FUEL METERING EFFECTS ON HYDROCARBON EMISSIONS AND ENGINE STABILITY DURING CRANKING AND START-UP IN A PORT FUEL INJECTED SPARK IGNITION ENGINE

by

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To my parents,
for their endless support, encouragement, and guidance.

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Brigitte Marie Paule Castaing

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ABSTRACT

To gain a better understanding of engine behavior in starting, cranking tests were performed on a 2.0L 4-cylinder Ford Zetec engine. The tests consisted of varying the initial engine position, starting temperature, and fueling strategy during the first two cycles in order to elucidate the effects of these variables on IMEPg and HC emissions. The engine stopping position is a function of the gas load balance on the piston after the engine has been shut off. The most important effect of that position is on the instantaneous engine speed during cranking, which varies according to the sequence of firing events in the engine during a given cycle. Increased speed increases engine output through improved combustion phasing and heat release schedule. For a given starting position, IMEPg is fairly insensitive to in-cylinder λ within the range of 0.65 to 1.1; this is due to the combination of decreased combustion efficiency and improved heat release phasing. For all temperatures and starting positions, the firing threshold is $\lambda \sim 1.1$; tests run under colder temperature conditions require more fuel to reach this limit. Hydrocarbon emissions are sensitive to in-cylinder λ outside the range of 0.65 to 1.1; lean and rich mixtures result in high HC levels due to misfires and partial burns, respectively. In addition, for a given in-cylinder λ, lower temperatures result in higher HC levels. Open valve injection causes non-uniform in-cylinder mixtures, liquid fuel deposits in-cylinder, and spark plug wetting, and is therefore detrimental to engine stability and HC emissions.

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NOMENCLATURE

PFI	Port-fuel injected
SI	Spark ignition
ECU	Electronic control unit
FFID	Fast Flame Ionization Detector
UEGO	Universal exhaust gas oxygen sensor
λ	Relative air-fuel ratio
IMEPg	Gross indicated mean effective pressure
EOHC	Engine-out Hydrocarbons
CAD	Crank angle degree
IVO	Intake valve open
IVC	Intake valve closed
OVI	Open valve injection
CVI	Closed valve injection
BDC	Bottom dead center
ABC	After bottom center
BBC	Before bottom center
BTC	Before top center
ATC	After top center

Chapter 1: INTRODUCTION

1.1 MOTIVATION

This project has revolved around one main concern: the need to optimize cranking engine control strategies. Driven by customer demands and stricter emissions regulations, the need to improve start-up strategies has recently become increasingly important. During normal operation, catalytic converters are able to control emissions and maintain relatively low tailpipe levels. However, converters are ineffective until they have reached operating temperatures and are therefore essentially ineffective during start-up. Consequently, engine out (EO) emissions in that period translate directly into tailpipe emissions. With the advent of fast light-off catalysts, the emissions from the first few cycles in the cranking process will be a substantial portion of the total trip emissions. The fueling strategy for the first few cycles thus plays an important role in maintaining acceptable emissions levels.

The three regulated emissions are carbon monoxide (CO), oxides of nitrogen (NO_x), and hydrocarbons (HC). Of the three, HC is the most directly affected by the fuel metering strategy and is thus the subject of this investigation.

In typical current start-up strategies for port fuel injected engines, the Engine Control Unit (ECU) employs a bank-injection method. Once the ECU senses crankshaft rotation, it waits approximately 30° of crank rotation and then simultaneously injects a large amount of fuel to each cylinder. During normal engine operation, fueling is based on the estimated amount of air inducted to each cylinder and, in the case of engine transients, on the calibrated fuel transport behavior. During cranking, which may be considered an extreme case of transient, much of the engine parameters are unknown. For example, the engine stopping, and therefore starting, piston position is not known: the air induction, fuel delivery, and engine combustion behaviors depend critically on such. The method of over-fueled bank-injections is necessary to ensure that at least one cylinder fires on the first cycle, and that the remaining cylinders quickly follow.

There are several problems with this method. Because of the incomplete knowledge of the engine's initial state, the first few cycles are far from optimized: the fuel is not injected at the right time relative to valve and piston position, and the cylinder is often substantially over-fueled. The method results in both uncertainty in combustion behavior, and a large potential for high emissions. To satisfy customer demands and emissions requirements, it is imperative that the engine fires quickly and stably; sub-optimal starting may lead to driveability problems. In addition, a large amount of fuel may not be consumed, based on the fact that some cylinders will either partial burn or misfire as a result of the inappropriate fueling. This behavior carries several negative consequences. The fuel efficiency tends to be fairly low; quite a bit of fuel is wasted in the first two cycles. There are also very high levels of engine out unburned HC (EOHC), which cause both health and government regulation problems.

