ratio				Report		14.624			
Oxygen	ASTM D4815	wt%			0.05	<0.01			
Sulfur	ASTM D5453	wt%	0.0025		0.0035	0.0031			
Lead	ASTM D3237	g/gal			0.01	<0.01			
Phosphorous	ASTM D3231	g/gal			0.005	< 0.0001			
Silicon	ASTM 5184	mg/kg			4	<1			
Composition, aromatics	ASTM D1319	vol %			35	28			
Composition, olefins	ASTM D1319	voi %			10	1			
Composition, saturates	ASTM D1319	voi %		Report		71			
Particulate matter	ASTM D5452	mg/l			1	1			
Oxidation Stability	ASTM D525	minutes	240			1000+			
Copper Corrosion	ASTM D130				1	la			
Gum content, washed	ASTM D381	mg/100mis			5	<0.5			
Fuei Economy Numerator/C Densit	ASTM E191	_	2401		2441	2418			
C Factor	ASTM E191			Report	I	0.9967			
Research Octane Number	ASTM D2699		96.0			96.8			
Motor Octane Number	ASTM D2700			Report		88.5			
Sensitivity			7.5			8.3			
Net Heating Value, btu/lb	ASTM D3338	btu/lb		Report		18497			
Net Heating Value btu/lb	ASTM D240	btu/ib		Report		18439			
Color	VISUAL			Report		Undved			
L	n/n								
APPROVED BY:	- front	$\square$	ι	AN	IALYST	PLI/PH/DIX/GP			

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Figure 3.1: Specification sheet for the gasoline used in the study.



Figure 3.2: Distillation curve for the gasoline used in the study



Figure 3.3: Schematic of fuel pathway tracking via carbon tracking



Figure 3.4: Implementation of fuel pathway tracking via carbon tracking



Figure 3.5: Fuel pathway deduction from the measurement of carbon-containing species.


Figure 3.6: Cycle-resolved engine speed for motoring and 150 fired cycles



Figure 3.7: Intake manifold pressure at intake valve closing versus cycle number and time for a typical experiment



Figure 3.8: Intake airflow versus cycle number and time for same experiment as Figure 3.7



Figure 3.9: Scatter plot of net imep and exhaust HC emissions versus port injection timing



Figure 3.10: Net imep and exhaust HC emissions versus batch number (i.e. time in experiment) for the data of Figure 3.9.



Figure 3.11: Engine parameters versus spark timing, using only isopentane



Figure 3.12: Spray residence time (tip travel time + injection duration) versus injector pressure differential for various injected fuel quantities



Figure 3.13: Spray vaporization relative to spray vaporization at 30 psi injector pressure differential, for the range of exponents in the characteristic diameter versus injector pressure differential scaling



Figure 3.14: Weber and Ohnesorge numbers relative to their values at 30 psi injector pressure differential, for the range of exponents in the characteristic diameter versus injector pressure differential scaling



Figure 3.15: Sample images of sprays that are and are not deflected by the intake flow, for the "head and upper liner" targeting



Figure 3.16: Momentum flow rate through the intake valve gap



Figure 3.17: Images 36° after spark for "early" liquid deposition (left) and "late" liquid deposition (right). With "late" deposition, blackbody radiation from soot particles is visible; chemiluminescence from the flame is overpowered by a halogen lamp that is illuminating the combustion chamber. (See text for further explanation.)



Figure 3.18: Anticipated unburned hydrocarbon emissions versus burned gas relative air-fuel ratio for the conditions of Table 3.2



Figure 3.19: Typical exhaust burned gas relative air-fuel ratio versus cycle number, for stable cycles only. Cycles with liquid deposition are shown in red; those without liquid deposition are black.



Figure 3.20: Typical exhaust mass-averaged hydrocarbon emissions versus cycle number, for stable cycles only and the same experiment as Figure 3.19. Cycles with liquid deposition are shown in red; those without liquid deposition are black.



Figure 3.21: Exhaust mass-averaged hydrocarbon emissions versus exhaust burned gas relative air-fuel ratio with and without liquid deposition, for the data of Figure 3.19 and Figure 3.20. Cycles with liquid deposition are shown in red; those without liquid deposition are black.

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### 4. ANALYSIS METHODOLOGY

This chapter focuses on processing, using, and interpreting the raw species concentration data in the exhaust and volume beneath the piston in order to determine the liquid fuel film behavior and its impact on the unburned hydrocarbon emissions. These methods were used to produce the results in Chapter 5.

#### 4.1. EQUIVALENT FUEL MOLECULE

In these experiments, a mixture of two fuels (gasoline and isopentane) was used. Subsequent calculations can be simplified and approximations can be made by defining an "equivalent" molecule for the mixture. This molecule has the same mass- <u>and</u> molar-based properties as the binary mixture.

Consider a mixture of two different fuels: mass  $m_1$  of  $C_{a_1}H_{b_1}$  and mass  $m_2$  of  $C_{a_2}H_{b_2}$ . The moles of each in the mixture can be computed as:

$$n_i = \frac{m_i}{MW_i} \tag{4.1}$$

where  $n_i$  is the number of moles of species *i* and  $MW_i$  is the molecular weight of species *i*. The mole fraction of each fuel in the mixture is thus:

$$\bar{x}_{1} = \frac{n_{1}}{n_{1} + n_{2}}$$

$$\bar{x}_{2} = \frac{n_{2}}{n_{1} + n_{2}}$$
(4.2)

The carbon and hydrogen content for the equivalent fuel molecule  $C_{a_{equiv}} H_{b_{equiv}}$  is given by:

$$a_{equiv} = \overline{x}_1 a_1 + \overline{x}_2 a_2$$
  

$$b_{equiv} = \overline{x}_1 b_1 + \overline{x}_2 b_2$$
(4.3)

For the same total mass of the equivalent molecule  $C_{a_{equiv}} H_{b_{equiv}}$  as the binary mixture (=  $m_1 + m_2$ ), the total number of moles of the equivalent molecule are the same as the total number of moles of the binary mixture (=  $n_1 + n_2$ ). Therefore, this equivalent fuel molecule has the same massand molar-based properties as the binary mixture (e.g. the stoichiometric air fuel ratio or the molecular weight). Thus, it can (and will) be used in subsequent calculations instead of handling the mixture components separately<sup>1</sup>.

The major utility of using the equivalent fuel molecule is that it greatly simplifies the calculations. Moreover, it simplifies the process of looking at the effect of varying fuel proportions in the mixture and permits approximations to be made. In fact, because of the specific fuels chosen in this study and the proportions of them that were used, the effect of changes in the mixture proportions on the calculations is small.

The molecular weight of the gasoline used in the study was not available. It was assumed to have a typical [29] molecular weight of 110 g/mol; for its reported carbon to hydrogen ratio its equivalent fuel molecule was thus  $C_{7.911}H_{14.865}$ . Isopentane is  $C_5H_{12}$ . Using the methodology above, a mixture of 20% gasoline and 80% isopentane has an equivalent molecule of  $C_{5.410}H_{12.404}$ . This mixture composition is a slight upper bound on the gasoline content used in the actual experiments when gasoline was added to the isopentane.

The impact of this change in fuel composition, from 100% isopentane to 20% gasoline and 80% isopentane, on both the experiments and the subsequent calculations, is small. Where appropriate these impacts are mentioned below.

#### 4.2. CYCLE-RESOLVED MASS-AVERAGED EXHAUST SPECIES CONCENTRATIONS

The fast gas analyzers measure molar species concentrations: in order to interpret and use this data it is desirable to convert these concentrations to masses. The mass of a given species can be determined from the integral of the product of the mass flow and the species' mass-based concentration. The challenge in performing this integration is determining the <u>appropriate</u> flow and mass-based concentrations, and then integrating over the <u>appropriate</u> period of time. The approach used here was to first determine a mass-averaged (molar) exhaust concentration which could then be used in subsequent calculations.

Flow in the exhaust runner only occurs during the exhaust event, and thus during most of the engine cycle the fast gas analyzers in the exhaust runner are sampling stationary gas. The first step in using the fast gas analyzer signals is the identification of the portion of the concentration data corresponding to the period of exhaust flow. There is both a flow delay to the analyzer sampling point and a transit time delay within the analyzer which must be accounted for in order to properly phase the signals.

The flow delay was estimated by the engine simulation code WAVE for this particular sampling location and experimental conditions, and was roughly 1 crank angle degree. (Recall that the engine speed is 976 RPM; see section 3.3.1.) The transit delay in the analyzer was estimated using software provided by the manufacturer. In practice, this estimated transit delay must be manually adjusted due to minor variations in probe geometry as a result of the machining processes, as well as due to simplifying assumptions made in the analysis software [30]. The values for the transit delays of each of the fast gas analyzers in this study agreed very well with those of another investigator [31] who used the exact same hardware and analyzer settings. Figure 4.1a shows a properly phased exhaust hydrocarbon concentration profile.

In order to compute the mass-averaged exhaust species concentration for each cycle, the mass flow rate during the exhaust event must be estimated. Two common approaches to do this are either to use a cycle simulation code to generate an average or typical exhaust flow profile, or to estimate the flow profile using compressible flow based upon the in-cylinder and exhaust port pressures. The former was difficult to implement for this setup because of the increased blowby mass loss in this engine which could not easily be modeled in the simulation code. The latter requires very accurate knowledge of the pressures on either side of the exhaust valve, the charge temperature (in order to determine its density) and moreover the discharge coefficients for the exhaust valve and port system. These quantities were challenging to determine as well, due to both the blowby mass loss and the "nonstandard" valve and port configuration.

The approach used here to estimate the exhaust mass flow, although simplified, could easily be implemented on a cycle-by-cycle basis and more importantly was not affected by the unknown blowby mass loss. This approach was to assume isentropic processes for the gases that stay in the combustion chamber during the exhaust event. That is, heat transfer to/from the gases is neglected, during blowdown the gases that stay in the combustion chamber are assumed to undergo free expansion, and during the displacement portion of the exhaust event the gases are assumed to be reversibly displaced.

The major advantage of this approach is that it does not require knowledge of the state of the charge at exhaust valve opening, and moreover that it only requires cylinder pressure and volume data. Though some of the assumptions made above are simplified, this method most importantly provides a means of appropriately scaling or weighting the concentration profile by its corresponding (even if approximate) mass flow. This method was found to result in very good agreement with other direct measurements and calculated consistency checks

for this dataset.

Since there is negligible blowby flow during the exhaust event (the valve open area is much greater than the blowby flow area), all of the mass lost from the combustion chamber flows out the exhaust valve. By determining the change in the mass in the combustion chamber each crank angle, the exhaust flow rate can be estimated. In particular, for the isentropic process from one crank angle to the next,

$$m_{i} = m_{i-1} \left(\frac{p_{i}}{p_{i-1}}\right)^{\frac{1}{\gamma}} \left(\frac{V_{i}}{V_{i-1}}\right)$$
(4.4)

where *m* denotes the mass in the combustion chamber, *p* denotes the cylinder pressure, and *V* denotes the cylinder volume. The subscript *i* refers to a given crank angle; the subscript *i*-1 refers to the previous crank angle.

The exhaust mass flow rate  $\dot{m}_{exh}$  can then be estimated as

$$\dot{m}_{exh} = \frac{m_i - m_{i-1}}{\text{time per crank angle}}$$
(4.5)

The important thing about this calculation is that it does not depend on the cylinder mass at exhaust valve opening – all subsequently calculated masses are expressed as a fraction of it. A typical plot of this estimated mass flow is shown in Figure 4.1b. Roughly half of the mass is exhausted during the blowdown process and half is exhausted during displacement flow. The dimensions of this flow are not shown since, as discussed above and below, it is only the shape of this curve that matters in order to determine the mass-averaged species concentration. And furthermore, for emphasis, any uncertainty in the blowby mass loss in these experiments does <u>not</u> affect this estimated flow rate, and thus does <u>not</u> affect the calculated mass-averaged emissions concentrations.

Ultimately, for using and working with the data, a cycle-resolved mass-averaged species concentration is desired. This concentration  $\bar{x}_{\text{species k}}^{\text{mass average}}$  (which is on a molar basis) is given by

$$\bar{x}_{species k}^{mass average} = \frac{\int\limits_{EVO}^{EVC} \bar{x}_{species k}(t) \dot{m}_{exh}(t) dt}{\int\limits_{EVO}^{EVC} \dot{m}_{exh}(t) dt}$$
(4.6)

where the integrand in the numerator is the product of the molar species concentration profile and the exhaust flow rate, the denominator is just the net mass exhausted that cycle, and the integration is performed from exhaust valve opening to exhaust valve closing. This molar-basis concentration is later converted to a mass-based concentration using the ratio of molecular weights of the given species and the exhaust gas.

From the above equations, it can be seen that the cylinder mass at EVO is a constant multiplier in both the numerator and denominator, and thus does not affect the calculation. It is for this reason that the exhaust flow profile (which is the integrand in the denominator of the above equation) was normalized to have integral 1 as shown in Figure 4.1b.

The product of the species concentration and this normalized exhaust flow rate is shown in Figure 4.1c. It is the integrand in the numerator in the above equation, and since as stated above the integral of the denominator is 1, the integral of the curve in Figure 4.1c is the massaveraged exhaust species concentration for that particular cycle.

The utility of this mass-averaged concentration is that it removes the need to perform integration of the flow in subsequent calculations and assigns a single appropriately averaged value to the species concentration each cycle – it can then be used in a straightforward way in the calculations. For example, the mass of each species exhausted in a particular cycle is given by:

$$m_{\rm speciesk}^{\rm exhausted} = \overline{x}_{\rm speciesk}^{\rm mass\ average} \frac{MW_{\rm speciesk}}{MW_{\rm exh}} m_{exh}$$
(4.7)

where the  $MW_i$ 's are the respective molecular weights and  $m_{exh}$  is the mass exhausted that cycle. In the actual analysis, the exhaust burned gas relative air-fuel ratio was used to determine its composition, and then its molecular weight on a cycle-by-cycle basis.

## 4.3. BURNED GAS RELATIVE AIR-FUEL RATIO FROM FAST EXHAUST NDIR MEASUREMENTS

Ultimately, one quantity that is desired to aid in the interpretation of the data is the mass of fuel burned each cycle. If one trusts the exhaust CO and  $CO_2$  concentration measurements, the mass of fuel that was burned each cycle can directly be determined from them using the above equation and a simple chemical reaction equation.

However, problems were encountered with these experiments when using this particular fast CO and CO<sub>2</sub> analyzer sampling the exhaust gases. It is believed this was due to the temperature (greater than 500 °C) of the exhaust gases. The reported concentrations were greater than expected, and at times greater than physically possible, when sampling the exhaust gases. When sampling ambient temperature calibration gas blends, however, the analyzer exhibited very good accuracy. Figure 4.2 shows a plot of the expected and reported concentration behavior for exhaust gas sampling.

This discrepancy was a <u>major</u> challenge in the study. The precise cause of the problem was never determined, but a means to use the data from the analyzer that was both physicallybased and consistent with other measurements was devised. As stated above, it is suspected that the discrepancy encountered with the analyzer is due to sampling high temperature gases.

The possible causes of this discrepancy, in which the exhaust concentrations were higher than expected, can be thought of as being of two types. The first type are those causes that would scale both the CO and CO<sub>2</sub> concentrations in the same way – for example, condensation or more mass than expected being drawn into the sample chamber (the analyzer responds to the absolute mass of species in its sample chamber: if more mass than expected was drawn in, the reported concentration would be higher). The second type would affect the CO and CO<sub>2</sub> differently. The most likely cause of this type of discrepancy is infrared interference from another species present in the exhaust gas. The possibility of this interference was tested offline (at ambient temperature) by sampling numerous species present in the exhaust gas, and the infrared interference was insignificant.

Thus, it appeared the most likely effect was that the CO and  $CO_2$  concentrations were scaled in the same way. Whatever the source of this scaling, using the <u>ratio</u> of the two concentrations in the subsequent analysis would eliminate it. In fact, using the ratio of the massaveraged exhaust CO and CO<sub>2</sub> concentrations to determine the burned gas relative air-fuel ratio resulted in excellent agreement with other measurements and calculations.

However, in order for the ratio of the CO and CO<sub>2</sub> concentrations to accurately determine the relative air-fuel ratio the mixture must be sufficiently rich (e.g., see Fig 4-20 in reference [29]). For these experiments, the method that follows was determined to be valid for relative airfuel ratios of 0.96 or less, as will be shown below. This limitation dictated the experimental conditions that could be examined in order to obtain cycle-resolved data.

Assuming rich combustion, there is negligible oxygen and hydrocarbons in the burned gases. Moreover, any species from the crevices will simply dilute the CO and  $CO_2$  concentrations – and thus the ratio of the exhaust CO and  $CO_2$  concentrations is the same as the

ratio of the burned gas CO and  $CO_2$  concentrations. Therefore, the chemical reaction in the burned gases, neglecting any oxides of nitrogen, can be cast as:

$$C_a H_b + \lambda \left(a + \frac{b}{4}\right) \left(O_2 + \beta N_2\right) \rightarrow n_p \left(\overline{x}_{CO} CO + \overline{x}_{CO_2} CO_2 + \overline{x}_{H_2} H_2 + \overline{x}_{H_2O} H_2 O + \overline{x}_{N_2} N_2\right)$$

where  $\beta$  is the proportion of nitrogen to oxygen in atmospheric air (typically taken as 3.773), n<sub>p</sub> is the total number of moles of the products, and the  $\bar{x}_i$ 's are the mole fractions of the respective species. The goal is to determine the burned gas relative air-fuel ratio  $\lambda$  from this equation.

There are 7 unknowns in this equation:  $\lambda$ , n<sub>p</sub>, and the 5  $\bar{x}_i$ 's. And there are 7 equations available: 4 atom balances (on C, H, O, and N), the fact that the sum of the  $\bar{x}_i$  mole fractions must be 1 by definition, the water-gas shift reaction equilibrium constant, and the measured ratio of the exhaust CO and CO<sub>2</sub> concentrations. (A typical value for the water-gas shift equilibrium constant of 3.6 was used for these calculations; varying it among typical values has a negligible impact on the value of  $\lambda$  as will be evident below.) Since there are 7 equations and 7 unknowns this system of equations can be solved for  $\lambda$ . The closed-form solution can be expressed as:

$$\lambda = \frac{2(1+2R)(1+KR) + \frac{a}{b}KR(1+R)}{(1+R)(1+KR)(4+\frac{a}{b})}$$
(4.8)

where R is the ratio of the exhaust CO<sub>2</sub> concentration to the exhaust CO concentration  $(R = \frac{\bar{x}_{CO_2}}{\bar{x}_{CO}})$ and K is the value of the water-gas shift equilibrium constant. As mentioned above, the concentrations used for this analysis were the mass-averaged ones.

The impact of the fuel molecule's carbon to hydrogen molar ratio (= a/b), and thus the impact of the varying mixture compositions in these experiments, on the computation of lambda is negligible. This is a further advantage of this method for determining the burned gas relative air-fuel ratio. Figure 4.3 shows a plot of the relative air-fuel ratio versus the ratio of the CO to

 $CO_2$  concentrations (note that in the notation used above the x-axis is 1/R, not R). The relative air-fuel ratio is plotted for two different fuel H-C molar ratios: that of 100% isopentane and that of 100% gasoline. For this change in mixture composition, the impact on the calculated air-fuel ratio is negligible; in the actual experiments the mixture composition ranges from 100% isopentane to roughly 80% isopentane and 20% gasoline, and thus its impact on the calculation of the relative air-fuel ratio is even smaller than plotted in Figure 4.3.

Figure 4.4 shows a scatter plot of the burned gas relative air-fuel ratio as computed from the above method versus the relative air-fuel ratio reported from the UEGO. The agreement is excellent – the scatter is due to the fact that the UEGO has a slow response. This establishes the consistency and validity of this method of determining the cycle-resolved relative air-fuel ratio.

As a further consistency check, Figure 4.5 shows 3 different relative air-fuel ratios: the delivered estimate (computed from the injector calibration, engine speed, and intake airflow each cycle), the cycle-resolved burned gas relative air-fuel ratio computed from the fast CO and CO<sub>2</sub> data, and the exhaust relative air-fuel ratio as reported by the UEGO. As mentioned above and shown below, the  $\lambda$  computed from the CO to CO<sub>2</sub> ratio is not valid for  $\lambda > 0.96$ . Furthermore, the delivered  $\lambda$  estimate is just that: an estimate. It is calculated from several quantities and a high degree of accuracy is not expected. However, the shape of and trend in the delivered estimate agrees very well with the other  $\lambda$ 's.

In particular, not all of the fuel injected in a given cycle is consumed in that cycle (due to displacement into the intake port at intake valve closing and liquid fuel film dynamics) and thus the burned gas  $\lambda$  determined from the cycle-resolved CO and CO<sub>2</sub> data lags behind the delivered  $\lambda$  estimate. Furthermore, the "slow" response of the UEGO is evident as it lags behind the cycle-resolved lambda as determined from the CO and CO<sub>2</sub> concentrations.

Finally, Figure 4.6 shows the discrepancy between the relative air-fuel ratio  $\lambda$  as computed from the CO to CO<sub>2</sub> ratio and the relative air-fuel ratio as reported by the UEGO versus the UEGO value. The UEGO reading is valid over the entire  $\lambda$  range shown in the plot but is not fast enough for cycle-level resolution, thus the scatter. From this plot (and many others like it), the computation of  $\lambda$  from the CO to CO<sub>2</sub> ratio appears to begin to be invalid near  $\lambda > 0.96$ . It is for this reason that the limit of  $\lambda \le 0.96$  was imposed for this method of determining the cycle-resolved burned gas  $\lambda$  from the cycle-resolved CO and CO<sub>2</sub> concentrations to be valid.

#### 4.4. FUEL PATHWAY MASS ACCOUNTING

As described in section 3.2, to quantify the behavior of the liquid fuel films, the approach used here is to track all of the fuel: by knowing the amount of fuel delivered as a vapor, deductions can be made about the liquid fuel films. Figure 3.5 shows the various pathways that fuel (liquid or vapor) can follow in a given cycle. The fast gas analyzers in the exhaust runner and the volume beneath the piston measure the <u>concentrations</u> of carbon-containing species in each location. The challenge addressed in this section is the conversion of these concentrations to the corresponding masses of fuel burned and unburned that are either exhausted or lost from the combustion chamber in the blowby gases, on a cycle-by-cycle basis.

Numerous castings of such a methodology were attempted: the one described here worked very well and provided very good agreement with consistency checks both within a given experiment and between all of the experiments.

