Reducing Cold Start Hydrocarbon Emissions from Port Fuel Injected
Spark Ignition Engines with Improved Management of Hardware & Controls

by

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Submitted to the Department of Mechanical Engineering on May 5, 2006 in Partial Fulfillment of the Requirements for the Degree of Doctor of Philosophy in Mechanical Engineering

ABSTRACT

An experimental study was performed to investigate strategies for reducing cold start hydrocarbon (HC) emissions from port fuel injected (PFI) spark ignition (SI) engines with better use of existing hardware and control systems. Engine experiments and computer simulations were used for three major phases of the project: the effect of variable valve timing on first cycle mixture preparation, the interaction between fuel injection and valve events, and the development of a flow reactor exhaust manifold for fast catalyst light-off.

For the first cycle of cranking, delaying the intake valve opening (IVO) creates a pressure difference across the valve, resulting in strong but brief shear flow to facilitate atomization; delayed IVO also produces a cooler charge due to the expansion process before IVO. It was observed that the in-cylinder equivalence ratio increased with delayed IVO, primarily by displacing the lean portion of the stratified cylinder charge back into the port. However, HC emissions for the first cranking cycle increased with delayed IVO.

With closed valve injection, injection timing has no significant impact on mixture preparation or emissions. With open valve injection, however, HC emissions scale with both valve lift and mass flow because of increased cylinder wall wetting. By timing split injection such that the second injection event hits the overlap back flow, a small mixture preparation and emissions benefit was achieved. Earlier IVO results in a longer back flow period, however the impact on mixture preparation is small. The observed reduction in HC emissions resulted from a higher residual gas fraction due to early IVO, which yielded later combustion phasing, which in turn yielded increased post-flame oxidation.

Under steady-state cold coolant conditions, operation of a 4-cylinder engine with three cylinders running rich and the fourth used to pump air into the exhaust manifold resulted in near total oxidation of CO and HC at sufficiently retarded spark timing. Exhaust gas temperatures and enthalpy flow rates were significantly higher than for the conventional engine configuration at fast idle. Using this strategy to perform real cold starts proved challenging without the additional hardware needed for sufficient control over air flow to the engine.

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<th>Abbreviation</th>
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<tr>
<td>AFR</td>
<td>Air Fuel Ratio</td>
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<tr>
<td>ATDC</td>
<td>After Top Dead Center</td>
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<tr>
<td>BDC</td>
<td>Bottom Dead Center</td>
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<tr>
<td>BTDC</td>
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<td>Closed Valve Injection</td>
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<td>Constant Pressure</td>
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<td>Engine-Out Hydrocarbons</td>
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<td>Federal Test Procedure</td>
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<td>Hydrocarbon</td>
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<td>Hydrocarbon Sampling Module</td>
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<td>Main Control Unit</td>
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<td>MFB</td>
<td>Mass Fraction Burned</td>
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<td>Noise, Vibration, and Harshness</td>
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<td>Sauter Mean Diameter</td>
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<td>Start of Injection</td>
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<td>Abbreviation</td>
<td>Description</td>
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<td>SULEV</td>
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<td>Top Dead Center</td>
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<tr>
<td>VVT</td>
<td>Variable Valve Timing</td>
</tr>
<tr>
<td>UEGO</td>
<td>Universal Exhaust Gas Oxygen (Sensor)</td>
</tr>
</tbody>
</table>

- **m**: Mass
- **$\dot{m}$**: Mass flow rate
- **n**: Hydrogen to carbon ratio of fuel, i.e. CHₙ
- **nₐ**: Moles of air
- **nᵣ**: Moles of fuel
- **p**: Pressure
- **q**: Average number of C atoms in a fuel molecule
- **T**: Temperature
- **W**: Molecular weight
- **X**: Mole fraction

- **λ**: Lambda, normalized AFR
- **Φ**: Phi, fuel air equivalence ratio
- **ρ**: Density
- **τ**: Tau, time scale
- **θ**: Theta, crank angle degrees
- **R**: Universal gas constant
Chapter 1 - INTRODUCTION

1.1 Background and Motivation

As emissions regulations in the U.S. have become increasingly stringent, the primary means of achieving the necessary reductions in tailpipe emissions of unburned hydrocarbons (HC), carbon monoxide (CO) and nitric oxides (NOx) has been the three-way catalyst. Figure 1-1 shows the trend in U.S. emissions regulations over the past several decades, noting in particular the emissions levels possible with catalysts that operate at 90% conversion efficiency and 99% conversion efficiency, respectively. On the most current, cleanest applications, close-coupled, fast light off catalysts can reach their operating temperature, about 250°C, in roughly 10-20 seconds. However, during those first seconds of engine operation, the cold catalyst has almost no effect on engine-out emissions, resulting in high tailpipe emissions. Particularly for HC emissions, this period before catalyst light off yields the majority of total trip emissions. In a modern vehicle that meets the Super Ultra-Low Emissions Vehicle (SULEV) emissions standards, the first 20 seconds of the Federal Test Procedure (FTP) can account for 80% to 90% of the total tailpipe HC emissions. [1] Thus, with such highly effective catalysts, reducing HC emissions is fundamentally a problem of reducing cold start HC emissions.

HC emissions during this period before the catalyst is operational are high primarily due to the presence of a large amount of liquid fuel in the engine as a result of poor mixture preparation. [2] To reduce cold start HC emissions, mixture preparation for the crank, run-up, and initial idle must be improved. Beyond this, catalyst light-off time must be reduced as much as possible. Figure 1-2 shows the cumulative tailpipe emissions for a Low Emissions Vehicle (LEV) based on a FTP city drive cycle. Catalyst light-off is achieved in about 30 seconds, at which point roughly 70% of the cumulative emissions have been produced. The LEV emissions standard is an average of 75 mg/mile of HC. This can be contrasted with the SULEV standard of 10 mg/mile, which requires catalyst light-off closer to 10 seconds.

1.2 Engine Behavior During Cold Start

Figure 1-3 depicts a typical cold start. During cranking, the starter motor spins the engine at low speed (200-300 Revolutions Per Minute or RPM), and thus the port air velocity is low compared to typical engine operation at several thousand RPM. Coupled with cold port wall
and valve temperatures, near-atmospheric Manifold Absolute Pressure (MAP), and the absence of the reverse blow back flow to atomize and redistribute the fuel at Intake Valve Opening (IVO), this constitutes an unfavorable environment for fuel evaporation.

Thus in the first cycle, much fuel is injected, of which only a small portion (typically 20-30% at 20°C ambient temperature) is vaporized to create a combustible mixture. [3] The significant residual liquid fuel is a major source of HC emissions for both the first and the subsequent cycles. [4] Therefore, achieving low HC emissions depends strongly on improving the fuel delivery process in the first cycle. After the engine first fires, the speed increases rapidly; this is known as the run-up or flare. During this brief period of several cycles, conditions in the engine – RPM, MAP, etc. – are changing rapidly, and this extreme transient makes achieving low HC emissions difficult. Next, the engine speed settles to a quasi-steady, “fast idle” period, typically 1200-1400 RPM. It is in this period that most of the exhaust gas enthalpy flow for catalyst light off is produced, however engine surfaces are still relatively cold and thus HC emissions are still significant.

While the catalyst warms up relatively quickly, the engine itself warms up on a much longer time scale, typically several minutes. In port fuel injected (PFI) engines, the fuel is injected toward the back of the intake valve, so that during normal engine operation the liquid fuel vaporizes quickly due to heating from the valve. When the engine is cold, however, vaporizing the fuel becomes much more difficult, and thus excess liquid fuel must be injected in order to yield a combustible air-fuel mixture. Since the liquid fuel is a significant source of HC emissions, this practice results in very high HC emissions until the catalyst reaches its operating temperature.

Under normal operating conditions, the engine is controlled by feedback from an oxygen sensor to the Engine Control Unit (ECU). The oxygen sensor effectively monitors the air-fuel ratio, and adjusts the injected mass of fuel in order to maintain a stoichiometric mixture. For the first 10-20 seconds of a cold start, however, engine control is open loop, and the ECU must predict how much fuel to inject for the given conditions. To ensure a robust startup under a variety of conditions and allowing for differences in consumer fuels, engine calibrations are generally very conservative and inject more than enough fuel to start the engine. This also results in higher HC emissions.
1.3 HC Emissions Mechanisms

Several mechanisms contribute to HC emissions in port fuel injected engines, as shown in Figure 1-4. Liquid fuel and/or air-fuel mixture may become trapped in crevices in the combustion chamber, including the piston ring pack. Fuel in these crevices escapes combustion, and may be released during the expansion and exhaust strokes to exit the combustion chamber as HC emissions. For fully-warm engines, this is believed to be the single largest source of unburned HC in the exhaust. [1] Liquid fuel may also be absorbed into oil layers on the cylinder walls and deposits on the piston crown, but these are believed to contribute only 5% to 10% of the total HC emissions even for cold engines. [1] Particularly when the engine is cold and the combustion chamber surfaces are far below their fully-warm temperatures, flame quenching will contribute to HC emissions, since there will be a boundary layer of air-fuel mixture close to these surfaces that will escape combustion. However, this is not considered a major source of HC emissions. [1] For cold engines, liquid fuel entering the combustion chamber is by far the single largest source of HC emissions. Takeda et al found that most engine-out hydrocarbons (EOHC) result from the cylinder wall fuel film evaporating during the expansion stroke. [4]

Another cause of high HC emissions is the problem of partial burns and misfires, although these are the result of poor mixture preparation and/or engine control, rather than a fundamental source of HC emissions. [1] Additionally, if the air-fuel ratio (AFR) is extremely rich, some HC will inevitably escape combustion, but this is an engine control issue rather than a fundamental source of unburned HC. [2] Another practical source of HC emissions rather than a fundamental one is exhaust valve leakage; as deposits build up on the exhaust valve seat, the valve will not fully close during the compression stroke and unburned air-fuel mixture will be exhausted as emissions. [2]

An additional, critical source of HC emissions during cold start is the rapid speed/load transients experienced by the engine during the speed flare. [2] Injecting precisely the correct amount of fuel under rapidly changing conditions is extraordinarily difficult, resulting typically in conservative fueling strategies in order to avoid misfires due to overly lean mixtures.

Of the fundamental sources of HC emissions, only one can be directly mitigated. By improving mixture preparation, a greater fraction of the injected fuel mass will enter the combustible charge, and thus less injected mass is required to achieve a stoichiometric AFR.
This will reduce the amount of liquid fuel in the combustion chamber and thus reduce EOHC. Good mixture preparation is also essential to avoiding misfire and partial burns.

Direct injection (DI) gasoline engines offer some further insight into the sources for HC emissions. Under normal operation, direct injection engines offer very low HC emissions because fuel is vaporized in-cylinder from very small droplets produced by high-pressure injection; there is effectively no liquid fuel mass in the form of a port wall film. However, during the cold start period, DI engines suffer from high HC emissions due to poor mixture preparation. Since the high pressure fuel pump is typically cam driven, the first few seconds of engine operation do not benefit from the same high injection pressure. DI injectors operating at low pressure produce very large droplets, resulting in significant wetting of the cold cylinder walls and piston crown. [5] The droplets too large to vaporize will experience diffusion burning. For PFI engines, studies have shown that liquid fuel on the cylinder walls is a major source of EOHC, and the same is true for DI engines. Additionally, direct injection can result in charge stratification, with overly rich, quenched regions near the walls, and leaner combustion in the center of the cylinder leading to lower temperatures and reduced post-flame oxidation.

1.4 Previous Work

Numerous strategies to improve HC emissions have been investigated since cold start emissions started becoming more critical in the early- to mid-1990s. These strategies fall into two major categories – reducing engine-out HC emissions and achieving faster catalyst light-off to reduce tailpipe emissions. [6] To reduce engine-out HC emissions, several areas have been investigated, including such broad categories as optimized cold start calibration, improved engine geometry and cylinder head design, variable valve timing, and improved mixture preparation. [6] [7] [8]

1.4.1 Mixture Preparation

Improved mixture preparation has been achieved by using fuel injectors at a higher fuel pressure, using air-assisted injection, and using a heated tip. Each of these results in finer droplets from the injector tip, and thus better mixture preparation and lower HC emissions. [6] [9] Takeda et al found that open valve injection (OVI) resulted in more cylinder wall wetting
and thus higher EOHC compared to closed valve injection (CVI). With fine-spray injectors, however, reduced wall wetting subsequently lowered emissions. [4]

While much work has looked at different injector types, less attention has been paid to the specific details of fuel injection strategy, including injection timing and duration with respect to valve timing. Variable valve timing (VVT) is now available on many production engines, but it is used primarily for fully-warm operation, and its potential to reduce engine-out HC emissions during cold start has only begun to be evaluated. FEV developed a prototype vehicle with a fully variable, electromechanical valvetrain, which allowed for unthrottled operation. With many degrees of freedom for intake and exhaust valve timing and duration and cycle-by-cycle cold start engine management, they were able to achieve dramatic reductions in cold start HC emissions. [10] However, such VVT systems remain impractical for current or near-future production vehicles because of cost, reliability, and noise, vibration, and harshness (NVH) issues. A previous study of simple intake cam phasing during cold start found that by delaying IVO, the coefficient of variation (COV) of net indicated mean effective pressure (NIMEP) in the first 200 cycles was reduced from the original timing while the exhaust HC remained unchanged. [11] A recent study used a simulated cold start procedure to evaluate the effect of exhaust valve timing and duration on HC emissions, finding that early Exhaust Valve Opening (EVO) combined with early Exhaust Valve Closing (EVC) reduced emissions by 27% prior to catalyst light off. [12] However, these studies did not investigate the details of how new valve timing strategies impacted mixture preparation.

1.4.2 Catalyst Light-Off

Numerous methods for reducing catalyst light-off time have been studied, ranging from simple calibration changes to radical hardware such as electric catalyst heaters. Heavy spark retard results in hotter exhaust temperatures, with the additional benefit of yielding more post-flame HC oxidation, at the expense of combustion stability. [13] Secondary air injection (SAI) has been implemented for many years in production vehicles. With this strategy, the engine is run rich, and air is injected near the exhaust valve to oxidize the rich combustion products. SAI yields hotter exhaust temperatures as well as lower emissions into the cold catalyst. SAI systems use either passive injection of secondary air or an air pump to achieve better control of air injection. [14] Additional research has focused on using exhaust gas ignition (EGI) to rapidly
light off the catalyst. Research dating back as far as the 1960s has sought to reduce emissions by this method, using a spark or glow plug to ignite the mixture. However, the early systems only achieved ignition with warmed up exhaust components. The more recent system developed in the early 1990s by Ford and others achieved combustion at cold start temperatures. This method used an afterburner to burn extremely rich (fuel-air equivalence ratio $\Phi$ as high as $\sim 2.0$) exhaust gases from the engine with additional air pumped into the exhaust stream. The quantity of secondary air is thus nearly equal to the air entering through the intake, requiring a very large pump. The presence of hydrogen in the mixture is critical to achieving combustion with cold exhaust system components. Once the flame was established, the mixture was leaned out somewhat, but still rich enough to sustain the flame until the catalyst was hot. \[15\] EGI was further investigated in \[16\] with regard to potential failure modes and negative impacts on durability. However, the primary drawback of this system is that it requires a very large air pump to provide enough air for a stoichiometric mixture, which presents problems for cost, packaging, and other practical concerns. One proposed alternative is to intentionally misfire the engine for several seconds during cranking. The unburned air-fuel mixture is then ignited upstream of the catalyst by glow plugs, thus heating the catalyst. \[17\] Obviously, this creates drivability problems due to an excessively long crank. A later study by the same group misfired only two cylinders, and used the other two running cylinders to both start the engine and provide excess air. \[18\]

Hardware changes to the exhaust manifold and catalyst can also reduce catalyst light-off time, including use of a close-coupled catalyst near the engine to minimize heat losses, higher precious metal content, and thinner substrate walls to reduce the thermal inertia of the catalyst brick(s). \[19\]

1.5 Objectives - Overview

The overall objective of this project is to achieve lower HC emissions using existing, current-production engine hardware and control systems, such that any benefits could be quickly and cost-effectively realized in production vehicles. The following methods of reaching said objective were investigated:

- Understand the effect of intake valve timing on first cycle mixture preparation and emissions.
- Clarify the interaction between injection strategy and intake valve timing during the fast idle period.
  - For closed valve injection, the effect of valve timing on over-lap backflow and the subsequent effects on HC emissions.
  - For open valve or split injection, the interaction between the overlap backflow and injection timing and subsequent effects on emissions.
  - For the above injection/valve strategies, the effect of injector behavior.
- Achieve faster catalyst light off using a flow reactor exhaust manifold.

1.5.1 First Cycle Mixture Preparation

The first phase of research looked at the effect of valve timing on mixture preparation for the first cycle of cranking. One potential method for promoting better fuel vaporization for the first cycle of cranking is to delay the intake cam phasing. [11] The result is sub-atmospheric cylinder pressure at IVO, which drives a much stronger initial forward flow past the valves. This stronger flow aids strip atomization of the fuel film in the port, resulting in finer droplets and enhanced evaporation. The lower initial cylinder pressure would also help evaporation. In addition, the flow work done by the pressure drop across the intake valves would increase the charge temperature. All of these effects would favor fuel vaporization, thus reducing the required injected mass and lowering emissions from liquid fuel.

There are several key drawbacks to this strategy. First, the expansion of the trapped air in the cylinder between EVC and IVO cools the charge, such that the first fuel entering the cylinder hits a mass of cold air, which inhibits vaporization. For fixed cam duration, late IVO also results in late Intake Valve Closing (IVC), thus reducing both the effective compression ratio and the compression duration. This will result in less vaporization during the compression process. Late IVC also reduces volumetric efficiency, and the first cycle torque will be subsequently reduced. With less torque, the engine will start more slowly, a condition which is undesirable both for emissions and for the customer.

Thus, this phase of the project sought to determine, through both simulation and experiments, the effects of changing IVO timing (with a fixed cam profile) on engine behavior. Simulations were used to assess the thermodynamic effects on charge temperatures, pressures,
and flows, and engine experiments were conducted to determine the impact on mixture preparation and HC emissions.

1.5.2 Fuel Interaction with Valve Events During Fast Idle

The second phase of the project again looked at the effect of valve events, specifically the interaction between flow past the intake valves and the fuel injection event during the fast idle period. Unlike the first cycle of cranking, which has no residual burned gases from a previous cycle, mixture preparation during the fast idle period depends heavily on back flow into the port. When the intake valve opens, hot, burned gases from the previous fired cycle flow into the intake port, contributing to fuel vaporization – since the intake valve and port surfaces are still relatively cold, this back flow contributes significantly to mixture preparation. During the intake stroke, forward flow into the cylinder also contributes to mixture preparation, by strip atomizing fuel droplets off the port film(s).

With this in mind, this phase of the project sought to understand the interaction between valve timing, back and forward flow, and mixture preparation. To study this under more controlled circumstances than a 4-cylinder crank start, and to avoid cylinder-to-cylinder variations, the engine was modified to operate as a single-cylinder engine in addition to the existing, stock multi-cylinder mode. For the closed valve injection case, the effect of variable valve timing and the subsequent effect on backflow were evaluated in terms of mixture preparation, engine out HC emissions, and combustion quality. For the case of open valve or split injection, where the injection event is divided into closed and open valve portions, the interaction between back and forward flow and injection timing were investigated.

1.5.3 Flow Reactor Exhaust Manifold

The third phase of the project sought to reduce HC emissions from the exhaust side, by achieving rapid catalyst light-off with a flow reactor exhaust manifold. This concept uses one non-firing cylinder and three rich cylinders to oxidize rich combustion products from the latter in the exhaust manifold, in order to both minimize catalyst-in emissions and maximize catalyst-in enthalpy flow. A related concept was previously studied in gasoline direct injection (GDI) engines. [20] The potential benefits of this method include fewer sources for liquid fuel (i.e. fuel is injected to only three cylinders), rich operation and higher load for better combustion quality,
hot exhaust for high catalyst-in enthalpy flow, and low catalyst-in CO and HC emissions. Perhaps most critically, this concept requires no additional hardware, whereas SAI or EGI require large, loud air pumps and associated plumbing. The primary drawbacks of this concept include rougher idle due to only three fired cylinders, and potentially lower catalyst-in enthalpy flow than operating four cylinders with SAI.
Figure 1-1 - Trends in U.S. passenger car emissions, HC and NOx on log-log scale.

Figure 1-2 - Cumulative FTP tailpipe HC emissions for a LEV. Also shown is vehicle speed versus time.
Figure 1-3 - Typical engine behavior during a cold start.

Figure 1-4 - Sources of HC emissions for a PFI engine.
Chapter 2 - EXPERIMENTAL APPARATUS AND PROCEDURES

2.1 Engine Systems

This project consists of three major phases of experiments: the effect of intake cam phasing on first cycle fuel delivery, the interaction between fuel injection strategy and valve events during fast idle, and the investigation of a flow reactor exhaust manifold. All experiments were conducted using a Chrysler 2.4L four-cylinder engine, however the first set of experiments was performed using a 2000 model year (MY) engine, while the second two sets were performed using a newer, 2003 MY engine in a different test cell. Both engines feature four valves per cylinder with dual overhead cams and are nearly identical in design; however there are subtle differences as listed in Table 2-1.

In both cases, the engines were fully set up and instrumented for testing. The second engine in particular was set up to allow for numerous potential test variables: the injection system, both in terms of injector design and injection strategy; valve timing; spark timing; and fuels. In addition, a heater/chiller system allowed control of the engine coolant, oil, and fuel temperatures to simulate various cold start conditions. The spark timing and fuel injection were controlled by a custom, MIT-developed programmable computer controller. Moreover, the second engine was set up to run either as a normal, 4-cylinder engine, or as a motored single-cylinder engine. For quasi-steady- or steady-state testing, the use of the single-cylinder mode allowed greater control and isolation of test variables. In this mode, the engine was driven by a 10 hp electric motor coupled to the dynamometer.

A set of indexed cams allowing the valve timing to be varied in +/- 1 crank angle degree (CAD) increments was available for use on both engines. By shifting the cam pulley relative to the timing belt in addition to using the special cams, the valve timing can be varied with great precision over a wide range.