Figures 1.1 and 1.2 demonstrate the above description in a production vehicle. The data were taken from the Federal Test Procedure for a light-duty vehicle. Figure 1.1 shows that over 80% of EOHC emissions recorded in the test cycle are emitted in the first 100 seconds since the catalytic converter is ineffective during this time. For future fast light off catalysts, the first five seconds will contribute to a major part of the tailpipe emissions. Figure 1.2 illustrates the potential for improving the Engine Out Hydrocarbon

(EOHC) emissions in the first few seconds of start-up. It is apparent that in this period, the emissions index (the ratio of the EOHC flow to the fuel flow) is substantially higher than in normal operating cycles. Therefore, there is great opportunity to improve.

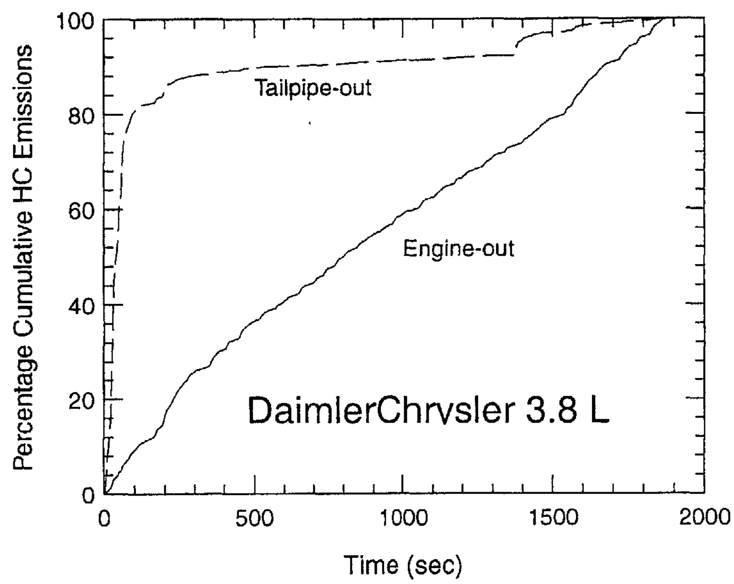


Figure 1.1: Percentage of cumulative HC emissions

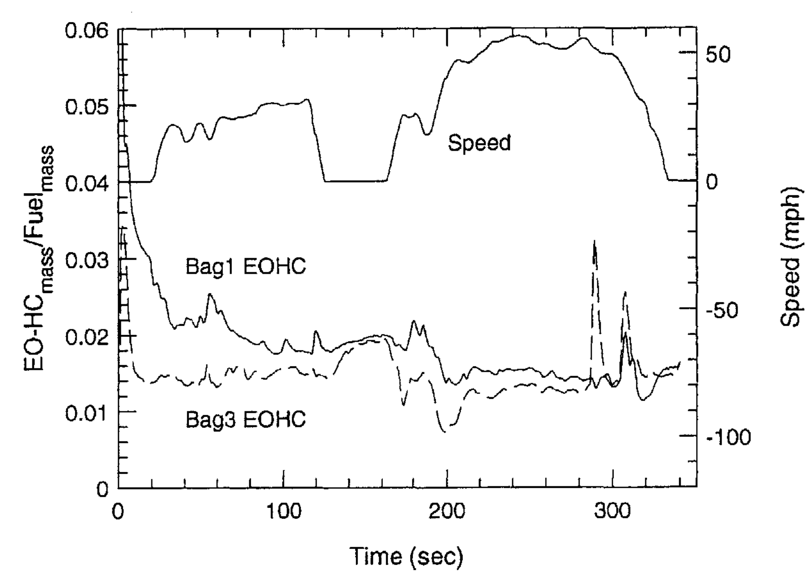


Figure 1.2: EOHC emissions during cranking and start-up

1.2 PREVIOUS WORK

While engine stability and HC emissions during cranking have been of concern for some time, relatively little work has been done to elucidate the problem and look for solutions. In 1995, Henein, Asmus, et al. [1] performed tests in order to correlate the engine starting position with engine out Hydrocarbons. They found that, with an arbitrary but fixed fueling strategy, the engine starting position played an important role in determining EOHC behavior in the first cycle of operation. They also suggested that regulating MAP and spark timing could help reduce HC levels. While the test provided a starting ground for studying and understanding HC emissions during cranking, little was done to gain a better understanding of the necessary mixture preparation during start-up.