The mass of fuel burned and exhausted in a given cycle can be expressed as

$$m_{burned}^{exhaust} = \frac{m_{air}^{exhaust}}{\lambda \left(A/F\right)_{s}}$$
(4.9)

where  $m_{arr}^{echaust}$  is the mass of inducted air atoms that are exhausted each cycle,  $\lambda$  is the relative air-fuel ratio, and (A/F)<sub>s</sub> is the stochiometric air-fuel ratio. All quantities on the right hand side of this equation are either known with cycle-level resolution or change negligibly cycle-to-cycle, and thus  $m_{burned}^{echaust}$  can be evaluated on a cycle-by-cycle basis.  $m_{arr}^{echaust}$  can be estimated from the measured blowby flow rate. Recall from section 2.1.4 that the blowby mass flow rate was measured to be approximately 9% of the intake air flow rate under the fired conditions under consideration. Thus,  $m_{arr}^{echaust}$  is 91% of the trapped intake air mass at intake valve closing. The trapped air mass can be estimated from the intake air flow rate and engine speed. (The intake air flow rate and engine speed are essentially constant for the cycles under consideration here – see section 3.3.1). It should be noted that the impact of changing fuel compositions on the above calculation, from 100% isopentane to 80% isopentane and 20% gasoline, is small: the term  $\frac{1}{(d E)_{a}}$ changes by 0.9%. The burned gas relative air-fuel ratio  $\lambda$  is known for each cycle as described in the previous section, and therefore the mass of fuel that was burned and exhausted for a given cycle can be evaluated directly from equation 4.9.

The mass of fuel exhausted as unburned hydrocarbon emissions can be expressed as

$$m_{unburned}^{exhaust} = \bar{x}_{CH_{y_{x}}}^{exhaust, mass average} \frac{MW_{CH_{y_{x}}}}{MW_{exhaust}} \left( m_{air}^{exhaust} + m_{unburned}^{exhaust} \right)$$
(4.10)

where  $\bar{x}_{CH_{\frac{v}{s}}}^{exhaust,mass average}$  is the unburned hydrocarbon molar fraction as reported by the fast FID that has been mass-averaged as described in section 4.2. Because the FID counts carbon atoms in hydrocarbon molecules, the concentration is expressed on a "per carbon atom" basis. The ratio of molecular weights above converts this mole fraction to a mass fraction. For each cycle the exhaust gas composition was determined from the cycle-resolved burned gas relative air-fuel ratio and then used to determine the (wet) molecular weight of the exhaust gases  $MW_{exhaust}$ . The term in parentheses is the total mass exhausted each cycle, neglecting the mass of unburned fuel (which constitutes a negligible fraction of the total exhaust mass). The hydrogen to carbon ratio  $({}^{y}/_{x})$  of the unburned hydrocarbons was assumed to be the same proportion as the fuel. Any variation in this ratio (due to either mixture composition changes or the actual hydrocarbon emissions having a different hydrogen to carbon proportion than that of the fuel) is relatively insignificant on this mass calculation since carbon is roughly 12 times heavier than hydrogen. Thus, equation 4.10 provides a means for determining, for each cycle, the mass of fuel that was exhausted as unburned hydrocarbon emissions.

The above two equations provide a means of determining the mass of burned and unburned fuel that was exhausted each cycle. Next, the determination of the mass of burned and unburned fuel lost from the combustion chamber each cycle due to blowby is addressed.

As described in section 2.1.5, a ventilator device was designed to facilitate the tracking and measurement of species in the blowby gases. The mass of unburned fuel lost to the volume beneath the piston each cycle via blowby can be estimated from:

$$m_{unburned}^{blowby} = \int_{cycle} \bar{x}_{CH_{y'_{x}}}^{ventilator} \frac{MW_{CH_{y'_{x}}}}{MW_{air}} \rho \frac{dV}{d\theta} d\theta$$
(4.11)

This equation states that the net mass of unburned hydrocarbons exiting the combustion chamber each cycle is simply the integral over the entire cycle of the product of the measured concentration (converted to a mass basis) and the flow rate. Figure 2.6 shows for the ventilator a typical hydrocarbon concentration profile, an "ideal" flowrate, and their product (which is the integrand of equation 4.11) graphically. As above, the unburned hydrocarbons are assumed to have the same hydrogen to carbon ratio as the fuel; the impact of changing fuel composition on the mass calculation is, likewise, small as well. The bulk of the gas in the volume beneath the piston is assumed to be air given the amount of air used to refill the volume each up stroke and that most of the unburned mixture is air. Thus, the molecular weight of air is used to convert the measured molar fraction to a mass fraction. The  $\rho \frac{dV}{d\theta}$  term is the mass flow rate at the sample point, expressed per crank angle degree.

A first approximation to estimating this flow rate is to assume instantaneous reed valve opening and closing at top and bottom center (see section 2.1.5 for information on the ventilator design) and incompressible flow. With these assumptions, the flow rate is simply determined from the piston motion. In generating this estimated flow rate, the density  $\rho$  was assumed to be constant and taken as the value of air at the measured average ventilator exit temperature.

Analogous to equation 4.11 above for the unburned hydrocarbons, the mass of CO and  $CO_2$  lost from the combustion chamber each cycle via the blowby gases can be determined. Then, the mass of fuel that was burned and lost from the combustion chamber each cycle via the blowby gases is given by:

$$m_{burned}^{blowby} = \frac{1}{a} M W_{C_a H_b} \left( \frac{m_{CO}^{blowby}}{M W_{CO}} + \frac{m_{CO_2}^{blowby}}{M W_{CO_2}} \right)$$
(4.12)

where the fuel molecule is  $C_aH_b$ . The term in parentheses is number of moles of carbon burned. When it is multiplied by 1/a it is the number of moles of fuel molecules  $C_aH_b$  burned, which when multiplied by its molecular weight is the mass of fuel molecules burned.

In practice, however, these masses of burned and unburned fuel in the blowby gases each cycle as computed with the above assumptions were too large. The possible sources of this

discrepancy are that the concentration profile is incorrect and/or that the assumed flow is incorrect. The concentration profile could be incorrect due to improper signal phasing (because the concentration and flow rate have very similar shapes their product is very sensitive to phasing) or due to the response time of the instruments. The assumed flow could be incorrect because of compressibility effects (at maximum piston speed for these conditions the Mach number at the sample point is approximately 0.4) or because of "non-ideal" reed valve behavior (e.g. both sets of valves being open or both being closed at some point). The assumed density could also be incorrect, and it is in fact incorrect when the piston is moving up and drawing in fresh air – but during this time the concentrations are low since the analyzers are sampling fresh air, so this is not thought to have a significant impact.

A very simple adjustment to the masses computed using the above assumptions that provided excellent agreement and consistency was simply to scale their values by a factor of 0.5. While this is rather crude, the proof that it is reasonable is that as the fueling levels changed within an experiment, for stable fueling conditions the net mass exiting the combustion chamber each cycle is very close to the mass of fuel injected each cycle – which is to be expected when the fueling is stable.

Figure 4.7 shows, for ten stable cycles, the mass of fuel to each of the above pathways as computed by the methodology in this section, along with the mass of fuel injected each cycle. The agreement is excellent: the conditions are stabilized so the net mass of fuel exiting the combustion chamber each cycle agrees very well with the mass of fuel entering the combustion chamber each cycle (i.e. the amount injected).

Figure 4.8 shows the same data as Figure 4.7, but for <u>all</u> of the stable fueling cycles within a particular experiment. The agreement is excellent as fueling levels change – the

maximum difference between the injected fuel quantity and the total mass of fuel exiting the combustion chamber is 2.4%, which occurs around cycles 232 to 241. In these cycles liquid fuel is being deposited and it is believed that the liquid fuel film is still (slowly) building up and thus the conditions are not truly stabilized – therefore the mass of fuel exiting the combustion chamber is not quite equal to the mass of fuel entering. (The discrepancy in the first ten cycles plotted is believed to be due to the fact that the engine speed, airflow, and intake pressure are not completely stabilized – see section 3.3.1).

Two separate fuel injectors were used in the engine, and thus the value of the injected fuel mass in Figure 4.7 and Figure 4.8 is the sum of two independent quantities (the mass of fuel injected from each of the two fuel injectors). Over a wide range of conditions, the net mass of fuel exiting the combustion chamber (as computed from this methodology) agrees with the sum of these two independently varying quantities – which validates this methodology.

For reference, Figure 4.9 shows the same data as Figure 4.8, but with the mass of fuel to each pathway expressed as a percent of the injected fuel. Roughly 91% of the fuel molecules are exhausted, with most of them being burned. The remaining roughly 9% of the fuel molecules are lost from the combustion chamber in the blowby gases, with most of them being unburned – which is to be expected since it is mostly unburned mixture that constitutes the blowby gas as described in section 2.1.5 and Table 2.2.

The results of the methodology outlined in this section will be used subsequently to normalize the results and to make deductions about the amount of liquid fuel sticking to the wall, as will be described below.

#### **4.5. WALL FILM MASS ESTIMATION**

The transients that occur when liquid deposition begins and ends contain useful information about the amount of liquid fuel sticking to the wall. The first way this transient can be used is to deduce the amount of liquid fuel on the wall when stable deposition conditions are reached.

Figure 4.10 shows the cycles before, during, and after a sample liquid deposition. The vertical axis is the total mass of fuel molecules exiting the combustion chamber (the sum of the 4 pathways described above) each cycle. The horizontal axis is the cycle number: the ten cycles before up to the ten cycles after the liquid deposition are shown. (Recall from section 3.4 that these cycles without liquid deposition are all performed at the same vaporous fueling level, as indicated by the lower dashed line.)

When stable conditions are reached, the additional mass of fuel that is exiting the combustion chamber each cycle is by definition the amount of additional fuel injected as a liquid each cycle. (Otherwise the conditions would not be stable.) The stable amount with liquid deposition is indicated by the upper dashed horizontal line.

The mass of fuel on the wall under stable deposition conditions can be estimated from both the buildup profile when liquid deposition starts and the decay profile when liquid deposition ends. Looking at the buildup as indicated in Figure 4.11, when stable conditions are reached some total amount of additional fuel was injected as a liquid. However, some amount less than this – the sum of the "extra" masses above the vapor amount each cycle – has exited the combustion chamber. The difference between these two amounts – the amount that was injected minus the amount that has exited the combustion chamber – is the amount of fuel that is left on

the combustion chamber walls. By computing this difference when the liquid deposition has stabilized, the mass of fuel on the wall during the stable deposition cycles can be estimated.

Likewise, looking at the decay transient as shown in Figure 4.12, when the liquid deposition ends the total amount of fuel <u>in excess of the vapor amount</u> that exits the combustion chamber is another estimate of the amount of fuel that was on the combustion chamber walls during the stable liquid deposition cycles. This excess fuel that is observed is precisely the liquid fuel films "cleaning out". A minor nuance is that not all of the excess fuel consumed in the very last cycle of liquid deposition was on the wall (some of the injected liquid rebounds off the wall or vaporizes in transit to the wall) as will be described in the following section – this has a minor impact on the calculation and this effect was included in the calculations.

In implementing these calculations, the total mass of fuel exiting the combustion chamber each cycle was computed using the results of the previous section. Because the results of this methodology were somewhat "noisy", each numerical sum was terminated when two consecutive cycles were within 5% of the mass of liquid fuel injected each cycle and "excess" values that were negative were discarded.

In performing these calculations, the residual gas effect on the transient was neglected. From a WAVE cycle simulation, the residual fraction (on a mass basis) at the conditions of these experiments is roughly 10%. The impact of neglecting this effect is to <u>slightly</u> overestimate <u>both</u> of the methods for estimating the wall film mass.

In practice, the mass of fuel on the wall as computed from the decay transient was larger than that computed from the buildup transient, typically by 15-20%. This is believed to be due to the fact a larger fraction of fuel from each deposition sticks to the wall when impinging upon the wetted surface of the later cycles, and a smaller fraction of fuel sticks to the wall during the

initial depositions since the surface is dry. This effect would result, after the initial transient, in a very slow rise in the stable "with liquid deposition" level that can not accurately be observed in these numerical calculations.

Because it actually corresponds to the stable liquid deposition cycles, the value computed from the decay transient is subsequently used as the "mass of fuel on the wall" for those cycles.

A final nuance in the implementation of this estimate of the wall film mass is that, due to the limitations on the ability to determine the cycle-resolved burned gas relative air-fuel ratio (see section 4.3), only the "buildup" data from the second and fourth depositions in the quasisteady experiment scheme (see section 3.4) could be used to estimate the wall mass. These depositions were onto a lean overall vapor fueling. The end-of-deposition "decay" wall mass was estimated for these depositions by scaling up the "buildup" masses to account for the fact that the decay masses were larger as described above. However, as described in section 3.3.1, the engine speed is slightly unstable when operating fuel lean and transitioning to fuel rich operation, and the nature of these calculations accumulates errors rapidly. For all of these reasons, there is more uncertainty in the wall film masses for the second and fourth liquid depositions, as will be evident in the subsequent results presentation.

It should be emphasized that while the determination of these wall masses is somewhat approximate in nature, it provides a very useful means for looking at differences in the unburned hydrocarbon emissions data as will be evident subsequently.

# 4.6. ESTIMATE OF THE FRACTION OF EACH LIQUID INJECTION THAT STICKS TO THE WALL

A second way that the data during the transient that occurs when liquid deposition begins and ends can be used is to estimate the fraction of each liquid injection that actually sticks to the combustion chamber walls. Stated somewhat loosely, the previous section looked at the integral of the transient, whereas this section looks at the details of the shape of the transient.

It should be emphasized that the methodology in this section is very sensitive to errors in the cycle-resolved "net mass of fuel exiting the combustion chamber". In a situation in which these values are known more accurately, for example when blowby is negligible and thus the mass of fuel exiting the combustion chamber is dominated by the burned fuel (which is uniquely determined by the cycle-resolved burned gas relative air-fuel ratio), this methodology may be more accurate and useful. Nonetheless, the trends observed with this dataset (reported in section 5.7) are consistent with the expected behavior and other observed trends.

Not all of the liquid fuel that is sprayed at a combustion chamber wall will actually stick to the wall, and moreover the fuel that does stick to the wall will leave the wall over multiple subsequent cycles. This is shown schematically in Figure 4.13: liquid fuel was injected in cycle 1. The horizontal axis is the cycle number; the vertical axis the amount of this injected liquid fuel that exited the combustion chamber each cycle. For this figure, the vertical axis is normalized by the amount of liquid fuel injected in one cycle. That is, the sum of the amounts exiting the combustion chamber in Figure 4.13 is 1. As described in the previous section, any injected fuel that did not yet exit the combustion chamber remains on the combustion chamber walls.

In Figure 4.13, fuel that was consumed from the wall is shown with dark shading; the light shading for a portion of the cycle 1 amount represents fuel from the injection that did <u>not</u> stick to the wall. This "fuel that did not stick to the wall" consists of fuel that vaporized in transit to the wall and fuel that rebounded from the wall and subsequently vaporized, which will exit the combustion chamber in the same cycle as the injection and as stated above is represented

by the lightly shaded portion of the cycle 1 amount. The task of this section is thus the estimation of the size of this lightly shaded portion of the cycle 1 amount in Figure 4.13, which represents the fraction of injected fuel that did <u>not</u> stick to the wall. The fraction of injected fuel that <u>did</u> stick to the wall is simply 1 minus this number.

Assuming that the profile of Figure 4.13 is known, the fraction of fuel sticking to the wall can be estimated. (Later in this section, a means of estimating this profile from the experimental data is described.) As stated above, graphically this amounts to determining the size of the lightly shaded bar in cycle 1 of Figure 4.13.

Let s<sub>i</sub> denote the net mass of fuel exiting the combustion chamber in cycle i for the <u>single-injection</u> profile with injection in cycle 1 (such as the one in Figure 4.13). Also, for generality let S<sub>inj</sub> denote the mass of liquid fuel injected. As described above the amount injected in Figure 4.13 is 1 unit, so for it S<sub>inj</sub> = 1 unit. Note that by definition  $\sum s_i = S_{inj}$ .

As described above, for cycle 1 <u>only</u> some fraction  $\alpha$  of the fuel that exited the combustion chamber did <u>not</u> stick to the wall. (It either vaporized in transit to the wall or rebounded from the wall and vaporized.) Thus, the total amount of fuel on the wall from the injection is:

$$S_{wall} = (1 - \alpha)s_1 + \sum_{i=2}^n s_i = S_{inj} - \alpha s_1$$
(4.13)

That is, it is all but the amount that did not stick to the wall, which is  $\alpha s_1$ . The fraction  $\beta_{\text{stick}}$  of the injection that stuck to the wall is simply:

$$\beta_{stick} = \frac{S_{wall}}{S_{inj}} = 1 - \frac{\alpha s_1}{S_{inj}}$$
(4.14)

The challenge, then, is determining  $\alpha$ . A first bound on it can be obtained from the physical argument that the amount of fuel consumed from the wall must decrease each cycle (because the fuel film is shrinking). Graphically, this means that the dark shaded bars in Figure 4.13 must decrease in size each cycle. As a direct consequence of this, the amount of fuel consumed from the wall in cycle 1 must be greater than the amount of fuel consumed from the wall in cycle 2. Written symbolically this is:

$$(1-\alpha)s_1 \ge s_2 \tag{4.15}$$

Graphically, this equation states that the height of the darkly shaded region in cycle 1 must be <u>at</u> <u>least</u> as high as the cycle 2 amount. From equation 4.15, a <u>maximum</u> value of  $\alpha$  can be obtained. Namely,

$$\alpha_{\max} = 1 - \frac{s_1}{s_2}$$
(4.16)

This provides a means of estimating the <u>minimum</u> value of the fraction of fuel sticking to the wall using equations 4.16 and 4.14. (When  $\alpha$  is maximum, S<sub>wall</sub> and  $\beta_{stick}$  are minimum.)

Furthermore, estimated values of  $\alpha$  can also be obtained from the s<sub>i</sub> profile. Graphically, this amounts to using the later cycles to extrapolate back to cycle 1 to determine the height of the darkly shaded region in cycle 1 of Figure 4.13. In practice this can be challenging due to noise in the data and the high degree of accuracy required for these calculations. For example, a linear extrapolation from the second and third cycles back to the first cycle results in:

$$\alpha_{estimate} = \frac{s_1 - 2s_2 - s_3}{s_1}$$
(4.17)

From this estimated value of  $\alpha$  and equation 4.14, an estimate of the fraction of the injection that sticks to the wall can be determined.

The preceding analysis assumed that the profile for the behavior of a single injection was known. The task of what follows is the determination of this profile from the transient data that occurs when liquid deposition begins and ends in the quasi-steady experiment scheme of section 3.4.

The following assumptions are made in this analysis: first, the residual gas effect on the transient is neglected. Neglecting this effect greatly simplifies the accounting, and is justified given the approximate nature of these calculations. (For reference, the residual gas fraction under the conditions of these experiments is  $\sim 10\%$  on a mass basis.)

Second, linear superposition of the behavior of a single injection is assumed to be valid. That is, the behavior of many injections is assumed to be the sum of the response of the individual injections, and all of the injections are assumed to have the same behavior. This assumption is what enables this analysis approach to be applicable to the case where fuel is injected in consecutive cycles.

<u>Aside</u>: For this engine operating at the conditions of section 3.3, experimental data supports this linear superposition assumption. (A caveat is that this supporting data was obtained with "poor" engine sealing and with an injected liquid amount of roughly 8 mg per cycle compared to the 5.00 mg injected per cycle for the results of Chapter 5. Nonetheless it supports the notion that the assumption of linear superposition is reasonable.) This data is shown and discussed in Appendix A.

Figure 4.14 shows the result of linearly superposing the (arbitrary) profile of Figure 4.13 for injection in 8 consecutive cycles. As stated above, all of the injections are assumed to behave

identically. Each color tracks the contribution of a given cycle's injection to the total amount exiting the combustion chamber each cycle. What is actually observed exiting the combustion chamber is the total amount each cycle (graphically, the tops of the stacked bars) – the assumption of linear superposition enables the contribution of each injection to the total to be identified.

Recall that the amount of fuel injected each cycle for Figure 4.14 is 1 unit. When the fueling conditions stabilize, the amount of fuel exiting the combustion chamber each cycle is precisely the amount of fuel injected each cycle. (This is, after all, the definition of the conditions being stable.) For Figure 4.14, stable fueling is reached for cycles 5 through 8.

Through careful subtraction of the <u>total</u> amounts of injected liquid fuel exiting the combustion chamber each cycle (which is what the experimental data provides), the shape of the profile for a single injection (i.e. the s<sub>i</sub>'s in the notation used earlier in this section) can be deduced as described below.

Let  $S_i$  denote the <u>total</u> amount of injected liquid fuel exiting the combustion chamber in a given cycle of Figure 4.14. Graphically, these  $S_i$ 's are the total heights of the stacked bars in Figure 4.14. In particular, note that the <u>capital</u>  $S_i$ 's are the sum of <u>lower-case</u>  $s_i$ 's – which are components of the profile of a <u>single</u> injection that we are trying to determine from the <u>known</u>  $S_i$ 's.

Looking at the buildup data, if the first cycle of liquid injection is index 1 for the  $S_i$ 's,

$$s_1 = S_1$$
  

$$s_2 = S_2 - S_1$$
  
and in general for  $i \neq 1$ ,  $s_i = S_i - S_{i-1}$   
(4.18)

Likewise, looking at the decay that occurs when liquid deposition ends, another estimate of the single-injection profile (i.e. the  $s_i$ 's) can be obtained. In particular, if the index "cut" is

used to identify the cycle in which liquid deposition ended (for Figure 4.14, that cycle is cycle 9):

$$s_1 = \mathbf{S}_{inj} - \mathbf{S}_{cut}$$
  

$$s_2 = \mathbf{S}_{cut} - \mathbf{S}_{cut+1}$$
  
and in general for  $i \neq 1$ ,  $s_i = \mathbf{S}_{cut+i-2} - \mathbf{S}_{cut+i-1}$   
(4.19)

Thus, the buildup and decay data that occur when liquid deposition in consecutive cycles begins and ends can directly be used to deduce the behavior of a single injection (with some assumptions). To emphasize, it is the assumption that the profiles do not change and that they add linearly that enables this subtraction to be performed in order to estimate the single-injection profile. The methodology described earlier in this section can then be applied to this singleinjection profile in order to estimate the fraction of injected liquid fuel that actually sticks to the wall.

#### 4.7. UNBURNED HYDROCARBON EMISSIONS DUE TO LIQUID FUEL FILMS

Figure 3.21 shows and section 3.4.3 discusses the typical scatter plot of exhaust hydrocarbons versus relative air-fuel ratio for stable cycles with and without liquid deposition. At a given relative air-fuel ratio, the exhaust hydrocarbon emissions are higher when liquid fuel films are present, and the size of this impact appears to decrease as wall temperatures increase. The focus of this entire section is the interpretation of this scatter plot and the extraction of information from it.

### 4.7.1. Mechanisms of Unburned Hydrocarbon Emissions

To aid in this process, it is helpful to first consider the mechanisms by which fuel molecules are able to escape combustion and become exhausted as unburned hydrocarbon emissions. These mechanisms have been previously identified by Cheng et al [32]. They are summarized below, with some slight adaptations:

Mechanisms involving fuel only

- oil layers

- deposits

- liquid fuel films

Mechanisms involving unburned fuel-air mixture

- quench layers
- crevices
- exhaust valve leakage

The oil layer and deposits mechanisms both involve fuel that is sorbed (as in absorbed or adsorbed) into them and subsequently desorbed. When this desorption occurs after flame passage, the desorbed fuel can be exhausted as unburned hydrocarbon emissions.