2.1.1 Engine Dynamometer Systems

Both engines utilized eddy current, absorbing-only dynamometers controlled by Digalog 1022a controllers. The 2000 MY engine used a Froude Consine AG-80 dynamometer, while the 2003 MY engine used a Digalog AE-150 dynamometer. Both are essentially identical in design and functionality, with the AE-150 having higher maximum power absorption. In addition, the
2003 MY engine was equipped with a 10 hp Leeson WattSaver electric motor coupled to the opposite end of the dynamometer. This allowed the engine to be motored for single-cylinder testing, purging of residual liquid fuel, and other useful purposes.

2.1.2 Engine Control System

For the 2000 MY engine, a custom controller circuit was designed to inject fuel only once to cylinder #4, with the other injectors disabled. Thus the engine would not warm up significantly from a single fired cycle, and repeated test points could be taken much more quickly. The fuel pulse was triggered by using the stock engine controller to trigger a pulse generator, which allowed custom control of the pulse width. Fuel was injected with the intake valve closed, and spark timing was controlled by the ECU.

For the 2003 MY engine used in the second and third phases of the project, the engine was controlled using a custom C++ based code and electronics developed in-house. The code operates by reading in crank angle and bottom dead center (BDC) compression pulses in order to determine engine timing. Spark timing is set by prescribing the timing for spark discharge as well as the charging duration for the ignition coil. Fuel injection timing is prescribed in terms of injection timing and pulse width in microseconds. For some experiments, a feedback controller based on the universal exhaust gas oxygen (UEGO) sensor was utilized. Communication with a second PC allowed for on-the-fly changes in fuel pulse width, spark timing, or exhaust $\lambda$ values. For cold start experiments, fueling and/or spark were controlled cycle-by-cycle, in some cases with different fueling for different cylinders as needed.

2.1.3 Cylinder Pressure Measurements

Both engines were equipped with in-cylinder piezoelectric pressure transducers mounted through the back of the cylinder head in Cyl. #4. The 2000 MY engine used a Kistler 6051 transducer, while the 2003 MY engine used a larger Kistler 6125a transducer. In both cases, the transducer was coupled to a Kistler 5010B charge amplifier to yield a voltage output. For the 6051 transducer, response was calibrated using a dead weight tester, while the 6125a transducer was factory-calibrated and installed directly in the engine. To provide a reference pressure for pegging the in-cylinder pressure signal, a Data Instruments Model SA absolute pressure transducer was installed in the intake manifold of both engines. The in-cylinder pressure was
pegged assuming that the pressure was equal in both cylinder and manifold at BDC of the intake stroke. Correct crank angle phasing was verified by motoring the engine and plotting cylinder pressure and volume on a log-log scale.

2.1.4 Data Acquisition System

Data was acquired using a National Instruments PCI-6023E data acquisition board connected to a BNC-2090 analog input device. Data was recorded and processed using Labview 7.0, with post-processing in MATLAB. Data was sampled every CAD, and cycle data was recorded using a BDC compression pulse superimposed on the pressure trace.

2.1.5 Fast Response Emissions Measurements

Exhaust HC measurements were taken with a Cambustion HFR400 Fast Flame Ionization Detector (FFID). The FFID system consists of a vacuum pump connected to the FFID Main Control Unit (MCU), which connects to the Hydrocarbon Sampling Module (HSM) through a 2 m conduit containing electrical and gas handling lines. Connected to the HSM is a sampling probe which is inserted into the exhaust system through Swage-Lok fittings. The sampling probe is maintained at a high temperature, typically 180°C, in order to prevent condensation in the sample line. When the sample reaches the HSM, it is burned in a hydrogen-air flame, resulting in the production of ions. For HC fuels, the number of ions is very closely proportional to the number of carbon atoms in the fuel. Thus, by using an ion collector to produce a signal in response to ion formation, the output is generated in proportion to the concentration of HC in the sample. This output is amplified in the MCU and subsequently output to the data acquisition system as a voltage. See Figure 2-1 for schematic of FFID operation.

The FFID output depends on the mass flow rate entering the sample chamber through the probe; the HFR400 is designed to provide a constant mass flow rate irrespective of pressure fluctuations in the exhaust system. The MCU pressure controller regulates the pressure difference across the sample tube and the gauge pressure in the constant pressure (CP) chamber. In order to maintain mass flow during fluctuating pressure conditions in the exhaust manifold, the CP chamber must be operated at very low absolute pressure. For sampling close to the exhaust valve where pressure fluctuations are strongest, an additional volume was connected to the CP chamber to provide additional damping. No such additional volume was necessary.
further downstream where pressure fluctuations were smaller. Typical values were 100 mmHg vacuum for the pressure drop across the sampling tube and 400 mmHg for the CP vacuum.

The calibration of the FFID is not internally regulated, and thus it is subject to drifting slowly over time. As such, the unit was calibrated periodically to ensure consistent measurements. Calibration was performed by flowing a reference span gas over the tip of the probe, typically 2000 ppm propane (C$_3$H$_8$) with balance nitrogen. In this region the FFID is highly linear, and further calibration points were not necessary.

For in-cylinder FFID measurements, the probe was installed in a special sampling spark plug. Calibration for in-cylinder measurements was performed by flowing mixtures of air and propane across the probe tip outside the engine, using mass flow controllers to produce mixtures of 1%, 2%, 4%, and 8% propane in air.

Previous work found that exhaust FFID measurements were not sensitive to sampling location with respect to a given cross section of the exhaust system, thus the exhaust HC concentration was assumed for all tests to be spatially uniform in the directions normal to the flow. [21]

CO measurements were taken using a Cambustion NDIR500 fast response CO/CO$_2$ analyzer. The unit uses a small sample chamber operating below atmospheric pressure to draw in sample gas through small heated capillaries. Infrared (IR) light passes through the sample gas and then through optical filters mounted on a chopping wheel. By utilizing reference and blank filters on the wheel, changes in temperature and IR emitter performance can be corrected. Each individual gas in the sample will absorb IR at a particular frequency; by measuring the amount of IR absorbed by the sample gas at the characteristic frequencies for CO and CO$_2$, the concentration of CO/CO$_2$ in the sample can be determined. Time response is approximately 8 ms. [22] Figure 2-2 shows a schematic of the NDIR500.

The detectors are enclosed in remote sample heads connected to the main unit by 10 m conduits housing the electrical and gas handling lines. The analyzer is controlled by a computer with self-calibration software. Calibration is performed using calibration tubes fitted over the sample probes which pass zero gas and two span gases each for both CO and CO$_2$ over the tip of the probe(s). Since the response of the NDIR500 is non-linear, a linearized calibration is obtained by means of a transfer function fitted by the PC.
2.1.6 Air and Fuel Mass Flow Rates

On the 2003 MY engine, intake air mass flow was measured using an EGI Master-Touch Series 8000MP/NH laminar flow element mass airflow sensor (MAFS). Fuel flow rates for both engines were obtained by calibrating the fuel injectors at a range of fuel rail pressures to derive a correlation between injection pulse width, pressure differential across the injector, and injected mass. See Figure 2-3 below for the injector calibration for the 2000 MY engine, and Figure 2-4 for the calibration for the 2003 MY engine. Where necessary, corrections were used for fuel temperature. Calibrations differ due to differences in injector design between the two engines.

Additionally, for the 2003 MY engine an exhaust mass flow measurement system was added in order to estimate exhaust mass flow rates during the cold start transient, when the time response of the MAFS was too slow due to a large upstream damping tank. The system consisted of a pitot tube apparatus with an Omega PX163 fast response differential pressure sensor, along with a fast response thermocouple to measure temperature. From this data, exhaust mass flow rate could be estimated.

2.1.7 Relative Air/Fuel Ratio Measurements

Exhaust gas relative AFR was measured using an NTK L1H1 wide-band UEGO. For various tests, the sensor was located either in a single, isolated exhaust runner to measure $\lambda$ for a single cylinder, or further downstream to measure overall $\lambda$ for the engine. The sensor consists of a narrow-band oxygen sensor coupled with a pump cell and diffusion chamber. The pump cell, with catalytic reactions at the electrode surfaces, either consumes oxygen or oxidizes rich combustion products. Rich exhaust gases will produce a high voltage resulting in a pump current to oxidize the CO, H$_2$, HC, etc. Lean exhaust gases produce a low voltage and a pump current in the opposite direction to consume free oxygen. These actions continue until the reference voltage is achieved, and the pump current needed to produce this equilibrium is proportional to $\lambda$. The control electronics convert the pump current into a voltage output. [23]

2.1.8 Temperature Measurements

Temperatures were measured using chromel-alumel (type K) thermocouples. For measurements of intake port and valve surface temperatures, Omega CO cement-on thermocouples were affixed to the back of the intake valve and two locations on the intake port.
for Cylinder #1 of the 2003 MY engine. For exhaust gas measurements in the flow reactor experiments, Omega high-temperature 1/16” thermocouples with exposed junctions were used. Exhaust system surface temperatures were measured with Omega self-adhesive film thermocouples. Other temperatures (coolant, oil, etc.) were monitored using Omega 1/8” shielded junction thermocouples.

2.1.9 Test Fuel

All experiments, on both engines, were performed using Chevron Phillips UTG91 reference fuel. See Table 2-2 for fuel properties.

2.2 Methodology

The following section describes the test methodology used in each of the three major sets of experiments performed for this project.

2.2.1 First Cycle Mixture Preparation

The first cycle cranking tests made extensive use of the special indexed cam pulleys, which allowed the cam phasing can be varied +/- 7 CAD in 1 CAD increments, relative to stock valve timing. In addition, by slipping the timing belt relative to the pulley, the timing could be varied +/- 17.14 CAD per cam pulley tooth. Thus, with combinations of the cam indexing system and rotating the pulley relative to the timing belt, many different intake valve timings were possible.

In-cylinder FFID measurements were used to determine the mixture AFR and a second FFID probe was mounted in the exhaust manifold runner for Cylinder #4 at 12 cm from the exhaust valve seat. At this location, the strong jet flow of the blow down process was avoided so that the sample point was, in an averaged sense, exposed to the cumulative displacement flow of the exhaust gas. See Figure 2-5 for the locations of key instrumentation.

Given that this set of experiments dealt only with the first cycle of cranking, it was not desirable to do a full warm-up of the engine following each start, since the subsequent cool-down process would take several hours, and only a few seconds of data was desired. Thus, in order to ensure a repeatable starting condition, the following procedure was developed.
First, the intake port and cylinder were purged with clean, dry, compressed air fed directly into the intake runner of the test cylinder (#4), to remove any residual fuel from the previous test. This was done with the engine positioned such that the intake valves were open, thus cleaning out both port and cylinder. To verify the removal of residual fuel, the engine was cranked without injecting until the in-cylinder FFID reading was consistently below 1% of the value for a stoichiometric in-cylinder mixture. Then, the engine was positioned in the same starting position for each test, in the middle of the compression stroke for Cylinder #2, one of the engine’s natural stopping positions. This also ensured closed valve injection for Cylinder #4 at the same timing relative to IVO for each test. Finally, the battery voltage was maintained with an external charger, to keep cranking speed consistent.

For each test, five channels of data were recorded, including cylinder pressure, in-cylinder and exhaust FFID measurements, fuel pulse width, and a BDC marker. Data was sampled every CAD.

Table 2-3 shows the test matrix used; the two variables were intake cam phasing (relative to stock timing) and injected fuel mass. Cam timing was controlled using the indexed cam pulleys discussed above, while injected mass was varied using the commanded pulse width and injector calibration curve in Figure 2-3. Cam delay values refer to the change in IVO timing relative to stock, e.g. a cam delay of 17.14 (one tooth on the timing belt) refers to IVO 17.14 CAD after stock timing (1 CAD before top dead center or BTDC). All valve timings are defined based on the 0.15 mm lift points. Each point of the test matrix in Table 2-3 was repeated 5 times.

Spark timing was controlled by the ECU, with factory spark timing of 10 CAD BTDC. Preliminary investigations indicated that the engine would not fire with IVO delayed more than 22.14 CAD after stock timing, due to the reduction in effective compression ratio. Exhaust valve timing was not changed from the stock values.

Injected mass values in Table 2-3 correspond to fuel injector pulse widths of 40, 60, and 80 ms. The pulse width of 40 ms was chosen as the baseline value since it was close to the stock, ECU-controlled pulse for the first cycle at a nominal ambient temperature of 23°C. However, in a real-world engine, there will be residual fuel in the port and cylinder from the previous shutdown. Thus, with the purged engine used here for repeatability, a pulse width of 40
ms resulted in a considerably leaner in-cylinder mixture than what would normally be encountered in the first cycle of cranking.

The raw data acquired by the FFID and pressure transducer required several steps of post-processing in order to yield useful engine information. The FFID was calibrated against known mixtures of propane (C₃H₈) and air for the in-cylinder measurements, resulting in data for the HC mole fraction of C₁ (X₀). The fuel molecule was represented as (CHₙ)q. For the fuel used, n = 1.89 was the ratio of hydrogen to carbon atoms; the value of q = 7.90 was to match the average molecular weight of the fuel. Using W to denote the molecular weights, the measured value of X₀ was related to the AFR by:

\[ X₀ = \frac{q}{1 + (AFR) \frac{W(CHₙ)}{W_{air}}} = \frac{1}{1 + (AFR) \frac{W(CHₙ)}{W_{air}}} \]  

(1)

The fuel equivalence ratio Φ can then be obtained from the AFR solved from the above equation.

\[ Φ = \frac{(AFR)_{stoch.}}{\left(1 - \frac{1}{q} \frac{W_{air}}{W(CHₙ)} \right) \left(1 - \frac{1}{X₀} \frac{W_{air}}{W(CHₙ)} \right)} \]  

(2)

The total exhaust HC mass per cycle (m_HC) was computed by integrating the exhaust FFID measurement weighted by a calculated exhaust mass flow rate (per crank angle), dm_e/dθ.

\[ m_HC = \int dm_e(\theta) X₁(\theta, 0) \frac{W(CHₙ)}{W_b} d\theta \]  

(3)

where \(W_b\) is the burned gas molecular weight. The C₁ mole fraction at the cylinder exit (X₀(θ, 0)) is related to the measured value (X₀(θ, L)) at distance L from the valve by:

\[ X₀(\theta, 0) = X₀(\theta + \theta_i + \theta_d, L) \]  

(4)

where \(\theta_i\) is the internal delay (in crank angles) of the FFID, and \(\theta_d\) is flow delay from the exhaust valve to the sampling point. The value of \(\theta_d\) is obtained from solving the following equation:

\[ Ve = \int_{\theta}^{\theta + \theta(\theta)} \frac{dm_e(\theta')}{d\theta} \frac{RT_e(\theta')}{W_b P_e} d\theta' \]  

(5)
where $V_e$ is the volume of the exhaust port from the cylinder exit to the sampling point, $R$ is the universal gas constant, $p_e$ is the exhaust pressure and $T_c$ is the charge temperature computed by assuming adiabatic expansion in the blow-down process and constant temperature in the displacement process.

2.2.2 Fuel Interaction with Valve Events During Fast Idle

Fast idle testing was performed using the 2003 MY engine in single-cylinder mode. The UEGO was relocated to the exhaust runner of Cylinder #4, which was isolated from the non-firing cylinders. Intake cam timing was varied using the same method described above for first cycle mixture preparation experiments. FFID data was recorded primarily in a mixing/damping tank located downstream from the exhaust manifold, to provide steady-state cold coolant data.

For the purposes of these experiments, the fast idle test condition was selected to be 1400 RPM and 1.25 bar average NIMEP, with engine coolant controlled to a nominal value of 22°C, +/- 2°C. This corresponds to the fast idle condition about 7-10 seconds into the cold start, once the speed flare has settled and the MAP has stabilized. At this point in the cold start, the engine coolant temperature (ECT) has not changed significantly, however intake valve surfaces warm up on a much shorter time scale.

In order to accurately match the fuel puddle development process of a real cold start, the valve and intake port temperatures were measured using fast-response thermocouples for several real, 4-cylinder cold starts. Figure 2-6 shows typical engine RPM, MAP, and the temperatures on the back of the valve, the intake port floor, and the intake port wall. At 7 seconds into the cold start, the back of the intake valve has reached 43°C, and is warming up at approximately 10°C every 3 seconds. By contrast, the intake port has only reached about 25°C by this point, i.e. it is only marginally higher than the ECT. Note that in the first second or so, the fast-response thermocouple signals were not valid due to high levels of leakage current from the starter motor.

Motoring at 1400 RPM with cold coolant, the intake valve surface reached a steady state temperature of 44°C, while the port wall and floor temperatures stabilized at 26°C and 27°C, respectively. Thus, the fast idle test condition could be reasonably approximated by motoring the engine at 1400 RPM, then injecting fuel and firing for a given number of cycles; subsequent valve and port warm-up depended on the duration of firing. Simulated fast idle tests could then
be repeated by motoring the engine for several thousand cycles in order to purge out any residual fuel from the port and cylinder while also cooling the valves back down to the steady-state motored temperature.

Prior to running simulated fast idle tests as described above, the interaction between injection timing and valve events was first investigated by running steady-state cold coolant tests. ECT was fixed at ~22°C, and for a given test condition the engine was fired continuously at 1400 RPM, $\lambda = 1.00$, and the desired NIMEP (1.25 bar for most tests). Spark timing was fixed at 35 CAD BTDC; for the conditions tested, this is a few degrees retarded from maximum brake torque (MBT) timing, however the torque curve is quite flat around MBT for this engine.

Under steady-state cold coolant conditions, the intake valves will reach a steady temperature of approximately 100°C, considerably higher than the fast idle value, although port wall temperatures remain representative of cold start conditions. Thus, while fuel was injected onto a hot surface, these tests nonetheless provided valuable information about how the back and forward flow past the intake valves affect EOHC emissions. All tests were performed in comparison to the following baseline case:

- 1400 RPM
- 1.25 bar NIMEP
- Stock cam phasing

Variables investigated included intake cam phasing, load, RPM, and injector design/rail pressure.

This phase of the project also sought to determine whether there might be some benefit to split injection, where the total injected fuel mass is divided into two parts: a CVI pulse and an OVI pulse. With appropriate injection timing, the OVI fuel spray could “hit” the backflow in such a way that it improves vaporization and mixing. The effect of split injection was first tested using the same steady-state cold fluids method discussed previously. Test conditions were 1400 RPM, 1.25 bar NIMEP, $\lambda = 1.00$, stock 4-hole injector at 4 bar rail pressure, and stock valve timing. Tests were performed for a range of injection timings centered on the back flow period, with sweeps performed for 25%, 50%, and 75% split fractions\(^1\) along with a single pulse. The end of injection (EOI) timing of the single pulse and the second pulse for split injection was

---

\(^1\) Split fraction is defined here as the percent of the total injected mass that is injected in the second pulse, nominally during the back flow period.
varied from 300 to 400 CAD after top dead center (ATDC), in 10 CAD increments. For split injection, the EOI timing for the first pulse was fixed at 200 CAD ATDC.

To simulate an actual fast idle condition, the engine was motored to 1400 RPM for many minutes, to ensure the valves reached their steady-state motoring temperature (44°C). The dynamometer was engaged in constant torque absorption mode to pre-load it for better control, while the electric motor was programmed for the desired speed. MAP was controlled by an external throttling valve. Fuel metering was prescribed in order to give $\lambda = 1.00 \pm 0.05$ within about 5 cycles, as measured by the UEGO for Cylinder #4. After 10 cycles, the engine was typically at $\lambda = 1.00 \pm 0.02$ and the NIMEP value did not change substantially. Average NIMEP was 1.25 bar. Achieving a steady AFR required several iterations of the prescribed cycle-by-cycle fueling for each test case.

The desired operating conditions were achieved quickly and robustly by injecting a large first pulse (typically five times the nominal pulse width), which was followed by gradually decreasing the injected mass each cycle. If the injected mass were kept constant for all 50 fired cycles, the mixture would become increasingly rich, partly due to the valve warming up but also attributable to the growing fuel puddles in the port and cylinder. (See discussion of fuel puddle mass in §4.7.)

After 50 cycles were fired, fueling was cut off, and the engine was motored for many cycles to purge out the residual fuel and to allow the valves to cool back down to their steady-state motored temperature. See Figure 2-7 for a schematic of the breakdown of fired cycles versus motored cycles. In addition, Figure 2-7 shows the 20 cycles used to integrate the mass of EOHC emissions. Using FFID measurements in the exhaust runner near the port exit, the mass of EOHC for each cycle was calculated; these values were then summed for the 20 cycles shown, in order to mitigate the impact of any one cycle with particularly high or low EOHC.

To further understand the mixture preparation process during a cold start, measurements were taken in order to estimate the development of the fuel puddle. The method involved connecting the exhaust flow to a buffer tank, so that the cyclic flow pulsations were averaged out. The steady downstream HC concentration in the tank was measured and integrated to obtain the cumulative HC leaving the engine. The "puddle" mass was obtained by this measurement in the process of purging the residual fuel from the engine. With this set up, the residual fuel measured was the sum of that in the port and in the cylinder.
The test procedure was similar to that used for the split injection tests described above. The engine was motored at 1400 RPM to steady-state then fired for a specified number of cycles. Fueling was controlled cycle-by-cycle to maintain a stoichiometric AFR for up to 300 cycles. Two cases were observed: single pulse CVI and 25% split pulse injection with EOI for the second pulse at 360 CAD after top dead center (TDC) compression. The latter value was chosen because it gave the lowest EOHC. The fueling schedule for both cases is shown in Figure 2-8. A large first pulse is used in order to establish quasi-steady condition quickly and avoid having misfired cycles.