Other work has been done to investigate fuel transport phenomena during cranking to improve engine stability. These studies include determining the minimum fuel and EOHC for cold starts [2]; visualizing first cycle port fuel transport [3]; sampling measurements of in-cylinder mixtures [4,5]; the trapping and measuring of first cycle fuel and HC [2,7]. This project focuses on gaining a better understanding of the effects of starting positions and fuel metering on both HC emissions and fuel transport behavior during cranking.

1.3 OBJECTIVES

This project was designed to elucidate the effects of fuel metering at different temperatures on engine stability and HC emissions during the first three cycles of operation. The goal is to gain a better understanding of the fuel preparation and combustion process during cranking, and in doing so, to demonstrate the benefits of incorporating initial engine conditions in the fueling strategy during these cycles.

1.4 METHOD

The project consists of a series of tests in which both the initial piston position and the initial engine temperature varied. For the different starting conditions, various fuel strategies were tested to optimize cranking behavior. The focus was placed on obtaining good engine stability and reducing HC emissions.

All tests were performed on one cylinder in a 2.0L 4-cylinder engine; the other three cylinders were controlled by the engine ECU. Engine starting position was monitored with an absolute encoder mounted on the camshaft, and in-cylinder and exhaust HC measurements were recorded using a fast Flame Ionization Detector.

Chapter 2: EXPERIMENTAL APPARATUS

The experiments in this project were conducted on a Ford four-cylinder four valve per cylinder port fuel injected Zetec engine mounted to a Froude eddy current water cooled dynamometer. Engine specifications can be found in Appendix A. In order to obtain a consistent data set, all data were taken from cylinder number 4 (at the transmission end of the engine block), which was controlled by a separate computer; cylinders 1, 2, and 3 were controlled by the Ford ECU. Fuel injection and spark timing for the first two cycles of cylinder 4 were programmed by the user. Subsequent cycles were controlled through a computer program; the code determines the fuel pulse width to create a stoichiometric mixture by using the intake manifold pressure information for the speed-density calibration. For certain tests, the spark in cylinder 4 was also controlled. The exhaust runner for cylinder 4 was separated from the other three to enable accurate single-cycle (cylinder) exhaust HC measurements.

2.1 FAST FLAME IONIZATION DETECTOR (FID) [7]

2.1.1 General concept

Unburned Hydrocarbon measurements were taken using a Cambustion Fast Flame Ionization Detector (FID). The FID device was first introduced when the importance of understanding and controlling vehicle and engine emissions grew due to increasingly stricter government regulations. The FID uses a hydrogen flame to burn Hydrocarbons. This releases ions and in turn creates a voltage proportional to the HC introduced.

The device consists of a sampling tube, a gas handling system, and the electronic controls. A sample of exhaust gas is collected through the sampling tube and fed through to the FID chamber where it passes through a diffusion flame, created by hydrogen in an air stream. The flame causes the Hydrocarbon bonds to break, releasing ions which then migrate to a negatively biased electrode, creating a voltage. This voltage, proportional to the number of Carbon atoms burned in HC form, can be converted to read a HC concentration in ppm C1.

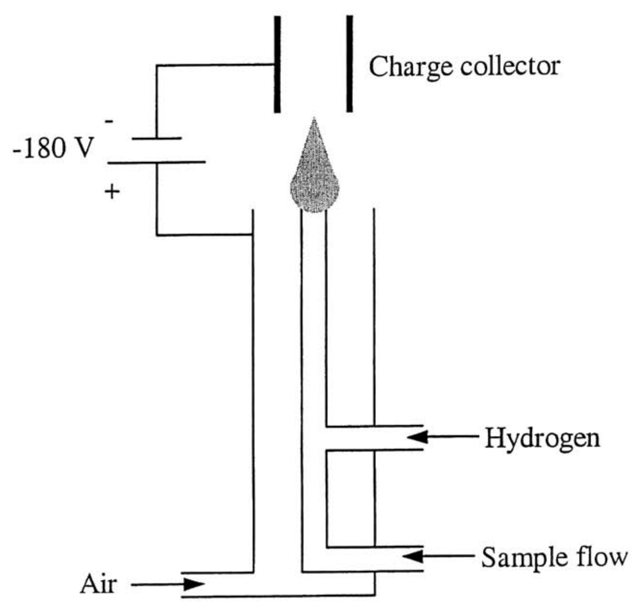


Figure 2.1: Basic FID Diagram [7]