The liquid fuel films mechanism involves liquid fuel present on the combustion chamber surfaces (but not sorbed into the oil layer or deposits), and is the focus of this study. Any fuel from these films that escapes combustion can subsequently be exhausted as unburned hydrocarbon emissions.

The quench layer and crevice mechanisms involve unburned fuel-air mixture that escapes combustion because the flame does not reach them. And like the above mechanisms, if this fuel escapes the combustion chamber it can become unburned hydrocarbon emissions.

Finally, the exhaust valve leakage mechanism involves unburned fuel-air mixture that is exhausted due to the exhaust valve not closing completely.
In this study, there is no lubrication oil (the piston sealing is achieved via graphite bars – see section 2.1.4). Moreover, there are virtually no deposits: the combustion chamber is cleaned after each experiment, each of which contains roughly 45 seconds of firing. Thus, the only "fuel only" emissions mechanism in these experiments is that of the liquid fuel films.

The exhaust valve leakage mechanism is an abnormality and not considered here, as there was no evidence to suggest it was occurring. Thus the only "unburned mixture" mechanisms in these experiments are that of the quench layers and crevices.

The unburned fuel emissions sources that are present in these experiments can thus be recast: the quench layer and crevice mechanisms <u>affect fuel in vapor form</u>, and the liquid fuel films mechanism by definition <u>affects the liquid fuel films on the walls of the combustion</u> <u>chamber</u>. Thus, when there is no liquid deposition only these "vapor" mechanisms are present; when there is liquid deposition both these "vapor" and "liquid" mechanisms are present. In comparing the exhaust hydrocarbon emissions with and without liquid fuel films present, it is essential to consider how these underlying mechanisms may change as fueling conditions change. This is done subsequently in section 4.7.3. First, however, the behavior of the engine with vapor-only fueling is addressed.

#### 4.7.2. Hydrocarbon emissions characterization with vapor-only fueling

In order to infer the impact of the liquid fuel films, the behavior of the engine when operating on vapor-only (i.e. isopentane) fueling was characterized. Ultimately, a means of <u>predicting</u> the exhaust hydrocarbon emissions for a particular cycle due to fuel in vapor form was devised, which is used subsequently to infer the contribution of the liquid fuel films to the exhaust hydrocarbon emissions.

Experiments were conducted at the conditions of section 3.3 with vapor-only (i.e. isopentane-only) fueling in which the fueling level was rapidly varied within a range of relative air-fuel ratios. The purpose of rapidly varying the fueling level was to obtain data over a range of wall temperatures at the fueling levels of interest, and moreover to establish the validity and robustness of the method for predicting, on a cycle-by-cycle basis, the hydrocarbon emissions due to vaporous fuel.

Figure 4.15 shows the cycle-resolved burned gas and UEGO-measured relative air-fuel ratios versus cycle number for such an experiment. The "slow" response of the UEGO is evident in Figure 4.15 – it is shown as a consistency check on the cycle-resolved relative air-fuel ratio. Figure 4.16 shows the cycle-resolved mass-averaged exhaust hydrocarbon emissions versus cycle number for the same experiment. Figure 4.17 is a scatter plot of the cycle-resolved hydrocarbon emissions versus the cycle-resolved relative air-fuel for this same data.

There is a linear trend in the exhaust hydrocarbon emissions with relative air-fuel ratio, with some scatter. Examination of the data shows that *most* of this scatter is due to variation in the combustion phasing, which can be accounted for as described below.

A physically-based predictive fit provided excellent agreement for the data of Figure 4.17, and other similar data using vapor-only fueling, and was able to account for much of the apparent scatter in the trend of Figure 4.17. This fit was motivated by the underlying mechanisms that cause the unburned hydrocarbon emissions mechanisms with vapor-only fueling, as described above. Most of the unburned hydrocarbon emissions due to vaporized fuel is from the crevices [32], and to a lesser extent the quench layer (which can conceptually be thought of as a thin crevice). At a given fuel-air mixture composition (i.e. a fixed relative air-fuel ratio), the amount of fuel in a particular crevice region that escapes combustion scales as

$$\frac{P_{flame arrival} V_{crevice}}{RT_{wall}}$$
(4.20)

where  $p_{flame\ arrival}$  is the cylinder pressure corresponding to flame arrival at that location,  $V_{crevice}$  is the volume of that crevice region, R is the gas constant for the fuel-air mixture, and  $T_{wall}$  is the crevice wall temperature. Here, the crevice gases are assumed to be at the local wall temperature.

In reality, the flame arrives at different crevice regions at different times and the wall temperatures of the different crevice regions are different. However, by using a spatially averaged wall temperature and an approximate average cylinder pressure corresponding to flame arrival at the crevices, a very good predictive fit was possible. In particular, a fit of the form

$$\left(\overline{x}_{HC}^{exhuast}\right)_{predicted} = \frac{\left(p_{\max}\right)^{0.5}}{T_{wall}^{average}} a_1 \left(a_2 - \lambda\right)$$
(4.21)

provided an excellent <u>prediction</u> of the exhaust hydrocarbon emissions for a particular cycle. Here, the square root of the maximum cylinder pressure  $p_{max}$  was used as a representative pressure at flame arrival at the crevices (little impact on the "goodness" of the fit was observed for exponents ranging from 0.4 to 0.6) and the value used for  $T_{wall}^{average}$  was simply the arithmetic mean of the 6 combustion chamber surface thermocouple (described in section 2.2.2) for that particular cycle. A linear trend in exhaust hydrocarbon emissions with relative air-fuel ratio  $\lambda$ was assumed. The average value of two fit parameters  $a_1$  and  $a_2$  were  $1.16 \cdot 10^5 \frac{ppmCl}{(kra^{0.5}/k)}$  and 1.095, respectively, for these experimental conditions. The coefficient of determination (R<sup>2</sup>) for this fit was 0.91. Figure 4.18 shows the actual and predicted (from the above fit) exhaust unburned hydrocarbon emissions versus cycle number for the data of Figure 4.16. Despite the simplifying assumptions and only two fit parameters, the agreement is excellent over a wide range of conditions. Figure 4.19 shows the same data as Figure 4.18, but with the actual and predicted hydrocarbon emissions scattered against one another. 95% of the points have a relative error less than 13.5% and an absolute error less than 350 ppmC1. The disagreement is largest for the "most rich" cycles ( $\lambda \sim 0.80$ ); however, conditions only as rich as  $\lambda \sim 0.825$  are actually encountered with liquid fuel films present in the subsequent data.

Thus, in summary, for this particular engine operating at the conditions of section 3.3, the unburned hydrocarbon emissions with vapor-only fueling are both well-behaved and predictable <u>on a cycle-by-cycle basis</u> based upon the cylinder pressure, combustion chamber wall temperatures, and the burned gas relative air-fuel ratio. At fixed pressure and temperature conditions, the unburned hydrocarbon emissions exhibit a very good linear trend in relative air-fuel ratio for relative air-fuel ratios in the range 0.80 to 0.96.

The predictive fit described here is used subsequently in section 4.7.4 to infer the impact of the liquid fuel films on the exhaust hydrocarbon emissions by "subtracting off" the contribution of vaporous fuel to the total exhaust hydrocarbon emissions when liquid films are present.

It should be noted that because of the relatively small number of cycles (10) that are averaged over for each "batch" of liquid deposition cycles (see section 3.4), in the absence of a method such as the one described here that accounted for the impact of cycle-to-cycle combustion variability on the unburned hydrocarbon emissions, the trends in the data of chapter 5 were not as clear.

# 4.7.3. Effects of Changed Fuel Composition on Exhaust Hydrocarbon Emissions Sources and Measurements

A final factor to consider when examining scatter plots such as Figure 3.21 is the extent to which differences in the measured exhaust hydrocarbon concentration and the corresponding mass of hydrocarbons are simply due to the fact that the overall fuel composition is different with and without liquid deposition. The extent to which both the underlying unburned hydrocarbon source mechanisms, as well as the mass of fuel that corresponds to a given reported analyzer concentration, change as the fueling scenario changes from "no liquid deposition" (isopentane only) to "with liquid deposition" (isopentane plus gasoline) are addressed below.

Recall from section 3.3.1 that the experiments are performed such that for the stable fired cycles of interest the mass of air trapped per cycle is essentially constant. In comparing a blend of 80% isopentane and 20% gasoline (which as described earlier is a bound for the gasoline content in the liquid deposition cycles) to 100% isopentane, at the same mass of air inducted and the same relative air-fuel ratio there is 1.7% more fuel mass trapped with the 80/20 blend. That is, for scatter plots such as Figure 3.21, at the same relative air-fuel ratio there was <u>slightly</u> more fuel trapped in the liquid deposition cycles of the plot as compared to the "no liquid deposition" cycles.

However, based upon the fuels' lower heating values, the energy content on a per unit mass basis of the 80/20 blend (again, corresponding to the liquid deposition cycles) is 1.0% <u>lower</u> than 100% isopentane. Therefore, in comparing the liquid deposition cycles to the cycles without liquid deposition in plots such as Figure 3.21, at the same relative air-fuel ratio the total trapped chemical energy differs only very slightly (~0.7%) between the two fueling scenarios. The implication of this is that the cylinder pressure history, and thus the fraction of vaporized

fuel that is forced into the crevices, at a given combustion phasing would be expected to change negligibly between the two scenarios. This "fuel in the crevices" is the source of hydrocarbon emissions due to fuel in vaporous form. Therefore, at a given relative air-fuel ratio and combustion phasing, the source mechanisms for the unburned hydrocarbons that are due to fuel in vaporous form change negligibly between the two fueling scenarios.

Furthermore, as described earlier in section 4.4, at a given relative air-fuel ratio as the fuel composition changes the mass of fuel that corresponds to a particular concentration changes only slightly as well. The error in converting, at a given relative air-fuel ratio, a particular concentration to its corresponding mass of unburned hydrocarbons is 3% for an assumed fuel composition varying from 100% isopentane to 100% gasoline. That is, when looking at the concentrations of scatter plots such as Figure 3.21, the <u>maximum</u> error in the mass of unburned hydrocarbons corresponding to a given concentration is 3%. Thus, in working with such scatter plots it is valid to work with the concentration data and later the methodology of section 4.4 can be used to convert these concentrations to masses.

Thus, it is valid to directly look at differences in scatter plots such as Figure 3.21 in order to determine the impact of liquid fuel films on hydrocarbon emissions. Any differences that are the result of the underlying change in the fuel composition that occurs as fueling changes from isopentane-only to a mixture of isopentane and gasoline in both the source of hydrocarbon emissions due to fuel in vaporous form or in the mass of fuel that corresponds to a given concentration are small. This is a consequence of the particular fuels (and their proportions) chosen to be used in this study.

Finally, a further assumption that is made in this analysis is that as fuel composition changes from isopentane to the mixture of isopentane and gasoline, the oxidation behavior of the

fuel molecules does not change "much". There is a critical temperature of roughly 1300-1500 K [33][34][35][36] above which most hydrocarbon molecules will oxidize, so this assumption is reasonable. Due to the nature of these experiments, if the oxidation behavior of the gasoline molecules was substantially different than the oxidation behavior of isopentane there is no (straightforward) way to determine or account for such a difference.

#### 4.7.4. Deduction of Hydrocarbon Emissions Due to Liquid Fuel Films

The previous sections established the validity of looking at differences in scatter plots such as Figure 3.21. The final step is to "subtract off" the hydrocarbon emissions due to vaporous fuel from the total hydrocarbon emissions when liquid fuel films are present so that the contribution of the liquid fuel films to the hydrocarbon emissions can be deduced. The critical issue in this subtraction is the composition at which the vaporous fuel emissions mechanisms are assumed to be. As described earlier, the source of these emissions can conceptually be thought of as "the crevices". Thus, this issue then becomes: when there is liquid deposition, at what composition are the crevice gases?

The answer to this question can be bounded by two extremes: first, the crevices could be at the same overall or average composition as the burned gases. Second, the crevices could be at the prevailing "vapor-only" or isopentane-only fuel-air mixture composition. The former is equivalent to assuming that all of the injected liquid fuel <u>that vaporized and burned</u> was uniformly distributed throughout the combustion chamber and crevices. The latter is equivalent to assuming that the injected liquid fuel <u>that vaporized and burned</u> did not mix – for example, it remained stratified near the fuel film – so that the crevice gases are at the isopentane fuel-air mixture composition.

These two different and bounding assumptions are shown graphically in Figure 4.20 and Figure 4.21. These figures are schematic versions of the actual exhaust emissions versus relative air-fuel ratio scatter plots. The horizontal axis in each of these figures is the relative air-fuel ratio; the vertical axis is the exhaust hydrocarbon emissions. The hydrocarbon emissions behavior with and without liquid fuel films present are shown schematically. Focusing on the liquid deposition condition labeled "2" in the figures, under assumption 1 above the crevices are assumed to be at the same composition as the burned gases. Therefore, the open circle that is marked on Figure 4.20 indicates the contribution to the unburned hydrocarbons from fuel in vaporous form under this assumption. This amount must be subtracted off of the total emissions in order to deduce the amount of emissions that are due to the liquid fuel films. Thus, the vertical line segment in Figure 4.20 indicates the contribution of the liquid fuel films to the unburned hydrocarbon emissions under assumption 1.

Likewise, under assumption 2 above the crevices are assumed to be at the "vapor-only" composition. Recalling how the experiments were conducted (see section 3.4), the underlying isopentane-only fuel-air proportions are known: the unburned hydrocarbon emissions at the corresponding vapor-only relative air-fuel ratio is shown in Figure 4.21 as the open circle around the point labeled "1". The vertical line segment in Figure 4.21 then indicates the deduced contribution of the liquid fuel films to the unburned hydrocarbon emissions under assumption 2.

While the actual situation lies somewhere in between these two bounding extremes, in order to subtract off the vapor contribution to the hydrocarbon emissions and deduce the contribution of the liquid fuel films, the first assumption was used for all of the results presented subsequently. It is plausible that the actual physical scenario is closer to this extreme given that a portion of the spray rebounds from the wall and that the liquid injection is done very early in

the cycle (see section 3.3.4). That this assumption is reasonable is shown subsequently in section 5.6.

Since the trend in the vapor-only emissions is linear, the impact of varying the assumption about the crevice gas composition would effectively be to add an offset to all of the deduced liquid fuel film contributions to the unburned hydrocarbon emissions – assuming the same behavior for all of the liquid fuel film locations. That is, the trends in the subsequent results plots showing the impact of the liquid fuel films on the hydrocarbon emissions would be unchanged by a changed assumption about the crevice composition. They would simply be shifted upward by some offset.

In order to assign a single value to the contribution to the exhaust hydrocarbons for each "batch" of stable liquid deposition cycles (see section 3.4 for more information on the experimental scheme), the vaporous fuel contribution to the total exhaust hydrocarbon emissions was determined on a cycle-by-cycle basis using the methodology of section 4.7.2. Then, for each cycle, this amount was subtracted from the actual measured mass-averaged exhaust hydrocarbon concentration in order to determine the contribution of the liquid fuel films to the exhaust hydrocarbon emissions <u>for that cycle</u>. These values were then averaged – this average value is plotted subsequently as the increase in hydrocarbon emissions due to liquid fuel films for each batch of stable liquid deposition cycles. By handling these calculations on a cycle-by-cycle basis, the impact of any combustion phasing variability is greatly reduced (since the methodology of section 4.7.2 implicitly accounts for combustion phasing).

These increases in hydrocarbon emissions due to the liquid fuel films are the focus of this study. The next chapter looks at these impacts, along with other relevant data, for liquid fuel films at various locations in the combustion chamber and various wall temperature conditions.

<sup>&</sup>lt;sup>1</sup> In fact, gasoline itself is a mixture of many components and an extension of this logic shows that it is valid to use a single "equivalent" molecule for representing it.



Figure 4.1: Typical (a) exhaust hydrocarbon concentration, (b) normalized exhaust flow, and (c) the product of the exhaust hydrocarbon concentration and normalized exhaust flow



Figure 4.2: Expected and actual meter reading mass-averaged exhaust CO &  $CO_2$  concentration behavior



Figure 4.3: Effect of fuel composition on the calculation of relative air-fuel ratio from the ratio of CO to  $CO_2$  concentrations (assuming fuel-rich combustion). Note that in the notation of equation 4.8 the horizontal axis is 1/R, not R.



Figure 4.4: Comparison of relative air-fuel ratio as calculated from cycle-resolved CO and CO<sub>2</sub> concentration ratio and as reported by the UEGO meter



Figure 4.5: Relative air-fuel ratio – delivered estimate, UEGO reading, and calculated from cycle-resolved CO and CO<sub>2</sub> concentrations.



Figure 4.6: Discrepancy between  $\lambda$  calculated from exhaust CO and CO<sub>2</sub> ratio and as reported by the UEGO versus the UEGO reading



Figure 4.7: Total mass of fuel exiting the combustion chamber via the various pathways and the mass of fuel injected each cycle (typical data for 10 stable fueling cycles)



Figure 4.8: Total mass of fuel exiting the combustion chamber via the various pathways and the mass of fuel injected each cycle (typical data for stable fueling cycles in one experiment) (Color coding is identical to Figure 4.7).



Figure 4.9: Percent of total fuel injected to each of the various pathways, for the same data as Figure 4.8.



Figure 4.10: Mass of fuel molecules exiting the combustion chamber for the cycles before, during, and after a sample liquid deposition.



Figure 4.11: The "buildup" method for estimating the mass of fuel on the wall



Figure 4.12: The "decay" method for estimating the mass of fuel on the wall



Figure 4.13: Schematic of amount of injected liquid fuel exiting the combustion chamber versus cycle number, for liquid injection in cycle 1.



Figure 4.14: Schematic of amount of injected liquid fuel exiting the combustion chamber versus cycle number, for liquid injection in cycles 1 through 8 and using linear superposition of the profile of Figure 4.13. Each color tracks the contribution of a given injection to the total.



Figure 4.15: Relative air-fuel ratio versus cycle number for vapor-only fueling.



Figure 4.16: Mass-averaged exhaust hydrocarbon concentration versus cycle number for vaporonly fueling, for the same experiment as Figure 4.15.



Figure 4.17: Cycle-resolved mass-averaged exhaust hydrocarbon concentration versus cycleresolved relative air-fuel ratio for vapor-only fueling, for the same data as Figure 4.15 and Figure 4.16.



Figure 4.18: Actual and predicted (from fit) exhaust unburned hydrocarbon emissions versus cycle number for the data of Figure 4.17.



Figure 4.19: Predicted (from fit) versus actual exhaust unburned hydrocarbon emissions for the data of Figure 4.18.



Figure 4.20: Schematic of method 1 for subtracting off the vapor contribution to the exhaust hydrocarbon emissions: crevices are assumed to be at the burned gas composition. The vertical line segment shows the contribution of the liquid fuel film to the emissions under this assumption for point "2".



Figure 4.21: Schematic of method 2 for subtracting off the vapor contribution to the exhaust hydrocarbon emissions: crevices are assumed to be at the "vapor-only" composition. The vertical line segment shows the contribution of the liquid fuel film to the emissions under this assumption for point "2".

## 5. RESULTS AND DISCUSSION

#### 5.1. OVERVIEW OF EXPERIMENTS

Table 5.1 summarizes the experiments that were ultimately used to produce the results in this study. The ambient conditions, as well as the motored net mean effective pressure and the stable intake airflow (which change slightly as ambient conditions change), are shown for reference. For all of these experiments, the motored net mean effective pressure is within 0.9% of the average, and the intake airflow is within 0.8% of the average. Furthermore, all of these experiments were misfire-free. Additional details on the operating conditions for the experiments are described in section 3.3.

Unless indicated, all of the experiments in Table 5.1 used the quasi-steady experimental scheme described in section 3.4. The only experiments that did not use this scheme are indicated and were used to determine and verify the emissions behavior of the engine with vapor-only fueling as described in section 4.7.2.

The experiments above the double line were all performed on the same day; those below the double line were performed roughly three weeks later, within a single day. As described in section 3.3, except as noted each experiment started with the combustion chamber wall temperatures at 30 °C. The engine cooled roughly 75 to 90 minutes between experiments. (For further information on the experimental scheme and procedures see sections 3.4 and 2.4.)

Six different targetings of the liquid fuel deposition were used. The intention was for these locations to be on the head, liner (in the clearance volume), and the piston on both the intake and exhaust sides of the engine. Because of spray rebounding and the actual spray cone angle being slightly larger than its nominal value, the actual wetted footprints for these nominal

targetings were slightly more complex as reflected in the nomenclature used in the first column of Table 5.1 and shown below.

#### 5.2. WETTED FOOTPRINTS

Figure 5.1 shows two labeled views of the combustion chamber, one looking up at the head and liner and the other looking down at the piston and liner. These images were obtained simultaneously using a mirror: the piston and liner view has been flipped and repositioned for ease in viewing and interpretation. In both of these images the intake side of the engine is closest to the observer and the DI fuel injector used to deposit the liquid fuel films is in the right-hand side metal wall (this wall is not visible in either view).

The images of Figure 5.1 are used below (in Figure 5.2 through Figure 5.7) to indicate the wetted footprint for each of the deposition locations. The area wetted by the gasoline deposition is indicated by the checkered pattern.

<u>Note</u>: For all of the deposition locations, any reference to "liner" wetting is completely in the clearance volume. In order to keep the nomenclature somewhat compact, the term "liner" alone is used; a more accurate phrasing would be "liner in the clearance volume". The reader is reminded that any time the term "liner" is used in referring to a liquid deposition location, the region being referred to is <u>the liner surface that is in the clearance volume of this engine</u>. (Recall that the shape of this combustion chamber is a rectangular prism.) For example, "head and upper-liner" wetting means that the head and upper portion of <u>the liner in the clearance volume</u> are wetted.

#### Wetted Footprint Identification

Figure 5.2 and Figure 5.3 show the "head and upper liner" depositions on the intake and exhaust sides of the engine, respectively. These two depositions are very nearly mirror images of one another, with the notable exception being that in high speed videos it appeared that for the exhaust side wetting more liquid fuel accumulated at the locations indicated near the head-liner interface. (This is suspected to be due to the fact that the exhaust valve protrudes slightly from the head and that the exhaust valve was so hot that fuel could not stick to it, and is further discussed in subsequent sections.)

Figure 5.4 and Figure 5.5 show the "liner" depositions on the intake and exhaust sides of the engine, respectively. For both of these depositions, the wetted area on the liner is above the piston seal bar travel. That is, the liquid fuel film is <u>not</u> scraped by the seal bars. Most of the spray ends up on the liner, with a small amount wetting the piston. From the high speed videos of the spray impingement for this wetting it appeared that the wetted film thickness and quantity on the piston was small in comparison to the wetting on the liner. For this reason, and to simplify the nomenclature and avoid confusion, this wetting is referred to subsequently as "liner" wetting. As before, the two wetted footprints are very nearly mirror images of one another.

Finally, Figure 5.6 and Figure 5.7 show the "piston and mid-liner" depositions on the intake and exhaust side of the engine, respectively. Most of the spray wets the piston, but some rebounds off of it and wets the liner in the middle of the clearance volume. And as above, the two wetted footprints are very nearly mirror images of one another.