Figure 2-9 shows a typical trace of HC concentration versus cycle number for 300 fired cycles. The HC level begins at some baseline value, which is non-zero due to background HC levels from the oil. As the engine fires the HC quickly rises, and begins to plateau as the fueling and conditions in the engine stabilize. When fueling and spark are cut off, the HC concentration rises rapidly, due to the large fuel puddle mass remaining in the port. The HC reaches a peak value and then begins to decay asymptotically toward the original baseline value. Figure 2-9 shows in blue the area of the HC trace that is integrated to determine the puddle mass. The integration begins when fueling is shut off, and ends when the HC level returns to the baseline value. This baseline value is then deducted, based on the assumption that the background HC level from the oil is constant throughout.2

The mass of HC was calculated from the measured HC concentration and the measured mass flow rate of air through the engine as follows. This assumes that the air flows equally through all four cylinders. For each cycle:

\[
kg \ HC = \left( \frac{PPM \ C_1}{1 \times 10^6} \right) \left( \frac{W_{C_1}}{W_{air}} \right) \left( \frac{kg \ air}{hour} \right) \left( \frac{1 \ hour}{60 \ min} \right) \left( \frac{1}{RPM} \right) \left( \frac{2 \ rev}{cycle} \right)
\]

(6)

Where \( W_{air} \) is the molecular weight of air and \( W_{C_1} \) is the molecular weight of a \( C_1 \) molecule of fuel, based on a H/C ratio of 1.89 for UTG91 fuel; thus \( W_{C_1} = 13.89 \) kg/kmol.

---

2 Data was taken with the PCV valve both connected and disconnected, with no significant impact on the calculated puddle mass. At low MAP, the PCV valve remains closed, and thus any additional fuel that enters the oil should remain trapped in the sump.
2.2.3 Flow Reactor Exhaust Manifold

The key hardware change for the flow reactor exhaust manifold experiments was the installation of the exhaust manifold itself. The manifold, pictured fully instrumented in Figure 2-10, was a prototype unit designed by Chrysler to promote strong mixing between cylinders. The exhaust from all four cylinders was sent to a downstream mixing tank to allow steady-state cold coolant tests. Fast-response CO and HC measurements were taken in the exhaust port of Cylinder #4 as well as downstream in the mixing tank. “Catalyst-in” measurements were also taken at the location shown in Figure 2-10 below, however these were negligibly different from mixing tank measurements. Thus, the data presented was measured in the downstream tank.

For the steady-state cold coolant tests, coolant temperature was maintained at 20°C. The engine was motored to 1400 RPM using the electric motor, at which point the dynamometer was engaged at low load. Firing was then started using the custom 4-cylinder controller code, with the desired spark timing and fueling controlled by feedback from the UEGO. Dynamometer torque was then adjusted to achieve the desired speed and load, 1400 RPM with an average NIMEP for the four cylinders of 1.25 bar. The non-firing cylinder contributed a negative NIMEP; this value was determined by motoring tests as a function of MAP, and thus for a given test at a given MAP, the throttle was adjusted to give the correct average NIMEP for all four cylinders. For example, at a MAP of 0.497 bar, the negative NIMEP from motoring is -0.65 bar. Thus, to achieve to right torque at the crankshaft, the NIMEP from each of the three fired cylinders must be approximately 1.88 bar.

Test variables for steady-state cold coolant tests were spark timing and overall AFR. Spark timing was varied from 20 CAD BTDC to 15 CAD ATDC. Testing was performed for two different methods of calculating overall $\lambda$, however for both methods overall $\lambda$ was held at 1.00, 1.05, 1.10, and 1.15. First, tests were performed based on calculated overall AFR based on total air mass flow and total injected fuel mass. However, detailed investigation revealed that the combustion $\lambda$ was considerably leaner than the calculated $\lambda$. Thus, experiments were repeated holding the UEGO $\lambda$ constant. For both cases, tests were conducted with no fuel to either Cylinder #1 or Cylinder #2. Fueling to the each of the three firing cylinders was identical.

With this concept, the firing $\lambda$ constrains the overall $\lambda$ since the air mass pumped through the non-firing cylinder cannot be independently varied. The air mass in the non-firing cylinder

3 See §5.1.2 for detailed discussion.
was estimated from motoring data for MAFS versus MAP, assuming no residual burned gases in
the non-firing cylinder. With short valve overlap and relatively high MAP for heavy spark
retard, this assumption should be reasonably accurate; however this cannot be easily
independently verified.

Based on this assumption, the following relations were used to determine firing and
overall $\lambda$.

\[
\dot{m}_{\text{air, fired}} = \dot{m}_{\text{air, total}} - \dot{m}_{\text{air, non-fired}}
\]  
(7)

\[
\lambda_{\text{fired}} = \left( \frac{\dot{m}_{\text{air, fired}}}{\dot{m}_{\text{fuel, total}}} \right) \left( \frac{1}{AFR_s} \right)
\]  
(8)

\[
\lambda_{\text{overall}} = \left( \frac{\dot{m}_{\text{air, total}}}{\dot{m}_{\text{fuel, total}}} \right) \left( \frac{1}{AFR_s} \right)
\]  
(9)

For sufficiently early spark timing, with relatively small injected mass, UEGO measurements
agreed well with calculations based on fuel and air mass. For later spark timing, however, the
higher injected mass and reduced back flow due to higher MAP resulted in a larger portion of
fuel lost to the sump, and thus UEGO measurements were considerably leaner than calculated
values. Thus, a carbon balance based on CO, CO2, HC, and mass air flow measurements was
used to determine the actual combustion $\lambda$.

For crank start tests, spark and fueling was prescribed for each of the first 200-300
cycles, due to the slow time response of the UEGO during the startup transient. Fueling was
prescribed as shown below in Figure 2-11. The transition to 15 CAD ATDC spark timing was
achieved within seven cycles from cranking. Fueling was calibrated in order to give good RPM
response, i.e. a minimal speed flare, followed by a quick settle to a quasi-steady fast idle near
1400 RPM, with average NIMEP for four cylinders close to 1.25 bar NIMEP. The engine was
started on only three cylinders in order to avoid a fuel puddle in the non-firing cylinder, which
would have resulted in a HC spike when fueling was cut off.

In order to provide enough air for idle operation with heavy spark retard and only three
fired cylinders, substantially more airflow than could be provided by the idle bypass motor was
required. Thus, the throttle plate was locked partially open before the start. While not a real-
world solution, it provided the desired startup conditions, and does not significantly affect the
speed flare. An alternator was used to provide sufficient engine load without using the
dynamometer, which did not have adequate transient response.

In addition, the production catalytic converter was installed in order to measure catalyst
light-off times and tailpipe emissions. Temperatures were recorded at the catalyst inlet, between
the first and second bricks, and at the catalyst outlet. HC and CO data was recorded at the
catalyst inlet and catalyst outlet. See Figure 2-12 for locations of key instrumentation.
Table 2-1 - Engine specifications for 2000 MY and 2003 MY 2.4L test engines.

<table>
<thead>
<tr>
<th></th>
<th>2000 MY</th>
<th>2003 MY</th>
</tr>
</thead>
<tbody>
<tr>
<td>No. of cylinders</td>
<td>4</td>
<td>4</td>
</tr>
<tr>
<td>Total Displacement (L)</td>
<td>2.4</td>
<td>2.4</td>
</tr>
<tr>
<td>Bore/Stroke (mm)</td>
<td>87.5/101</td>
<td>87.5/101</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>9.54:1</td>
<td>9.5:1</td>
</tr>
<tr>
<td>Stock IVO/IVC (CAD After TDC Exhaust)</td>
<td>-1/231</td>
<td>-1/231</td>
</tr>
<tr>
<td>Stock EVO/EVC (CAD After TDC Exhaust)</td>
<td>488/728</td>
<td>493/727</td>
</tr>
<tr>
<td>Valve Overlap (CAD)</td>
<td>9</td>
<td>8</td>
</tr>
<tr>
<td>Max. Valve Lift Intake/Exhaust (mm)</td>
<td>8.3/6.5</td>
<td>8.25/6.6</td>
</tr>
</tbody>
</table>

Figure 2-1 - Schematic of Cambustion HFR400 fast response flame ionization detector.

Figure 2-2 - Schematic of Cambustion NDIR500 fast response CO/CO₂ analyzer.
Figure 2-3 - Fuel injector calibration for 2000 MY engine.

\[ \frac{mg}{\sqrt{\Delta P}} = 1.6671\tau + 0.1036 \]

Figure 2-4 - Fuel injector calibration for 2003 MY engine.

\[ \frac{mg}{\sqrt{\Delta P}} = 1.4613\tau - 1.3939 \]

Table 2-2 - Fuel properties for Chevron Phillips UTG91 test fuel.

<table>
<thead>
<tr>
<th>Property</th>
<th>Typical Value</th>
<th>Specification</th>
<th>Test Method</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copper Corrosion, 3 h at 50 °C</td>
<td>1</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Specific Gravity at 60/60 °F</td>
<td>0.735</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>API Gravity at 60 °F</td>
<td>61</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Distillation Stability</td>
<td>1440+</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Existent Gum, mg/100 mL</td>
<td>2</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Lead Content, g/gal</td>
<td>0.001</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Sulfur Content, wt %</td>
<td>0.013</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Phosphorus, g/gal</td>
<td>0.001</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Total Alcohol Content, vol %</td>
<td>0.00</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Reid Vapor Pressure at 100 °F, psi</td>
<td>9.0</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Research Octane Number</td>
<td>90.8</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Motor Octane Number</td>
<td>83.0</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Sensitivity</td>
<td>7.8</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Distillation Range at 760 mmpg, °F</td>
<td>7.8</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Initial Boiling Point</td>
<td>89</td>
<td>ASTM D 130</td>
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</tr>
<tr>
<td>10%</td>
<td>122</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>50%</td>
<td>212</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>90%</td>
<td>321</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>End Point</td>
<td>369</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Composition, vol %</td>
<td>10 max</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Cetane</td>
<td>24</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Aromatics</td>
<td>24</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Naphthenes</td>
<td>70</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Heat of Combustion, Net, Btu/lb</td>
<td>18500</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Carbon Content, wt %</td>
<td>86.3</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Hydrogen Content, wt %</td>
<td>13.7</td>
<td>ASTM D 130</td>
<td></td>
</tr>
<tr>
<td>Anti-Knock Index, (R+M)/2</td>
<td>96.6</td>
<td>ASTM D 130</td>
<td></td>
</tr>
</tbody>
</table>
Figure 2-5 – Locations of FFID probes and in-cylinder pressure transducer for first cycle cranking tests.

Table 2-3 - Test matrix for tests on intake cam phasing effects on first cycle mixture preparation.

<table>
<thead>
<tr>
<th>Cam delay (CAD)</th>
<th>IVO</th>
<th>IVC</th>
<th>Injected fuel mass (mg)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-17.14</td>
<td>-18.14</td>
<td>213.86</td>
<td>132.9</td>
</tr>
<tr>
<td>-7</td>
<td>-8</td>
<td>224</td>
<td>132.9</td>
</tr>
<tr>
<td>Stock</td>
<td>-1</td>
<td>231</td>
<td>132.9</td>
</tr>
<tr>
<td>10.14</td>
<td>9.14</td>
<td>241.14</td>
<td>132.9</td>
</tr>
<tr>
<td>17.14</td>
<td>16.14</td>
<td>248.14</td>
<td>132.9</td>
</tr>
<tr>
<td>22.14</td>
<td>21.14</td>
<td>253.14</td>
<td>132.9</td>
</tr>
</tbody>
</table>

Table 2-3 - Test matrix for tests on intake cam phasing effects on first cycle mixture preparation.

Figure 2-6 – Intake valve and port wall temperatures versus time for ECU cold start.
Figure 2-7 - Schematic of firing vs. non-firing cycles for simulated fast idle cold start tests.

Figure 2-8 - Injected mass versus cycle number for fuel puddle measurement tests.

Figure 2-9 - Integration technique used to calculate liquid fuel mass in port and cylinder.
Figure 2-10 - Location of key exhaust instrumentation for flow reactor exhaust manifold tests.

Figure 2-11 - Fueling schedule for three-cylinder flow reactor manifold cold starts.
Figure 2-12 - HC/CO sampling locations before and after catalyst for cold start testing.
Chapter 3 - FIRST CYCLE MIXTURE PREPARATION RESULTS

The objective of this phase of the project was to examine the effect of IVO timing on the amount of delivered fuel that forms the combustible charge. In the data presented below, each data point represents the mean of five repeat tests, and the error bars represent the minimum and maximum in the range of the data.

3.1 In-Cylinder FFID Measurements

The amount of fuel delivered to the combustible charge is estimated from the HC mole fraction found from in-cylinder FFID measurements. Typical in-cylinder HC and pressure traces are shown in Figure 3-1. The HC during the intake and the early compression process were quite non-uniform. However, a plateau region just prior to combustion could be identified and the average value there was used to calculate the in-cylinder fuel equivalence ratio of the combustible charge.

The fuel equivalence ratio ($\Phi$) as a function of the IVO timing for the different injected fuel mass is shown in Figure 3-2. When the IVO timing was retarded from the stock value of 1 CAD BTDC, there was a slight increase of the in-cylinder $\Phi$ value. However, $\Phi$ also decreased with advanced IVO timing.

3.2 Explanation of Mixture Preparation Results

For IVO retarded from the stock value, the increase in $\Phi$ might be attributable to all the positive aspects of the retarded IVO strategy: increased strip atomization due to the strong initial forward flow and the lower initial cylinder pressure to help fuel vaporization. These effects, however, were expected to be modest because of the confounding effects of a lower initial charge temperature into which the fuel drops were introduced, and of the decrease in compression temperature and duration. Furthermore, the strong forward flow period was brief. Thus, of the total injected fuel mass, the strip atomization was effective for only a small portion of the fuel located near the valve seat.

For IVO advanced from stock timing, since the exhaust valve was still open at IVO, there should not be a significant pressure difference between the cylinder and the intake manifold. At IVO, however, there was a small back flow to the intake port because of the piston displacement.
Also compared to the stock timing of IVO close to TDC, the initial forward flow with a significant shear (because of the low lift) was absent. These factors may contribute to a less favorable condition for preparing the mixture and hence lower $\Phi$ values were attained.

An alternative explanation which encompasses the data in both the retarded and advanced IVO timing is shown schematically in Figure 3-3. During the intake stroke, the in-cylinder HC is likely to be stratified. For the range of valve timing used, IVC always occurs before BDC; thus, if the stratification is such that the mixture is leaner in the upper part of the cylinder, relatively more air than fuel would be displaced as the piston came up before IVC. [24] The net effect would be a higher in-cylinder $\Phi$ value with a retarded IVC timing, as a consequence of retarded IVO timing since the cam profile was fixed.

This explanation was supported by the signature of the in-cylinder HC signal in Figure 3-4. Since the sample was drawn from the spark plug, the HC values were representative of those from the upper part of the cylinder. High HC values from the early part of the intake stroke decreased to values significantly lower than those of the final combustible mixture at the end of intake and the beginning of compression. Thus the displaced charge in the period between BDC and IVC would be significantly leaner than the remaining part of the charge, and the process increased the $\Phi$ value of the trapped charge.

The continuous nature of the $\Phi$ dependence on IVO timing in Figure 3-2 suggests that charge stratification is likely the dominant explanation, although mixture preparation effects due to flow differences may also play a role.

3.3 Engine-Out HC Emissions Measurements

The HC emissions for the first cycle are shown in Figure 3-5. Those from the 40 ms injection pulse – injected mass ($m_i$) of 133 mg – are not shown because there were significant misfired events. For the same cam phasing, the HC emissions increased with the amount of fuel injected because of a larger amount of liquid fuel entered into the cylinder. Since a substantial amount of the in-cylinder liquid fuel survives combustion and escapes the cylinder as engine-out emissions, the larger amount of in-cylinder liquid fuel would lead to higher HC emissions. [4]

For a given injected mass, the emissions increased with later IVO timing. The increase was approximately 40% when IVO timing was changed from 18 CAD BTDC to 21 CAD ATDC for both the 199 mg and 266 mg injected fuel masses.
3.4 Explanation of Emissions Results

Many factors affect the HC emissions, thus a cycle-simulation with combustion represented by a Wiebe function was used to assess the different effects. When the IVO timing changed from 18 CAD BTDC to 21 CAD ATDC, the engine behavior changed as follows.

For $m_i = 199$ mg, $\phi$ increased from 0.78 to 0.85, with a corresponding increase in burned gas temperature of $\sim 70$ K due to this enrichment. The temperature change was approximately the same throughout the expansion and exhaust process. This increase, however, was offset by a 20 K decrease due to the combined effect of a lower compression ratio and a lower volumetric efficiency (which affected the impact of heat transfer). The net increase of 50 K shown in Figure 3-6 should help post-flame HC oxidation and reduce emissions. For $m_i = 266$ mg, $\phi$ increased from stoichiometric to 1.10. The net effect was to decrease the burned gas temperature by $\sim 80$ K throughout the expansion process; see Figure 3-6. The lower temperature and lack of excess air should hinder HC oxidation, thus increasing emissions.

Although for the case of $m_i = 266$ mg, the increase in HC emissions with later IVO timing was consistent with the expected change in post-flame oxidation, the post-flame temperatures would predict an opposite trend from that observed for the case of $m_i = 199$ mg (Figure 3-5). However, valve timing may significantly affect the liquid fuel distribution in the cylinder and hence the HC emissions. Even in the absence of strong back flow from a previous fired cycle, the displacement flow into the port with early IVO might help prevent some liquid fuel from entering the cylinder due to wall film flow. With late IVO and the strong shear flow it creates, the fuel film immediately near the valve seat might be strongly drawn into the cylinder, increasing wall wetting and thus emissions.
Figure 3-1 - Determination of the in-cylinder HC value for calculation of the fuel equivalence ratio ($\Phi$) for the combustible charge.

Figure 3-2 - Fuel equivalence ratio of the combustible charge as a function of IVO timing for different injected mass.
INCOMING MIXTURE INCREASINGLY LEAN AS PISTON DRAWS IN CHARGE

INCREASE FLOW

LEAN

PISTON DISPLACES MORE LEAN CHARGE AS IVC DELAYED

Figure 3-3 - Schematic of charge stratification during first cycle intake stroke.

Figure 3-4 - In-cylinder FFID trace and pressure versus crank angle for the first cycle of cranking. Shaded area indicates IVO period. Transport delay between intake port and FFID sample location results in delay between IVO and spike in HC due to initial rich mixture. Mixture becomes leaner during the intake stroke until BDC, then lean charge near top of cylinder is displaced back into port.
Figure 3-5 - HC emissions for the first cycle of cranking as a function of IVO timing.

Figure 3-6 - Contributions to the change in expansion burned gas temperature (defined as the value at mid stroke expansion) as cam phasing was changed from IVO at 18 CAD BTDC to 21 CAD ATDC.
Chapter 4 - FUEL INTERACTION WITH VALVE EVENTS DURING FAST IDLE RESULTS

The objective of this phase of the project was to examine the effect of injection timing relative to the intake air flow events on HC emissions.

4.1 HC Emissions versus Injection Timing for Stock Intake Cam Phasing

Figure 4-1 shows the exhaust HC concentration for the baseline case in terms of PPM of propane (C3H8), the gas used to calibrate the FFID, as a function of EOI timing. EOI timing was used rather than start of injection (SOI) because EOI has been shown to correlate well with HC emissions over a wide range of conditions. [25] Also shown in Figure 4-1 are the traces for intake valve lift, intake mass flux, and valve curtain velocity, as found by using the WAVE model. The black arrow shows the duration of a nominal fuel pulse.

Several key features are worth noting. First, EOHC emissions are essentially constant for CVI. This indicates that the residence time of the fuel in the port has little effect on mixture preparation, if any. Second, there is a substantial increase in EOHC for OVI, with peak emissions roughly 50% higher for OVI than for CVI. As shown in Figure 4-1, the EOHC concentration scales primarily with valve lift, but also with the forward intake flow which pulsates because of the line acoustics. The flow dynamics result in two peaks in forward flow, as seen from the intake mass flow curve. However, the emissions corresponding with the second peak are significantly higher than those for the first peak; this is because the second peak coincides with peak valve lift. With a large open area and strong forward flow, more liquid fuel is thrown onto the cold cylinder walls and/or piston crown, leading to higher EOHC emissions. This is consistent with previous work. [25] [26]

Also worth noting is the small increase in EOHC around 320 CAD after TDC. After this small increase, the EOHC level returns back to the nominal closed valve value of ~1000 PPM C3 in a small window of EOI timing, before increasing rapidly as the injection shifts to full OVI. Clearly at least some portion of the fuel arriving at the valve curtain in this small window is

\[ 4 \text{ For fixed } \lambda, \text{ PPM C3 is proportional to emissions index; 1000 PPM C3 corresponds to an emissions index of 2.2\%.} \]

\[ 5 \text{ See §6.2.1 for details of the WAVE modeling. Model-derived flow results are presented here for sake of clarifying the emissions results.} \]
being injected open-valve. Based on results from the WAVE simulation, it would appear that this fuel is hitting the strong backflow. This effect will be discussed further with the simulated cold-valve fast idle tests.

4.2 HC Emissions versus Injection Timing for Varied Intake Cam Phasing

Steady-state cold fluids tests were also performed for three different values of intake cam phasing: stock, 20 CAD advanced, and 20 CAD retarded. The results are shown below in Figure 4-2, along with WAVE simulation data for the valve lifts, intake mass fluxes, and valve curtain velocities for the three cases.

EOHC is significantly lower for the 20 CAD advanced case versus the stock case, and EOHC is significantly higher for the 20 CAD retarded case. This could partly be explained by the fact that the back flow is stronger as IVO is advanced and weaker as IVO is retarded, as shown in Figure 4-2. While the peak back flow velocity is not significantly changed, the duration of the back flow changes. If the back flow is assumed to assist in mixture preparation by helping to vaporize and distribute the liquid fuel in the port, this would lead to lower EOHC emissions for advanced IVO and vice versa for retarded IVO. The impact of flow past the valves is also seen for the 20 CAD retarded case, where the OVI emissions are somewhat lower than expected – the flow dynamics with retarded IVO result in less flow during the peak valve lift period when compared to the stock IVO timing. Thus, one would expect less wall wetting. Also worth noting is how the trends shift with valve timing – the small increase noted above shifts about 20 CAD earlier as expected, and for the 20 CAD retarded case the OVI peak shifts along with the valve lift and peak forward flow.