The Fast FID differs from the conventional FID in that it allows for cycle-by-cycle HC measurements due to the high sampling frequency. Once the sample has been collected, it is fed directly into the FID head and through the flame rather than through a pump as in conventional design. This minimizes mixing and transit time in the sample tube, hence decreasing the error caused by both of these behaviors. Because the pump has been eliminated, a system is needed to maintain a pressure below atmospheric in the FID head in order to draw the sample in. This is achieved with a constant pressure chamber between the FID head and the sampling tube, which provides the necessary pressure difference to pull a sample through the line. The addition of a vacuum chamber also reduces error caused by pressure variations in the sampling system. When collecting from the exhaust stream, the pressure in the exhaust port varies throughout the cycle as the valve opens and closes. This variation causes sample flow rate variations, which affect the reading. Introducing a constant pressure chamber regulates the sample flow rate into the FID head.

The FID device is capable of measuring both in-cylinder and exhaust HC. Measuring in-cylinder HC provides information on different operating characteristics that are useful in understanding the behavior and performance of the engine. It is possible to determine:

- air/fuel ratio at time of ignition;
- variation in air/fuel ratio within a cycle and from cycle to cycle;
- the residual gas fraction of the charge;
- the in-cylinder storage and release of HC mechanisms.

2.1.2 Typical sampling signal

For a typical engine cycle, the FID reading starts low due to the residual gases left in the cylinder. As the new mixture comes into the cylinder and mixes with the remaining gases, the signal slowly rises. However, this portion of the data is not very accurate due to the low sampling pressures. The signal continues to rise through most of the compression stroke, eventually reaching a plateau, which represents the in-cylinder mixture conditions prior to combustion. When the flame travels across the cylinder and over the sampling tube, the FID signal drops to zero as it is surrounded by burnt gases. Toward the end of the cycle, the signal increases slightly as a result of HC release from crevices and the oil layer. The cycle then repeats itself.

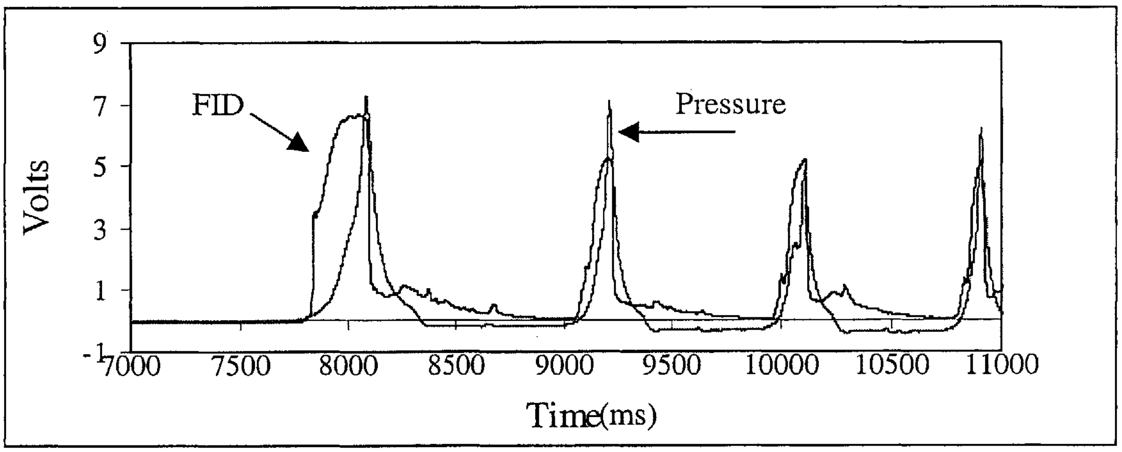


Figure 2.2: Typical cranking in-cylinder FID signal

Measuring HC in the exhaust port proves to be easier than in-cylinder for two reasons. First, the system can be calibrated in-situ, which is not feasible in-cylinder. Second, the sampling pressure is approximately atmospheric at the exhaust valve and in the port, and this results in smaller pressure variations that can be compensated for by the FID system. In a typical single-cycle sample, the signal remains constant while the exhaust valve is closed, representing stagnant air in the exhaust port. If the FID was calibrated just prior to the test, the reading will correspond to the span gases used. The first peak in the signal can be an indication of two things: either unburned fuel that leaks past the closed valve during compression and the early stages of combustion, or unburned fuel that was stored in the crevices around the valve and is released into the runner. This initial peak is typically followed by a small peak, caused by the flow of gases around the sampling tube being swirled by the flow reversal. As the exhaust valve opens, the signal drops sharply as exhaust gases rush past the sampling tube. Towards the end of the exhaust stroke, before the valve closes, the level briefly rises as unburned gas escapes the crevices, oil layers, and ring grooves. The cycle then repeats itself.