#### Wetted Area Estimate and Distibution

Table 5.2 shows estimates of the areas of the wetted footprints (as marked with the checkered patterns in Figure 5.2 to Figure 5.7) for each of the deposition location types. These

areas were calculated from post-processing of the image files. Because portions of the wetted footprints are not visible in some of the images and the wetted footprints are very nearly symmetrical, the intake and exhaust side wettings of a given type are assumed to have the same area for this table. The wetted areas are approximately the same for the "liner" and "piston and mid-liner" wettings (roughly 1000 mm<sup>2</sup>), with the "head and upper liner" wetting being slightly larger (roughly 1200 mm<sup>2</sup>).

Also shown in Table 5.2 are the approximate distributions of each of the depositions on the combustion chamber surfaces. Generally speaking, roughly <sup>3</sup>/<sub>4</sub> of each wetted footprint is on its corresponding "primary" target location and <sup>1</sup>/<sub>4</sub> is on another combustion chamber surface. This fact is reflected in the nomenclature used for the wetted locations; the primary location is always stated first. As noted above, for the "liner" wetting both the liner and the piston are wetted, but it appeared the piston wetting wasn't as significant – for this reason and to simplify the nomenclature this wetting is referred to subsequently as "liner" wetting.

#### Corresponding Fuel Film Thicknesses

Finally, for reference, 10 mg of gasoline <u>uniformly</u> distributed over a wetted area of 1000  $\text{mm}^2$  corresponds to a 13 µm film thickness. Estimates of the actual mass of liquid fuel on the wall are in section 5.6; the corresponding wall film thicknesses in this study are of the same order and/or typical of those observed in-cylinder in other studies during the warmup phase of operation [37][38][39][40].

#### Summary

This section graphically identifies the wetted footprints for each of the liquid deposition locations referred to subsequently. All of the liquid depositions wet multiple combustion chamber surfaces: roughly <sup>3</sup>/<sub>4</sub> of each fuel film is on the "primary" targeted surface and the

remaining  $\frac{1}{4}$  is on other combustion chamber surfaces. The wetted area for these fuel films ranged from approximately 1000 mm<sup>2</sup> for the "liner" and "piston and mid-liner" wettings to approximately 1200 mm<sup>2</sup> for the "head and upper liner" wettings.

#### 5.3. REPRESENTATIVE WALL FILM TEMPERATURES

In this section the combustion chamber wall temperatures are revisited and a representative temperature history for each of the wetted footprints of the previous section is determined.

#### Combustion Chamber Surface Temperature Data

Figure 5.8 shows the combustion chamber wall temperatures for a typical quasi-steady liquid deposition experiment using the scheme of section 3.4. The vertical axis is the temperature measured by each of the combustion chamber thermocouples; the horizontal axis is the time in the experiment. Also shown along the horizontal axis is the corresponding batch number in the scheme (see section 3.4 for more details). On each curve a symbol marks the average temperature during the stable fueling cycles of each batch. As described earlier, it is these stable cycles that are used to examine the hydrocarbon emissions with and without liquid fuel films present.

Depending upon the location, these temperatures vary from roughly 50 to 55 °C for the first batch of the scheme. Different locations heat up at different rates though: in the last batch the temperatures range from 65 °C for the piston-intake thermocouple to 110 °C for the head-exhaust thermocouple.

The maximum deviation in a given temperature of Figure 5.8 among <u>all</u> of the liquid deposition experiments in this study was  $\pm 2$  °C. These deviations can be attributed to minor

variations in both the initial combustion chamber wall temperatures and the ambient conditions, as well as the fact that when the surfaces were wetted by liquid fuel they were cooled slightly.

#### Location of Combustion Chamber Surface Thermocouples

Figure 5.9 shows the locations of these thermocouples: it contains the same images that were used previously to identify the wetted footprints (e.g. see Figure 5.1), but with the thermocouples highlighted in yellow. (When printed in black and white, the thermocouples are the white spots in Figure 5.9 that are not present in Figure 5.1). Comparing Figure 5.9 to Figure 5.2 through Figure 5.7, with the wetting of the piston the thermocouple is not actually wetted by the liquid fuel film. When the head is wetted, the thermocouple is wetted by the liquid fuel film, but much of the film is on portions far away (in a relative sense) on the head. Finally, for the wetting of the liner surface in the "head and upper liner" and "liner" depositions, the thermocouple is wetted by the liquid fuel film; for the "piston and mid-liner" deposition the liner thermocouple is not wetted by the liquid fuel film. Thus, in some cases the thermocouples are actually wetted by the liquid fuel films, but in others they are only near the liquid fuel films. This is due both to physical constraints on where the thermocouples could be installed and to the fact that, for the piston wetting, the deposition timing ultimately used in the study (see section 3.3.4) was earlier than was presumed when originally installing the piston thermocouples and thus the piston was higher than anticipated in the combustion chamber when the spray impinged upon it.

#### Determination of Representative Wall Film Temperature

As described above, the goal of this section is the determination of an average, even if approximate, fuel film temperature history for each of the wetted areas of the previous section. Although the combustion chamber wall temperature is a field that varies continuously and the

combustion chamber surface thermocouples are a point measurement (that is close to if not within the actual wetted locations), the approach described here provided a means to examine the data and account for wall temperature differences. Because the variation in temperatures between experiments was small, the average temperature history was used to compute a "representative wall film temperature" history for each of the liquid deposition locations. This temperature was computed for each wetted footprint by simply weight-averaging the temperatures of Figure 5.8 by the corresponding wetted area proportions of Table 5.2. For example, the "head and upper liner – intake" representative wall film temperature was computed as <sup>3</sup>/<sub>4</sub> of the "head – liner" temperature.

These representative wall film temperatures can be thought of as the average wall temperature each of the liquid fuel film depositions is subjected to. The term "representative" is used to emphasize the approximate nature of using weighted averages of the point measurements (that are in some cases only "near" the films) to assign a value to the average wall temperature each liquid fuel film is subjected to.

Figure 5.10 is a plot of this representative wall film temperature versus time. There is one curve for each liquid deposition location, with a symbol marking the average temperature for the stable, last ten cycles of each liquid deposition batch. For example, the 4<sup>th</sup> liquid deposition (batch 10) for the "head and upper liner – exhaust" wetting had a representative wall film temperature of just over 100 °C for its stable cycles. The symbols in this plot are connected for ease in viewing, though strictly speaking they are discrete (one point for each liquid deposition "batch").

## Summary

Special surface thermocouples were installed in the combustion chamber of this engine in order to provide a measure of the temperatures the liquid fuel films were subjected to. Because the liquid depositions wet multiple combustion chamber surfaces, a representative average temperature history for each wetting was determined by the methodology described in this section.

For the initial liquid depositions of each wetting experiment these "representative wall film temperatures" ranged from approximately 50 to 60 °C, depending upon the particular location that is wetted. For the final liquid depositions of each wetting experiment these temperatures ranged from approximately 70 to 100 °C, again depending upon the particular location that was wetted.

These representative temperatures are used later to compare the various liquid depositions as a function of wall temperature. Despite their approximate nature, they provide a very useful way to examine the data that accounts for differences in wall temperature among the various locations.

# 5.4. INCREASE IN HYDROCARBON EMISSIONS DUE TO LIQUID FUEL FILMS – EFFECT OF FUEL FILM LOCATION

The main goal of this study is the assessment of the hydrocarbon emissions impact of having liquid fuel films at various locations in the combustion chamber, and the role that the wall temperature plays in that impact. This data is presented below, and was obtained using the experimental scheme of section 3.4 and the analysis methodology of section 4.7.

The increase in hydrocarbon emissions due to liquid fuel films for stable liquid deposition cycles are plotted in Figure 5.11 for each of the liquid deposition locations described

in sections 5.1 and 5.2. As described in section 4.7, the primary vertical axis is the contribution of the liquid fuel films to the mass-averaged measured hydrocarbon emissions concentration. The horizontal axis is the time in the experiment. Also shown for reference along the horizontal axis is the corresponding batch number in the quasi-steady scheme of section 3.4. The <u>shape</u> and <u>color</u> of the symbols indicate the wetted locations; the <u>shading</u> of the symbols indicates whether the wetting was on the intake (open symbols) or exhaust (filled symbols) of the engine. As time in the experiments increase, the wall temperatures increase. For all of the data in this plot, the same mass of gasoline (5.00 mg) was sprayed at the combustion chamber walls each cycle, and the combustion chamber wall temperatures at the start of each experiment was 30 °C. Further information on the operating conditions is in section 3.3.

Recall that, by construction, the different liquid depositions in the experimental scheme occur at different relative air-fuel ratios (see section 3.4). For this reason, the first and third depositions are at an overall relative air-fuel ratio of 0.83 and are shown with <u>thick outlined</u> symbols. The second and fourth depositions are at an overall relative air-fuel ratio of 0.87 and are shown with <u>thin outlined symbols</u>.

If the methodology of section 4.4 is applied to this data, the maximum error in scaling these concentrations by the same factor in order to convert them to *masses* of unburned hydrocarbons is 3% in a relative sense. (This error is due to their different relative air-fuel ratios, and thus different exhaust gas molecular weight and total moles of products per mole of fuel.) Since this error is small, the secondary vertical axis of Figure 5.11 is shown as the mass of this increase in hydrocarbon emissions that are due to the liquid fuel films each cycle, normalized by the amount of liquid fuel injected each cycle. That is, if one trusts the assumptions and methodology used to determine the contribution of the liquid fuel films to the total hydrocarbon

emissions<sup>1</sup>, the secondary vertical axis can be thought of as the fraction of the liquid fuel that is sprayed at the combustion chamber wall that ultimately becomes unburned hydrocarbon emissions. This fraction is actually quite low, ranging on a mass basis from roughly 10% down to less than 1% of the injected fuel for all of this data.

# Effect of alternate assumption about how to "subtract off" the vaporous fuel contribution to the hydrocarbon emissions

If the assumption about how the vaporous fuel emissions are "subtracted off" the total emissions in order to determine the emissions due to the liquid fuel films is changed to the alternate extreme (see section 4.7.4), the effect is to simply shift the values in Figure 5.11 upward by an amount corresponding to roughly 5% of the injected fuel quantity. That is, on the secondary axis the points of Figure 5.11 would instead range from roughly 5% to 15%. Even with this alternate assumption only a small fraction of the injected liquid fuel ultimately becomes hydrocarbon emissions. It should be emphasized, however, that it is shown later (in section 5.6) that the assumption of section 4.7.4 that was used to generate Figure 5.11 and all of the subsequent "hydrocarbon emissions due to liquid fuel" plots appears to be the more accurate assumption.

# General trends: time in experiment, intake versus exhaust wetting

An initial observation of Figure 5.11 is that as time in the experiment (and thus wall temperatures) increase, the general trend is that the impact of the liquid fuel films on exhaust hydrocarbon emissions decrease. The only exception to this trend is the "head and upper liner – exhaust" wetting: it is believed that for this location the wetted footprint is changing as temperatures increase. This is further discussed in subsequent sections.

Furthermore, when the liquid fuel film is on the exhaust side of the engine the impact on hydrocarbon emissions is larger than when the film is at the same location on the intake side of the engine. (Recall that the exhaust side wettings have filled symbols and the intake side wettings have open symbols.) The size of this difference varies by location and has several contributing factors that are expanded upon in the sections that follow. The fact that wetting on the exhaust side of the engine has a larger impact is plausible as, for a given location, with wetting on the exhaust side of the engine the liquid fuel film is closer to the exhaust valve and thus presumably any fuel that vaporizes from it has a higher likelihood of being exhausted.

#### Scatter plot versus representative wall film temperature

As described above, as time in the experiment increases all of the combustion chamber temperatures increase, and moreover the various surfaces increase in temperature at different rates. Thus, a useful way to look at the data of Figure 5.11 is to scatter it against the representative wall film temperature that was determined earlier for each liquid deposition. This is done in Figure 5.12.

Again, as this is the same hydrocarbon emissions data as Figure 5.11, the fraction of injected liquid fuel that ultimately becomes hydrocarbon emissions in this plot ranges from roughly 1% to 10% depending upon both the location of the liquid film and the wall temperature. How these impacts vary with both location and temperature are expanded upon below.

#### Trends with wall temperature; extrapolation to temperature for which liquid fuel has no impact

With the aforementioned exception of the "head and upper-liner – exhaust" wetting, when plotted against this estimate of the fuel film temperature all of the wetted locations exhibit decreasing slopes of comparable order. Looking at the depositions at the same relative air-fuel ratio, these slopes range from a drop of approximately 20 ppmC1 per °C for the "liner – exhaust"

wetting to a drop of approximately 45 ppmC1 per °C for the "head and upper-liner – intake" wetting. Moreover, again excluding the "head and upper liner – exhaust" wetting, for each location there appears to be a temperature (ranging from roughly 85 to 110 °C depending upon the location) above which an extrapolation of this trend suggests that <u>at these conditions this</u> <u>particular amount of liquid fuel</u> would have no impact on exhaust hydrocarbon emissions. This would presumably be because above this temperature all of the liquid fuel that was on the combustion chamber walls is able to vaporize and burn prior to being exhausted. (The validity of such an extrapolation is addressed in section 5.11.)

The fact that this critical temperature varies for the different wetted locations could be due to the fact that different quantities of liquid fuel are on the combustion chamber walls for each of the wetted locations, or that the representative wall film temperatures are not an accurate enough representation of the temperatures the films experience (for reasons described above, or even due to non-uniform film thicknesses).

# "Low" temperature data and piston wetting impact differences

Because of the need to motor the engine and fire it for a brief period of time prior to obtaining data with liquid deposition, the lowest wall temperatures for which data on the liquid fuel behavior was obtained were roughly 55 °C. This data corresponds to the very first liquid deposition for each location. Referring to Figure 5.11 or Figure 5.12, for the very first liquid deposition (batch 2) of each experiment, for both the "head and upper liner" and the "liner" wettings, the impact on hydrocarbon emissions is virtually the same for the intake and exhaust side wettings of the same type. For example, the "liner – exhaust" and "liner – intake" wettings have virtually the same hydrocarbon emissions impact. (Note that the "head and upper liner –

exhaust" wetting is <u>not</u> an exception here; one possible explanation is that the wall temperature was low enough so that the wetted footprint mirrored that of the intake side wetting.)

However, for the "piston and mid-liner" wetting, the exhaust side wetting has roughly double the hydrocarbon emissions impact as the intake side wetting – both initially <u>and</u> as temperatures increase. This is believed to be due to the fact that with the intake side piston wetting, a substantial portion of the fuel that vaporizes post-flame from the liquid fuel film is trapped in the combustion chamber at exhaust valve closing. This is supported by evidence in sections 5.9 and 5.10.

<u>Aside</u>: this substantial difference in the "piston and mid-liner" wetting is most certainly <u>not</u> due to calibration errors or differences. In fact, in using the maximum and minimum fast FID calibrations from all of these experiments, the resulting difference obtained in a 2000 ppmC1 value is  $\pm$  25 ppmC1.

#### Effect of relative air-fuel ratio; evidence of oxidation differences

As described earlier, by design the first and third liquid depositions at each location are at an overall relative air-fuel ratio of 0.83 and the second and fourth depositions are at an overall relative air-fuel ratio of 0.87. This difference is indicated by the thickness of the symbol outlines in Figure 5.11 and Figure 5.12.

Except for the aforementioned exception of the "head and upper liner – exhaust" wetting, for all locations the depositions that are at the overall "less fuel rich" condition (that is, the second and fourth depositions) exhibit a smaller impact on the unburned hydrocarbons than would be expected from an interpolation and extrapolation of the data for the depositions at the "more rich" fueling (the first and third depositions). The size of this difference varies by location, with the intake side wettings exhibiting a larger difference than the exhaust side

wettings and the "liner" wettings exhibiting the largest difference. These differences are presumably due to oxidation differences – more post-flame oxidation of any fuel vaporized from the liquid films apparently occurs in the "less rich" depositions because more oxygen is present (for example, coming from the crevices gases).

#### "Liner" wetting impact too small?

In Figure 5.12, the impact of the "liner" wettings is small in comparison to the other wetted locations, with only the "piston and mid-liner – intake" wetting (which has already been described above as being an exception) being comparable. This is somewhat peculiar in that, referring to the wetted footprints of Figure 5.2 to Figure 5.7, the "liner" wettings are "between" the other depositions. (Since the "piston and mid-liner – intake" wetting has been previously identified as a different type of exception, the apparent discrepancy being described here is most evident when looking at the data for the exhaust side wettings.)

This apparent discrepancy is due to the fact that less of the injected liquid fuel is actually on the combustion chamber walls for the "liner" depositions as compared to the other locations, which will be shown in section 5.6.

## Summary

In this section the hydrocarbon emissions due to the liquid fuel films were presented for all of the liquid deposition locations. These results were presented as a fraction of the liquid fuel injected each cycle as a function of both time and the representative wall film temperature. Depending upon the wetted location, the hydrocarbon emissions from the liquid fuel films were 1 to 10% of the amount of liquid injected. In general, as time and temperature increased the contribution of the liquid fuel films to the total hydrocarbon emissions decreased. Additionally, for all wetted locations the exhaust side wetting exhibited a larger impact on the hydrocarbon
emissions than its corresponding intake side wetting. This data was obtained at two different relative air-fuel ratios: a minor effect of relative air-fuel ratio was observed, presumably due to slightly increased postflame oxidation at the "less fuel rich" condition.

Two notable exceptions were present in this data: the first was the "head and upper-liner – exhaust" wetting. It did not exhibit the same behavior as the other locations when plotted against temperature. For this wetting, its wetted footprint is thought to be changing as wall temperatures increase, which will be further supported by data in subsequent sections. The second exception was the "piston and mid-liner – intake" wetting. Its hydrocarbon emissions impact was roughly half that of its corresponding exhaust side wetting. For this wetting, it is thought that fuel vaporized from the piston film postflame is trapped in the combustion chamber at exhaust valve closing. This is supported by data in subsequent sections.

Finally, it was observed that the impact of the "liner" wettings appear too small relative to the other locations. This is due to the fact that less fuel is sticking to the combustion chamber walls for these particular liquid depositions. In fact, many of the apparent discrepancies or exceptions in this section can be explained by differences in the amount of liquid fuel on the combustion chamber surfaces, as will be evident subsequently. First, however, some additional data showing the effect of the initial combustion chamber wall temperatures, the effect of the amount of liquid sprayed at the walls each cycle, and the repeatability of the data and analysis methods is presented before addressing these differences in the amount of liquid fuel actually on the walls.

# 5.5. INCREASE IN HYDROCARBON EMISSIONS DUE TO LIQUID FUEL FILMS – ADDITIONAL EFFECTS AND REPEATABILITY

# 5.5.1. Effect of Initial Combustion Chamber Wall Temperature and Repeatability

Figure 5.13 shows the increase in hydrocarbon emissions due to the liquid fuel films for "liner – intake" wetting with initial combustion chamber wall temperatures (at the start of motoring in each experiment) of 25 °C and 30 °C. As described before, the experimental scheme of section 3.4 and analysis methodology of section 4.7 were used to obtain these results. Analogous to Figure 5.11, the vertical axis is the increase in hydrocarbon emissions due to the liquid fuel films by the methodology of section 4.7. The horizontal axis is the time in the experiment. Also shown for reference is the corresponding batch number in the scheme of section 3.4.

# Effect of Initial Combustion Chamber Wall Temperature; Hydrocarbon Decrease with Temperature

Roughly speaking, throughout the entire experiment all of the wall temperatures for the 25 °C experiment were 5 °C lower than the corresponding temperatures in the 30 °C experiment (which are shown in Figure 5.8). The 25 °C experiment exhibits a roughly 200 ppmC1 offset in the impact of the liquid fuel films on the unburned hydrocarbon emissions relative to the experiment with the 30 °C initial wall temperature for all four depositions. This corresponds to a drop of approximately 40 ppmC1 per °C for the wall temperature change <u>between</u> the two experiments. In contrast, <u>within</u> each of the experiments the drop in the hydrocarbon emissions impact (comparing liquid depositions at the same relative air-fuel ratio) is approximately 20 ppmC1 per °C. (Figure 5.12 shows a plot of the 30 °C initial wall temperature experiment's hydrocarbon impact versus representative wall film temperature – recall that the wetted location here is "liner – intake".) It is not known why these two slopes differ – one possible explanation

is that the representative wall film temperature may not be a completely accurate representation of the temperatures the films are subjected to. These two slopes are, however, of the same order.

#### Repeatability

Finally, the consistency in the "offset" between the two experiments of Figure 5.13 demonstrates the repeatability of both the experiments as well as the robustness of the analysis methodology used to determine the contribution of the liquid fuel film to the exhaust hydrocarbon concentration.

## 5.5.2. Effects of Amount of Liquid Injected and Initial Wall Temperature, Repeatability

Similar to Figure 5.13, Figure 5.14 shows the increase in hydrocarbon emissions due to the liquid fuel films obtained from the analysis methodology of section 4.7 for the "head and upper liner – intake" wetting. It compares different initial combustion chamber wall temperatures, amounts of injected liquid fuel, and a repeat of identical conditions on different days. Because of the number of closely-spaced data points in this plot, lines have been added that connect data points of a given experiment. Because of the variation in relative air-fuel ratio no trend is implied by the addition of these lines: they are solely an identification/labeling aid. The "baseline" data points to which the others should be compared are labeled "Day 1, 30 °C, full amount" and are shown with a dark orange fill and connecting line.

#### Effect of Initial Combustion Chamber Wall Temperature

Comparing the 25 °C to the 30 °C initial combustion chamber wall temperature experiment, the trend with initial combustion chamber wall temperatures is very similar to the behavior described above, but slightly less consistent. As expected, the liquid fuel films in the lower initial wall temperature experiment exhibit a larger impact on the unburned hydrocarbon

emissions. Furthermore, the offset is of the same order (~200 ppmC1 drop for a 5°C temperature change) as previously for the "liner – intake" wetting.

#### Repeatability (data taken on different days)

Comparing the data taken on different days (and thus slightly different ambient conditions and gas analyzer operating conditions), the behavior is very similar and agrees within 200 ppmC1 for all of the liquid depositions. This level of agreement is acceptable given typical variations in measured hydrocarbon concentrations day-to-day. (Note that for this reason, with the exception of this particular comparison, all of the direct comparisons made in this study are made between experiments performed on the same day in order to reduce this variability.)

# Effect of amount of liquid fuel injected

Finally, the three orange-shaded data sets compare three different amounts of injected liquid fuel: a "full" amount,  $\frac{7}{8}$  of that amount, and  $\frac{3}{4}$  of that amount. The full amount is 5.00 mg, which as mentioned previously is the quantity of liquid gasoline injected in each cycle of liquid deposition in all of the other experiments in the study. The  $\frac{7}{8}$  and  $\frac{3}{4}$  amounts correspond to 4.38 mg and 3.75 mg, respectively. (Hardware limitations on the minimum possible injection duration precluded amounts much smaller than this.)