However, several significant effects must also be considered. The effective compression ratio is highest for the 20 CAD advanced case and lowest for the 20 CAD retarded case. Figure 4-3 below shows how the EOHC emissions for nominal closed valve injection timing scale linearly with both IVO timing and effective compression ratio. Thus, one might expect that HC emissions decrease with earlier IVO due to higher compression ratios and higher peak temperatures. However, Figure 4-4 shows the in-cylinder pressure traces for the stock and 20 CAD advanced cases (average of 200 cycles). The 20 CAD advanced case actually has a lower peak pressure than the stock case due to the fact that the advanced IVO case has higher residual gas fraction, 24.5% versus 19.3% for stock cam phasing. This higher residual fraction is
reflected in the COV of NIMEP, typically 13.5% for the 20 CAD advanced case vs. 8.9% for the stock case. See Table 4-1 below.

As a result of the higher residual gas fraction with earlier IVO, the combustion phasing is later for the 20 CAD advanced case, as seen in Figure 4-4. Since NIMEP for both cases is 1.25 bar, with later combustion phasing the 20 CAD advanced case would require more fuel, since more fuel energy goes into burned gas enthalpy instead of work. Indeed, as shown in Table 4-1, the 20 CAD advanced case requires higher MAP and more injected mass per cycle. The residual fraction decreases with late IVO due to negative valve overlap, and the earlier, faster burn shown in Table 4-1 reflects this effect. Thus, these factors indicate that the primary reason for lower EOHC emissions with advanced intake cam phasing is later combustion and thus greater post-flame oxidation due to higher residual gas fraction.

This effect was confirmed by performing a spark timing sweep with stock intake cam phasing; the results for HC emissions versus combustion phasing are shown in Figure 4-5. For the same combustion phasing, HC emissions are within the limits of repeatability for the different cam timing cases. This indicates that the change in valve timing has a small effect if any on mixture preparation and that post-flame oxidation is the dominant effect for these two valve timing cases. Advancing the intake cam opening by 20 CAD has approximately the same effect as retarding the spark timing by 9 CAD, while retarding IVO by 20 CAD has roughly the same effect as advancing spark timing by a few degrees.

4.3 HC Emissions versus Injection Timing for Varied Load

Additional steady-state injection timing sweeps were performed at higher NIMEP. Increasing the NIMEP (i.e. running at higher MAP and with higher injected mass) does not result in substantial change in the backflow, since the major part of the back flow is choked, and hence independent of intake pressure.

Figure 4-6 below shows EOHC as a function of EOI timing for the baseline case of 1.25 bar NIMEP along with higher load cases at 2.05 bar NIMEP and 3.30 bar NIMEP. Also shown are WAVE simulation data for each of the three cases. Note that the change in valve curtain velocity during the backflow period is relatively small. However, the mass flux during the peak forward flow increases significantly with higher MAP. Moreover, significantly higher injected mass is required to achieve higher load, thus there is considerably more liquid fuel in the
port/cylinder. As such, the EOHC increases across the board with higher load, most noticeably for OVI. Note also that the location of the EOHC peak during OVI correlates well with EOI timing.

4.4 HC Emissions versus Injection Timing for Different Fuel Rail Pressures and Injectors

Further steady-state cold fluids tests were conducted to evaluate the effect of injector droplet size on EOHC emissions. Injector droplet sizes were changed both by varying the fuel rail pressure and by using a prototype 12-hole injector provided by DaimlerChrysler.

Figure 4-7 shows EOHC as a function of EOI timing for the stock, 4-hole injector at various rail pressures. Note that the fuel pulse widths are longer at lower rail pressure, since a longer pulse is needed to inject the same fuel mass. There was no significant effect for closed valve injection, since the fuel droplets all hit the port walls and/or valve and thus droplet size was not critical. [27] [28]

For open valve injection, a benefit was seen for using a higher rail pressure; however the OVI emissions were still substantially higher than the CVI emissions. Tests run at a low rail pressure (about 25 psi or 1.72 bar) resulted in a small increase in OVI emissions. The observed small effects are likely attributable to a relatively insignificant difference in droplet size; data from DaimlerChrysler for a different injector indicated that increasing fuel rail pressure from 4 bar to 6 bar resulted in 15% smaller droplets, however for the relatively large droplets at these low pressures, such a difference is unlikely to affect emissions significantly. For the stock 4-hole injector, the Sauter Mean Diameter (SMD) is 70 μm and the \( D_{0.9} \) droplet size is 203 μm.

Additional data was taken using the prototype 12-hole injector. For all cases, no improvement was found versus the 4-hole injector for closed valve injection. Figure 4-8 below shows the EOHC data for the 12-hole injector versus the 4-hole injector at 4 bar fuel rail pressure. There is some improvement for most of the OVI test points. For CVI, the same small increase is seen around 320 CAD after TDC. For 6 bar fuel rail pressure, as shown below in Figure 4-9, there is no significant difference between the 4-hole injector and the 12-hole injector. Note that the fuel injector calibrations are nearly identical for the two injectors, so the nominal fuel pulse width is effectively the same for a given rail pressure. For the prototype 12-hole injector, it is not known if the injector targeting was optimized for use in this engine. Thus, the expected benefits from smaller droplet sizes might not have been realized.
4.5 HC Emissions versus Injection Timing for Different Idle Speeds

In discussions with DaimlerChrysler it was noted that some earlier work had found a "sweet spot" at normal idle conditions with lower RPM than that being investigated here as a simulated fast idle. Thus, a limited injection timing sweep was performed at 1000 RPM, using the stock injector at 4 bar rail pressure with stock intake cam timing. Results are shown below in Figure 4-10 for both a steady-state cold condition and a warmed-up (80°C) case. Spark timing was 35 CAD BTDC for both cases. Even fully-warm, emissions are substantially higher at 1000 RPM versus 1400 RPM. Several phenomena explain this difference: the crevice gas oxidation in the later part of the expansion stroke is reduced because of the higher cumulative heat transfer from the charge to the wall, and the reduced oxidation in the exhaust port/runner due to a cooler gas temperature because of the lower average flow rate.

For the fully-warm condition, the engine surface temperatures are sufficiently hot that there is negligible difference between OVI and CVI emissions, with the exception of a small spike right near the period of peak forward flow. For the cold condition, the same small increase in EOHC around 320 CAD after TDC is seen, with a very narrow window of EOI timing afterward in which emissions are marginally lower. However, with cold engine surfaces, EOHC increases dramatically as injection timing is shifted later into the OVI period. Thus, no significant "sweet spot" was found.

4.6 HC Emissions for Split Injection

Figure 4-11 below shows the trend in EOHC emissions for split injection (two pulses) under steady-state fast idle conditions, with EOI timing based on the second pulse. The first, closed-valve pulse is always injected with EOI timing of 200 CAD ATDC. There is no significant difference between the different injection strategies until the intake valve begins to open significantly. With fully open-valve injection timing, emissions are highest for a single pulse, as expected, and emissions decrease as the split fraction is reduced, i.e. a smaller fraction of the injected mass is injected open-valve. However, these results might obscure the potential benefit of split injection. The intake valve under steady-state cold fluids conditions will be considerably warmer than under actual fast idle conditions, about 100°C versus about 40°C. As split injection is intended to overcome the problem of injecting liquid fuel onto a cold valve, these steady-state tests might not reveal the
benefits of split injection. Therefore, it was desirable to perform split injection tests under conditions that match actual fast idle conditions as closely as possible.

Figure 4-12 shows the emissions results for the simulated cold-valve fast idle tests described in §2.2.2 in terms of specific EOHC emissions, expressed in g/kWhr, averaged over 20 cycles. For comparison, the blue horizontal line represents the average value for closed valve injection. EOI timing for the second pulse was varied from 330 to 370 CAD after TDC-compression. These values put the EOI in line with the window of the intense backflow; see Figure 4-1.

There is a small benefit in EOHC for a 25% split fraction with the correct injection timing. The EOHC was about 6% lower than the single-pulse CVI value. The benefit for larger split fractions is smaller still, but still dependent on correct injection timing. In general, emissions were higher with a larger OVI fraction and highest for the single pulse (100% OVI) in the back flow period.

Figure 4-13 shows the fuel delivery fraction for each test case, defined here as the percent by mass of the injected fuel that constitutes the combustible mixture. This value was estimated based on an estimate of the trapped air (from a speed density calibration and the MAP measurement), the residual gas fraction (from a previously obtained correlation developed with a similar engine), and the AFR (from the UEGO). The injected mass was determined from the fuel pulse width. The values shown in Figure 4-13 represent an average for the test, given that the injected mass is constantly decreasing over the course of the 50 fired cycles. The figure shows that fuel delivery improves versus the closed valve baseline for the 25% and 50% split fractions, while the 75% split fraction and OVI cases had typically worse fuel delivery.

Figure 4-14 shows the EOHC data for the 25% split fraction case along with the valve flow simulation data. Note that there is some transit delay between the fuel leaving the injector tip and either hitting the back flow or arriving at the valve/port wall; based on a fuel pressure difference of approximately 4.75 bar across the injector tip, this would yield an initial tip velocity of approximately 35 m/s, and an injector transit delay of approximately 3.1 ms. This is shown in Figure 4-14. Fuel droplet breakup will be driven by the dynamic pressure at the valve seat, which will scale with \( pv^2A \), where \( v \) is curtain velocity and \( A \) is valve open area; this is proportional to \( \dot{m}v \), which is also shown in Figure 4-14. The middle portion of the fuel injected into the backflow period arrives at the valve almost exactly at the peak of the \( \dot{m} v \) curve. This
indicates that fuel injected with this specific split pulse timing hits the peak back flow and is thus most likely to see the best mixture preparation. Fuel injected later will largely miss the strongest back, and the result is a sharp increase in emissions compared to slightly earlier EOI timing, which sees only somewhat weaker back flow and only somewhat higher emissions. Higher split fractions will result in the majority of the OVI fuel pulse missing the $m_v$ peak, resulting in higher emissions.

The same set of tests was run using the 12-hole injector at 6 bar (gauge) fuel rail pressure; in theory this combination would yield the smallest liquid fuel droplets and thus lower EOHC emissions. However, as shown in Figure 4-15 below, this is not the case. The closed valve baseline is significantly worse than that for the 4-hole injector at 4 bar rail pressure. Moreover, there seems to be little correlation between injection timing and emissions. Figure 4-16 shows the fuel delivery fraction for the 12-hole injector tests. Only the 100% OVI case had fuel delivery close to CVI; all of the split injection cases had worse fuel delivery. Moreover, no single case with the 12-hole injector was as good as the worst fuel delivery case for the 4-hole injector. This data might indicate poor injector targeting for the 12-hole injector tests; if the fuel spray is hitting the port walls over a large area, the valve events would have less impact than for spray targeted at the back of the valve.

4.7 Fuel Puddle Mass Measurements

Figure 4-17 shows the results for the integrated fuel puddle mass versus the number of injected cycles. The data shown is for the stock 4-hole injector with 4 bar rail pressure, operating at 1.25 bar NIMEP, with stock valve timing. The data in red is for CVI, and the data in black is for the split injection case that gave the lowest emissions – 25% split fraction, with EOI for the second pulse 360 CAD ATDC. The solid lines show the least squares fit of the data (in the form of $m = m_0[1-\exp(-t/\tau)]$). For CVI, the puddle mass reaches steady state in approximately 300 cycles; this is consistent with the data taken by Takeda et al.\textsuperscript{6} using the trapping method. The steady-state puddle mass values found here, however, are only valid if the port temperatures remain cold due to external cooling. This value is expected to decrease substantially as the port warms up in normal engine operation.

\textsuperscript{6} The Takeda study was based on open valve injection. As such, the data showed a large initial cylinder puddle that decayed steadily. By contrast, the port puddle increased until it peaked around 300 cycles, after which it began to decay as the engine warmed.
The data indicate quite clearly the time involved in the fuel puddle development. The fuel puddle is not fully formed by the beginning of the fast idle period (approximately 7-10 seconds since cranking), since according to Figure 4-17, a steady state is not reached until approximately 26 seconds at 1400 RPM. Moreover, the mass of liquid fuel in the engine is significant – at steady state with closed valve injection, there is roughly 16 injections worth of liquid fuel in the port and cylinder.

It is difficult to explain the small difference in time scales for the two injection cases. CVI may result in faster puddle development because all of the fuel is injected in a single pulse, thus providing more liquid fuel mass to the back of the valve and port wall, and for the portion of the fuel not delivered in OVI, there is more time in each cycle for that liquid fuel to equilibrate in the port.

The difference in puddle mass between the two cases changes significantly with time. For the first 10 cycles, there is less than 1% difference in cumulative fueling for the two cases, but there is a significant difference in puddle mass. The CVI case shows about 10 mg more fuel in the puddle; that amounts to about 25% more mass than in the split injection case. This fact suggests that the CVI case had considerably lower fuel delivery for the first 10 cycles; this observation is consistent with the lower average delivery ratio value (relative to the 25% split injection) shown in Figure 4-13. In Figure 4-18 and Figure 4-19, the fuel mass distributions are shown for the two different injection cases. Since the air flow and λ values were approximately constant for the period of interest, with some uncertainty for the first few cycles, the delivered fuel mass in the combustible charge per cycle was taken as constant. The puddle masses, i.e. the residual liquid fuel mass in the engine, are the values shown in Figure 4-17. The remaining fuel mass must either exit the engine as EOHC or be deposited in the oil sump; these two masses of fuel are lumped together. These various masses are shown as a percentage of the cumulative injected fuel mass for the given number of injected cycles.

For the split injection case, the emissions/sump mass becomes unrealistically small when there are more than 150 injected cycles. This data would imply erroneously that there were virtually zero cumulative EOHC emissions in the case of 300 injected cycles. A possible explanation for this discrepancy is that the pulse width for the second pulse of split injection becomes very small for many fired cycles, as small as 1.6 ms. This value is near the linearity limit for this injector. Because of the finite calibration precision, there may actually be more
injected fuel mass than was prescribed by the commanded fuel pulse width. Moreover, a significant portion of the fuel mass injected in that second pulse was delivered while the injector was in the process of opening or closing; this is likely to result in a higher uncertainty in the fuel delivered for a fixed pulse width.

Based on mass airflow data and the measured AFR, about 8.03 mg per cycle must be delivered to the combustible mixture. Since this does not include any HC emissions or fuel loss to the sump, the actual injected mass can be assumed to be somewhat higher. For comparison, with a single injection pulse, the delivered fuel mass per cycle is estimated to be 8.13 mg, compared to an injected mass of 8.31 mg/cycle once the fueling has reached steady state. For split injection, however, after about the 200th injected cycle the fueling calculations based on pulse width and the injector calibration predict that less fuel is injected than is found to be in the combustible mixture; by the time the test reaches steady state as it approaches the 300th cycle, the total injected mass for the split injection case would be only 7.90 mg per cycle.

For the first tens of cycles of data (first few seconds of firing) for which the error of the cumulative fuel injected was small, the above accounting procedure shows that the split injection case does result in a smaller fuel puddle. If we further assume that because of the intense blowback, the 25% fuel delivered as OVI did not enter the cylinder as liquid fuel, then the smaller fuel puddle in the split injection case would imply a lesser amount of liquid flow into the cylinder in the forward flow period. This would result in less in-cylinder wall wetting, which would be consistent with the earlier results that split injection yields lower EOHC emissions.

Figure 4-20 shows the HC level measured in the downstream mixing tank for the two cases. Emissions are significantly lower for the split injection case during firing, however once firing ceases, the HC concentrations during purging are nearly identical. This result is consistent with the convergence in residual liquid fuel mass for the two injection strategies beyond 200 fired cycles.

With the very short fuel pulses for split injection beyond 200 fired cycles, injector targeting also becomes questionable, and it is plausible that fuel is being sprayed in locations that contribute minimally to mixture preparation. This could include fuel that hits the port walls too far from the intake valves, or alternatively large droplets that enter the combustion chamber and hit the piston crown or cylinder walls. Thus, AFR remains roughly constant even though the
total liquid fuel mass in the port and cylinder increases. However, knowledge of the actual liquid fuel behavior in the port and cylinder is too limited to draw any meaningful conclusions.
Figure 4-1 - HC emissions versus end of injection timing for stock intake cam phasing, 1400 RPM 1.25 bar NIMEP, $\lambda = 1.00$. For fixed $\lambda$, PPM C3 is proportional to emissions index; 1000 PPM C3 corresponds to an emissions index of 2.2%.

Figure 4-2 - HC emissions versus end of injection timing for stock intake cam phasing, 20 CAD advanced intake cam phasing, and 20 CAD retarded intake cam phasing. 1400 RPM, 1.25 bar NIMEP, $\lambda = 1.00$. 
Figure 4-3 - HC versus IVO timing and effective compression ratio for 1400 RPM, $\lambda = 1.00$, 1.25 bar NIMEP, closed valve injection.

Figure 4-4 - Average pressure traces for stock intake cam timing and 20 CAD advanced intake cam timing.
Table 4-1 - Effective compression ratio, residual gas fraction, location of 50% mass fraction burned, 10-90% burn duration, MAP, fuel pulse width, and HC emissions for stock, 20 CAD retarded, and 20 CAD advanced intake cam timing.

<table>
<thead>
<tr>
<th></th>
<th>Ret. 20</th>
<th>Stock</th>
<th>Adv. 20</th>
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<tr>
<td>Residual Gas Fraction</td>
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</tr>
<tr>
<td>Fuel PW (ms)</td>
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<td>3.46</td>
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</tr>
<tr>
<td>HC (PPM C3)</td>
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<td>1002.3</td>
<td>903.3</td>
</tr>
</tbody>
</table>

Figure 4-5 - Steady state downstream HC concentration versus location of 50% mass fraction burned, 1400 RPM, 1.25 bar NIMEP, 20°C ECT. The dark blue points are for stock cam phasing at a range of spark timings, the red triangles are for 20 CAD advanced and 20 CAD retarded intake cam phasings with spark timing 35 CAD BTDC.
Figure 4-6 - HC emissions versus end of injection timing for 1400 RPM, $\lambda = 1.00$, NIMEP = 1.25 bar, 2.05 bar, 3.30 bar. Stock intake cam phasing.

Figure 4-7 - HC emissions versus end of injection timing for 1400 RPM, $\lambda = 1.00$, 1.25 bar NIMEP, varying fuel rail pressure, stock 4-hole injector. Stock intake cam phasing.
Figure 4-8 - HC emissions versus end of injection timing for 1400 RPM, $\lambda = 1.00$, 1.25 bar NIMEP, stock 4-hole injector and prototype 12-hole injector at 4 bar rail pressure. Stock intake cam phasing.

Figure 4-9 - HC emissions versus end of injection timing for 1400 RPM, $\lambda = 1.00$, 1.25 bar NIMEP, stock 4-hole injector and prototype 12-hole injector at 6 bar rail pressure. Stock intake cam phasing.
Figure 4-10 - HC emissions versus end of injection timing for 1000 RPM, $\lambda = 1.00$, 1.25 bar NIMEP, cold coolant and fully warm coolant. Stock intake cam phasing.

Figure 4-11 - HC emissions versus end of injection timing for 1400 RPM, $\lambda = 1.00$, 1.25 bar NIMEP, different split injection strategies. EOI timing is for second pulse for split injection cases. EOI timing for CVI pulse 200 CAD ATDC.
Figure 4-12 - Simulated cold start HC emissions for different injection strategies, 4-hole injectors, 4 bar gauge rail pressure. Different color/pattern bars represent different EOI timing for given split fraction.

Figure 4-13 - Simulated cold start fuel delivery fraction for different injection strategies, 4-hole injectors, 4 bar gauge rail pressure. Different color/pattern bars represent different EOI timing for given split fraction.
Figure 4-14 - HC emissions versus end of injection timing for 25% open valve split fraction. Also shown injector pulse width and transit delay, valve lift, curtain velocity, mass flux, and $\dot{m}_V$.

Figure 4-15 - Simulated cold start HC emissions for different injection strategies, 12-hole injectors, 6 bar gauge rail pressure. Different color/pattern bars represent different EOI timing for given split fraction.
Figure 4-16 - Simulated cold start fuel delivery fraction for different injection strategies, 12-hole injectors, 6 bar gauge rail pressure. Different color/pattern bars represent different EOI timing for given split fraction.

Figure 4-17 - Fuel puddle mass versus number of injected cycles for simulated fast idle condition, 1400 RPM, $\lambda = 1.00$, 1.25 bar NIMEP.
Figure 4-18 - Fuel accounting for single pulse closed valve injection.

Figure 4-19 - Fuel accounting for split injection, 25% split fraction, EOI for split pulse 360 CAD ATDC.
Figure 4-20 - Downstream HC emissions for simulated cold starts with closed valve injection and split injection. First 300 cycles are fired, subsequent cycles represent purging of residual HC.
Chapter 5 - FLOW REACTOR EXHAUST MANIFOLD RESULTS

The objective of the third phase of the project was to examine the feasibility of operating a 4-cylinder engine with three cylinders running rich with the fourth cylinder acting as an air pump to provide air for exhaust oxidation of the rich combustion products. Experiments were carried out in the course of investigating the flow reactor exhaust manifold concept. For steady-state cold fluids tests, variables included the choice of non-firing cylinder, spark timing, and overall $\lambda$. Overall $\lambda$ was determined by two different methods – first, using air and fuel mass flow rates, and second, using the UEGO – which yielded very different results. In addition, experiments were carried out running four cylinders rich, in order to better establish the state of the exhaust gases. Crank starts were also performed to investigate real cold start behavior, particularly catalyst-light off and tailpipe emissions.

5.1 CO Emissions and Exhaust Temperatures versus Spark Timing for Varying $\lambda_{overall}$

Results for CO concentrations and exhaust gas temperature (EGT) are presented first to explain the onset of secondary oxidation, since CO oxidation provides most of the heat release and is the best indicator of oxidation in the manifold. In all cases, $\lambda_{overall}$ refers to the total relative AFR of the three fired cylinders plus the non-firing cylinder.