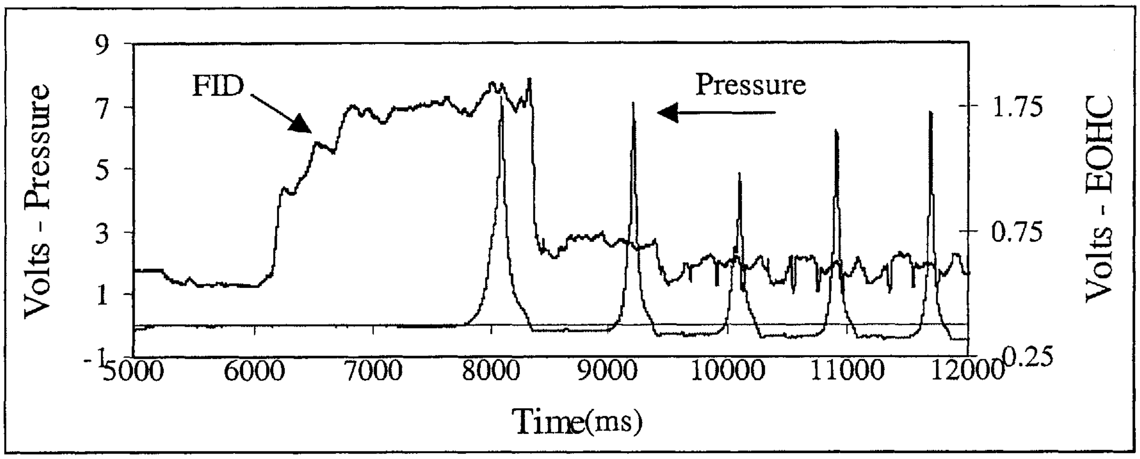


Figure 2.3: Typical cranking exhaust FID signal

2.1.3 Calibrating

Calibrating the system is fairly simple when measuring exhaust HC, but has proven to be more difficult in-cylinder. To calibrate the system, a gas of known composition is introduced into the system. The voltage corresponds to a certain reading in ppm C1; this correlation can then be used to convert the recorded HC voltage into a usable reading. For exhaust measurements, Propane/Nitrogen mixtures are typically used. Calibration can be done in-situ, which helps to decrease errors caused by drift. In-cylinder calibration is more difficult because it is not possible to complete in-situ.

2.1.4 Common problems with the Fast FID

There are two main problems that can arise when using the FID to measure HC: condensation in the vacuum line and in the sampling line. In the vacuum line, condensed water vapor can alter the FID chamber pressure by entering and clogging the vacuum-regulating valve. This problem is no longer an issue due to a new design in which only air is allowed to pass through the valve. It is only problematic at very low experimental temperatures or at very high ambient humidity.

Water vapor from the exhaust gases can clog the sampling line, as the dew point of these gases is 50°C. This causes the signal to fade or disappear as the droplets enter (burst) the flame in the FID head. Heating the sampling line, and hence decreasing the water vapor condensation and increasing heat transfer between the gases and the tube, can typically solve this problem.

2.2 BEI ABSOLUTE ENCODER

In order to determine the starting position of the engine, a 256-bit BEI absolute encoder was mounted on the camshaft. With this device it is possible to instantaneously read the engine's position at engine start and stop, enabling experiments at various starting positions.

2.3 DATA ACQUISITION

All data were taken using a National Instruments LabView data acquisition system. The following channels were programmed and used during all experiments:

in-cylinder HC, exhaust HC, in-cylinder pressure, speed, MAP, fuel delivery, UEGO, and crank angle degrees.

However, results focus on in-cylinder and exhaust HC measurements, in-cylinder and manifold pressures, and speed.

2.4 Fuel Type

California Phase II fuel was used throughout the project. For hot and ambient starts, summer fuel was used (RVP 7), and a winter blend (RVP 11) was used for the cold starts.

Chapter 3: TEST VARIABLES

There were four main variables involved in this experiment: fuel metering, spark timing, temperature, and piston starting position.