Note that the experiments with decreased amounts of liquid injected (i.e. the " $^{7}/_{8}$ " and " $^{3}/_{4}$ " experiments) used the same <u>vaporous</u> fueling amounts as the other experiments in the study (see section 3.4), and thus the overall relative air-fuel ratio for the liquid depositions with these decreased amounts of liquid injected do <u>not</u> match the other experiments (or each other). This is indicated by the broken symbol outlines for these experiments in Figure 5.14. However, as is evident within each dataset of this plot and the other results plots, for this wetted location the

effect of relative air-fuel ratio variation on the hydrocarbon emissions impact of the liquid fuel films is small.

As expected, as the amount of liquid gasoline that is sprayed at the combustion chamber walls decreases, the impact of the liquid fuel films on the exhaust hydrocarbon emissions decreases as well. Furthermore, the size of these <u>differences</u> appears fairly linear in the amount of liquid fuel injected. That is, the difference between the "full" amount and the "<sup>7</sup>/<sub>8</sub>" depositions is roughly the same as the difference between the "<sup>7</sup>/<sub>8</sub>" and "<sup>3</sup>/<sub>4</sub>" depositions.

## Extrapolation to the amount of injected liquid that has no impact

Interestingly, at a given time (or equivalently, at a given batch number) and thus wall temperature condition, an extrapolation of this data suggests that at each temperature there is some nonzero amount of injected liquid fuel that would have no impact on the exhaust hydrocarbon emissions. For example, a linear extrapolation of the average trend indicates for the very first deposition of each experiment (i.e. the batch 2 data) that at its corresponding wall temperature conditions a deposition amount of the order 25% of the full amount would have no impact on the exhaust hydrocarbon emissions. Presumably this would be because for an injected amount less than this, the liquid fuel film would completely vaporize and oxidize prior to being exhausted.

Furthermore, the amount of liquid for which this extrapolation suggests that there is no impact of the injected liquid fuel on the exhaust hydrocarbon emissions appears to decrease as wall temperature (i.e. time in the experiment) increases – as one would expect. For example, a linear extrapolation of the third deposition data (batch 7) indicates that an amount of the order 50% of the full amount would have no impact on exhaust hydrocarbon emissions.

While there is insufficient data here to make firm conclusions, it suggests that <u>at a given</u> <u>wall temperature condition</u> there is an amount of injected liquid fuel which would have no impact on the exhaust hydrocarbon emissions. As stated previously, presumably this is because all of the liquid fuel from the injection that sticks to the combustion chamber walls vaporizes and oxidizes prior to being exhausted. Moreover, as wall temperature increases the amount of injected liquid fuel that would have no impact on the hydrocarbon emissions appears to increase as well. (Conversely, recall that extrapolation of the data of section 5.4 suggested that <u>at a fixed</u> <u>amount of liquid injected</u> there was a wall temperature, which varied slightly for the different locations, above which that mass of injected liquid fuel would have no impact on the exhaust hydrocarbon emissions.)

#### Summary

Some additional effects on the hydrocarbon emissions impact of the liquid fuel films were examined in this section. The effect of the initial combustion chamber wall temperature was found to simply be an offset in the hydrocarbon emission impact of the liquid fuel films. An increase of approximately 40 ppmC1 increase per °C decrease was observed.

The effect of the amount of liquid fuel sprayed at the combustion chamber walls was similarly well-behaved (for the particular wetted location tested). As the amount of injected liquid fuel decreased the hydrocarbon emissions impact of the liquid fuel films at a given temperature decreased as well. Extrapolation of this data suggested that there is an amount of liquid fuel (which increases as wall temperature increases) below which the liquid fuel would be expected to have no impact. This is presumably because below this critical amount all of the liquid fuel that sticks to the combustion chamber walls vaporizes and oxidizes prior to being exhausted.

Finally, the repeatability of the experiments and analysis methods was assessed and found to be very good, especially for experiments performed on the same day.

#### 5.6. WALL FILM MASS

As has been alluded to earlier, much of the data on the hydrocarbon emissions impact of the liquid fuel films can be understood in terms of the amount of liquid fuel that is actually on or sticking to the combustion chamber walls – this data is presented below.

Plotted in Figure 5.15 is the estimated mass of liquid fuel on the combustion chamber walls for the stable cycles of each of the liquid depositions of Figure 5.11 and Figure 5.12. These masses were obtained by the analysis methodology described in section 4.5. The vertical axis is the mass of fuel on the wall during the stable liquid deposition cycles; the horizontal axis is the time in the experiment (and as before, the corresponding batch number for the scheme of section 3.4 is also shown for reference). Note that because of the close spacing and overlapping of the data points in this plot, lines connecting data points of a given experiment (i.e. wetted location) have been added. No trend is implied by the addition of these lines; they are solely to aid in identifying and locating each of the data points in the plot.

As described in section 4.5, there is uncertainty in the estimates of second and fourth deposition masses. That is, the first and third deposition values are more accurate and reliable.

#### Independence of the exhaust hydrocarbon and wall film mass measurements and calculations

In looking at the subsequent data that uses these estimates of the mass of liquid fuel on the wall, <u>the most important thing to realize about these estimated wall film masses is that their</u> <u>determination is essentially independent of the exhaust unburned hydrocarbon emissions</u>. As explained in section 4.5, the estimate of the wall film mass is achieved by looking at the net mass of burned and unburned fuel that leaves the combustion chamber each cycle during the transient that occurs when liquid deposition begins and ends. The net mass of fuel leaving the combustion chamber is dominated by the burned fuel (e.g. see Figure 4.9) – which is determined from the CO and CO<sub>2</sub> measurements of the fast NDIR (see section 4.4). In contrast, the unburned hydrocarbon emissions measurements are from a <u>separate</u> fast gas analyzer (the fast FID). And furthermore, as can be seen in Figure 4.9, the unburned hydrocarbon emissions in the exhaust are a very small fraction of the net mass of fuel leaving the combustion chamber each cycle, so they contribute negligibly to the determination of the wall film mass. *Thus, the mass of exhaust unburned hydrocarbons and the wall film mass can be considered essentially independent calculations from independent measurements*.

#### General observations

Recall that the mass of liquid gasoline injected each cycle was 5.00 mg. Thus, for the initial liquid depositions of each experiment, the mass of fuel on the wall under stable deposition conditions ranges from 2 to 4 times the amount injected each cycle. As time (and thus wall temperature) increases, the size of the stable fuel films decrease, becoming roughly <sup>1</sup>/<sub>2</sub> to 2 times the amount injected each cycle for the final liquid depositions.

A first observation about the wall film mass behavior is that for both the "liner" and "piston and mid-liner" wettings the wall film masses are essentially identical between the intake and exhaust side wettings. (Recall as noted above that there is uncertainty in the calculated values for the second and fourth depositions.) In contrast, the "head and upper-liner" wettings start with essentially the same wall mass, but for the later depositions the exhaust side wetting has roughly double the mass of fuel on the wall as compared to the intake side wetting. (Interestingly, and as will be examined more quantitatively in section 5.8.2, for the "head and

upper liner – exhaust" wetting the shape of the wall film mass curve is very similar to the shape of the unburned hydrocarbon emissions that are due to the liquid fuel films shown in Figure 5.11.)

The "piston and mid-liner" wetting wall film masses are the largest and the "liner" wettings are the smallest. (The behavior of the "head and upper-liner" wettings are too complicated to generalize.) In general, the "piston and mid-liner" wetting wall masses are double that of the "liner" wettings at the same time in the experiment. One natural question is whether or not the wall temperatures are the reason for this substantial difference.

### Scatter versus representative wall film temperature; reason that the liner masses are small

Thus, Figure 5.16 shows this same data of Figure 5.15, but scattered against the representative wall film temperature of section 5.3. As before, the values for the second and fourth deposition are less certain than the values for the first and third depositions. Furthermore, as with Figure 5.15, due to the closely spaced and overlapping data points, connecting lines have been added between data points of a given wetted location. Due to the relative air-fuel ratio variation, no trend is implied by these lines.

Referring to Figure 5.16, note that even at comparable wall temperatures, the wall masses for the "piston and mid-liner" wettings are still roughly double those of the "liner" wettings. This is, in fact, plausible: due to the location of the DI fuel injector that was used to spray the liquid gasoline at the combustion chamber surfaces, the "liner" depositions involve essentially "head-on" impingement in which any drops that rebound from the wall are not directed at another combustion chamber surface and are more likely to remain suspended in the charge. This was, in fact, observed in high speed videos of the impingement. In contrast, the "piston and mid-liner" deposition involves impingement at an oblique angle on the piston in which any fuel

that rebounds off of it then impinges upon the liner. This effect was not anticipated when designing and selecting the means for depositing the liquid fuel films on the combustion chamber surfaces. As will be evident subsequently, this effect explains why in the previous section the impact of the "liner" wettings was smaller than expected.

# Trend with temperature; extrapolation to temperature at which wall mass is zero

Furthermore, with the exception of the "head and upper-liner – exhaust" wetting, all of the wetted locations have roughly the same change in wall mass with temperature. (Again, recall that the masses for the first and third depositions are most accurate.) Moreover, an extrapolation of the trend in this data suggests that for a wall temperature greater than 80 to 100 °C (depending upon the fuel film location), for this amount of liquid gasoline sprayed at the combustion chamber walls the mass of fuel on the walls would be zero. As described previously, differences in this critical temperature for each location may be due to nonuniformities in the wall film thickness or that the representative wall film temperature may not be an accurate enough estimate of the actual average temperature the wall film is subjected to.

## "Head and upper liner - exhaust" exception

The fact that the "head and upper-liner – exhaust" wetting exhibits significantly different behavior than the other deposition locations suggest that its underlying physical situation is changing. As mentioned earlier, it is suspected that for this deposition location the wetted footprint is changing as the wall temperatures increase. In particular, as the temperatures increase it is thought that fuel is not sticking to the exhaust valve and instead a thicker film is being formed on the head and upper-liner at the locations indicated in Figure 5.3. Further evidence supporting this notion that the wetted footprint is changing for this deposition location is in section 5.9.

# Validation of assumption about "subtracting off" vaporous fuel hydrocarbon emissions contribution

Finally, the fact that an extrapolation of this data suggests a wall temperature of 80 to 100 °C (depending on the deposition location) would have zero mass of liquid fuel on the combustion chamber surfaces supports the assumptions made in section 4.7 about how to "subtract off" the vaporous fuel contribution to the total exhaust hydrocarbon emissions. In particular, with the assumption made in that section, the results of section 5.4 suggest that at a wall temperature of 85 to 110 °C there would be no impact of the liquid fuel on the exhaust hydrocarbon emissions. This is consistent with the observation in this section that for wall temperatures greater than 80 to 100 °C there would be no mass of liquid fuel expected on the wall. Thus, the data in this section (which is essentially independent of the exhaust hydrocarbon emissions) validates the assumptions made in section 4.7 regarding how to "subtract off" the contribution of vaporous fuel to the total exhaust hydrocarbon emissions. This validity may very well depend upon the particular choice made for the timing of the liquid deposition (described in section 3.3.4).

#### Summary

In general, as wall temperatures increased the amount of liquid fuel actually on the combustion chamber surfaces when stable fueling conditions were reached decreased. Depending upon the particular wetted location, the stable amount of liquid fuel on the combustion chamber surfaces was 2 to 4 times the amount of liquid fuel injected each cycle for the initial depositions of each experiment, and ½ to 2 times the amount of liquid fuel injected each cycle for the final liquid depositions of each experiment. With the exception of the "head and upper-liner" wettings, intake and exhaust side wettings at a given location exhibited very similar wall mass behavior.

Finally, an extrapolation of the trend in the wall masses as wall temperature increases validated the assumptions made earlier about how to deduce the hydrocarbon emissions that were due to the liquid fuel films.

These estimated masses of fuel on the combustion chamber walls for each of the liquid depositions can be used to normalize the hydrocarbon emissions for each deposition. This is done subsequently in section 5.8.2. First, however, an alternate way of thinking of "the amount of fuel that sticks to the wall" is presented in the following section.

# 5.7. FRACTION OF EACH LIQUID INJECTION THAT STICKS TO THE WALL

Rather than look at the amount of fuel that is actually on the combustion chamber walls, an alternate way to think of and look at differences in the amount of fuel sticking to the wall is to look at <u>the fraction of the liquid that is sprayed at the wall that actually sticks to the wall</u>. This amount and the amount on the wall when the film stabilizes should correlate with one another, but each provides its own paradigm for looking at the data.

Table 5.3 shows the results of applying the methodology of section 4.6 to the data of Figure 5.11, Figure 5.12, Figure 5.15, and Figure 5.16. Shown are the minimum and estimated (from a linear extrapolation) fractions of injected liquid fuel sticking to the wall for the first and third liquid depositions of each experiment. For situations in which linear extrapolation was not valid (in the notation of section 4.6, when  $s_3 \ge s_2$ ), the estimated value was assigned the minimum value. Because it corresponds to the cycles of stable liquid deposition that were used to generate the other data in the study, the "decay" profile that occurs when liquid deposition ends was used with this methodology. (For further information, see section 4.6). Also, as described earlier, in the transients that occur before and after each set of liquid depositions cycles, the engine speed is more stable for the first and third depositions – and thus for them the fuel accounting is much more accurate. For this reason, and given the sensitivity of the methodology of section 4.6 to accumulated errors, only the results for the first and third depositions of each experiment are shown in Table 5.3.

As described earlier, the analysis methodology of section 4.6 is <u>very</u> sensitive to errors in the net mass of fuel exiting the combustion chamber. For this reason, suspected outliers thought to be the product of accumulated errors affecting the calculation are italicized in Table 5.3. Furthermore, given the approximate nature of these results, Table 5.4 shows average values for each of the deposition location types in which the suspected outliers were excluded. For these average values it was assumed that the intake and exhaust depositions of a given type behave similarly. This assumption may not be completely accurate, especially for the "head and upperliner" wettings.

Despite all of these simplifications and assumptions, the trend in this data is consistent with the previous data in section 5.6 and the expected behavior. The "liner" wetting has the smallest fraction of injected liquid fuel sticking to the wall (approximately  $\frac{1}{2}$  to  $\frac{3}{5}$ ), and also in general has the smallest mass of fuel on the wall for each deposition. As described previously, with the "liner" wettings, fuel that rebounds off of the liner surface is not directed at any of the other combustion chambers surfaces, so this behavior is plausible.

In contrast, the piston wetting has the largest fraction of injected liquid fuel sticking to the wall (approximately  $\frac{4}{5}$ ), and in general also has the largest mass of fuel on the wall for each deposition. This is also plausible, because as described earlier, for this deposition fuel is targeted

at the piston and any fuel that rebounds off of it impinges upon the liner where it has another opportunity to stick to a combustion chamber surface.

Finally, the "head and upper liner" wetting has an intermediate fraction of injected fuel sticking to the wall (approximately  $^{2}/_{3}$ ), and also in general has an intermediate mass of fuel on the wall for each of the depositions.

# Comparison to other data

The numerical values for the fractions of the fuel spray that stick to the wall are of the same order as those reported in [37] using a completely different methodology. For an injection at similar injection pressure and injected fuel quantity, they reported for a direct (normal) impingement on a dry, flat surface that approximately 44% of the injected fuel stuck to the wall. The "liner" depositions are most comparable to this condition since they are nearly a normal impingement and furthermore fuel that rebounds from the wall is not directed at another combustion chamber surface. Recall that for the "liner" wettings, the estimated fraction of each injection sticking to the wall was approximately  $\frac{1}{2}$  to  $\frac{3}{5}$ . The fact that, using the methodology of this section, the estimated fraction of fuel that sticks to the walls is higher than that observed in [37] could be due to the fact that for the estimates of this section the surface was wet, and thus more fuel would be expected to stick to it, as well as the fact that some of the fuel that rebounds from the liner surface could stick to another combustion chamber surface. Or, this discrepancy could simply be due to the simplifying assumptions and sensitivity to errors of the methodology of section 4.6 or inaccuracies in either of the methodologies. Nonetheless, the two estimates are of the same order, providing a consistency check on the results of this section.

Summary

As has been emphasized earlier, these estimates of the fraction of each injection that stick to the combustion chamber surface are approximate and not as robust as the wall film mass estimates. They are, however, consistent with the wall film mass estimates as described above. Depending upon the wetted location, the fraction of each liquid injection estimated to actually stick to the combustion chamber surfaces ranged from approximately 50% (for the "liner" wettings) to 80% (for the "piston and mid-liner" wettings).

\* \* \*

Both the fraction of injected fuel that sticks to the wall for each deposition location and the mass of fuel actually on the wall for the stable fueling conditions of each deposition can be used to normalize the unburned hydrocarbon emissions that are due to the liquid fuel films. These results are presented next.

# 5.8. NORMALIZED INCREASE IN HYDROCARBON EMISSIONS DUE TO LIQUID FUEL FILMS

Figure 5.17 shows a simplified schematic of the pathways injected liquid fuel can follow. In the figure, fuel in liquid form is shown in blue and fuel in vaporous form is shown in pink. The <u>widths</u> of the arrows indicate typical relative proportions for each of these pathways for the data in this study. For reference, the wall film mass is typically 1 to 4 times the injected fuel mass (e.g., see Figure 5.15).

Some fraction of the liquid fuel that is sprayed at the wall vaporizes in transit to the wall or rebounds from the wall and subsequently vaporizes. As shown above (in section 5.7), on average this amount that does not stick to the wall is approximately 1/3 of the injected quantity for this data. The remaining fraction (on average, approximately 2/3) of injected fuel sticks to the

wall, establishing a wall film. Under stable film conditions, the mass of fuel vaporizing from the film each cycle is equal to the mass of liquid sticking to the wall each cycle. Thus, in Figure 5.17 the widths of these two arrows are identical. The fuel that vaporizes from the film can either burn in the flame or after flame passage, or become hydrocarbon emissions. For the data in this study, this amount that becomes hydrocarbon emissions each cycle is approximately 1 to 10% of the amount injected each cycle (see Figure 5.11); the remaining fuel burns.

There are three different amounts of liquid fuel in Figure 5.17: the amount injected, the amount that sticks to the wall, and the amount that is on the wall. This suggests three ways to normalize the unburned hydrocarbon emissions under stable fueling conditions; namely, by each of these fuel amounts. No one approach is necessarily better than the other: they are simply three different viewpoints from which to look at the data.

## 5.8.1. Normalized by the amount of liquid fuel injected each cycle

Figure 5.11 and Figure 5.12 were normalized by the amount of liquid injected each cycle as shown on their secondary axes. Thus, section 5.4 discusses and interprets the hydrocarbon emissions impact of the liquid fuel films when normalized in this way. One drawback of this particular normalization method is that it does not account for the (unintended) differences in the amount of fuel sticking to the combustion chamber surfaces that were a byproduct of the particular method chosen for depositing the liquid fuel films in this study.

#### *5.8.2. Normalized by the stable amount of liquid fuel on the wall*

Plotted in Figure 5.18 is the mass of hydrocarbon emissions due to the liquid fuel films (the data of Figure 5.12), normalized by the corresponding mass of liquid fuel on the combustion

chamber surfaces for each deposition (the data of Figure 5.16), versus the representative wall film temperature for each of the depositions. That is, if one trusts the assumptions and methodology used to determine the contribution of the liquid fuel films to the total hydrocarbon emissions<sup>2</sup>, the vertical axis of this plot can be thought of as the fraction of fuel that was on the wall that ultimately became unburned hydrocarbon emissions. As before, the horizontal axis is the representative average temperature the liquid fuel films are subjected to. Recall that the first and third depositions of a given experiment are at an overall relative air-fuel ratio of 0.83 and are shown with <u>thick outlined symbols</u>. The second and fourth depositions are at an overall relative air-fuel ratio of 0.87 and are shown with <u>thin outlined symbols</u>. Also recall that the values of the wall film masses for the second and fourth depositions are more uncertain than those of the first and third depositions. (For this reason, the apparent outlier of the second deposition for the "head and upper liner – intake" wetting is not thought to be meaningful.)

## Range of values, trend with temperature

The value of the hydrocarbon emissions due to the liquid fuel films normalized in this way, which as described above can be thought of as the fraction of the wall film that ultimately becomes hydrocarbon emissions, is small and ranges from approximately 1% to 6% for these conditions.

Except for the "liner – intake" wettings at the "less rich" condition (for which the determination of the wall film mass is less accurate as mentioned above), for all of the deposition locations when comparing depositions at the same overall relative air-fuel ratio the fraction of the wall film that becomes unburned hydrocarbon emissions <u>increases</u> as wall temperatures increase. This is, in fact, to be expected: higher wall temperatures would be expected to result in higher vaporization rates. Thus, on a fractional basis, more fuel would be expected to vaporize

from the wall films both before and after flame passage as the wall temperatures increase. Fuel that vaporizes after flame passage can become hydrocarbon emissions. Therefore, as wall temperatures increase, a higher fraction of the fuel on the wall would be expected to become hydrocarbon emissions, which is precisely what is observed here.

## Head and upper liner impact largest

The "head and upper liner" wettings exhibit a much larger fraction of the wall film that becomes hydrocarbon emissions than the other locations (ranging from 2.4 to 5.7% of the wall film mass, compared to 0.75 to 2.3% for the other locations). One possible explanation for this trend could be that these locations experience flame passage earlier than the other locations, and thus more of the vaporization from the film occurs after flame passage resulting in more vaporized fuel that can become unburned hydrocarbon emissions. This, however, is not likely: referring to Figure 5.9, the spark plug is located between the two head surface thermocouples. Referring then to Figure 5.2 through Figure 5.7, the wetted footprints for the "head and upper liner" and "liner" wetting are of similar distances from the spark plug.

Another possible explanation for why the "head and upper liner" wetting exhibits a much higher fraction of the wall film that becomes hydrocarbon emissions is that fuel molecules vaporized from these films have not yet finished oxidizing at the sample location in the exhaust runner (since these wetting locations are closest to the exhaust valve). This possibility cannot, unfortunately, be completely ruled out: cycle simulation results at these operating conditions (which, however, do not include blowby leakage) suggest that at exhaust valve opening the charge temperature is close to temperatures at which oxidation could still occur. However, the fact that for the "head and upper liner" wettings there is no significant effect of the different relative air-fuel ratios (neglecting the aforementioned outlier of the second deposition in the "head and upper liner – intake" wetting) suggests that significant oxidation of the fuel vaporized from these films is not occurring. Otherwise, one would expect to see a drop in the hydrocarbon emissions trend for the "less fuel rich" cases because more oxygen (e.g. from the crevices) was present. Thus, the possibility of oxidation of the unburned hydrocarbons from the liquid fuel films still occurring past the sample point in the exhaust runner appears unlikely.

From this one can conclude that, for a fixed mass of fuel on the wall and a fixed wall temperature, in this study wetting the head results in the largest impact on the unburned hydrocarbon emissions. (Because all of the other depositions involve liner wetting, one can deduce that it is the wetting of the head that results in this impact.) A plausible explanation for why this occurs is that, due to the flow field details, fuel vaporized after flame passage from the films on the head simply does not mix well with oxygen from the crevices during the expansion and exhaust strokes. In particular, it likely does not mix well with oxygen from the piston top land crevice, which is by far the largest crevice in the combustion chamber. Presumably, the wettings on the liner and piston <u>do</u> mix and react with these crevice gases, resulting in their decreased hydrocarbon emissions impact.