5.1.1 Results for $\lambda_{overall}$ Based on Air Mass Flow Rate and Injected Mass Calculations

Figure 5-1 shows the results for steady-state cold coolant tests with $\lambda_{overall} = 1.00$, based on air and fuel mass flows. Plotted against the left axis, the temperature in the exhaust manifold increases steadily with retarded spark, and the temperature downstream at the catalyst-in location is typically 30-50 K cooler due to heat transfer to the exhaust system surfaces, for spark timing up to 7 CAD ATDC. CO concentrations, plotted against the right axis, are fairly constant until they begin to decrease slightly for spark timing after TDC. With spark later than 7 CAD ATDC, the CO rapidly oxidizes and exhaust temperatures increase between the manifold and catalyst inlet. A similar trend is seen in Figure 5-2, with $\lambda_{overall} = 1.05$, however the baseline CO concentration is lower, as expected.
For leaner overall AFR, however, the trends are subtly different. While there is still a critical point beyond which CO is oxidized and temperatures increase in the direction of flow, the temperature jump is much less pronounced for the leaner overall mixtures. Figure 5-3 shows the results for $\lambda_{\text{overall}} = 1.10$ and Figure 5-4 shows the results for $\lambda_{\text{overall}} = 1.15$. For each case, the temperature trends shift for $T_{\text{manifold}} > 900$ K, however for the leaner mixtures the trend in temperature is much more linear. This can be attributed to the fact that the leaner mixtures have far less CO and H$_2$ to oxidize in the exhaust manifold. In addition, since the mixture is more dilute, oxidation will be slower; as a result, more oxidation will occur further downstream in the manifold than for richer cases, resulting in lower manifold EGT but still high catalyst-in EGT.

Steady-state exhaust manifold surface temperature measurements shown in Figure 5-5 also show signs of secondary oxidation in the manifold. For early spark timing, the surface temperatures increase with EGT, with the temperature difference between the two increasing with higher EGT due to increased radiation heat transfer from the manifold surfaces. However, with spark timing sufficiently late to yield significant CO oxidation, the temperature at the body of the manifold is hotter than the temperature of the exhaust runner surface, as strongly exothermic reactions occurring in the manifold raise the EGT above the average engine-out temperature.

5.1.2 Mass Flow Rate AFR Calculations versus Actual Combustion AFR

It was initially assumed that by holding the ratio of total air mass flow to total fuel mass flow constant, the rich fired cylinders would have a constant AFR. However, detailed investigation proved this to be untrue. The CO levels, particularly for early spark timing where relatively cold EGT should not yield any chemical reactions, are far below the equilibrium mole fractions expected for the combusted $\lambda$ values calculated based on mass flow. This was at first believed to be the result of some secondary CO oxidation occurring far upstream in the manifold, i.e. where the very hot blowdown gases might interact with air. However, regardless of the high level of mixing achieved by this manifold, it is highly unlikely that air could be drawn that far upstream. The valve overlap is only 8 CAD and thus the back flow from the exhaust port is small. Calculations from WAVE indicate that during the valve overlap period, when sub-atmospheric pressure in the intake manifold draws exhaust gas back into the cylinder, a volume approximately equal to one half the volume of an individual exhaust valve’s port (i.e. before the
two ports join to exit the cylinder. By contrast, the preceding forward flow past the exhaust valves displaces more than ten times this volume. Thus, the back flow cannot reasonably be expected to include oxygen, and only burned exhaust gases are likely to be present in the exhaust port and runner. This indicates that all significant mixing occurs in the body of the manifold.

In addition, fast-response CO measurements in the port for both 3-cylinder operation and 4-cylinder rich operation confirmed that no reactions were occurring upstream; in-cylinder measurements further confirmed that CO levels were never reaching the expected concentrations.

Further investigation found that “lost fuel” resulted in the discrepancy. At early spark timing, where a relatively small injected mass is required to achieve the desired torque, the mass flow rates predict $\lambda$ reasonably well, although a small portion of the fuel mass is lost into the sump, due to the cold operating condition. This, combined with HC emissions, results in a leaner combustible mixture than predicted from the mass flow rates. This disparity increases at higher injected mass, i.e. at later spark timing. This effect was verified by running four cylinders rich, and comparing the steady state CO/CO$_2$ values with equilibrium mole fractions, shown in Figure 5-6. The measurements were taken with the UEGO mounted far downstream of the mixing/damping tank to ensure a uniform, steady mixture, and data was taken with a range of spark timings, including spark 15 CAD ATDC to produce hot exhaust temperatures, to ensure there was no skewing of the UEGO reading due to overheating.

In general the UEGO $\lambda$ values agree very well with the equilibrium mole fractions for CO, however there is an increase in CO for very rich combustion, most likely due to partial burns and/or incomplete combustion. The offset in CO$_2$ values for rich combustion was consistent across many tests with frequent recalibration of the CO meter; no such offset was present for stoichiometric or lean combustion.

The data shown in Figure 5-6 verifies that the measured UEGO $\lambda$ value is representative of the actual combusted mixture. However, for the same test points with 4-cylinder rich operation, calculations of the overall AFR based on air mass flow and injected fuel mass showed consistently richer $\lambda$ than that measured by the UEGO, and the difference between UEGO and MAFS/injected mass measurements increased for later spark timing, i.e. higher mass air flow and higher injected mass per cycle. The injected fuel mass was previously verified during calibration of the fuel injector. To verify the correct operation of the MAFS, data was taken at several
different engine speeds, to yield much higher air flow with the same injected mass per cycle. This data verified that the discrepancy in λ measurements was a function of injected mass, not air flow.

Since the measured HC emissions were only a small fraction of the mass discrepancy between burned fuel mass and injected fuel mass, it was concluded that, due to running steady state with cold coolant, a significant portion of the injected fuel mass was being lost into the oil. At cold temperatures, more fuel is absorbed into the oil however the oil remains cold enough that it will not be released back into the crankcase ventilation system. [26] Anecdotal evidence suggested this was occurring, as a significant fuel odor and substantial viscosity reduction was detectable whenever the oil was changed without running the engine fully warm beforehand, however quantifying this would not have been possible except on a bulk scale, over many hours of operation for various tests. [7] Further supporting this hypothesis, an earlier visualization study found that liquid fuel in the cylinder during cold conditions would be transported into the piston ring pack. [29]

With additional data taken for constant AFR based on UEGO measurements, CO, CO₂, and HC data was used to perform a carbon balance and estimate the mass of fuel lost into the sump per cycle. The percentage lost was found to be proportional to injected mass, since the primary mechanism for liquid fuel transport into the cylinder is wall film flow, and higher injected mass will result in substantially larger fuel puddles near the valve. [30] See Appendix A for a summary of the carbon balance results. Lost fuel versus injected mass is shown in Figure 5-7.

Thus, while CO levels were indeed reduced by secondary oxidation in the manifold, the reductions were not nearly as dramatic as first thought. Using data from 4-cylinder rich operation, an estimate for the CO mole fraction in 3-cylinder operation was made by diluting the rich combustion products with the estimated mass flow of air through the non-firing cylinder. Figure 5-8 shows the result for λ<sub>overall</sub> = 1.00 and λ<sub>overall</sub> = 1.10. Note that the calculation agrees well with the measured results up to spark timing at 7 CAD ATDC for the λ<sub>overall</sub> = 1.00 case, and 3 CAD ATDC for the λ<sub>overall</sub> = 1.10 case. Thus, it can be concluded that beyond these points, secondary oxidation is occurring. However, for the λ<sub>overall</sub> = 1.00 case, at most only 2%

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7 To verify that HC was not being vaporized from the oil and contributing to emissions, the same test condition was run immediately before and after an oil change, with negligible differences in the emissions results.
CO by mole fraction is being oxidized, not the roughly 6% one would expect based on diluted equilibrium mole fractions for $\lambda_{\text{overall}} = 1.00$ based on mass flow rates. Figure 5-8 shows clearly how, for constant $\lambda_{\text{overall}}$ based on mass flows, the mixture becomes increasingly lean at later spark timings which require higher injected mass.

5.1.3 Results for $\lambda_{\text{overall}}$ Based on UEGO Measurements

With the actual combustion $\lambda$ maintained at a constant value, the results for CO oxidation and temperature rise are similar to those presented above, but with several significant differences. First, the case for $\lambda_{\text{overall}} = 1.00$ based on UEGO measurements is shown in Figure 5-9. For spark timing 7 CAD ATDC or earlier, the CO concentration is virtually unchanged, i.e. there is no secondary oxidation in the exhaust manifold. This is due to the relatively cool exhaust gas temperatures, which increase steadily with spark timing yet remain too low to oxidize the significant mole fraction of CO in the exhaust. For later spark timing, however, the trend in the catalyst-in temperature shifts abruptly, indicating the initiation of secondary oxidation. With further spark retard, there is negligible temperature difference between the two cases, despite significant heat transfer off the manifold and catalyst inlet pipe, and CO measurement confirm significant oxidation. However, a significant portion of this oxidation must be occurring upstream in the manifold, rather than in the catalyst inlet pipe, since there is no temperature increase between manifold and catalyst inlet.

For $\lambda_{\text{overall}} = 1.05$ based on UEGO measurements, shown in Figure 5-10, the temperatures are not significantly different for early spark timing, however the onset of CO is more abrupt, and the temperatures jump sharply after 5 CAD ATDC spark timing. For heavy spark retard, there is a temperature jump between manifold and catalyst-in measurement locations, indicating that rich combustion products are being oxidized between the two locations. [The sensible enthalpy increase from secondary oxidation is larger than the heat transfer losses between these two points. See discussion of heat release in §5.5.]

The rapid transition between the non-oxidizing regime and the oxidizing regime can be seen in the CO and temperature data. Figure 5-11 shows the exhaust gas temperatures in the manifold and at the catalyst inlet, along with the downstream CO percentage, for 200 fired cycles with $\lambda_{\text{overall}} = 1.05$ and spark timing 5 CAD ATDC. At this spark timing, the first signs of secondary oxidation are visible as spikes in the temperatures, which correspond with dips in the
CO. For this spark timing, some cycles, but not a majority, produce exhaust gases that are hot enough to oxidize CO when mixed with air; thus the oxidation reactions are not sustained.

This can be contrasted with Figure 5-12, which shows the same measurements for $\lambda_{\text{overall}} = 1.05$ and spark timing 9 CAD ATDC. Note the rapid drop in CO for this point seen in Figure 5-10. At this spark timing, secondary oxidation has not yet reached a stable operating condition (such as it does for spark timing 15 CAD ATDC), but many cycles are now hot enough to oxidize the CO, which is now consistently below the levels seen for much earlier spark timing.

For the two leaner cases with $\lambda_{\text{overall}} = 1.10$ and $\lambda_{\text{overall}} = 1.15$ (based on UEGO measurements), the trends are similar. For the leaner cases, CO oxidation begins earlier, as the less-rich combustion yields slightly hotter engine-out EGT and the presence of additional oxygen aids oxidation. However, with less total CO and H$_2$ to oxidize, the temperature increase versus that which would be achieved simply by retarding the spark is less significant. Figure 5-13 shows the results for $\lambda_{\text{overall}} = 1.10$, and Figure 5-14 shows those for $\lambda_{\text{overall}} = 1.15$. For $\lambda_{\text{overall}} = 1.15$, the CO at 15 CAD ATDC spark timing is only 0.115% by mole fraction, indicating nearly complete oxidation of the combustion products.

5.1.4 Differences Between Choice of Non-Firing Cylinder

What was originally thought to be a difference in mixing between the two non-firing cylinders tested was found to be related to the lost fuel issue. The manifold geometry shown in Figure 5-15 implies that the rich combustion products and air would mix better with Cyl. #2 as the non-firing cylinder, due to its shorter runner, central location, and since the runner exit for Cyl. #1 is closer to the manifold exit.

Initially, these geometry differences were thought to be responsible for the results in Figure 5-16, which shows consistently lower CO for cases with Cyl. #2 non-firing. However, further investigation revealed that for a given spark timing and overall AFR, the difference between the two non-firing cylinders could be correlated to slight differences in UEGO measurements, i.e. cylinder-to-cylinder differences between the two resulted in skewed calculations of overall AFR based on mass flow rates. This might be attributable to differences in the injector or air flow to a particular cylinder however it cannot be attributed to the location

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8 The high sensitivity of CO to AFR results in fairly large differences in CO concentration for very small differences in combustion AFR.
of the non-firing cylinder, especially since a difference was seen for very early spark timing when no reactions are occurring.

Figure 5-17 shows the comparison between Cyl. #1 non-firing and Cyl. #2 non-firing for overall AFR controlled based on the UEGO measurement. AFR was controlled to within 0.01 λ; see Appendix A. Note that for all four cases, the CO mole fractions are essentially constant for spark timing 5 CAD ATDC or earlier; there are no reactions occurring in this region, and there is no slow decrease in CO concentration as seen in Figure 5-16, where the combusted mixtures were becoming steadily leaner with later spark timing. Figure 5-17 also shows that there is only a small difference between the two non-firing cylinders, and only at heavy spark retard where secondary oxidation begins to take effect, particularly for the case where \( \lambda_{\text{overall}} = 1.00 \). This might be attributable to mixing effects, as there are only small differences in temperature between the two cases for late spark timing. See Appendix A for comparisons of EGT for the two cases.

5.2 Combustion Analysis for Fired Cylinders

Combustion variability and burn rate analysis was performed based on the cylinder pressure data for Cylinder #4, one of the three rich cylinders. As expected the COV of NIMEP increases with later spark timing, from about 4% with spark 20 CAD BTDC to about 12% for spark 15 CAD ATDC. There is no clear variation with overall AFR, since each of the four cases are rich of stoichiometric but not so rich as to yield incomplete combustion. In general, the COV for these cases even with very late spark timing is roughly equivalent to that for normal fast idle conditions (1.25 bar NIMEP, 1400 RPM) with earlier spark timing (35 CAD BTDC), since the firing cylinders are operating at higher load with rich combustion.

Figure 5-18 shows the location of 50% fuel mass fraction burned (MFB) versus spark timing for each of the four cases, for AFR based on mass flow rates. The location of 50% MFB scales linearly with spark timing, and the leaner mixtures burn more slowly, particularly at late spark timing. The later combustion phasing for the leaner cases was due to the fact that, for very late spark timing, combustion \( \lambda \) was in fact at or even above stoichiometric for \( \lambda_{\text{overall}} = 1.15 \) based on mass flow.

By comparison, the location of 50% mass fraction burned for the four cases with AFR controlled based on UEGO measurements has almost no variation with overall \( \lambda \), as shown in
Figure 5-19. This is due to the relatively small difference in combustion $\lambda$ for the four cases; $\lambda$ varies between 0.825 and 0.91, all of which are fairly rich but not so much so that combustion will be incomplete. As expected, 50% burn location varies linearly with spark timing.

WAVE simulation results shown in Figure 5-20 show that the burned gas temperature in the cylinder at EVO is proportional to the location of 50% MFB. As expected, later combustion phasing results in much hotter post-flame temperatures, which in turn enable the oxidation of CO seen for late spark timing.

5.3 CO Mass Flow Rates and Exhaust Gas Enthalpy

Figure 5-21 shows the steady-state catalyst-in CO flow rate versus the catalyst-in sensible enthalpy flow rate for each of the four 3-cylinder cases, along with data for 4-cylinder stoichiometric operation. The data in Figure 5-21 is based on AFR calculated from mass flow rates. For the 3-cylinder case with $\lambda_{\text{overall}} = 1.00$, the CO flow rate initially increases due to the rise in mass flow rate and relatively constant $\lambda_{\text{firing}}$, however as the combusted mixture becomes leaner and the CO begins to oxidize the flow rate decreases dramatically. In addition, for each case there is a large increase in enthalpy flow rate, due to both higher EGT and higher mass flow rates, since later combustion phasing requires more air and fuel to achieve the same torque. Compared to the 4-cylinder case, which has inherently low CO due to stoichiometric combustion, the CO flow rate is higher for the 3-cylinder cases unless the oxidation rate is high due to very late spark timing. However, the cases with late spark timing not only achieve lower CO but also higher enthalpy flow, which will result in faster catalyst light-off.

Figure 5-22 shows the same results for $\lambda_{\text{overall}}$ based on UEGO measurements. Note the different trends between the two methods; for the data in Figure 5-22, the firing $\lambda$ was held constant, thus as the mass flow rate increases with spark retard, the CO flow rate increases until CO oxidation begins to occur. After this transition point, CO/H$_2$ oxidation results in heat release while further spark retard increases EGT additionally, and the CO mass flow rate drops dramatically even as mass flow rate continues to increase.

Figure 5-23 shows the data for AFR based on mass flow rates normalized by what is considered normal fast idle, 4-cylinder stoichiometric operation with spark timing 20 CAD BTDC. For the best 3-cylinder cases, CO is reduced by approximately 40%, but more importantly the enthalpy flow rate is roughly 400% higher; this dramatic increase in enthalpy...
flow rate would have significant implications for catalyst light-off. Figure 5-24 shows the same results for AFR based on UEGO measurements. The trends are unchanged from the non-normalized data, and the enthalpy flow rate is still approximately 400% higher. However, only the leanest case with very late spark timing results in CO levels approaching those for stoichiometric operation, since the engine-out CO levels are significantly higher than for the unintentionally-lean combustion of the cases shown in Figure 5-23.

For normal 4-cylinder engine operation at a given speed and torque, exhaust gas enthalpy flow rate will increase with later spark timing. The same is true from 3-cylinder operation with the flow reactor exhaust manifold; however the oxidation of CO provides an additional heat release which increases the enthalpy flow more rapidly. Figure 5-25 shows enthalpy flow rate versus exhaust mass flow rate for the four 3-cylinder cases, with AFR based on mass flow rates. For the two richer overall AFR cases, the jump in enthalpy flow is pronounced, yet beyond this jump the trend is similar in slope to the leaner cases; i.e. once the bulk of the CO has been oxidized, further temperature increases are due primarily to higher mass flow from later spark timing. For the two leaner cases, enthalpy scales almost linearly with mass flow rate, since the change in the $\Delta T$ term is closely tied to the change in the $\dot{m}$ term. Moreover, the two leaner cases with very late spark timing have very low levels of CO and H$_2$ to oxidize, thus the primary means of increasing enthalpy flow rate is higher mass flow rate due to spark retard.

The trends in enthalpy flow rate versus mass flow rate are quite different for the cases where AFR was based on UEGO measurements, as shown in Figure 5-26. For low mass flow rates (i.e. early spark timing), enthalpy flow rate increases almost linearly with mass flow rate, as the temperature increases are relatively small. However, the transition into the CO/H$_2$ oxidation regime is readily apparent, as the jump in temperature produces a sharp change in the slope of the trends. Note that for the case of $\lambda_{overall} = 1.00$, which had the latest onset of CO oxidation, the transition in Figure 5-26 occurs at higher mass flow (i.e. later spark timing).

5.4 HC Emissions Measurements

Steady-state HC emissions were also investigated for each of the four 3-cylinder cases and the 4-cylinder stoichiometric baseline. Figure 5-27 shows the HC concentration versus spark timing for AFR based on mass flow rates. In each case, the HC steadily decreases as spark timing is retarded; as the post-flame temperatures increase, more of the unburned air-fuel
mixture is oxidized. Beyond the critical point where significant secondary oxidation occurs, the majority of the HC is oxidized, and for late spark timing the HC is on the order of only 10-20 ppm. For successively less-rich 3-cylinder cases, the HC concentration is lower for a given spark timing. However, HC was also reduced as the mixtures became increasingly lean with later spark timing.

As expected for the stoichiometric 4-cylinder case, HC is always lower for a given spark timing; the mixture is not burning rich, and there is less liquid fuel since each fired cylinder operates at lower load (constant net crankshaft torque). For spark timing 5 CAD ATDC, the HC was within the limits of measurement for the FFID.

Figure 5-28 shows the HC concentrations for AFR based on UEGO measurements. Note that concentrations are consistently higher in the region with no reactions, since the combusted mixtures were richer (requiring more injected mass). HC oxidation remains significant at heavy spark retard. Even with constant combustion $\lambda$, in the region where no significant secondary oxidation is occurring, HC still steadily decreases with later spark timing, i.e. hotter exhaust gas temperatures. However, the effect of this relatively small reduction in HC concentration (e.g. 1000 PPM reduction between spark 20 CAD BTDC and spark 5 CAD BTDC) is minimal in terms of heat release or CO levels; the reduction in mole fraction of $C_1$ is only 0.1%, and the corresponding increase in CO would be difficult to detect reliably.

Figure 5-29 shows the HC data for AFR based on mass flows in terms of mass flow rate versus enthalpy flow rate. The onset of secondary oxidation in the flow reactor manifold results in a rapid drop in HC flow rate as well as a rapid jump in enthalpy flow rate, particularly for the case of $\lambda_{\text{overall}} = 1.00$, which has the most HC and CO to oxidize and thus the greatest energy release. While HC levels never quite reach that of stoichiometric 4-cylinder operation with late spark timing, the enthalpy flow rates are significantly higher.

Figure 5-30 shows the same HC mass flow data for AFR based on UEGO measurements. Compared to Figure 5-29, where the mixture was becoming leaner with later spark timing, the HC mass flow rates are much higher for a given enthalpy flow rate, until the onset of secondary oxidation. The increase in HC with sensible enthalpy produced by later spark timing has to do with the increase in the amount of fuel injected. The rate of HC reduction with increasing enthalpy flow rate is slower for constant UEGO $\lambda$, as expected compared to cases with leaner mixtures. However, with sufficiently late spark timing, HC mass flow rates are negligibly
different from the leaner cases in Figure 5-29, indicating that nearly all of the EOHC is being oxidized. For $\lambda_{\text{overall}} = 1.00$ based on UEGO measurements, however, HC mass flow rates are still significant even for very late spark timing, since the injected mass values required to achieve a richer mixture at such late spark timings result in more EOHC than can be oxidized before the reactions are quenched.

Figure 5-31 shows the HC and enthalpy flow rate data for AFR based on mass flow rates normalized again by the numbers for 4-cylinder stoichiometric operation with spark timing 20 CAD BTDC, and Figure 5-32 shows the results for AFR based on UEGO measurements. Compared to this baseline, HC flow rate is reduced by an order of magnitude, while enthalpy flow rate increases by a factor of four. Thus, in the limiting, ideal case of hot exhaust system surfaces, the flow reactor manifold concept would yield a significant reduction in catalyst-in HC emissions as well as a significant reduction in catalyst light-off time.

5.5 Exhaust Oxidation Heat Release

To quantify the heat release from secondary oxidation, calculations were performed for each of the four values of $\lambda_{\text{overall}}$ (based on UEGO measurements), for spark timing 15 CAD ATDC. This was selected since it yielded by far the highest secondary oxidation.

5.5.1 Heat Release Contributions By Species

In order to estimate the heat release from oxidation of CO and H$_2$ in the exhaust manifold, the mass flow rates of CO and H$_2$ exiting the rich cylinders were estimated, assuming equilibrium combustion products for the rich $\lambda_{\text{combustion}}$ for a given case. The downstream CO mass flow rate was known based on measurements, and this yielded an estimate for the percent of engine-out CO oxidized; the same percentage was assumed for H$_2$.

These numbers would not take into account heat release from the oxidation of HC in the exhaust. HC measurements for 4-cylinder rich operation could not be used to estimate engine-out HC levels from the firing cylinders, since 4-cylinder operation yields substantially hotter EGT and thus does not result in the higher, “un-diluted” HC values one would expect, since even for rich operation some of the unburned HC is mixed with air and can undergo post-flame oxidation. Thus, in order to estimate the engine-out HC mass flow rates, the trends in HC mass
flow rates for early spark timing with non-reacting exhaust flow were extrapolated. While hardly a precise estimate, the heat release contribution from HC oxidation is relatively small.

5.5.2 Heat Release Contributions By Location

Next, calculations were performed to estimate where, physically, oxidation was occurring in the exhaust system. To estimate this, the exhaust system was broken down into two regions, hereafter referred to as “upstream” and “downstream,” as shown in Figure 5-33. T0 represents the average engine-out temperature (which is unknown), T1 is the measured manifold EGT, and T2 is the measured catalyst-in EGT. The total exhaust mass flow rate is \( m \), and the chemical enthalpy flow rate of CO, H2, and HC is represented as \( m_{c} h_{c} \). In the upstream region, some fraction, \( x \), of the CO, H2, and HC (which are lumped together) will be oxidized; the remainder, \( (1-x) \), will be oxidized in the downstream region.\(^9\) For each region, there is a net heat loss to the surroundings, \( \dot{Q}_{L} \). Thus, the energy balances for the two regions are as follows.

\[
\begin{align*}
mc_p (T_1 - T_0) &= x (m_c h_c) - \dot{Q}_{L,1} \\
mc_p (T_2 - T_1) &= (1-x) (m_c h_c) - \dot{Q}_{L,2}
\end{align*}
\]

In order to solve for \( x \), the heat loss terms from Equation (11) must be known. In order to estimate \( \dot{Q}_{L,2} \), an average heat transfer coefficient \( \bar{h} \) was estimated based on the known heat loss between the manifold and catalyst-inlet measurement locations for the cases with no secondary oxidation. The temperatures for no oxidation, \( T_1' \) upstream and \( T_2' \) downstream, were measured for numerous cases where no secondary oxidation occurred. Thus, an average value for \( \bar{h} \) was calculated from Equation (12), and found to be 1.51 W/K. Radiation, natural convection, and the total heat transfer area are all lumped into \( \bar{h} \). \( T_{r,x} \) is the log-mean temperature difference between the two sampling locations for the non-reacting cases. With a known value for \( \bar{h} \), Equation (13) below could be solved for \( x \).

\[
\dot{Q}_{L,2} = mc_p (T_1' - T_2') = \bar{h} (T_{r,x})
\]

\(^9\) CO measurements at the catalyst inlet and downstream mixing tank, along with temperature measurements a few cm downstream of the catalyst inlet, suggest that all oxidation is complete by the catalyst inlet location. For these tests no catalyst was present, as it would have been damaged by sustained high EGT.
5.5.3 Heat Release Contributions to Catalyst-In Sensible Enthalpy

In order estimate how much of the heat release from secondary oxidation actually contributed to an increase in catalyst-in sensible enthalpy, the temperatures that would have been present at the catalyst inlet in the case of no oxidation were estimated based on the data for early spark timings. Four-cylinder data shows that EGT trends are fairly smooth over the range of 20 BTDC to 15 ATDC spark timing, thus the extrapolation method should yield approximately the right temperatures. The enthalpy flow rates were approximated using these temperatures along with the calculated exhaust mass flow rate, compared to the actual measured catalyst-in temperatures.

5.5.4 Summary of Heat Release Calculations

Figure 5-34 summarizes the results of the heat release calculations and estimates. The total heat release decreases with increasing $\lambda_{\text{overall}}$, since the exhaust contains less rich combustion products to oxidize. CO oxidation accounts for nearly 2/3 of the total heat release, with $\text{H}_2$ oxidation yielding an additional ~20% and HC accounting for the remainder.

The manifold heat release versus catalyst-in heat release offers some insight into where oxidation is occurring. For all four cases, the majority of the heat release occurs upstream, with up to 81% upstream oxidation for $\lambda_{\text{overall}} = 1.00$, and as little as 53% upstream oxidation for $\lambda_{\text{overall}} = 1.15$. For leaner $\lambda_{\text{overall}}$, a smaller fraction of the total heat release occurs upstream, since the leaner mixtures will have longer ignition delay times. (See §6.3.2.) This explains why there is no temperature increase between manifold and catalyst-inlet seen for $\lambda_{\text{overall}} = 1.00$, but such a jump is seen for leaner mixtures.

For each case, roughly half of the total heat release contributes to increased catalyst-in sensible enthalpy, while the remaining half is lost as additional heat transfer to the environment. Indeed, heat transfer losses were observed to be significant, as the exhaust manifold and catalyst inlet pipe were glowing red hot, and significant convective currents could be felt directly above the exhaust system.

\[
\dot{m}c_p (T_2 - T_1) = (1 - x)(\dot{m} c_p) - h(T_{1/2})
\]  

(13)
5.6 Idle Quality for Three-Cylinder Operation

Of significant concern regarding the real-world applicability of this concept is the feel and quality of engine idle with only three fired cylinders. Customer perceptions of noise and vibration are critical, and running a 4-cylinder engine on three cylinders might create a problem. Figure 5-35 shows typical engine speed versus crank angle for both a 3-cylinder case and baseline 4-cylinder case; due to the loss of torque from Cyl. #1 not firing, shown in Figure 5-36, the engine speed varies significantly within each cycle compared to the 4-cylinder case.

An attempt to relate 3-cylinder engine testing to real-world idle quality parameters used by the industry found the crankshaft speed variations to be unusually small; this was attributable to the high inertia of the drivetrain in this test setup. Whereas a normal idling engine has only the flywheel mass external to the engine, this system has the flywheel, a driveshaft with steel coupling plates at either end, the dynamometer, and the electric motor. An attempt was made to model the engine and drivetrain as a simple dynamic system, as shown in Figure 5-37. However, the system is complex; the engine, with torque source and internal friction, is attached to a driveshaft with a compliant coupling designed to reduce impact loading to the drive shaft. This is in turn connected to the dynamometer, which itself is connected to the electric motor via a second compliant coupling. Thus the system consists of three unknown moments of inertia, three unknown friction terms, and two unknown compliance terms. Attempts to simplify this model and match engine results were unsuccessful. This difficulty might also be due to the fact that the dynamometer was engaged and actively controlling the torque.

Consultation with industry engineers suggests that at sufficiently high idle speed, customers would not perceive a missed firing event; in one anecdotal example, a 6-cylinder engine was idled with only three cylinders firing, and only at low speeds (below 900 RPM) was the difference perceptible. Thus, idle quality is not anticipated to be a real-world limitation to using this strategy.

5.7 Three-Cylinder Crank Starts

While the flow reactor exhaust manifold was found to yield substantial oxidation of rich combustion products with sufficiently late spark timing under steady-state, cold coolant conditions, oxidation was promoted by the hot exhaust system surface temperatures. With cold surfaces, such as those encountered in a real cold start, the thermal boundary layer would be
significantly thicker and oxidation reactions might be quenched. Thus, to evaluate the real-world feasibility of the flow reactor manifold concept, crank starts were performed with initial conditions of 20°C ECT and oil temperature. It should be noted that this test was solely carried out as a feasibility study and was by no means optimized for HC emissions.

5.7.1 Engine Speed and Torque Behavior

With sufficient, if not necessarily optimal, fuel and spark calibration on a cycle-by-cycle basis, an acceptable crank start was achieved by running on three cylinders. Figure 5-34 shows the engine RPM and MAP versus time for a typical 3-cylinder start as well as a 4-cylinder ECU start. Even with only three fired cylinders, the engine starts up rapidly, accelerating from crank as quickly as the ECU cold start. The 3-cylinder case yielded a small speed flare, after which speed settled to a fast idle speed of approximately 1300 RPM. For the ECU start, the fast idle period lasts from approximately 5-10 seconds into the start, after which the engine controller slows the engine to around 900 RPM. Speed control was not possible for the 3-cylinder start, since the throttle plate was locked open and no feedback control of airflow was possible with this engine.

Net crankshaft torque for the two cases was nominally the same, as shown in Figure 5-39; the first five seconds of engine operation were not shown due to the rapid transient and change in NIMEP for both cases. Once idle stabilized, however, the average NIMEP for 3-cylinder operation, calculated as shown, and COV of NIMEP were roughly equivalent to those for the ECU cold start.

5.7.2 Exhaust Temperature Response and Catalyst-In Enthalpy Flow

Figure 5-40 shows temperatures measured at the catalyst inlet, between the two catalyst bricks, and the catalyst outlet versus time for both a 3-cylinder cold start and ECU cold start. Nominal ambient temperature was 20°C. Catalyst-in temperatures quickly rise for the 3-cylinder case due to late spark timing, reaching 600 K in approximately 3 seconds; the ECU cold start does not reach 600 K after 20 seconds, due to its significantly earlier spark timing. The effect of catalyst light-off is seen as well, as the mid-catalyst measurement shows signs of light off for the 3-cylinder case.
Exhaust manifold surface temperatures, along with the EGT measured in the body of the manifold, are shown in Figure 5-41 for both cold starts. The rapid rise in EGT is clearly visible for the 3-cylinder case compared to the 4-cylinder ECU start. This translates into much faster warm-up of the manifold surfaces, both at the runner for Cyl. #4 (a fired cylinder) and in the middle of the manifold body. However, this data shows that heat transfer from the hot burned gases in the 3-cylinder case will be significant; the temperature difference between the bulk gases and the outer surface of the manifold is typically 300-400 K for much of the cold start. This will result in significant quenching of any secondary oxidation reactions near the manifold surfaces.

Based on the catalyst-in temperatures and the exhaust mass flow rate, cumulative catalyst-in sensible enthalpy was calculated for both cases, shown in Figure 5-42. The 3-cylinder case has dramatically higher enthalpy flow, due to both higher temperature and significantly higher mass flow as a result of retarded spark.

5.7.3 Cold Start Hydrocarbon Emissions and Catalyst Light-Off

Figure 5-43 shows the catalyst-in and catalyst-out HC concentrations for both the three-cylinder case and a 4-cylinder ECU cold start. The behavior of the two cases is quite different. For the ECU case, this engine uses bank fueling to simultaneously inject a large mass of fuel to each cylinder during cranking, since it cannot detect engine position within the cycle to know which cylinder will fire first. As a result, there is a substantial mass of unburned fuel entering the catalyst during cranking and the first few seconds of operation. However, this fuel does not simply pass through the catalyst; rather, much of it hits the front catalyst face and is adsorbed into the catalyst surface. However, it is not desorbed until roughly 15 seconds later, when the catalyst warms sufficiently for it to be released from the surface. This results in the large catalyst-out HC spike seen in the figure. Other work shows a similar effect. [21]

For the 3-cylinder case, with sequential fueling, catalyst-in emissions reach a much smaller HC peak, but are maintained at a fairly high level even 20 seconds into the cold start. This is because the engine does not move out of 3-cylinder heavy spark retard mode, and thus a significant amount of liquid fuel must be injected to maintain engine torque. This results in high EOHC relative to the ECU case later in the start.
Several definitions of “catalyst light-off” are used in the industry, but one of the more common is that of 50% conversion efficiency. Figure 5-44 shows the ratio of catalyst-out to catalyst-in HC concentration for both cases; with its high enthalpy flow rate, the 3-cylinder case achieves 50% conversion efficiency in only 9.1 seconds, compared to 28.0 seconds for the ECU case. The ECU result is somewhat skewed by the second spike seen in Figure 5-43 due to the release of liquid fuel from the catalyst surface. The cumulative catalyst-in enthalpy at the light-off time is considerably higher for the 3-cylinder case, suggesting that light-off might also be heat transfer rate-limited.

Figure 5-45 shows the cumulative catalyst-out (i.e. tailpipe) HC emissions versus time for the two cold start cases. The 3-cylinder case has a high initial rate of increase due to the very high injected mass needed to achieve a rich mixture at high load with heavy spark retard so soon after cold start; fuel delivery fraction will be quite low, particularly for the first few seconds. This, coupled with the high mass flow rates needed with heavy spark retard, results in high HC mass flow rates into and out of the catalyst prior to light-off. In addition, while secondary oxidation of HC is difficult to quantify under transient cold start conditions, there will inevitably be significantly more quenching than for the steady-state cold fluids case. Thus, despite the rapid light-off, only a small improvement in cumulative emissions is seen over the ECU cold start.

5.7.4 Cold Start CO Emissions

Figure 5-46 shows the catalyst-in and catalyst-out CO concentrations for both the three-cylinder case and a 4-cylinder ECU cold start, along with cumulative tailpipe CO mass. The ECU cold start results in very high CO levels since it uses a conservative fueling calibration; to ensure robust start-up under a variety of conditions and fuels, the calibration is intentionally rich. For the three-cylinder case, there is an initial CO spike, and subsequently catalyst-in CO levels are quite low. Figure 5-47 shows a light-off time based on CO conversion efficiency; for the 3-cylinder case, light-off is significantly faster.

5.7.5 Signs of Secondary Oxidation Under Real Cold Start Conditions

It is difficult to be certain whether secondary oxidation is occurring with a crank start, due to the rapid transients and high variability between individual starts. However, several signs
indicate that some secondary oxidation is occurring. First, there is a significant temperature jump between the manifold and catalyst inlet, about 100 K, despite significant heat transfer to cold manifold and catalyst inlet pipe surfaces. In addition, the catalyst-in CO is significantly lower than expected for the given fueling, and the downstream CO₂ is higher. However, it is very difficult to ascertain for these conditions the value of the actual, combustion λ. The lost fuel could be significant enough under true cold conditions that the mixture is leaner than expected.

5.8 Ideal Three Cylinder Cold Start Strategy for Fast Catalyst Light-Off

Achieving a robust, clean cold start with 3-cylinder operation presented a significant challenge given the limitations of the engine control system available. While fuel and spark were prescribed on a cycle-by-cycle basis, there was no control over air flow; the throttle plate was fixed partially open prior to the start, and subsequent MAP was governed solely by engine speed. However, there was no suitable means to provide feedback control of engine speed either. Thus, depending on the variability inherent in a cold start, the fast idle speed would settle at very different values. With 1300 RPM fast idle, as shown above, the engine ran considerably richer than for another start where idle speed was 1200 RPM, with the same injected mass per cycle and higher speed resulting in lower MAP.

In addition, emissions results were skewed by the inability to transition back to normal engine operation following catalyst light-off. Without an actuator to reduce air flow such that spark could be advanced and the fourth cylinder fired without a speed flare, the engine remained in 3-cylinder, high mass flow operation longer than necessary.

Thus, the following is presented as an idealized 3-cylinder cold start. The engine is crank started with three cylinders, and spark is quickly retarded to 15 CAD ATDC, as was achieved. The engine is run rich with heavy spark retard for approximately 5 seconds, until the catalyst starts to light off. Then, the engine is quickly transitioned back to normal 4-cylinder operation, with normal spark advance (~20-30 CAD BTDC) and stoichiometric fueling. The result would be a quick shift to low catalyst-in HC flow rates, which would level off the cumulative HC trace much sooner.

Using real data from the 3-cylinder starts and ECU starts, an idealized, composite cold start was assembled. Figure 5-48 shows the engine RPM and MAP behavior; the transition
between operating modes should be imperceptible to the customer, thus engine speed does not experience a rapid transient. The rate of change in MAP would be limited by transient fueling, i.e. too rapid a decrease in airflow would result in HC spikes. Figure 5-49 shows the composite for catalyst-in and catalyst-out HC emissions. The catalyst-in values are based simply on those obtained for 3-cylinder and ECU operation, while the catalyst-out values are based on conversion efficiency for fast light-off attained in three-cylinder mode.

Figure 5-50 shows the cumulative tailpipe HC emissions based on this idealization of the 3-cylinder start. Also shown is the HC standard for SULEV/PZEV vehicles, which equates to a total of 110 mg. Vehicles meeting SULEV/PZEV standards need not only fast catalyst light-off but also low catalyst-in emissions and extremely effective catalysts. Thus, the idealized 3-cylinder cold start case is not nearly clean enough to meet SULEV or PZEV standards, however it is not quite a fair comparison given that the 2.4L engine was discontinued in favor of a lower-emissions design to meet SULEV standards, and the associated catalyst is not designed for fast light-off.
Figure 5-1 – Steady-state catalyst-in CO percentage, manifold gas temperature, and catalyst-in temperature versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, \( \lambda_{\text{overall}} = 1.00 \) based on mass flow rates. Non-firing cylinder is Cyl. #1.

Figure 5-2 – Steady-state catalyst-in CO percentage, manifold gas temperature, and catalyst-in temperature versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, \( \lambda_{\text{overall}} = 1.05 \) based on mass flow rates. Non-firing cylinder is Cyl. #1.
Figure 5-3 – Steady-state catalyst-in CO percentage, manifold gas temperature, and catalyst-in temperature versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.10$ based on mass flow rates. Non-firing cylinder is Cyl. #1.

Figure 5-4 - Steady-state catalyst-in CO percentage, manifold gas temperature, and catalyst-in temperature versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.15$ based on mass flow rates. Non-firing cylinder is Cyl. #1.
Figure 5-5 - Steady state exhaust system surface temperatures and gas temperatures versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.00$. Runner temperature measured at center of upper surface of exhaust manifold runner for Cyl. #4. Surface temperature measured at center of upper surface of manifold body. Inset diagram shows measurement locations.

Figure 5-6 - Mole fractions of CO and CO$_2$ versus measured UEGO $\lambda$ values for a range of $\lambda$ and spark timings, normal 4-cylinder operation.
Figure 5-7 - Percentage of injected fuel mass lost into oil sump versus injected mass per cycle for 1400 RPM, 20°C ECT, steady-state cold coolant operation. Average NIMEP 1.25 bar. Cylinder #1 non-firing.

Figure 5-8 - Catalyst-in CO concentration versus spark timing for two different cases of $\lambda_{\text{overall}}$ based on mass flow calculations. Solid lines are for 3-cylinder operation, the dashed lines are for 4-cylinder rich operation plus a dilution calculation.
Figure 5-9 - Steady-state catalyst-in CO percentage, manifold gas temperature, and catalyst-in temperature versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.00$ based on UEGO measurements. Non-firing cylinder is Cyl. #1.

Figure 5-10 - Steady-state catalyst-in CO percentage, manifold gas temperature, and catalyst-in temperature versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.05$ based on UEGO measurements. Non-firing cylinder is Cyl. #1.
Figure 5-11 - Steady-state CO percentage measured in downstream mixing tank, manifold EGT, and catalyst-in EGT for 200 fired cycles at steady state, 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.05$ based on UEGO measurements, spark timing 5 CAD ATDC. Cyl. #1 non-firing.

Figure 5-12 - Steady-state CO percentage measured in downstream mixing tank, manifold EGT, and catalyst-in EGT for 200 fired cycles at steady state, 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.05$ based on UEGO measurements, spark timing 9 CAD ATDC. Cyl. #1 non-firing.
Figure 5-13 - Steady-state catalyst-in CO percentage, manifold gas temperature, and catalyst-in temperature versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.10$ based on UEGO measurements. Non-firing cylinder is Cyl. #1.

Figure 5-14 - Steady-state catalyst-in CO percentage, manifold gas temperature, and catalyst-in temperature versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.15$ based on UEGO measurements. Non-firing cylinder is Cyl. #1.
Figure 5-15 – Location of different exhaust runners for non-firing cylinders. Cyl. #1 exhaust runner is significantly longer, has more surface area, and is less centrally located.

Figure 5-16 – Steady-state catalyst-in CO percentage versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{\text{overall}}$ based on mass flow rates. Solid lines are for Cyl. #1 non-firing, dashed lines are for Cyl. #2 non-firing.
Figure 5-17 – Steady-state catalyst-in CO percentage versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{\text{overall}}$ based on UEGO measurements. Solid lines are for Cyl. #2 non-firing, dashed lines are for Cyl. #1 non-firing.

Figure 5-18 - Location of 50% mass fraction burned versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{\text{overall}}$ based on mass flow rates.
Figure 5-19 - Location of 50% mass fraction burned versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on UEGO measurements.

Figure 5-20 - Cylinder gas temperature at EVO versus location of 50% mass fraction burned for $\lambda_{overall} = 1.00$, for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT. Results from WAVE engine model, with combustion profiles from actual engine data.
Figure 5-21 - Steady-state catalyst-in CO flow rate in mg/s versus catalyst-in sensible enthalpy flow rate in kW for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on mass flow rates. Lowest curve is for 4-cylinder stoichiometric operation. Cyl. #2 non-firing.

Figure 5-22 - Steady-state catalyst-in CO flow rate in mg/s versus catalyst-in sensible enthalpy flow rate in kW for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based UEGO measurements. Lowest curve is for 4-cylinder stoichiometric operation. Cyl. #1 non-firing.
Figure 5-23 – Normalized steady-state catalyst-in CO flow rate versus normalized catalyst-in sensible enthalpy flow rate for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{\text{overall}}$ based on mass flow rates. Data normalized by values for 4-cylinder stoichiometric operation with spark timing 20 CAD BTDC. Cyl. #2 non-firing.