And interestingly, <u>again for the same mass of fuel on the wall and the same wall</u> <u>temperature</u>, the impact on the hydrocarbon emissions would be predicted to be virtually the same whether the fuel was wetting the head on the intake or exhaust side of the engine. This fact supports the notion that it is the overall flow field and the decreased mixing with the crevice oxygen (and not, for example, "just" the distance from the exhaust valve) that results in wetting of the head having the largest fraction of the fuel film result in unburned hydrocarbon emissions.

# Collapse of data; "head and upper-liner – exhaust" wetting is no longer an exception

Furthermore, there is a remarkable collapse evident in Figure 5.18 when the hydrocarbon emissions from the liquid fuel films are normalized in this way and plotted against the representative wall film temperature. Except for the "piston and mid-liner" wettings (whose reason for not collapsing in this way is addressed below and subsequently in section 5.10), locations of the same type exhibit very similar behavior with intake versus exhaust side wettings. The hydrocarbon emissions, wall film mass, and representative wall film behavior of the "liner" wettings were previously shown to be very similar for the intake versus exhaust side wettings, so the collapse seen here is to be expected for them.

However, recall that the "head and upper-liner – exhaust" wetting was an exception that exhibited very different hydrocarbon emissions and wall film mass behavior than other wetted locations. (See Figure 5.12 and Figure 5.16.) But when the hydrocarbon emissions from the liquid fuel film are normalized by the mass of fuel on the wall, the "head and upper-liner – exhaust" wetting is no longer an exception! That is, in Figure 5.18 the "head and upper-liner – exhaust" wetting behaves very similarly to the "head and upper-liner – intake" wetting.

It is argued subsequently in section 5.9 that the different hydrocarbon emissions and wall film mass behavior for the "head and upper-liner – exhaust" wetting is due to changes in the wetted footprint as wall temperature increases. Even if this is not the case and some other effect is occurring that is causing its hydrocarbon emissions and wall film mass behavior, the collapse seen here suggests that whatever mechanism results in fuel from the liquid films (at a given location) becoming exhaust hydrocarbon emissions depends upon the absolute amount of fuel present at that location. Thus, the impact on the hydrocarbon emissions for the "head and upper-liner – exhaust" wetting observed earlier in Figure 5.11 and Figure 5.12 was so large precisely because something (believed to be a changing of the wetted footprint) was happening that was

causing a larger mass of liquid fuel to be on the wall. When this is accounted for, the "head and upper-liner – exhaust" wetting behaves similarly to the "head and upper-liner – intake" wetting.<sup>3</sup>

## "Piston and mid-liner" exception

From Figure 5.18, the "piston and mid-liner" wetting on the exhaust side has roughly double the hydrocarbon emissions (as a fraction of fuel on the wall) as the intake side wetting for at all times/temperatures. The wall film masses for these two depositions are similar (see Figure 5.15), which suggests that something else is occurring that results in the hydrocarbon emissions to be lower with the intake side wetting. As described earlier and also supported by data in section 5.10, this is believed to be due to the fact that for the intake-side piston wetting (only) a substantial portion of the fuel vaporized from the film after flame passage is trapped in the combustion chamber at exhaust valve closing.

#### Summary

This section normalized the hydrocarbon emissions due to the liquid fuel films by the corresponding mass of liquid fuel on the combustion chamber surfaces. If one trusts the analysis methodology, this ratio can be thought of as the fraction of the wall film that ultimately becomes hydrocarbon emissions. The values of this fraction ranged from 1 to 6% on a mass basis depending upon the particular wetted location and the wall temperature. As wall temperatures increased this ratio <u>increased</u>. This is best understood by the fact that higher temperatures result in increased vaporization rates, both before and after flame passage.

The wetting of the head exhibited the most significant impact on the exhaust hydrocarbon emissions. On a mass basis, the hydrocarbon emissions due to the liquid fuel films normalized by the corresponding wall film mass ranged from 2.4% at the lowest wall temperature condition to

5.7% at the highest wall temperature condition, and were approximately 50% higher than observed for other locations at the same wall temperature.

The apparent earlier exception of the exhaust-side head wetting behaved very similarly to the intake-side head wetting when compared at the same wall temperature. This suggests that the exhaust hydrocarbon emissions impact of a liquid fuel film at a given location in the combustion chamber depends directly on the mass of fuel in that film. Moreover, it suggests that something – suspected to be a changing of the wetted footprint with wall temperature – was happening causing the wall mass for the exhaust-side head wetting to increase. When this is accounted for, the intake and exhaust side head wettings behave similarly.

Finally, an interesting exception was observed for the "piston and mid-liner – intake" wetting. Whereas the other wetted locations resulted in similar behavior for the intake and exhaust side wettings, for "piston and mid-liner" wettings the intake side wetting exhibited roughly half the impact of the exhaust-side wetting at all temperature conditions. This was surmised to be due to the trapping, for this particular wetted location, of fuel vaporized from the piston fuel film in the combustion chamber at exhaust valve closing.

## 5.8.3. Normalized by the amount of each liquid injection that sticks to the walls

Figure 5.19 plots, for each of the liquid deposition locations, the hydrocarbon emissions from the liquid fuel films (the data of Figure 5.12), normalized by the estimated mass of each liquid injection that sticks to the combustion chamber surfaces, versus the representative wall film temperature. The mass of each injection that sticks to the wall was obtained using the average estimated values of the sticking fractions which are discussed in section 5.7 and shown in Table 5.4. Recall that these values assume that depositions of the same type on the intake and

exhaust side of the engine have the same fraction of fuel sticking to the wall, which may not be accurate as described in section 5.7. Further, recall that these estimated sticking amounts are approximate in nature (due to the sensitivity of the calculations to accumulated errors) but are nonetheless consistent. Thus, looking at the data in this way may be useful in that it attempts to account for differences in the amount of fuel sticking to the combustion chamber surfaces.

When fueling conditions are stable, as they are for the data of Figure 5.19, the amount of fuel sticking to the wall each cycle is equal to the amount of fuel leaving the wall each cycle (see Figure 5.17). Thus, if one trusts the assumptions and methodology used to determine the contribution of the liquid fuel films to the total hydrocarbon emissions<sup>4</sup>, the vertical axis of Figure 5.19 can be thought of as the fraction of injected liquid fuel that is added to the wall film each cycle that becomes hydrocarbon emissions, or equivalently, as the fraction of fuel that leaves the film each cycle that becomes hydrocarbon emissions.

Unfortunately, no significant collapse is seen in the data when plotted in this way. This is could be due to the approximate nature of the calculations and the simplifying assumptions that were described above. More or less, the trends of Figure 5.19 are the same as those described in section 5.4 for Figure 5.12 – which is to be expected since on a relative basis the amounts of fuel sticking to the wall each cycle for the various depositions are not significantly different.

As stated earlier, the estimates of the mass of fuel in the wall film are much more robust than the estimates of the fraction of each liquid injection that sticks to the combustion chamber surfaces. Moreover, these two estimates correlate with one another: more fuel sticking to the wall each cycle was shown, in general, to correspond to a larger stable mass of fuel on the wall.

More data, or at least more accurate data, from which the fraction of each liquid injection that sticks to the combustion chamber surfaces could be determined, is required in order to make

definitive conclusions about the role it plays in the hydrocarbon emissions impact of the liquid fuel films.

If indeed the amount of liquid fuel that sticks to (or is consumed from) the combustion chamber surfaces each cycle does <u>not</u> play a role in the hydrocarbon emissions, which this data appears to suggest since there is no collapse evident, the implication would be that the means by which fuel vaporized from the liquid films becomes hydrocarbon emissions depends upon the amount of fuel in the film (as shown in section 5.8.2) and not necessarily on the how much fuel is gained or lost from the film each cycle. That is, the hydrocarbon emissions impact of liquid fuel at a given location would depend only on the size of the film, and not necessarily on how the film came to be the size it is and/or how much fuel is consumed from the film each cycle.

Ultimately, though, more data is required in order to make a definitive conclusion about how the hydrocarbon emissions behave when expressed as a fraction of the amount of fuel added to (or consumed from) the liquid films each cycle.

## Summary

This section normalized the hydrocarbon emissions due to the liquid fuel films by the corresponding mass of liquid fuel vaporized from (or added to) the combustion chamber liquid fuel films each cycle. If one trusts the analysis methodology, this ratio can be thought of as the fraction of the fuel vaporized from the liquid films that ultimately becomes hydrocarbon emissions.

No collapse in the data was observed when examined in this way. It is unclear if this observation is real or if it is an artifact of the approximate nature of the estimates of the amount of each liquid injection that sticks to the combustion chamber surfaces. More data, or more accurate data, is required in order to make a definitive conclusion.

# 5.9. EXHAUST HYDROCARBON PROFILES WITH AND WITHOUT LIQUID FUEL FILMS

The hydrocarbon emissions data presented up to this point have focused on the <u>total</u>, or equivalently the <u>mass-averaged</u>, hydrocarbon emissions each cycle. The exhaust mass flow rate and the fast hydrocarbon concentration in the exhaust runner were used to generate massaveraged concentrations as described in section 4.2. These mass-averaged concentrations were then used to determine the total mass of hydrocarbons exhausted each cycle as described in section 4.4. This section examines the exhaust hydrocarbon profiles for differences based upon if a liquid fuel film is present, and if a film is present, where it is. The goal in this analysis is to deduce when fuel vaporized from the liquid films is actually exhausted.

<u>Aside</u>: no discernable differences or trends were observed in the exhaust CO and  $CO_2$  profiles with and without liquid fuel films.

In order to reduce the impact of cycle-to-cycle variability, average exhaust hydrocarbon profiles within a given relative air-fuel ratio range are compared to one another below. That is, the cycles of interest in each experiment were classified by what particular range, or "bin", of relative air-fuel ratios they fell into. Then, an average exhaust hydrocarbon profile was generated for each of these "bins". In this process, cycles with and without liquid fuel films present were distinguished from one another.

# 5.9.1. Vapor-only fueling exhaust hydrocarbon profiles

Figure 5.20 shows average hydrocarbon concentration profiles for vaporous only (i.e. isopentane only) fueling from the experiments mentioned previously in section 4.7.2 used to

determine and baseline the hydrocarbon emissions behavior of the engine with vaporous only fueling – in these experiments no liquid fuel was ever deposited. The vertical axis is the measured fast FID concentration. The horizontal axis is the crank angle: the dashed vertical lines indicate exhaust valve opening and closing. The average curve for each of the relative air-fuel ratio bins is plotted; the spacing of the bins was 0.03 units. The particular relative air-fuel ratio ranges, along with the number of cycles found in the dataset within each range and their average relative air-fuel ratio, are shown in the legend.

Looking at a particular profile, the concentration during the relatively flat portion in the middle of the exhaust event is attributed to quench layer hydrocarbons – which are present throughout the entire exhaust event. The rise in the profile near exhaust valve opening is attributed to the outgassing of head and upper combustion chamber crevices during the blowdown process. The rise in the profile near exhaust valve closing is attributed to the exhausting of unburned hydrocarbons from the piston crevices. (For reference, a plot of the typical exhaust mass flow rate is shown in Figure 5.21 and also in Figure 4.1; blowdown constitutes the first 40% of the exhaust flow period and roughly half of the total exhaust flow.)

The general trend is that as the mixture becomes more fuel-rich, the hydrocarbon emissions increase throughout the entire exhaust event. Upon closer inspection, this increase is smallest in the middle of the exhaust event, which as described above corresponds to fuel that was in the quench layer. The increase is largest during the initial and final portions of the exhaust event, which as described above correspond to the exhausting of fuel that was in the quench layer <u>and</u> fuel that was in a crevice region.

#### 5.9.2. Validation of methodology

As a validation of the subsequent comparisons, the stable fueling cycles without liquid deposition, but from experiments in which liquid deposition was performed, are compared below to their corresponding profiles in Figure 5.20 (which were obtained from an experiment in which liquid fuel films were never present). The average exhaust hydrocarbon profiles for these stable cycles without liquid deposition are shown as the solid, colored curves in Figure 5.22 and Figure 5.23. The title of each subplot indicates the particular deposition experiment from which the average profile was generated, along with the number of cycles within that particular relative airfuel ratio range that were found and used to generate the average profile, and their average relative air-fuel ratio. Figure 5.22 shows the relative air-fuel ratio range of 0.93 to 0.96; Figure 5.23 shows the relative air-fuel ratio range of 0.78 to 0.81. (Note that the scales of Figure 5.22 and Figure 5.23 are not identical.) The dashed black line in each of the subplots is the corresponding average exhaust hydrocarbon profile from Figure 5.20 (again, recall that for the experiment from which its data was obtained liquid fuel films were never present). The note at the bottom of the plot lists the number of cycles it is the average of, along with their average relative air-fuel ratio.

The agreement is quite good for all of the experiments and both relative air-fuel ratio ranges. The largest difference between any two curves being compared is roughly 500 ppmC1, which is evident for the 0.93 to 0.96 relative air-fuel ratio range (Figure 5.22) with portions of both the "piston and mid-liner – intake" and "liner – exhaust" experiments and for the 0.78 to 0.81 relative air-fuel ratio range (Figure 5.23) with portions of the "liner – intake" experiment. Most of the experiments, however, exhibit much better agreement, with the maximum discrepancy less than 200 ppmC1 throughout the entire exhaust event. As will be evident

subsequently, the sizes of these differences are small in comparison to those seen when liquid fuel films are present. First, however, the next section addresses the sizes of differences that could be expected due to various sources. Afterward, the differences seen here are reconsidered and then the exhaust hydrocarbon profiles <u>with</u> liquid fuel films are examined.

# 5.9.3. Size of differences due to various sources

The one factor that cannot be accounted for easily is the effect of cyclic variability, especially given the relatively small number of cycles averaged over to generate each profile. For example, variations in the combustion phasing, and thus the cylinder pressure history, could affect the unburned fuel that is in (and outgassed from) the crevices. This impact appears to be small as evident in Figure 5.22 and Figure 5.23, but it could explain the largest differences noted above. Three other potential sources of differences in comparing the exhaust hydrocarbon profiles to one another, all of which can be accounted for, are possible errors in the fast FID calibration, relative air-fuel ratio differences, and wall temperature differences.

The fast FID, the instrument used to measure the hydrocarbon concentrations (see section 2.2.4), was calibrated after every experiment. As mentioned previously in section 5.4, the variation in these calibrations was small: for the range of calibrations obtained in all of the experiments, the variation in a 2000 ppmC1 value is  $\pm$  25 ppmC1. Thus, the possibility of an error in the fast FID calibration causing significant differences in the exhaust hydrocarbon profile is unlikely.

The various hydrocarbon profiles being compared to one another in each subplot here are at different overall average relative air-fuel ratios. As shown in Figure 5.20, the hydrocarbon concentrations do increase throughout the entire exhaust event as the air-fuel mixture becomes

more fuel-rich. However, the size of this increase is relatively small for the relative air-fuel ratio differences between the various profiles that are being compared to one another. The data of Figure 5.20 (which again was for vaporous only fueling) can be used to get a sense of the size of the differences expected in the hydrocarbon concentrations for a given change in relative air-fuel ratio. Curve-fitting the hydrocarbon concentration data in the flat portion of the profiles in the middle of the exhaust event against their corresponding average relative air-fuel ratios results in a very good linear trend: there is a roughly 67 ppmC1 increase in the hydrocarbon concentration for a decrease in relative air-fuel ratio of 0.01 units. Thus, given the relative air-fuel ratio differences present in all of the previous and subsequent comparison plots (which are of the order 0.01 units of relative air-fuel ratio) the effect of a slight difference in relative air-fuel ratio between the profiles being compared is expected to be small.

Finally, there is the possibility of wall temperature (and thus also charge temperature) variation resulting in differences in the exhaust hydrocarbon profiles. The "vaporous only" data of Figure 5.20 to which the other data is compared was obtained in a slightly shorter experiment (280 fired cycles) than was used for the liquid deposition experiments (340 fired cycles). Because the fueling was rapidly varied throughout the entire "vaporous only" experiment (see section 4.7.2), it is difficult to obtain an average profile at a particular temperature condition. Thus, when averaged over the entire dataset the "vaporous only" data of Figure 5.20 can be thought of as corresponding to wall temperature conditions after 140 fired cycles. This number of fired cycles roughly corresponds to the stable cycles in batch 4 of the quasi-steady liquid deposition experiments (see section 3.4). Therefore, depending upon what particular relative airfuel ratio range and/or particular batch of cycles from the liquid deposition experiments are being

examined, there is the possibility of slight temperature differences affecting the hydrocarbon concentrations.

The size of these differences that could be expected can be estimated from data in the study. As shown in section 5.5, a drop in the cycle-average hydrocarbon concentration with wall temperature in the range of 20-40 ppmC1 per °C is expected. Further, from Figure 5.8, a rise in the average wall temperature of the order 7 °C is expected in 60 cycles, which is the length of 2 batches in the quasi-steady liquid deposition scheme. Depending upon the particular relative airfuel ratio range and/or the particular batch in the liquid deposition experiments that is being compared to the vaporous only profile, temperature differences can affect the hydrocarbon profiles approximately 100 to 600 ppmC1. For example, a difference of approximately 150 ppmC1 would be expected when looking at the data of Figure 5.23, which is for the relative airfuel ratio range 0.78 to 0.81 and corresponds to batches 3 and 8 of the quasi-steady scheme. The maximum difference that would be expected would be in comparing the 4th liquid deposition (batch 10) to its corresponding vapor-only profile: the difference expected would a <u>decrease</u> in the exhaust hydrocarbon concentration of approximately 600 ppmC1. <sup>5</sup>

\* \* \*

In light of all of these effects, the agreement seen in Figure 5.22 and Figure 5.23 comparing the stable cycles <u>without</u> liquid deposition, but from the liquid deposition experiments, to cycles from an experiment in which liquid fuel was never deposited, is quite good and validates the subsequent comparisons. The likely cause of the largest discrepancies discussed above (for the "liner – intake" experiment in Figure 5.23 and the "piston and mid-liner – intake" and "liner – exhaust" experiments in Figure 5.22) is thought to be due to variations in combustion phasing and the relatively small number of cycles being averaged over. Most

importantly, the differences observed subsequently when liquid fuel films <u>are</u> present are much larger than could be expected solely due to the aforementioned effects (or are in the opposite direction) – and thus those differences can be attributed to the liquid fuel films. But in light of these effects and the fact that a few experiments exhibit behavior slightly outside the range of what would be expected due to the effects that can be accounted for, when drawing conclusions about the liquid fuel films either large numeric differences or substantial shape differences in the exhaust hydrocarbon profiles are required in order to make firm conclusions about the liquid fuel films.

## 5.9.4. Exhaust Hydrocarbon Profiles with Liquid Fuel Films Present

In Figure 5.24 and Figure 5.25 the average exhaust hydrocarbon profiles for stable cycles <u>with</u> liquid deposition (and that fall within the particular relative air-fuel ratio range being considered) are compared to their corresponding average exhaust hydrocarbon profile with vaporous-only fueling (from Figure 5.20). Note that because of the restriction on the relative air-fuel ratio range, these profiles do <u>not</u> necessarily use or represent all of the cycles used to generate the data in section 5.4. The title of each subplot indicates the wetted location (for images of each wetting, see section 5.2). Although it results in averaging over even fewer cycles, the average profiles for each individual liquid deposition batch are shown in order to show the time/temperature behavior. The legend of each subplot indicates which deposition is which: the earlier deposition at each relative air-fuel ratio is shown as the solid colored line; the later deposition is the dashed colored line. Also shown in the legend is the number of cycles averaged over in order to obtain each average profile, and their average relative air-fuel ratio. Figure 5.24 shows the relative air-fuel ratio range 0.81 to 0.84, corresponding to cycles from the

first and third liquid depositions. Figure 5.25 shows the relative air-fuel ratio range 0.87 to 0.90, corresponding to cycles from the second and fourth depositions. The dashed black line in each subplot is the average exhaust hydrocarbon profile with vapor-only fueling for that particular relative air-fuel ratio range. The number of cycles used to generate it and their average relative air-fuel ratio are shown in the note at the bottom of the figure.

In order to aid in the comparison of the various profiles to one another, with the exception of the "head and upper liner – exhaust" wettings, all profiles <u>within</u> a given figure use the same scale and furthermore the full scale <u>range</u> on the plots in Figure 5.24 and Figure 5.25 are identical (they all span 9000 ppmC1). Thus, one can directly compare differences relative to the vapor-only profiles between plots in the same figure and between the two figures (again, excluding the "head and upper liner – exhaust" wetting). The scale range for the "head and upper liner – exhaust" wetting in both figures is identical, however: it spans 12000 ppmC1.

As mentioned earlier, significant increases in the profiles relative to the vapor-only profile can be attributed to vaporized fuel from the liquid films being exhausted.

It should be emphasized in examining the subsequent hydrocarbon profiles that varying masses of liquid fuel are on the combustion chamber walls for each of the liquid depositions. The mass of liquid fuel on the combustion chamber walls was shown in section 5.8.2 to play a major role in the hydrocarbon emissions from the liquid fuel films, but is not in any way evident or accounted for in the presentation of these exhaust hydrocarbon profiles.

# Trend with Temperature; "Head and upper liner - exhaust" exception

With the exception of the "head and upper liner – exhaust" wetting, the general trend for both relative air-fuel ratio ranges (in Figure 5.24 and Figure 5.25) is that the later liquid deposition at each relative air-fuel ratio exhibits a similar shape but a lower hydrocarbon concentration than the earlier deposition at each relative air-fuel ratio. This is to be expected given the discussion in the previous section regarding the effect of temperature on the hydrocarbon emissions due to fuel in vaporous form.

For the "head and upper liner – exhaust" wetting, however, this behavior is different: during the initial portion of the exhaust event (corresponding to blowdown), for both relative airfuel ratio ranges the later, higher temperature deposition exhibits a higher concentration than the earlier, lower temperature deposition. Furthermore, as shown in Figure 5.24 for the relative airfuel ratio range 0.81 to 0.84 (corresponding to the first and third depositions), the later, higher temperature deposition exhibits a higher hydrocarbon concentration throughout the entire exhaust event.

The fact that the behavior for this particular wetted location is very different than the behavior of the other wetted locations is further evidence that something physically different is occurring with the "head and upper liner – exhaust" wetting. It is thought that there is a change in the wetted footprint with the "head and upper liner – exhaust" wetting. In particular, it is thought that as the exhaust valve temperature rises, liquid fuel that is sprayed at it eventually cannot stick to it: film boiling occurs and a vapor "cushion" is formed that both prevents the fuel spray from sticking to the exhaust valve and further redirects the fuel spray toward the upper liner / head interface, resulting in a larger accumulation of fuel at this location. This notion is supported by the fact that, as described above, for the later liquid depositions more fuel from the liquid films appears to be exhausted during the blowdown process. Fuel on the head / liner interface is expected to be more likely to vaporize and be entrained in the blowdown flow than fuel on the valve head due to the nature of the flow and the inherent stagnation zone below the valve head.

#### Intake versus exhaust wetting

For both relative air-fuel ratio ranges (Figure 5.24 and Figure 5.25), the exhaust side wettings exhibit a larger difference relative to the vapor-only hydrocarbon profile for the "head and upper-liner" and "piston and mid-liner" wettings. This observation is completely consistent with the observations in section 5.4 which examined the cycle-average hydrocarbon emissions each cycle due to the liquid fuel films: for the intake versus exhaust side wettings at these locations, the exhaust side wettings exhibited a larger amount of hydrocarbon emissions from the liquid fuel films each cycle.

It is difficult to generalize for the "liner" wettings as during some portions of the exhaust event the intake side wetting exhibits a larger difference relative to the vapor-only profile, while during other portions of the exhaust event the exhaust side wetting exhibits a larger difference.

# Shapes of profiles - comparing the same locations at different relative air-fuel ratios

For the most part, at a given location the shapes of the exhaust hydrocarbon profiles with liquid fuel films present, as well as the difference in them relative to the vapor-only profile, are very similar at different relative air-fuel ratios as evident in comparing Figure 5.24 and Figure 5.25. The implication of this is that the behavior of the fuel vaporized from the liquid films does not appear to significantly depend on the mixture composition (for this particular range of relative air-fuel ratios).

Upon close examination, however, there is a slight impact of different relative air-fuel ratios: the profiles at the "less fuel rich" condition (Figure 5.25) exhibit a slightly smaller difference relative to the vapor-only profile than the profiles at the "more fuel rich" condition (Figure 5.24). This is clearly evident, for example, with the "liner – intake" wetting. This observation is completely consistent with the observations earlier in section 5.4 that depositions

at the less fuel rich condition exhibit slightly less hydrocarbon emissions from the liquid fuel films than would be expected from an interpolation and extrapolation of the behavior for the more fuel rich condition. This is presumably because more oxidation of the fuel vaporized from the liquid films is occurring with the less fuel rich condition.

#### When is fuel vaporized from the films exhausted?

The ultimate goal of this section is the deduction of when, in the exhaust event, fuel vaporized from the liquid films is actually exhausted. This question is addressed below for the various combustion chamber surfaces. Because every experimental liquid deposition actually wetted multiple combustion chamber surfaces, deductions about the role each of the combustion chamber surfaces play in the observed profiles is necessary. By considering the depositions in turn as is done below, this deduction is possible. To that end, the exhaust hydrocarbon profiles in Figure 5.24 and Figure 5.25 are interpreted below.

# "Head and upper-liner - intake" wetting

For this particular wetting, a liquid fuel film was present mostly on the head on the intake side of the engine, and also to a lesser extent on the upper portion of the liner in the clearance volume of the engine on the intake side of the engine. The exhaust hydrocarbon emissions profile with liquid fuel more or less is offset by a constant amount relative to the vapor-only profile. Upon closer inspection, the offset is slightly larger during the early part of the exhaust event (corresponding to blowdown). The interpretation of this is that fuel from these films is exhausted throughout the entire exhaust event, and further that the fuel in the film is vaporized at a mostly constant rate, with perhaps slightly more vaporization during the blowdown process.

## "Head and upper-liner - exhaust" wetting

The behavior of this wetting is similar to the "head and upper-liner – intake" wetting, with the notable exception being that the concentration with the liquid fuel films present is markedly higher during the strong initial phase of the blowdown process. As described earlier in section 5.8.2, it is thought that wetted footprint is changing with time for this particular targeting. However, this rise during the initial blowdown process is present in all of the depositions, and is roughly of the same magnitude for all of them. Moreover, it is important to recall that when expressed as a fraction of the fuel on the wall this wetting behaved very similarly to the "head and upper-liner – intake" wetting.

An interpretation of these collective facts is that for liquid fuel films near the exhaust valve <u>most</u> of the fuel vaporized from the film is vaporized and exhausted during the initial blowdown process when flow velocities over the film are largest. Since the exhaust side wetting initially has much higher hydrocarbons emitted, but when integrated over the entire cycle behaves similarly to the intake side wetting, there could be a limit on the fraction of the fuel that can be vaporized from the film during each cycle at a given wall temperature condition. This is speculative, but suggested from this data. (It is also supported by the fact that the "liner" wettings exhibit similar behavior when integrated over the entire cycle, but exhibit different shapes relative to the vapor-only profile during the exhaust event.) This effect could be due to the volatility of the various components of the gasoline: for example, all of the "light ends" in the fuel film could be destined to be vaporized and exhausted each cycle – for the "head and upper liner – exhaust" wetting that vaporization would happen during the initial phase of blowdown, whereas for the intake side wetting it takes slightly longer.
Nonetheless, even if this slightly speculative explanation is not correct, it is clear from this data that most of the fuel vaporized from films near the exhaust valve is vaporized and exhausted during the initial blowdown process.

#### "Liner" and "Piston and mid-liner" wettings

Recall that, as described in section 5.2, the "liner" wettings also slightly wet the piston and thus the fact that these two wetted locations exhibit similar exhaust hydrocarbon profiles features is completely expected. The hydrocarbon profiles for the "liner" liquid depositions exhibit an offset relative to the vapor-only profile that grows toward the end of the exhaust event. The "piston and mid-liner" wettings exhibit similar behavior, with the notable difference being that the rise toward the end of the exhaust stroke is more substantial.

The interpretation of this is that fuel vaporized from the liner leaves the film and is exhausted more or less uniformly throughout the exhaust event, which is plausible as it always has a flow over it. Fuel vaporized from the piston films, however, more or less stays stratified near the piston and can only be exhausted when the piston approaches top-center, which is evident from the rise toward the end of the exhaust event. This explains why the "piston and mid-liner" wettings exhibit a steeper rise than the "liner" wettings: more liquid fuel was on the piston for them, and thus more fuel vaporized from the piston film is exhausted toward the end of the exhaust event.

That the fuel vaporized from the piston films could stay stratified near the piston and only be exhausted when the piston is near top-center is supported by data in the next section. First, however, the fact that the hydrocarbon profile for the "piston and mid-liner – intake" wetting is significantly less than its corresponding exhaust side wetting is addressed below.

## "Piston and mid-liner" - intake assessment

The "piston and mid-liner – intake" wetting is a significant exception to the trends observed earlier in the study: its hydrocarbon emissions impact is roughly half that of its corresponding exhaust side wetting at otherwise identical conditions. As described earlier, no error is thought to have occurred in the calibration of the fast FID. This is further supported by the agreement observed in section 5.9.2 in Figure 5.23 and to a lesser extent in Figure 5.22.

Thus, this particular wetting appears to have a decreased impact on hydrocarbon emissions for two reasons: first, like the "head and upper-liner" wettings the blowdown contribution of the liquid fuel film on the liner is less with the intake side wetting. (The "liner" wettings are difficult to generalize: during the initial phase of blowdown the intake side wettings actually have a larger contribution from the liquid fuel films, but during the later phase of blowdown the exhaust side wettings have a higher hydrocarbon contribution from the liquid fuel films.) This, however, appears to be only part of the effect.

Second and more importantly, for the intake side wetting it appears that during the displacement flow period late in the exhaust event that fuel vaporized from the piston film simply is not being exhausted. This is surmised to be due to stratification of fuel vaporized from the piston film near the piston: this vaporized fuel appears to remain near the piston and not be exhausted. That is, it is trapped in the combustion chamber at exhaust valve closing. The possibility of this stratification and subsequent trapping is addressed in the next section.

#### Summary

This section examined the hydrocarbon profiles during the exhaust event in order to deduce when fuel vaporized from the liquid films at a given location was actually exhausted. This methodology was validated by comparing cycles without liquid fuel films present from each

of the experiments to a baseline. This size of differences expected in the profiles due to normal variation or experimental inaccuracies was estimated and found to be small relative to the size of differences observed with liquid fuel films present.

It was possible to identify when (within the exhaust event) fuel vaporized from the liquid films was actually exhausted.

a. Fuel that was vaporized from films on the head was exhausted throughout the entire exhaust event. Slightly more fuel from the film was exhausted during the blowdown process for intake side head wetting, and substantially more fuel was exhausted during the blowdown process for exhaust side wetting.

b. Fuel that was vaporized from films on the liner (in the clearance volume) was more or less exhausted uniformly throughout the entire exhaust event.

c. Fuel that was vaporized from films on the piston was exhausted only in the later portion of the exhaust stroke when the piston approached top-center. That is, fuel vaporized from the piston films remained stratified near the piston – only when the piston approached top center could it be exhausted. For the intake side piston wetting, significantly less fuel was evident as being exhausted, further supporting the notion that for this wetting some fuel vaporized from the piston fuel film remained trapped in the combustion chamber at exhaust valve closing.

## 5.10. NATURAL LUMONISITY IMAGES

It was observed (by eye) during the experiments that in cycles without liquid deposition blue light was emitted from the combustion chamber, whereas in cycles <u>with</u> liquid deposition this same blue light was observed along with orange light near the location of the liquid fuel

film. (Recall that the engine used in the study was a visualization engine.) Although not the focus of this study, a brief exploration of the natural luminosity emanating from the combustion chamber during the experiments was performed. The high-speed camera was used to observe this luminosity in an otherwise dark room for three of the wetted locations in this study.

Figure 5.26 through Figure 5.29 show selected frames from the high speed videos from spark to exhaust valve closing for one sample cycle. The wetted locations are the "head and upper-liner – intake", the "liner – exhaust", and the "piston and mid-liner – intake". (Note that the aperture was one f/ stop more open for the "piston and mid-liner – intake" video, and thus its slight overexposure. Also note that there is residue from the graphite of the piston seal bars that slightly obscures the view into the combustion chamber, especially near the top of the combustion chamber. This residue is most clearly evident in the "piston and mid-liner – intake" images.)

The blue light that was described above can be attributed to the chemiluminescence of the propagating flame; the orange light that was described above can be attributed to radiating soot particles<sup>6</sup> that were formed from fuel vaporized from the liquid fuel film. At the frame corresponding to 62° aTC-expansion these soot particles are first evident. By the frame corresponding to 104° aTC-expansion they are clearly evident after having been "stretched" by the expansion of the combustion chamber.

For the "head and upper-liner – intake" wetting, these soot particles are last evident in the 54° aTC-exhaust frame; prior to the next frame they are all exhausted. For the "liner – exhaust" wetting the soot particles are last evident in the 11° aTC-exhaust frame; as above, prior to the next frame they are all exhausted. For the "piston and mid-liner – intake" wetting, however, the soot particles are evident in the combustion chamber even 2° before exhaust valve closing.

These images and their corresponding videos can be used in two ways: first, the radiating soot particles are effectively tracers for the charge motion in the combustion chamber, providing a sense of the flow pattern in the combustion chamber. Second, and more importantly, the soot particles (which appear to be formed from fuel vaporized from the liquid fuel films) are a likely indicator of the location of other molecules vaporized from the liquid fuel films. For example, one would expect fuel vaporized from the film after the soot formation to "trail" the soot particles in the combustion chamber.

The most relevant observation from this data, then, is for the "piston and mid-liner – intake" wetting: at exhaust valve closing radiating soot particles are evident in the combustion chamber. Thus, it is very likely for this particular wetting that a substantial portion of the molecules vaporized from the liquid fuel films are trapped in the combustion chamber at exhaust valve closing as well. This observation supports the earlier observations that for the "piston and mid-liner – intake" wetting the hydrocarbon emissions due to the liquid fuel films were lower than expected.

# Summary

This section discussed a brief exploration of the natural luminosity emanating from the combustion chamber during the experiments. The chemiluminescence of the normal flame propagation as well as the formation of soot particles from molecules vaporized from the liquid fuel films was observed. These soot particles are an indicator of where other molecules vaporized from the liquid films are likely to be, and thus help to explain the earlier exception of the "piston and mid-liner – intake" data.

### 5.11. THE LEIDENFROST EFFECT

## Overview

Figure 5.30 shows the lifetime of a 3.80 mg drop of gasoline on a metal surface as a function of the surface's temperature. The ambient conditions are 14.5 °C air temperature and 100 kPa ambient pressure. Interestingly, there is a temperature (roughly 160 °C) at which the vaporization rate peaks and the droplet lifetime on the surface is minimum. Below this critical temperature the droplet lifetime increases exponentially as wall temperature decreases, as one might expect.

However, above this critical temperature the droplet lifetime <u>increases</u> as wall temperature increases. This somewhat counter-intuitive observation is due to the so-called "Leidenfrost effect", which corresponds to the transition from the nucleate boiling regime to the film boiling regime. The temperature at which the vaporization rate is maximum (and thus the droplet lifetime is minimum) that marks this transition is commonly referred to as the "Nukiyama point".

The vaporization rate continues to decrease past the Nukiyama point until it reaches another critical temperature. Above this temperature, referred to as the "Leidenfrost point", further increases in wall temperature do result in increases in the vaporization rate. Thus, at the Leidenfrost point the droplet lifetime exhibits a local maximum. (The Leidenfrost point is not evident in Figure 5.30, but is evident in Figure 5.32.) These collective effects are commonly referred to as the "Leidenfrost effect".

The most important observation regarding the Leidenfrost effect is that higher surface temperatures do not necessarily imply higher vaporization rates.

Figure 5.31 shows a schematic of what is physically happening during this transition from nucleate boiling to film boiling. An insulating layer of vapor (from the liquid that

constituted the droplet) is formed that slows the vaporization rate. The presence of this insulating layer explains the decrease in vaporization rate above the Nukiyama point. Eventually, further increases in surface temperature do result in increased heat transfer rates through this vapor layer: as described above the temperature at which this transition occurs (and the droplet lifetime is a local maximum) is referred to as the Leidenfrost point.

To put this phenomenon into context, a common experience of the Leidenfrost effect in one's kitchen is the "dancing" of water droplets that do not stick to a hot skillet. For water at atmospheric pressure, the Leidenfrost point is approximately 200 °C or 400 °F [42].

## Effect of ambient pressure

The behavior of the Leidenfrost effect, and more generally the behavior of liquid drops and films above the saturation point, depends on ambient pressure as well. Figure 5.32 shows the lifetime of a 3.75 mg drop of gasoline as a function of surface temperature and ambient pressure. Both the Nukiyama and Leidenfrost points depend on ambient pressure. It is difficult to discern from this plot, but from other plots in [43] the Nukiyama point ranges from approximately 160 °C at 50 kPa ambient pressure to approximately 260 °C at 1101 kPa ambient pressure.

When in the "lower temperature" regime corresponding to wall temperatures less than approximately 160 °C, at a given wall temperature as ambient pressure increases the droplet lifetime increases. That is, in this regime, vaporization rates at a given wall temperature <u>decrease</u> as ambient pressure increases. Conversely, in the "higher temperature" regime corresponding to wall temperatures greater than approximately 240 °C, at a given wall temperature as ambient pressure increases the droplet lifetime decreases. In this regime, vaporization rates at a given wall temperature <u>increase</u> as ambient pressure increases. The lower temperature regime behavior

is explained by an increased partial pressure of the vaporized fuel molecules above the film as ambient pressure increases (the wall film is so thin that it is essentially at the wall temperature); the higher temperature behavior is explained by a compression of the vapor layer that hinders heat transfer as ambient pressure increases [43].

Previous studies investigating liquid fuel films in the combustion chamber of an internal combustion engine have either surmised or found that the Leidenfrost effect played an important role in explaining trends observed with liquid fuel films (e.g. [43] and [16]).

# Relevance to findings of current study

The gasoline deposited on the combustion chamber walls in this study is the same type of certification fuel as was used to obtain the data of Figure 5.32: all of the wall temperatures encountered in this study (see Figure 5.8 and Figure 5.10) thus appear to be in the "lower temperature" regime described above. However, the exhaust valve temperature was not directly measured in this study and it could be approaching the transition regime, which could explain the behavior observed with the "head and upper-liner – exhaust" wetting.

\* \* \*

Insomuch as the Leidenfrost effect could affect the data and conclusions of this study, as described above all of the wall temperature conditions encountered here (except for possibly the exhaust valve temperature which was not measured) are below the Nukiyama point at <u>all</u> combustion chamber pressures that were encountered. (Recall that the Nukiyama point is the temperature at which the droplet lifetime is minimum: above it the vaporization rate decreases as wall temperature increases due to the presence of an insulating vapor layer.)

As a result, one would expect the behavior encountered here to exhibit consistent trends with temperature. This consistency was in fact actually observed. However, great care must be exercised when extrapolating findings in this study to higher wall temperatures.

Since the Nukiyama point is the temperature at which the liquid fuel films exist for the minimum amount of time, any extrapolation of a trend in the data of this study to a wall temperature for which the liquid fuel films would be expected to no longer exist are more likely an estimate of some average Nukiyama point – and not necessarily a temperature above which liquid fuel in the combustion chamber would be expected to have no effect. As is evident from Figure 5.30, above the Nukiyama point the behavior changes and the liquid fuel films can still survive.

\* \* \*

It should be emphasized that none of the analysis methods used in this study require the fuel behavior to be in any particular regime. It is only in the extrapolation of the results of this study in which care must be exercised. Although an extrapolation of the data might suggest that liquid fuel films would have no effect above a specific temperature, as wall temperature increases the physical behavior of the liquid fuel films could change as described here.

<sup>2</sup> See note 1 above.

<sup>3</sup> The tacit assumption in this entire discussion is that the representative wall film temperature is "good enough" to provide a measure of the actual wall temperature the liquid fuel films are subjected to for <u>all</u> of the depositions in this study, which the evidence and agreement in this paragraph suggests is indeed the case.

<sup>4</sup> See note 1 above.

<sup>5</sup> Note that the methodology of section 4.7, which was used to deduce the hydrocarbon emissions resulting from the liquid fuel films, <u>does</u> take into account wall temperature variation.

<sup>6</sup> It is not clear whether or not the FID measures these soot particles, and thus if they are included in the unburned hydrocarbon emissions measurements or not. Moreover, the extent to which the amount of these soot particles could be different for the various locations cannot be assessed from these images since the camera is monochrome with a specific spectral response curve. A suitable fast response particulate matter analyzer that could have been used to independently measure the particulate matter during the experiments was not available for this study.

<sup>&</sup>lt;sup>1</sup> As described in section 4.7, if the assumptions are changed about how the hydrocarbon emissions due to the liquid fuel are determined, all of the locations would be affected similarly and thus the trends in the data would be unaffected.

Torgating Location	Data Filonomo	T <sub>ambient</sub>	Relative humidity	Pambient	Intake airflow [kg/hr] (averaged over	Motored imep <sub>net</sub> [kPa]	N
	Data Fhename		[%]		last 250 cycles)	(cycles 131-150)	INOTES
Liner – intake	di6a-linerint-lowertemp	22.3	44	99.90	12.13	-92.90	temperature 25°C
No depositon [Liner - intake plate installed]	di6a-linerint-DRY	22.4	46	99.96	12.15	-93.14	
Liner - intake	di6a-linerint-30degC	22.3	48	100.22	12.32	-93.08	
Piston and mid-liner - exhaust	di6a-pistexh	21.9	48	99.70	12.29	-94.27	
Head and upper liner - intake	di6a-headint	21.4	48	99.80	12.24	-94.30	
Head and upper liner - exhaust	di6a-headexh	21.0	52	100.26	12.24	-93.67	
Piston and mid-liner - intake	di6a-pistint	20.4	46	99.70	12.24	-93.98	
Liner - exhaust	di6a-linerexh	20.0	49	100.39	12.20	-93.78	
No depositon* [Liner - exhaust plate installed]	di6a-isoplambdasweep	19.5	49	100.23	12.16	-93.87	See note (*) below
Head and upper liner – intake	di6b-lowtemp_try1	22.8	42	100.47	12.28	-94.61	Initial metal temperature 25°C
Head and upper liner - intake	di6b-headint-30degC	23.6	44	100.31	12.28	-94.41	
Head and upper liner – intake	di6b-headint-lesseramt1	22.9	40	100.31	12.32	-94.36	Decreased amount of liquid deposited
Head and upper liner – intake	di6b-headint-lesseramt2	22.4	39	100.00	12.32	-94.22	Decreased amount of liquid deposited
No deposition* [Head and upper liner - intake plate installed]	di6b-isoplamsweep	22.2	39	100.19	12.29	-94.15	See note (*) below
* Did not use quasi-steady scheme: instead rapidly varied the fueling to baseline the vapor-only emissions.							

Table 5.1: Summary of experiments used in the study

	Approximate Wetted	
<b>Deposition Location</b>	Area [mm <sup>2</sup> ]	<b>Approximate Distribution</b>
Head and upper liner	1210 mm <sup>2</sup>	$\frac{3}{4}$ head, $\frac{1}{4}$ liner + window
Liner	$1020 \text{ mm}^2$	$^{3}/_{4}$ liner, $^{1}/_{4}$ piston
Piston and mid-liner	$1050 \text{ mm}^2$	$\frac{3}{4}$ piston, $\frac{1}{4}$ liner

Table 5.2: Approximate wetted areas	and their	distribution	on the	combustion	chamber	surfaces
for each of the deposition locations						

	Depos (bate	Deposition 1 (batch 2)		Deposition 3 (batch 7)		
	Minimum	Minimum Estimate		Estimate		
Head and upper liner						
Intake	0.81	0.98	0.59	0.70		
Exhaust	0.53	0.53	0.65	0.75		
Liner						
Intake	0.53	0.56	0.48	0.50		
Exhaust	0.60	0.68	0.50	0.54		
Piston and mid-liner						
Intake	0.81	0.87	0.66	0.98		
Exhaust	0.82	0.82	0.78	0.78		

Table 5.3: Estimated fraction of each injection	that sticks to the combustion chamber walls for
the first and third depositions at each location.	Outliers suspected to be due to accumulated
errors are italicized.	

	Minimum	Estimate
Head and upper liner	0.59	0.66
Liner	0.53	0.57
Piston and mid-liner	0.80	0.82

Table 5.4: Average values of the fraction of each injection that sticks to the combustion chamber walls for each deposition location type. Suspected outliers that were italicized in Table 5.3 were excluded from the averaging.

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Figure 5.1: Labeled views of the combustion chamber that are used to subsequently show the wetted footprints for each of the deposition locations.



Figure 5.2: Wetted footprint for the "head and upper liner – intake side" deposition



Figure 5.3: Wetted footprint for the "head and upper liner - exhaust side" deposition



Figure 5.4: Wetted footprint for the "liner - intake side" deposition



Figure 5.5: Wetted footprint for the "liner - exhaust side" deposition



Figure 5.6: Wetted footprint for the "piston and mid-liner - intake side" deposition



Figure 5.7: Wetted footprint for the "piston and mid-liner - exhaust side" deposition



Figure 5.8: Combustion chamber surface thermocouple readings versus time and batch number (for the scheme of section 3.4).



Figure 5.9: Location of the combustion chamber surface thermocouples. (This figure uses the same images as Figure 5.1, but the thermocouple locations are now highlighted in yellow. In black and white the thermocouple locations are the white spots not present in Figure 5.1).



Figure 5.10: Representative wall film temperature for each liquid deposition versus time and batch number (for the scheme of section 3.4).