Figure 5-24 - Normalized steady-state catalyst-in CO flow rate versus normalized catalyst-in sensible enthalpy flow rate for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{\text{overall}}$ based on UEGO measurements. Data normalized by values for 4-cylinder stoichiometric operation with spark timing 20 CAD BTDC. Cyl. #1 non-firing.
Figure 5-25 - Steady-state catalyst-in enthalpy flow rate in kW versus exhaust mass flow rate in kg/hr for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on mass flow rates. Cyl. #2 non-firing.

For leaner AFR, more of enthalpy flow increase comes from increased mass flow rate rather than from temperature rise - less CO oxidized.

Data from spark timing sweeps

Figure 5-26 - Steady-state catalyst-in enthalpy flow rate in kW versus exhaust mass flow rate in kg/hr for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on UEGO measurements. Cyl. #1 non-firing.
Figure 5-27 - Steady-state catalyst-in HC concentration in PPM C\textsubscript{1} versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{\text{overall}}$ based on mass flow rates. Cyl. #2 non-firing.

Figure 5-28 - Steady-state catalyst-in HC concentration in PPM C\textsubscript{1} versus spark timing for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{\text{overall}}$ based on UEGO measurements. Cyl. #2 non-firing.
Figure 5-29 - Steady-state catalyst-in HC mass flow rate in mg/s versus catalyst-in enthalpy flow rate in kW for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on mass flow rates. Cyl. #2 non-firing.

Figure 5-30 - Steady-state catalyst-in HC mass flow rate in mg/s versus catalyst-in enthalpy flow rate in kW for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on UEGO measurements. Cyl. #2 non-firing.
Data from spark timing sweeps

Normalized by values for 4cyl, 1.25 bar NIMEP, $\lambda = 1.00$, Sp. 20° BTDC

Figure 5-31 - Normalized steady-state catalyst-in HC flow rate versus normalized catalyst-in sensible enthalpy flow rate for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on mass flow rates. Data normalized by values for 4-cylinder stoichiometric operation with spark timing 20 CAD BTDC. Cyl. #2 non-firing.

Figure 5-32 - Normalized steady-state catalyst-in HC flow rate versus normalized catalyst-in sensible enthalpy flow rate for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on UEGO measurements. Data normalized by values for 4-cylinder stoichiometric operation with spark timing 20 CAD BTDC. Cyl. #2 non-firing.
Figure 5-33 – Schematic of exhaust system representation for heat release calculations. Exhaust manifold and catalyst inlet pipe are broken down into two regions.

Figure 5-34 - Steady-state heat release calculations for 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, varying $\lambda_{overall}$ based on UEGO measurements. Spark timing 15 CAD ATDC.
Figure 5-35 - Engine RPM versus crank angle for 3-cylinder and 4-cylinder operation, average NIMEP 1.25 bar for each case, average RPM 1400 for each, spark timing 20 CAD BTDC, steady state 20°C ECT. Average of 200 cycles.

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Figure 5-43 - Catalyst-in and catalyst-out HC concentration in PPM C1 for 4-cylinder ECU cold start and 3-cylinder custom start.
Figure 5-44 - HC conversion efficiency for 4-cylinder ECU cold start and 3-cylinder custom start. Catalyst light-off is defined as 50% conversion efficiency. 4-cylinder case reaches 50% conversion efficiency after 28.0 seconds and 32.6 kJ cumulative enthalpy flow into catalyst. 3-cylinder case reaches 50% conversion efficiency after 9.1 seconds and 57.7 kJ cumulative enthalpy flow into catalyst.

Figure 5-45 - Cumulative tailpipe HC emissions versus time for 4-cylinder ECU cold start and 3-cylinder custom start.
Figure 5-46 - Catalyst-in and catalyst-out HC concentration in PPM C1, and cumulative tailpipe CO mass for 4-cylinder ECU cold start and 3-cylinder custom start.

Figure 5-47 - CO conversion ratio for 4-cylinder ECU cold start and 3-cylinder custom start. Catalyst light-off is defined as 50% conversion efficiency. 4-cylinder case reaches 50% conversion efficiency after 16.3 seconds and 19.0 kJ cumulative enthalpy flow into catalyst. 3-cylinder case reaches 50% conversion efficiency after 6.6 seconds and 38.0 kJ cumulative enthalpy flow into catalyst.
Figure 5-48 - Engine RPM and MAP versus time for ideal 3-cylinder cold start. RPM response closely follows that of an ECU cold start. MAP is high for first five seconds when engine is operating in 3-cylinder mode, then is quickly lowered as fourth cylinder begins firing and spark is advanced back to normal idle values.

Figure 5-49 - Catalyst-in and catalyst-out HC concentration versus time for ideal 3-cylinder cold start. 50% light-off time is approximately 8 seconds.
Figure 5-50 - Cumulative tailpipe HC emissions for ECU cold start, actual 3-cylinder start, ideal 3-cylinder start, and SULEV/PZEV emissions standard.
Chapter 6 - Modeling

Each phase of the project featured an associated modeling effort to further understanding of the underlying phenomena. An MIT-developed engine simulation package was used for the 2000 MY engine, while DaimlerChrysler provided a highly detailed WAVE model of the 2003 MY engine. WAVE is a commercially available 1-D engine simulation program developed by Ricardo Consulting Engineers. Neither model can directly predict mixture preparation effects due to the complexities and details of the intake port and cylinder head geometry, which, coupled with imprecise knowledge of injector spray patterns and fuel film development along the intake port wall and valve, make it difficult to precisely know how the fuel is entering the mixture. The third phase of the project focused on modeling of ignition phenomena and mixing in the exhaust manifold.

6.1 Modeling of IVO Timing Effects on First Cycle Engine Behavior

An engine cycle simulation program developed at MIT [24] was used to study the effect of IVO timing on cranking behavior. Engine geometry, including valve timing, was that of the 2000 MY 2.4L engine described in §2.1. The engine and ambient temperatures were 298 K, intake pressure was 1 bar, and cranking speed was fixed at 200 RPM. IVO timing was varied from 0 to 30 CAD ATDC, to evaluate the effect of retarded IVO. Only one cylinder was simulated.

The initial engine position for the simulation was at IVC, such that the cylinder would have a trapped charge of air at ambient temperature and pressure. Most simulation results were not sensitive to engine starting position, provided the intake valve was not open, as this would negate the effect of delayed IVO. The temperature of the trapped charge at IVO was a strong function of starting position however, as will be discussed.

6.1.1 Flow Effects

The effects of IVO timing on flow past the intake valves are shown in Figure 6-1, Figure 6-2, and Figure 6-3. When IVO is later than 10 CAD ATDC, a substantial vacuum is created by the piston motion, seen in Figure 6-1. The pressure drop across the valve creates a high mean curtain velocity, as shown in Figure 6-2. For the case of IVO at 30 CAD ATDC, the flow is
choked; however, the mean curtain velocity is subsonic due to the use of a discharge coefficient to account for boundary layer effects. The combination of high curtain velocity and low valve lift when the valve first opens results in a very high shear rate, shown in Figure 6-3.

While such high shear flow is an effective means of strip-atomizing the fuel film, the duration of high shear flow is quite short and occurs at low valve lift, only a fraction of a millimeter. The case for IVO at 20 CAD ATDC is shown in Figure 6-4. Thus, fuel transport to the cylinder will be limited by the upstream feed, and only a small amount of liquid fuel located directly at the valve seat will benefit from high shear rate atomization.

6.1.2 Charge Temperature Effects

The charge temperature at IVO depends strongly on initial engine position when cranking begins. To illustrate the possible scenarios, the case of IVO at TDC is considered. In the engine simulation, EVC occurs at 8 CAD ATDC, so there is some valve overlap. There are four possible scenarios to consider.

If the engine starts out between EVO and IVO, then the charge temperature at IVO for the first cycle should be at ambient (298 K in the simulation), since the temperature should not be changed by the exhaust process; the gas temperature and wall temperature are both ambient, and the pressure drop across the valve should have a negligible effect at cranking speed (200 RPM).

In the second possible scenario, with the engine starting between IVO and IVC, the subsequent IVO event is considered to be IVO for the first cycle. In this case, the charge at IVC will be at ambient conditions. Thus, the charge pressure and temperature at EVO will be determined by the heat loss of the motored cycle, along with the relative compression and expansion ratios as determined by valve timing. If the cylinder pressure at EVO is sub-ambient, the result will be back flow from the exhaust manifold to the cylinder; this back flow will determine the temperature of the final mixture, since the subsequent exhaust stroke should not significantly change the charge temperature. Thus, this becomes the temperature at IVO. For this case, there is a single charge temperature at IVO which is different from ambient, but does not depend on the specific engine position at the start of cranking.

If cranking begins with the engine between IVC and TDC compression, the same conditions apply as in the second case, with the exception that the effective IVC point is the
starting point of cranking. Thus, the charge temperature depends on initial position. Alternatively, if the engine starts between TDC compression and EVO, the case is the same as above except there is no compression process. Thus, the ambient gas in the cylinder will be expanded until EVO, and the charge temperature at IVO depends on the starting point of cranking.

Figure 6-5 shows the charge pressures and temperatures as a function of crank angle for the different crank starting positions. Due to the piston expansion process, extremely cold cylinder temperatures are attained at EVO. For example, if cranking begins at TDC and the charge undergoes a full expansion stroke, the gas temperature at EVO is only 150 K. When the exhaust valve opens, the charge temperature rises due to three factors: mixing of the charge with back flow of ambient temperature gas from the exhaust port, heat transfer between the cold fluid and warmer surfaces, and flow work done by the pressure difference across the exhaust valve. This latter effect is similar to that of filling a vacuum chamber with ambient air; when the pressure equalizes, the chamber contents have a temperature equal to the ambient temperature multiplied by the ratio of specific heats, γ.

The charge temperature at IVO (taken to be TDC exhaust for purpose of illustration) is shown in Figure 6-6. Note that depending on initial position, this temperature can be above, at, or below ambient. For four-cylinder engines, the most likely starting positions (i.e. the natural stopping positions) are mid-stroke for each cylinder, at 90 CAD, 270 CAD, 450 CAD, and 630 CAD as shown on the x-axis in Figure 6-6. [31] Thus, with nominal intake cam phasing (IVO at TDC), two cylinders would have a charge temperature at approximately ambient for the first IVO event, while the other two would have temperatures approximately 25 K below ambient.

Figure 6-7 shows the charge temperatures for various IVO timings, with the crank start position assumed to be IVC. The expansion from EVC to IVO results in a significant drop in initial charge temperature as IVO timing is retarded – a 30 CAD delay in IVO resulted in a 75 K drop in charge temperature. After IVO, however, the temperature rises quickly due to flow work and mixing with the fresh charge; depending on the valve timing, the temperature might overshoot the ambient value. However, the period of low charge temperature corresponds to the period of high shear flow. Thus, the effects of delayed IVO are confounding: the high shear rate results in better strip atomization, but said atomized droplets are drawn into a colder charge.
6.1.3 Compression Ratio and Volumetric Efficiency Effects

With a fixed cam profile, retarding IVO results in an equivalent delay of IVC. Thus, the effective compression ratio and compression time will decrease for retarded IVO. Figure 6-8 shows the reduction in effective compression ratio as a function of intake cam delay. With lower compression, the in-cylinder droplets are exposed to a lower charge temperature for a shorter time, both of which will inhibit fuel vaporization. Delaying IVO by 30 CAD yields a charge temperature approximately 30 K cooler at a given point during compression, as shown in Figure 6-9. In practice, this also places a limit on the extent to which IVO timing can be delayed. In the engine experiments, ignition was not reliable when IVO was retarded beyond 22 CAD ATDC, even when in-cylinder FFID measurements showed a near-stoichiometric mixture.

Figure 6-10 shows the reduction in volumetric efficiency as a result of later intake cam phasing. When IVO was delayed by 30 CAD, the trapped mass was reduced by 20%. Thus there is a first-cycle torque penalty associated with delayed IVO. Lower torque is detrimental to the fast acceleration of the engine, and thus could contribute indirectly to HC emissions.

6.2 Modeling of IVO Fuel Interaction with Valve Events During Fast Idle

In order to facilitate the fundamental understanding of how injected fuel mass interacts with flow past the intake valves, the fast idle condition tested on the 2003 MY 2.4L engine was modeled using Ricardo WAVE v7.0. The model includes detailed geometry for the intake and exhaust systems, shown in a schematic of the model as represented in WAVE in Figure 6-11, and the detailed valve lift profiles.

Ambient conditions were 25°C and atmospheric pressure, and the simulation was started with cold fluids. To simulate the fast idle condition, the speed was fixed at 1400 RPM, and the throttle was trimmed to yield a NIMEP of 1.25 bar. The primary variable was the intake cam phasing, in order to evaluate the effect of variable intake cam timing on the backflow. The intake cam timing was varied from 20 CAD advanced to 20 CAD retarded relative to stock timing. The fuel was modeled as indolene.
6.2.1 WAVE Model Results for Valve Curtain Velocity

The model was run for each of the major test cases evaluated in the laboratory, including the various valve timings, loads, and speeds. To determine the backflow, the following parameters were measured in WAVE for Cylinder #1

- Pressure in cylinder and intake port
- Density in cylinder and intake port
- Velocity in intake port
- Mass flux in intake port

Using the mass flux, density, and valve lift data, the valve curtain velocity was calculated for each of the valve timing cases. This represents the average velocity past the valve, which indicates the relative strength of the backflow. The valve curtain velocity results are shown in Figure 6-12 for three different intake cam phasings. The x-axis is CAD after TDC compression, and the y-axis is curtain velocity in m/s, where negative velocities represent backflow into the port and positive velocities represent forward flow into the cylinder.

As soon as the intake valve begins to open even a fraction of a millimeter, the flow past the valve back into the port becomes choked, but this velocity decays as the valve lift increases. There is still backflow even after the piston has passed TDC — the backflow is dependent on the pressure difference between the cylinder and the port; the pressure difference does not reverse until some time later in the expansion stroke. As IVO is advanced, cylinder pressure is higher at IVO, and the period of strong backflow lasts longer. With heavily retarded IVO, there is still a short period of backflow, but there is also a strong forward flow relative to stock valve timing.

6.2.2 Implications of Valve Flow for Fuel Injection Strategy

Extensive results from the WAVE model were presented in Chapter 5 along with the emissions results. Steady-state injection timing sweeps showed the interaction between fuel spray and flow past the valves. However, the WAVE modeling also sought to investigate the window of opportunity for split injection, i.e. injecting some fraction of the total mass into the open valve period. While injecting into forward flow has been repeatedly shown in the literature to yield higher HC emissions, injecting into the back flow might yield a mixture preparation benefit. As shown in Figure 6-13, however, the window of opportunity is brief — for stock valve timing, the peak backflow period lasts only about 6 ms at 1400 RPM, and the valve is just beginning to open for about half of this period.
6.2.3 Previous Studies of Back Flow into the Intake Port

Several previous works have investigated valve flow interactions with fuel during the mixture preparation process. Shin et al performed extensive high-speed visualization work under fast idle cold conditions similar to those investigated here. [32] Shin observed the back flow stripping off and shear-atomizing the liquid fuel film at the port wall and on the back of the intake valve, resulting in droplets carried back up the intake port. Larger droplets produced this way were deposited on port walls and the valve stem, while smaller droplets were suspended in the flow. The net result is that fuel vaporization improved due to the fuel mass being distributed over a larger surface area, while also driving liquid fuel near the valve seat away from the cylinder; liquid fuel transport into the cylinder is a primary source of HC emissions. See Figure 6-14. Laser-induced fluorescence measurements of fuel film thickness found that increased overlap backflow resulted in a thinner film near the intake valve, and that fuel was blown further back up into the port with increasing valve overlap. [33]

With split injection, some portion of the fuel is injected into the back flow (with correct injection timing). Thus, the diagram in Figure 6-14 might change to look like that in Figure 6-15. Since most of the fuel is still injected with the valve closed, the same phenomena occur including fuel film transport away from the valve seat, strip atomization off the wall film, and blowback of small suspended droplets. However, since the fuel is being injected into the back flow, several new phenomena occur. First, since the closed valve injection mass is smaller, the wall films will be smaller than for the case of a single closed valve fuel pulse. In addition, sufficiently small fuel droplets from the injection event will be entrained in the back flow and pushed back into the port/runner. However, the largest droplets will have too much momentum to be significantly redirected by the back flow, and thus will impinge on the wall films, valve, etc., as shown.

Earlier work by Holthaus et al found that the arrival time for the first fuel droplets from the injector correlated linearly with start of injection timing; however the mass arrival rates depended heavily on injection timing due to interactions between different size droplets and the intake port flow. [34] Depending on injection timing, either large drops or small drops will arrive at the valve first. The Holthaus study, which was conducted at low RPM in a non-production engine, found almost no transit delay between SOI and the arrival of the first fuel
droplets. However there was significant arrival delay on a mass basis, i.e. there is no single characteristic transit time from injector tip to the back of the valve.

6.2.4 Modeling of Fuel Droplet Interaction with Back Flow

Detailed modeling of fuel spray interactions with back flow in the intake port is prohibitively complex. Unsteady flow, both in terms of velocity and temperature, interacts with a highly non-uniform distribution of evaporating liquid droplets that collide, coalesce, and break up. Even with detailed information about injector spray patterns, targeting, and tip velocities, such modeling is beyond the scope of this project, and detailed injector information beyond basic droplet size data was not available. Thus, the following simplified models were developed.

The WAVE model was used to calculate the temperature, density, and average velocity of the back flow gases as a function of both time and distance from the valve. Figure 6-16 shows the map for back flow velocity; the very high velocities at 0 cm from the valve are WAVE’s calculation of velocity past the intake valve, and are thus not representative of an average flow velocity across the port area. Figure 6-17 shows the same map for gas temperature. The hot back flow gases travel a significant distance back up away from the valves, especially considering the injector is only 11 cm upstream of the valves.

Using these maps of temperature, velocity, and density from WAVE, a simple simulation of the flow path for a single droplet was developed, based on the method in [35]. The droplet is assumed to leave the injector at an initial speed into the unsteady flow from WAVE; all calculations are 1-dimensional, and gravity is neglected. The drag force on the droplet is calculated from Equation (14), and the coefficient of drag from Equation (15).

\[
F_d = \frac{1}{2} C A_p \rho u^2
\]  

\[
C = \frac{24}{Re} \left(1 + 0.14 \text{Re}^{0.7}\right)
\]  

\(F_d\) is the drag force, \(C\) is drag coefficient, \(A_p\) is the projected droplet area, \(\rho\) is gas density, and \(u\) is relative velocity between the droplet and gas. The droplet is assumed to be perfectly spherical with constant diameter, i.e. vaporization during flight was not modeled. Also since a 1-D flow model was used for gas velocity, the local high velocity at the valve curtain during the reverse
blow down was not represented. Thus, the model will somewhat over-predict travel toward the valve, and under-predict droplet blowback.

The stock 4-hole injector droplet size distribution was predicted based on the SMD and \( D_{0.9} \) as shown in Figure 6-18 based on assuming a Rosin-Rammler distribution. [36] The production injector on the 2003 MY engine has a SMD of 70 \( \mu \text{m} \) and a \( D_{0.9} \) of 203 \( \mu \text{m} \), measured at 100 mm. In an earlier study, for an injector with approximately 100 \( \mu \text{m} \) SMD and 400 \( \mu \text{m} \) \( D_{0.9} \), the measured average spray velocity was found to be 19.8 m/s. [37] The maximum possible tip velocity given the fuel rail pressure is about 35 m/s; given this uncertainty, the model was run with a range of initial droplet velocities. Droplets as small as \( D_{0.1} = 40 \ \mu \text{m} \) and as large as \( D_{0.9} = 203 \ \mu \text{m} \) were modeled. The range of injection timings represents the timing sweep for split injection used in §4.6, with a 25% split fraction, from the start of injection for the earliest EOI timing to the end of injection for the latest timing, thus covering the full range of droplet entry into the intake port.

Figure 6-19 shows the predicted droplet paths for a 40 \( \mu \text{m} \) droplet (\( D_{0.1} \)) assuming an initial velocity leaving the injector of 20 m/s. The small droplets are quickly entrained in the back flow, and for injection timings at 350 and 360 CAD ATDC, the droplet is blown back further than it is for injection at 370 CAD ATDC. This indicates that the later injection timing misses the bulk of the back flow, as it is quickly entrained but not affected as strongly later in the cycle. Figure 6-20 shows the results for a 70 \( \mu \text{m} \) droplet (SMD). Once again, the latest injection timing results in less blowback. For the larger 108 \( \mu \text{m} \) droplets in Figure 6-21, the penetration distance is much greater for early injection timings, and for injection at 370 CAD ATDC, the droplet is just barely stopped by the back flow. Once again, the latest injection timing does not experience as strong back flow. For the large 203 \( \mu \text{m} \) droplets modeled in Figure 6-22, there is no reversal, only a minor slowdown, due to their high inertia.

The model results help to explain the emissions results seen in Figure 4-14, where emissions were lowest for a 25% split fraction with EOI at 360 CAD ATDC. The latest EOI timing (370 CAD ATDC) does not hit the back flow as strongly, as seen in the droplet path simulations. These results were consistent for a range of initial droplet sizes as well as initial droplet velocities. This indicates that droplets injected at 370 CAD ATDC are more likely to hit the walls close to the intake valve, and are less likely to be broken up and/or vaporized by the hot back flow gases. The model does not include in-flight fuel vaporization off the droplet surface;
however this would only compound the benefits of interaction with the back flow, as smaller droplets are more highly entrained. The model shows however that only the smallest perhaps 25% of the droplets are likely to have significant interaction with the back flow. Thus most of the fuel will not have a significant interaction with the back flow, perhaps explaining the relative small emissions benefit for split injection. However, the strong shear flow around the valve at IVO is still likely to prevent any fuel from entering the cylinder directly.