Figure 5.11: Increase in hydrocarbon emissions due to liquid fuel films versus time in the experiment for various liquid deposition locations and the same mass of liquid injected. The first and third points for each deposition location occur at an overall relative airfuel ratio of 0.83 and are shown with a <u>thick</u> outline; the second and fourth points occur at an overall relative airfuel ratio of 0.87 and are shown with a <u>thin</u> outline.



Increase in Exhaust HC Emissions due to Liquid Fuel

Figure 5.12: Increase in hydrocarbon emissions due to liquid fuel films versus representative wall film temperature for various liquid deposition locations and the same mass of liquid injected. The first and third points for each deposition location occur at an overall relative air-fuel ratio of 0.83 and are shown with a <u>thick</u> outline; the second and fourth points occur at an overall relative air-fuel ratio of 0.87 and are shown with a <u>thin</u> outline.



Figure 5.13: Increase in hydrocarbon emissions due to liquid fuel films versus time in the experiment for "liner – intake" wetting and different initial combustion chamber wall temperatures. The first and third points for each deposition location occur at an overall relative air-fuel ratio of 0.83 and are shown with a <u>thick</u> outline; the second and fourth points occur at an overall relative air-fuel ratio of 0.87 and are shown with a <u>thin</u> outline.



Figure 5.14: Increase in hydrocarbon emissions due to liquid fuel films versus time in the experiment for "head and upper liner – intake" wetting comparing different initial combustion chamber temperatures, amounts of injected liquid fuel, and identical conditions on a different day. The first and third points for each deposition location occur at "more rich" overall relative air-fuel ratio and are shown with a <u>thick</u> outline; the second and fourth points occur at a "less rich" overall relative air-fuel ratio and are shown with a <u>thin</u> outline. See text section 5.5.2 for further explanation of relative air-fuel ratio differences.



Figure 5.15: Estimated mass of liquid fuel on the combustion chamber surfaces during the stable fueling cycles for the liquid depositions of Figure 5.11 and Figure 5.12 versus time in the experiment. 5.00 mg of liquid fuel was sprayed at the combustion chamber walls each cycle of liquid deposition. The first and third points for each deposition location occur at an overall relative airfuel ratio of 0.83 and are shown with a <u>thick</u> outline; the second and fourth points occur at an overall relative airfuel ratio of 0.87 and are shown with a <u>thin</u> outline.



Figure 5.16: Estimated mass of liquid fuel on the combustion chamber surfaces during the stable fueling cycles for the liquid depositions of Figure 5.11 and Figure 5.12 versus representative wall film temperature. 5.00 mg of liquid fuel was sprayed at the combustion chamber walls each cycle of liquid deposition. The first and third points for each deposition location occur at an overall relative air-fuel ratio of 0.83 and are shown with a <u>thick</u> outline; the second and fourth points occur at an overall relative air-fuel ratio of 0.87 and are shown with a <u>thin</u> outline.



Figure 5.17: Schematic of the pathways injected liquid fuel can follow. The width of the arrows indicate typical proportions for the data of this study. For reference, the stable wall film mass was roughly 1 to 4 times the injected liquid mass for the data of this study.



Figure 5.18: Increase in hydrocarbon emissions due to liquid fuel films (per cycle) normalized by the mass of fuel on the wall during the stable deposition cycles for each deposition versus representative wall film temperature. The first and third points for each deposition location occur at an overall relative air-fuel ratio of 0.83 and are shown with a <u>thick</u> outline; the second and fourth points occur at an overall relative air-fuel ratio of 0.87 and are shown with a <u>thin</u> outline.



Figure 5.19: Mass increase in hydrocarbon emissions due to liquid fuel films (per cycle) normalized by the estimated mass of each injection that sticks to the wall versus representative wall film temperature. The first and third points for each deposition location occur at an overall relative air-fuel ratio of 0.83 and are shown with a <u>thick</u> outline; the second and fourth points occur at an overall relative air-fuel ratio of 0.87 and are shown with a <u>thin</u> outline.



Figure 5.20: Exhaust hydrocarbon profile for vaporous only fueling, at various relative air-fuel ratios. Exhaust valve opening and closing are 33° bBC and 1° bTC, respectively. Note that the relative air-fuel ratio difference between the curves is not uniform, but is very nearly 0.03 units of  $\lambda$ .



Figure 5.21: Typical normalized exhaust flow rate (the integral of the curve is 1) obtained via the methodology of section 4.2. Exhaust valve opening and closing are 33° bBC and 1° bTC, respectively, and indicated by the vertical dotted lines.



Dashed line: experiment without any liquid deposition, 44 cycles, avg  $\lambda$  = 0.9426

Figure 5.22: Comparison of average exhaust hydrocarbon profiles from stable cycles <u>without</u> liquid deposition in the liquid deposition experiments to the corresponding average profile from an experiment in which liquid fuel was never deposited, for the relative air-fuel ratio range 0.93 to 0.96. The dashed vertical lines indicate exhaust valve opening and closing.



Dashed line: experiment without any liquid deposition, 23 cycles, avg 2= 0.8026

Figure 5.23: Comparison of average exhaust hydrocarbon profiles from stable cycles <u>without</u> liquid deposition in the liquid deposition experiments to the corresponding average profile from an experiment in which liquid fuel was never deposited, for the relative air-fuel ratio range 0.78 to 0.81. The dashed vertical lines indicate exhaust valve opening and closing.



Figure 5.24: Comparison of average exhaust hydrocarbon profiles from stable cycles with liquid deposition to the corresponding average profile with vapor-only fueling, for stable cycles from the  $1^{st}$  and  $3^{rd}$  liquid depositions in each experiment that are in the relative air-fuel ratio range 0.81 to 0.84. The dashed vertical lines indicate exhaust valve opening and closing. Note that, in order to more clearly show the differences that are present, the scale in the "head and upper liner – exhaust" plot is different than the others.



Dashed line: No liquid deposition, 30 cycles, avg  $\lambda$ = 0.8836

Figure 5.25: Comparison of average exhaust hydrocarbon profiles from stable cycles with liquid deposition to the corresponding average profile with vapor-only fueling, for stable cycles from the  $2^{nd}$  and  $4^{th}$  liquid depositions in each experiment that are in the relative air-fuel ratio range 0.87 to 0.90. The dashed vertical lines indicate exhaust valve opening and closing. Note that, in order to more clearly show the differences that are present, the scale in the "head and upper liner – exhaust" plot is different than the others.



Figure 5.26: Natural luminosity for three different liquid deposition locations, frames 1-3.



Figure 5.27: Natural luminosity for three different liquid deposition locations, frames 4-6.

	Head and upper liner - intake	Liner - exhaust	Piston and mid-liner - intake
54° aBC exhaust			D
97° aBC exhaust			
140° aBC exhaust			

Figure 5.28: Natural luminosity for three different liquid deposition locations, frames 7-9.



Figure 5.29: Natural luminosity for three different liquid depositions, frame 10.



Figure 5.30: Lifetime of a 3.80 mg drop of gasoline on a metal surface as a function of the surface temperature. From [41].


Figure 5.31: Schematic of the Liedenfrost effect. From [42].



Figure 5.32: Lifetime of a 3.75 mg drop of gasoline on a metal surface as a function of the surface temperature and ambient pressure. From [43].

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# 6. SUMMARY AND CONCLUSIONS

#### 6.1. OVERVIEW OF STUDY

An experimental study was performed that examined the impact of combustion chamber liquid fuel films on engine-out hydrocarbon emissions for a spark ignition engine operating at conditions representative of the "warmup" phase of operation. The experimental setup was unique in that it combined direct visual observation of the liquid fuel films, direct measurements of the combustion chamber surface temperatures, measurements of the exhaust species concentrations with millisecond-level response time, and precisely targeted deposition of liquid fuel films at desired locations in the combustion chamber.

In order to examine the behavior of the liquid fuel films under well-controlled firing engine conditions, a fueling strategy was developed in which most of the fuel was delivered as a vapor and controlled amounts of liquid fuel were deposited at selected locations on the combustion chamber surfaces. The "vapor" fuel used in this study was isopentane; the liquid fuel used was gasoline. The amount of liquid fuel injected each cycle was approximately 17% of the stoichiometric fuel amount; for the data obtained in this study with liquid fuel films present the overall delivered relative air-fuel ratio was approximately 0.83 or 0.87. (Thus, the liquid fuel constituted approximately 14% or 15% of the total fueling.)

A fueling scheme for the experiments was devised that not only enabled the experiments to be tightly controlled but also facilitated the determination of the amount of liquid fuel actually on the combustion chamber surfaces. Means of processing the raw species concentration data were developed that enabled the determination of the amount of burned and unburned fuel molecules exiting the combustion chamber each cycle. Further, a physically-based methodology

for deducing the portion of the total engine-out exhaust hydrocarbon emissions that were due to the liquid fuel films was devised – this data is the main result of this study. A specific set of assumptions was used in order to deduce these hydrocarbon emissions that were due to, or from, the liquid fuel films. These assumptions were confirmed to be reasonable by other (independent) data in the study.

#### 6.2. SUMMARY OF RESULTS

1. At otherwise identical conditions and total delivered fuel amounts, an increase in exhaust hydrocarbon emissions was always observed when liquid fuel films were present in the combustion chamber.

2. As will be evident below, much of the data in this study collapses when compared at the same wall temperature and differences in the amount of liquid fuel actually on the combustion chamber surfaces for each liquid deposition are accounted for.

3. As stated above, for each of the targeted locations in the combustion chamber the same amount of liquid fuel was sprayed at the combustion chamber walls each cycle. When expressed as a fraction of the amount of liquid sprayed at the walls, the exhaust hydrocarbon emissions from the liquid fuel films ranged from 1 to 10% on a mass basis, depending upon the wetted location and the wall temperatures. In general, as wall temperatures increased the hydrocarbon emissions from the liquid fuel films decreased. An extrapolation of this trend suggested that for wall temperatures greater than 85 to 110 °C (depending upon the particular wetted location), the liquid fuel films would have no impact on the exhaust hydrocarbon emissions. Such an

extrapolation may not, however, be valid as the liquid fuel films may enter a different regime of behavior for wall temperatures greater than approximately 150 °C – see section 5.11.

The wetting of the head near the exhaust valve resulted in slightly different behavior than all of the other wetted locations: its hydrocarbon emissions impact had a different shape and moreover did not monotonically decrease with wall temperature. Furthermore, it also had the largest impact on the exhaust hydrocarbon emissions at all wall temperature conditions.

4. However, depending upon the wetted location, different amounts of liquid fuel actually stuck to the combustion chamber walls. These differences in the amount of fuel sticking to the wall are largely explained by differences in the impingement angle of the spray on the respective surfaces, which was consequence of the particular method chosen for depositing the liquid fuel films in this study. The difference in the amount of fuel on the combustion chamber walls was quantified in two ways: either as the fraction of each liquid injection that stuck to the wall or as the amount of liquid fuel on the wall when stable liquid deposition conditions were reached. These two quantities correlated with one another.

a. The targeting of the liner (which also slightly wet the piston) resulted in the smallest fraction of each spray sticking to the wall (about 50%) and the smallest masses of liquid fuel on the combustion chamber walls. These masses ranged from approximately 2 times (at the lowest wall temperature conditions) to approximately <sup>1</sup>/<sub>2</sub> (at the highest wall temperature conditions) the amount of liquid fuel injected each cycle. The intake and exhaust side wettings exhibited virtually identical behavior.

b. The targeting of the piston (which also wet the liner when the spray rebounded off the piston) resulted in the largest fraction of each spray sticking to the wall (about 80%) and the

largest masses of fuel on the combustion chamber walls. These masses ranged from approximately 4 times (at the lowest wall temperature conditions) to approximately 2 times (at the highest wall temperature conditions) the amount of liquid fuel injected each cycle. As above, the intake and exhaust side wettings exhibited virtually identical behavior.

c. The targeting of the head (which also wet the upper portion of the liner) resulted in an intermediate fraction of each spray sticking to the wall (about 65%) and in general an intermediate mass of fuel on the combustion chamber walls. The behavior of the wall masses for this targeting is difficult to generalize as, like the hydrocarbon emissions above, the exhaust side wetting exhibited different behavior than the other wettings.

5. By normalizing the exhaust hydrocarbon emissions from the liquid fuel films by the corresponding mass of liquid fuel on the walls, a collapse of the trends was observed.<sup>1</sup> The exhaust hydrocarbon emissions from the liquid fuel films each cycle <u>normalized by the corresponding amount of liquid fuel on the combustion chamber walls</u> ranged from approximately 1 to 6% on a mass basis. As wall temperatures increased this ratio <u>increased</u>. This is best understood by the fact that higher temperatures result in increased vaporization rates, both before and after flame passage.

When normalized in this way, the apparent earlier exception of the exhaust-side head wetting behaved very similarly to the intake-side head wetting when compared at the same wall temperature. This suggests that the exhaust hydrocarbon emissions impact of a liquid fuel film at a given location in the combustion chamber depends directly on the mass of fuel in that film. Moreover, it suggests that something – suspected to be a changing of the wetted footprint with

wall temperature – was happening causing the wall mass for the exhaust-side head wetting to increase. When this is accounted for, the intake and exhaust side head wettings behave similarly.

Further, the wetting of the head exhibited the most significant impact on the exhaust hydrocarbon emissions. On a mass basis, the hydrocarbon emissions due to the liquid fuel films normalized by the corresponding wall film mass ranged from 2.4% at the lowest wall temperature condition to 5.7% at the highest wall temperature condition, and were approximately 50% higher than observed for other locations at the same wall temperature.

An interesting exception was observed in this data with the wetting of the piston on the intake side of the engine. Whereas the wetting of the liner and head resulted in similar behavior for the corresponding intake and exhaust side wettings, for the piston wetting the intake side wetting exhibited roughly half the impact of the exhaust-side wetting at all temperature conditions. This was surmised to be due to the trapping, for this particular wetted location, of fuel vaporized from the piston fuel film in the combustion chamber at exhaust valve closing. This explanation was supported by natural luminosity images showing radiating soot particles formed from the piston film being trapped in the combustion chamber at exhaust valve closing.

6. By examining the hydrocarbon profiles in the exhaust runner, it was possible to identify when (within the exhaust event) fuel vaporized from the liquid films was actually exhausted. This was achieved by comparing the average exhaust hydrocarbon profiles with liquid fuel films present to corresponding profiles without liquid fuel films present.

a. Fuel that was vaporized from films on the head was exhausted throughout the entire exhaust event. Slightly more fuel from the film was exhausted during the blowdown process for intake side head wetting, and substantially more fuel was exhausted during the blowdown process for exhaust side wetting. b. Fuel that was vaporized from films on the liner (in the clearance volume) was more or less exhausted uniformly throughout the entire exhaust event.

c. Fuel that was vaporized from films on the piston was exhausted only in the later portion of the exhaust stroke when the piston approached top-center. That is, fuel vaporized from the piston films remained stratified near the piston – only when the piston approached top center could it be exhausted. For the intake side piston wetting, significantly less fuel was evident as being exhausted, further supporting the notion that for this wetting some fuel vaporized from the piston fuel film remained trapped in the combustion chamber at exhaust valve closing.

7. Finally, referring to Figure 5.17, the liquid fuel behavior observed in this study can be summarized as follows:

- a) When liquid fuel is injected:
  - i) some fraction of the spray, on average approximately  $^{2}/_{3}$ , stuck to the combustion chamber surfaces, establishing a wall film.
  - ii) the remaining fraction of the spray, on average approximately 1/3, did <u>not</u> stick to the combustion chamber surfaces. It either vaporized in transit to the wall or rebounded from the wall and subsequently vaporized.
- b) On average, the wall film stabilized at 2-3 times the amount of liquid fuel injected each cycle.
- c) When the fueling conditions are stable, the amount of fuel vaporized from the liquid film each cycle is equal to the amount of liquid fuel added to the film each cycle. Of the fuel vaporized from the film each cycle:
  - i) most of the fuel burned, either before or after flame passage
  - ii) a small fraction, for this study 1 to 10% of the amount of liquid fuel injected each cycle, contributed unburned hydrocarbon emissions.

#### 6.3. CONCLUSIONS

#### 6.3.1. Effect of fuel film location

When expressed as a fraction of the liquid fuel sprayed at the combustion chamber walls each cycle, wetting of the head on the exhaust side of the engine resulted in the largest impact on the exhaust hydrocarbon emissions. However, an intermediate effect was that a larger amount of liquid fuel was actually on the combustion chamber surfaces for this particular wetting relative to the other locations. When this is accounted for by normalizing the emissions due to the liquid fuel films each cycle by the corresponding stable amount of liquid fuel on the combustion chamber walls for each deposition, wetting of the head on both the intake and exhaust side of the engine behaved very similarly (when compared at the same wall temperature), and exhibited the largest impact on the exhaust hydrocarbon emissions. This is surmised to be due to the decreased likelihood of fuel from these films "finding" oxygen from crevice gases with which to react prior to being exhausted.

In contrast, the furthest location from the exhaust valve, the wetting of the piston on the intake side of the engine, resulted in the smallest impact on the engine-out hydrocarbon emissions.

For wetting of the head and liner, when comparing at the same wall temperature conditions and accounting for differences in the amount of fuel that was on the combustion chamber walls as described above, wetting the intake versus the exhaust side of the engine showed very similar impacts on the exhaust emissions.

For wetting of the piston, however, at these particular conditions wetting the piston on the intake side of the engine exhibited roughly half the impact as wetting of the piston on the exhaust side of the engine. As described above, this is surmised to be due to the trapping of fuel

vaporized from the film in the combustion chamber at exhaust valve closing. (Note that the compression ratio of this engine is 6, which is relatively low in comparison to modern engines.)

Given all of these facts, the effect of the fuel film location does not readily generalize, but it can loosely be interpreted as depending mostly on the distance from the exhaust valve. Perhaps more correctly, the two main factors involved in the impact the liquid film location has on the exhaust hydrocarbon emissions are: (1) how likely fuel vaporized from the film will find oxygen to react with (while charge temperatures are high enough for oxidation to occur), and (2) how likely any fuel vaporized from the film will be drawn out of the combustion chamber by the exhaust flow.

### 6.3.2. Effect of wall temperature

The effect of the wall temperature that the liquid fuel films are subjected to is more straightforward. An increased wall temperature increases vaporization rates. Thus, at a given location there is less fuel on the combustion chamber walls as wall temperatures increase, but a larger fraction of that fuel vaporizes after flame passage and subsequently contributes to exhaust hydrocarbon emissions. The decrease in wall mass is more rapid: for a constant amount of fuel sprayed at the combustion chamber surfaces, as wall temperature increased, the impact of the liquid fuel films on the exhaust hydrocarbon emissions decreased.

A secondary effect of the wall temperature was to affect the wetted footprint, in particular when the exhaust valve was wetted in this study. This is thought to have occurred at higher wall temperature conditions because the exhaust valve temperature was so high that film boiling occurred when the spray impinged upon it, resulting in no fuel sticking to the exhaust valve and a redistribution of the liquid fuel.

#### 6.3.3. Effect of the amount of liquid fuel sprayed at the combustion chamber walls each cycle

Although not the primary variable in this study, for one particular wetted location experiments were conducted that varied the amount of liquid fuel that was sprayed at the combustion chamber walls. All of the effects observed above were proportional to the amount of liquid fuel injected. Moreover, there appeared to be a threshold amount of injected liquid fuel below which an <u>extrapolation</u> of this data suggested would have no impact on the exhaust hydrocarbon emissions. Furthermore, this critical amount appeared to increase as wall temperatures increased. Presumably, for any amount less than the critical threshold amount, all of the injected liquid fuel in a given cycle would completely vaporize and oxidize in that cycle.

\* \* \*

A methodology was developed in this study for deducing the portion of the total engineout exhaust hydrocarbon emissions that were due to liquid fuel films in the combustion chamber. As is evident from the discussion above, much of the data collapsed when compared at the same local wall film temperature and differences in the amount of liquid fuel actually on the combustion chamber surfaces were accounted for. The combination of the unique capabilities of the setup and the particular methodology developed in this study enabled these quantities to be determined.

The insights gained from this study could be used to guide future engine cold start fueling strategies that are aimed at reducing engine-out hydrocarbon emissions. More generally, the methodology established by this study can be applied to future fuel accounting studies.

<sup>&</sup>lt;sup>1</sup> Estimates of the fraction of each injected liquid spray that actually stuck to the combustion chamber walls were not as accurate as the estimates of the mass of liquid fuel on the walls, and thus it was difficult to discern if they too could prove useful in understanding the hydrocarbon emissions impact of the liquid fuel films.

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Appendix A: Single- and Few-Shot Fuel Film Buildup and Decay Data (poor engine sealing)

The data in this section supports the assumption of linear superposition that was made in section 4.6. It was obtained at the conditions of section 3.3, but with "poor" engine sealing. The analysis methodology of section 4.4 was applied to determine the net mass of fuel exiting the combustion chamber each cycle: the "injected liquid fuel exiting the combustion chamber" each cycle was determined by subtracting the amount of vaporous fuel per cycle from the total mass of fuel exiting the combustion chamber each cycle.

For this experiment, the engine was operated on vaporous fuel (isopentane) and liquid gasoline was deposited on the combustion chamber walls for 1, 2, 4, and 7 consecutive cycles and allowed to clean out. Roughly 8 mg of liquid fuel was injected each cycle. Plotted in Figure A.1 is an overlay of the net mass of this injected liquid fuel exiting the combustion chamber each cycle versus cycle number for each of the liquid depositions. The first cycle of liquid deposition is cycle 6 for all of the profiles, and each length (i.e. number of consecutive cycles) of deposition was performed twice in the experiment.

The shape of both the buildup and decay are consistent for all number of consecutive injections, which is to be expected if linear superposition is valid. (This can be seen from examining Figure 4.14 for various number of consecutive injections.)

A further way to validate the assumption of linear superposition is to simply assume a profile for a single injection and actually linearly superpose it to generate "expected" profiles for 2, 4, and 7 consecutive injections, and then compare them to the actual data. This is done in Figures A.2 to A.4. The linear superposition of the proposed single injection profile is shown as the dotted line with square symbols.

As is most evident in the 7 consecutive injections of Figure A.4, the assumed profile is probably slightly too "fast" (it builds up and decays too rapidly) and possibly too "high" (stabilizing at too high a level). However, given these facts the agreement between the superposition of this assumed profile and the actual profiles is fairly good.

Thus, given the consistency in the shape of the buildup and decay of the liquid fuel films and the agreement obtained by assuming an approximate profile for a single injection and actually linearly superposing it and comparing it to the actual data, for this engine at these conditions the assumption of linear superposition made in section 4.6 is reasonable.



Figure A.1: Net mass of injected liquid fuel exiting the combustion chamber versus cycle number for varying numbers of consecutive injections. For all profiles the first cycle of liquid deposition is cycle 6.



Figure A.2: Actual and "generated from superposition" clean-out profiles for 2 consecutive injections.



Figure A.3: Actual and "generated from superposition" clean-out profiles for 4 consecutive injections.



Figure A.4: Actual and "generated from superposition" clean-out profiles for 7 consecutive injections.