The 25% split injection pulse had a nominal pulse width of 20 CAD, thus for EOI at 370 CAD ATDC some of the fuel would be injected earlier, when droplet/back flow interaction is strongest. However, for EOI at 360 CAD ATDC, a greater percentage of the spray distribution will hit the stronger back flow, thus the net result will be greater fuel vaporization/redistribution for EOI at 360 CAD ATDC. For still earlier injection timing, although back flow interaction is still strong, the maximum penetration depth is higher, thus the droplets are more likely to hit the wall closer to the valve; it is desirable from both an emissions and mixture preparation perspective to spread the fuel out over the largest area possible, to keep liquid fuel away from the cylinder and also to promote mass transfer off a thin, broad fuel film.

6.3 Modeling of Flow Reactor Exhaust Manifold

Building an accurate model of the flow reactor exhaust manifold concept developed in this project is no trivial task. The unsteady flow of partially reacted species from three firing cylinders mixes in a highly non-uniform, turbulent environment with an unsteady flow of air, resulting in exothermic chemical reactions. To fully capture these phenomena would require an engine cycle simulation with accurate exhaust composition coupled with a detailed 3-dimensional CFD model of the exhaust system – including detailed chemical kinetics. Such a model is far, far beyond the scope of this project. Thus, in attempting to understand and model the system, key concepts are retained in varying levels of detail.

6.3.1 Background on Post-Flame Oxidation

In order to understand the phenomena observed in the flow reactor exhaust manifold, the fundamentals of post-flame HC oxidation must be understood. Eng provides a thorough summary of HC oxidation chemistry in [1]. Post-flame consumption of HC species are primarily in the high-temperature chemistry regime, typically requiring temperatures greater than 1500 K.
In this regime, the HC oxidation reaction rates are controlled by $\text{H}_2\text{O}_2$ chain-branching reactions, and the overall oxidation process consists of three main steps: the larger fuel molecules are converted to smaller HC species and water, with minimal energy release; these small HC species are then oxidized to produce CO and water; and finally the CO is oxidized to $\text{CO}_2$, releasing the bulk of the energy. [1] This sequence can be represented conceptually by the following, where RH is the parent fuel, R' and R'' are smaller HC species, and X represents the O, H, or OH radicals that initiate the reaction. The specific initiation reactions depend on the parent HC species, and are different for paraffins and olefins.

$$RH + X \rightarrow R' + R'' + \text{H}_2\text{O} \rightarrow \text{CO} + \text{H}_2\text{O} \rightarrow \text{CO}_2 + \text{heat}$$ (16)

Since these reactions take place primarily at temperatures above 1500 K, where chemistry is fast, post-flame oxidation rates are primarily controlled by the diffusion of HC into the hot burned gas. Experiments and modeling have shown that the diffusion time scales from the boundary layer to the bulk burned gas are not small compared to the duration of the expansion stroke. [1]

Thus, given the attainable temperatures in the exhaust manifold, large amounts of unburned HC in the exhaust stream are undesirable – rather, what must occur is that spark timing must be late enough to ensure sufficient conversion of HC to CO and $\text{H}_2$ before EVO; the subsequent oxidation of CO and $\text{H}_2$ can take place at significantly lower temperatures such as those present in the exhaust manifold. In order for low temperature oxidation to occur, the presence of $\text{H}_2$ is critical. [15] [38] It is believed that $\text{H}_2$ can initiate the reactions that will subsequently sustain oxidation of CO and HC. [39]

Figure 6-23 shows the equilibrium mole fractions for combustion of air and fuel at six different AFR values, the leanest being stoichiometric. The engine must run significantly rich in order to produce even a few percent $\text{H}_2$ in the exhaust stream.

### 6.3.2 CFD Model of Exhaust Manifold

DaimlerChrysler was able to provide a 3-dimensional CFD model of the exhaust system, from the exhaust ports to the manifold exit. The simulation software used was Fluent v.6.2.16. The exhaust manifold was meshed using 50,000 cells from DaimlerChrysler's computer model.
of the production version of the prototype manifold used in testing; the interior surfaces are the same, only the wall material is different.

Boundary conditions for the model were determined from WAVE, to yield the unsteady flow of exhaust products into the exhaust system. However, since WAVE assumes equilibrium combustion products, the mole fractions closely resemble those in Figure 6-23, and thus do not include HC species. Thus, the model represents an idealized case with no liquid fuel and complete conversion to CO and H₂. Two cases were simulated, the first with Cyl. #1 non-firing and the second with Cyl. #2 non-firing.

Since the model does not include detailed chemistry, an alternative method was used to attempt to link the model output – mixture composition, temperature, and flow over a finely meshed 3-dimensional grid – to kinetics. Since the model can readily calculate temperature and mole fractions, it was originally intended to use ignition delay time, \( \tau_{\text{ign}} \), as a simple proxy for \( T \) and \( X_{\text{air}} \). However, since the exhaust temperature exiting the cylinder is unsteady, a given mixture can occur at a range of temperatures, and thus \( \tau_{\text{ign}} \) is not uniquely defined. This led to the mapping of \( \tau_{\text{ign}} \) as a function of \( T \) and \( X_{\text{air}} \), shown below in Figure 6-24. The numbers are \( 1/\tau_{\text{ign}} \) in units of 1/ms. Thus zero indicates “infinite” ignition delay – CHEMKIN calculations were carried out as long as 30 ms, and mixtures that did not ignite by 30 ms are considered non-reacting, given the residence time in the manifold (on the order of 10 ms).

These calculations show that for temperatures greater than 1000 K, as seen in the experiments for heavy spark retard, ignition delay times are on the order of 1 ms. Given the time scales for exhaust gases traveling through the manifold, this would indicate that, at heavy spark retard, secondary oxidation is mixing-limited rather than kinetics-limited.

Thus, this technique would indicate where oxidation of the rich combustion products is most likely to initiate; the locations with the shortest delay times are most likely to initiate the reactions. However, proceeding beyond this becomes increasingly difficult. As reactions and further mixing occurs, the exhaust gases are not simply a mixture of \( \lambda = 0.75 \) combustion products and air. (In theory, an overall stoichiometric mixture requires a firing \( \lambda = 0.75 \); in practice this is not the case.) Oxidizing the rich combustion products effectively changes the equivalence ratio, thus an entire range of \( \tau_{\text{ign}} \) maps would be required to include an energy generation term in the model. [For comparison, the ignition delay map for a firing \( \lambda = 0.83 \) (\( \lambda = 1.10 \) overall) is shown in Figure 6-25. As expected, ignition delay times are significantly
This quickly becomes computationally prohibitive, given the already intensive computing time required to simply compute the flow field.

Thus, for every given point in the model, if the model calculates the temperature and mole fraction of air \( (X_{\text{air}}) \), it could then calculate the ignition delay time (or \( 1/\tau_{\text{ign}} \)). This would indicate if in fact ignition is happening far upstream while also highlighting any differences between the two different non-firing cylinders.

6.3.3 CFD Model Results

Due to limitations of the combustion profiles available for use in WAVE, the simulation could not be run with the equivalent of spark timing 15 CAD ATDC. Thus, the exhaust gas temperatures were considerably lower than those necessary to initiate secondary oxidation. Nonetheless, the simulation results provided substantial insight into the mixing and flow within the exhaust manifold.

With heavy spark retard, the temperatures should be sufficiently high that ignition delay times are a stronger function of mixture composition than temperature. As shown in Figure 6-24 and Figure 6-25, an air mole fraction of 30% to 40% has the shortest ignition delay, and thus is most likely to initiate oxidation. Figure 6-26 shows a screen capture from the animation of transient air mole fraction for the case of Cyl. #1 non-firing. In general, oxidation appears most likely to occur at the outlet from the Cyl. #1 exhaust runner into the body of the manifold. In this location temperatures are still relatively high, and the mole fractions are in the appropriate range. A similar result is found for the case of Cyl. #2 non-firing, shown in a screen capture in Figure 6-27. Again, the runner exit is the most likely location for reactions to initiate; the simulations did not indicate the likelihood of significant reactions beginning upstream in the runners for the firing cylinders. For firing cylinders not directly adjacent to the non-firing cylinder, no mixing was observed until well downstream in the manifold. Firing cylinder(s) directly adjacent to the non-firing cylinder saw minimal mixing, driven largely by backflow pulsations.

The most interesting result from the simulations was the significant mixing that resulted from back flow into the non-firing cylinder. For each non-firing cylinder case, air is inducted at sub-atmospheric pressure, then compressed to TDC; as a result, there is heat transfer to the cylinder head, cooling the air. Thus, during the expansion stroke, the air expands to pressures significantly below atmospheric. As a result, at EVO, gas flows back into the cylinder from the
exhaust manifold. This effect was pronounced in animations of the transient mole fractions, indicating significant mixing.

Results from these CFD studies can be applied toward interpreting observed phenomena for the flow reactor exhaust manifold only to a point; once reactions are initiated, the species and temperatures begin to change rapidly, and the non-reacting simulation is no longer valid. From the simulation results, however, the difficulty in more detailed modeling is seen; the flow reactor exhaust manifold is by no means a "well-stirred" system. It falls into the intermediate region between perfect mixing and plug flow, where kinetics modeling is most challenging.
Figure 6-1 - Effect of IVO timing on cylinder pressure during the initial valve opening period.

Figure 6-2 - Effect of IVO timing on mean velocity at valve curtain during the initial valve opening period.
Figure 6-3 - Mean shear rate at the valve curtain during the initial valve opening period.

Figure 6-4 - Curtain velocity, mean shear rate and valve lift for IVO timing of 20 CAD ATDC. Top plot uses a linear scale, the bottom plot uses a log scale on the y-axis.
Figure 6-5 - Charge pressure and temperatures for the different crank starting positions from TDC exhaust: (a) 231° (same as the stock IVC position); (b) 270°; (c) 315°; (d) 360°; (e) 405°; (f) 450°. EVO at 488° and IVO at 720°.

Figure 6-6 - The charge temperature at IVO as a function of the crank start position. Ambient at 298K. IVO at TDC exhaust.
Figure 6-7 - Charge temperatures for different IVO timing, assuming initial piston position between IVO and IVC.

Figure 6-8 - Effective compression ratio versus IVO timing.
Figure 6-9 - Charge temperature during compression for different IVO timing. Insert – temperature at nominal spark timing which was at 10° BTDC.

Figure 6-10 - Relative trapped mass as a function of different IVO timing.
Figure 6-11 - Schematic of WAVE model for 2.4L engine.

Figure 6-12 - Valve curtain velocity calculated from WAVE simulation for three different intake cam phasings at 1400 RPM, 1.25 bar NIMEP, $\lambda = 1.00$, spark timing 35 CAD BTDC.

Figure 6-13 - Intake valve lift, curtain velocity, and mass flux versus crank angle for stock valve timing, 1400 RPM, 1.25 bar NIMEP, spark timing 35 CAD BTDC.
Figure 6-14 - Schematic of back flow at IVO driving closed valve injection fuel puddle away from valve seat, pushing small suspended droplets back up into intake port, and strip atomizing fuel off of wall film.

Figure 6-15 - Schematic of back flow at IVO with split injection. Wall films from closed valve injection are smaller since less mass is injected prior to IVO, small droplets from split fuel pulse are entrained in back flow, while large droplets are largely unaffected by back flow and hit walls/valve/etc. Other back flow effects, such as strip atomization off the wall film, are unchanged.
Figure 6-16 - Intake port/runner gas velocity versus CAD after TDC compression and distance from intake valve for stock valve timing, 1400 RPM, 1.25 bar NIMEP. High velocity at 0 cm from valve represents valve curtain velocity.

Figure 6-17 - Intake port/runner gas temperature versus CAD after TDC compression and distance from intake valve for stock valve timing, 1400 RPM, 1.25 bar NIMEP.
Figure 6-18 - Predicted droplet size distribution for 4-hole injector operating with 4 bar fuel rail pressure.

Figure 6-19 - Fuel droplet path from injector tip for various injection timings, 40 micron droplet ($D_{a,l}$). Period shown is extent of back flow into port/runner. Fuel leaving at 370 CAD ATDC has smallest interaction with back flow, as seen from final position closer to valve than any other case.
Figure 6-20 - Fuel droplet path from injector tip for various injection timings, 70 micron droplet (SMD). Period shown is extent of back flow into port/runner. Fuel leaving at 370 CAD ATDC has smallest interaction with back flow, as seen from slope following reversal.

Figure 6-21 - Fuel droplet path from injector tip for various injection timings, 108 micron droplet ($D_{0.5}$). Period shown is extent of back flow into port/runner. Fuel leaving at 370 CAD ATDC has smallest interaction with back flow, as seen from slope following reversal.
Figure 6-22 - Fuel droplet path from injector tip for various injection timings, 203 micron droplet \( (D_{90}) \). Period shown is extent of back flow into port/runner. Droplets are too large to have any significant interaction with back flow, and with early enough SOI timing hit valve.

Figure 6-23 - Exhaust gas equilibrium mole fractions for various \( \lambda_{\text{firing}} \). Minor species and radicals are not shown due to small mole fractions compared to major species.
Figure 6-24 - Inverse of ignition delay time for mixture of rich combustion products from firing cylinder with $\lambda_{\text{overall}} = 1.00$, for various initial mixture temperatures and initial mole fractions of air. Higher number indicates shorter ignition delay.

Figure 6-25 - Inverse of ignition delay time for mixture of rich combustion products from firing cylinder with $\lambda_{\text{overall}} = 1.10$, for various initial mixture temperatures and initial mole fractions of air. Higher number indicates shorter ignition delay.
Figure 6-26 – Screen capture of air mole fraction for Cyl. #1 non-firing, from animation of transient air mole fraction.

Figure 6-27 - Screen capture of air mole fraction for Cyl. #2 non-firing, from animation of transient air mole fraction.
Chapter 7 - CONCLUSIONS

7.1 First Cycle Mixture Preparation

The effects of delayed IVO timing on first cycle mixture preparation and HC emissions are complex. The delayed opening produces a pressure difference between the intake and cylinder which results in a strong shear flow that facilitates atomization of a portion of the fuel film. However, the expansion of the charge between EVC and IVO creates a significant temperature drop at IVO. For example, when IVO timing is changed from 0 CAD ATDC to 30 CAD ATDC, the peak valve curtain velocity varies from a few meters per second to choked flow; however, the temperature drops by 75 K. These effects may negate each other.

Furthermore, high shear flow lasts only briefly, thus only the liquid fuel located at the valve seat is affected. For fixed cam duration, late IVO implies late IVC. Thus, because the charge is stratified with a leaner mixture at the top of the cylinder, later IVC enriches the trapped mixture by displacing the leaner mixture between BDC and IVC. Experimentally, the first cycle in-cylinder equivalence ratio increased by approximately 10% when IVO timing was changed from 18 CAD BTD to 21 CAD ATDC, with EVC at 8 CAD ATDC. This observation was attributed mainly to the displacement effect since the flow effect was absent for cases with IVO earlier than EVC.

A 40% increase in the measured first cycle HC emissions was observed with retardation of the IVO timing from 18 CAD BTDC to 21 CAD ATDC. No conclusive explanation for this behavior was established; however it is most likely a result of stronger liquid fuel film flow into the cylinder.

7.2 Fuel Interaction with Valve Events During Fast Idle

Steady-state injection timing sweeps revealed a great deal about the interaction between liquid fuel in the port and flow past the intake valves. For CVI, EOHC emissions were independent of injection timing, and the residence time of the fuel in the port had little effect on the mixture preparation process. CVI results were insensitive to injection pressure and injector design, suggesting that droplet size does not affect CVI emissions.

With a single fuel pulse, OVI yielded substantially higher EOHC emissions irrespective of valve timing, engine RPM, fuel rail pressure, or injector design, for the two injectors tested.
The effects of flow past the valves on EOHC emissions were clearly visible for OVI. Emissions scale primarily with the convolution of fuel injection and valve lift, and strong forward flow during peak valve lift yields the highest emissions due to liquid fuel being thrown onto the cold cylinder walls and/or piston crown. Peak emissions were typically 50% higher than CVI emissions.

EOHC emissions were found to decrease with advanced intake cam phasing (for constant spark timing) due to later combustion phasing as a result of higher residual gas levels. Late combustion leads to a higher charge temperature late in the expansion process and results in more post-flame oxidation.

Split injection did not produce an emissions benefit under steady-state conditions with warm intake valves, however for simulated fast idle with cold valves and the second injection pulse aligned with the back-flow, a 6% reduction in EOHC was found by splitting the injected mass into a 75% CVI pulse and a 25% OVI pulse. At this second pulse timing and split fraction, the fraction of the injected fuel delivered to the combustible charge is larger than that found with single pulse CVI. This is the result of the hot back flow pushing the smaller fuel droplets from the OVI pulse back into the port and spreading the fuel over a larger area. This yields better mixture preparation and a smaller liquid fuel mass in the port and cylinder, as confirmed by puddle mass measurements. Split injection resulted in a smaller fuel puddle for the first few seconds of firing; however steady state measurements were unreliable due to the short duration of the second injector pulse.

7.3 Flow Reactor Exhaust Manifold

The flow reactor exhaust manifold concept was found to yield extremely high catalyst-in enthalpy flow with low catalyst-in HC and CO emissions, provided sufficiently retarded spark timing to produce exhaust gases hot enough to initiate oxidation. Under steady-state cold coolant conditions with hot, stabilized exhaust surface temperatures, enthalpy was increased by a factor of four while HC emissions were reduced by a factor of ten. However, operation in this mode was found to result in substantial fuel loss into the sump, due to the very high injected mass required to yield rich mixtures with heavy spark retard.

With real crank starts, the effectiveness of the concept is limited somewhat by cold exhaust system surfaces; however there is still evidence of significant CO oxidation. Very high
catalyst-in enthalpy flow resulted in a reduction in light-off time from 28.0 seconds to 9.1 seconds. However, high HC mass flow rate resulted in only a minor improvement in cumulative emissions.

In an ideal case, with full control over air flow to provide RPM/load feedback control, and with further optimization of fueling, light-off is believed to be achievable in only 5 seconds on the existing engine and catalyst, which were not designed for fast light-off. However, to realize SULEV-type emissions, lower engine-out HC would be required. This concept would benefit significantly from any devices that improve mixture preparation and reduce EOHC, since rich combustion products (CO and H₂) are desired far more so than liquid fuel in the exhaust.
REFERENCES


Appendix A - Additional Flow Reactor Exhaust Manifold Results

Figure A-1 through Figure A-4 shows the carbon balances for the four different cases of $\lambda_{\text{overall}}$ based on UEGO measurements. The total carbon mass was calculated based on the UEGO, CO, CO₂, and HC measurements. For early spark timing with no secondary oxidation, the percent mass into CO and CO₂ is relatively constant; however with the onset of secondary oxidation, the CO₂ percentage rapidly increases, while the CO and HC percentages decrease. In addition, with later spark timing, the percent of carbon mass as lost fuel (total carbon mass is proportional to total fuel mass) increases, as discussed in §5.1.2. Also notable is the lower mass percentages for CO, HC, and lost fuel with leaner overall mixtures.

In §5.1.2, it was noted that CO concentrations were consistent if UEGO values for $\lambda_{\text{overall}}$ were consistent. Figure A-5 shows the UEGO measurements for all of the cases: four different values of $\lambda_{\text{overall}}$ and two different choices of non-firing cylinder. This figure indicates the consistency of the UEGO measurements, typically within 0.01 $\lambda$ of the commanded value.

Figure A-6 through Figure A-9 shows the difference in exhaust gas temperatures for the two non-firing cylinder cases, with $\lambda_{\text{overall}}$ based on UEGO measurements. Differences between the two are small except for the manifold gas temperature with early spark timing; this is due to the location of the thermocouple relative to the manifold runner exit for Cyl. #2. The result is that for Cyl. #2 non-firing, the cold air is pumped directly at the thermocouple, whereas for Cyl. #1 non-firing the air does not hit as directly. For later spark timing, however, stronger blowdown flow from the firing cylinders results in more mixing in the manifold and negligible temperature differences between the two cases.
Figure A-1 - Carbon balance for steady-state 3-cylinder operation at 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.00$ based on UEGO measurements. Non-firing cylinder is Cyl. #1.

Figure A-2 - Carbon balance for steady-state 3-cylinder operation at 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.05$ based on UEGO measurements. Non-firing cylinder is Cyl. #1.
Figure A-3 - Carbon balance for steady-state 3-cylinder operation at 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.10$ based on UEGO measurements. Non-firing cylinder is Cyl. #1.

Figure A-4 - Carbon balance for steady-state 3-cylinder operation at 1400 RPM, 1.25 bar average NIMEP, 20°C ECT, $\lambda_{\text{overall}} = 1.15$ based on UEGO measurements. Non-firing cylinder is Cyl. #1.
Figure A-5 - UEGO $\lambda$ values for various cases of 3-cylinder operation. Engine controlled based on UEGO $\lambda$.

Figure A-6 - Exhaust gas temperatures measured in the manifold body and catalyst inlet for 1400 RPM, 1.25 bar average NIMEP, $\lambda_{\text{overall}} = 1.00$ based on UEGO measurements. Dashed line is for Cyl. #1 non-firing, solid line is for Cyl. #2 non-firing.
Figure A-7 - Exhaust gas temperatures measured in the manifold body and catalyst inlet for 1400 RPM, 1.25 bar average NIMEP, $\lambda_{\text{overall}} = 1.05$ based on UEGO measurements. Dashed line is for Cyl. #1 non-firing, solid line is for Cyl. #2 non-firing.

Figure A-8 - Exhaust gas temperatures measured in the manifold body and catalyst inlet for 1400 RPM, 1.25 bar average NIMEP, $\lambda_{\text{overall}} = 1.10$ based on UEGO measurements. Dashed line is for Cyl. #1 non-firing, solid line is for Cyl. #2 non-firing.
Figure A-9 - Exhaust gas temperatures measured in the manifold body and catalyst inlet for 1400 RPM, 1.25 bar average NIMEP, $\lambda_{\text{overall}} = 1.15$ based on UEGO measurements. Dashed line is for Cyl. #1 non-firing, solid line is for Cyl. #2 non-firing.