3.1 FUEL METERING

The amount of fuel injected and the timing of the injection relative to BDC were varied. Tests were conducted in two series: the first set consisted of using an optimized 1^{st} cycle FPW (optimized for a stoichiometric mixture) and varying the amount injected in the 2^{nd} cycle. This provided a baseline for the second set of tests. In this second set, only the 1^{st} cycle FPW was varied while the 2^{nd} and 3^{rd} cycle FPWs were maintained at constant levels, which provided a stoichiometric charge in-cylinder. This permitted an accurate examination of the amount of fuel necessary in the 1^{st} cycle to start the engine, as well as the effects of over-fueling and residual fuel on subsequent cycles.

The fuel injections in cylinders 1, 2, and 3 were determined by the Ford ECU and were not varied. The fueling for the first two cycles of cylinder 4 was specified by the operator. For the subsequent cycles, cylinder 4 fueling was controlled through a code, which determines the appropriate injection for a stoichiometric charge, calibrated on both speed and air density based on the pressure in the intake manifold.

3.2 SPARK CONTROL

During the majority of the experiments, spark timing was set to 10° BTC, which is the nominal setting for the Ford Zetec. A small set of tests were performed in order to gain a better understanding of the effects of spark timing on combustion phasing and IMEPg. For these tests, the spark was varied from 10° BTC to 15° ATC in 5° increments.

3.3 TEMPERATURE

Tests were conducted at three different coolant temperatures to compare the necessary fuel metering requirements and its effect on engine stability at various operating conditions. Tests were completed on a hot engine (90°C), at ambient conditions (22°C), and on a cooled engine (0°C). The manner in which the engine and test cell were prepared for these tests will be explained in further detail in a later section.

3.4 ENGINE STARTING PISTON POSITION

Five starting piston positions were chosen to represent typical engine stopping positions, as explained by Asmus and Henein [1]. The stopping position is determined by the balance of forces on the piston from the gases remaining in-cylinder and the connecting rod. As was explained in SAE #2000-01-2836, for a four-cylinder engine, this typically means that the starting piston position in one cylinder is within 15° of mid-stroke during compression (90° ABC Compression).

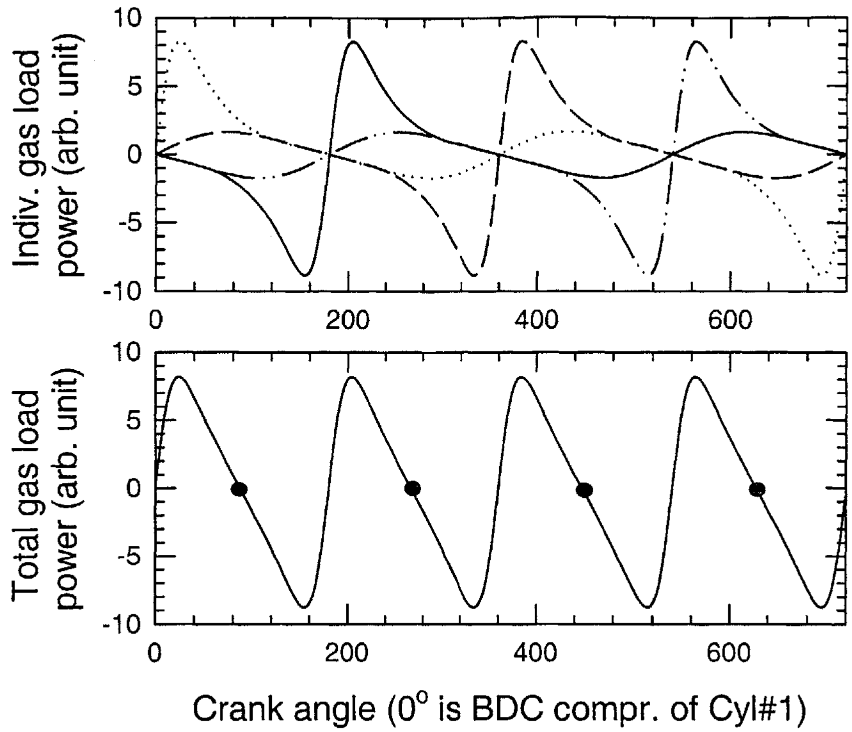


Figure 3.1: Force balance diagram

The five positions chosen for this project were as follows:

- Position A: 90° to 160°ABC Compression to represent very early closed-valve injection. Due to difficulty in the fueling code, position A was initially chosen late in the compression stroke at approximately 160° ABC Compression. This deviation from the typical stopping position was due to a programming error: the code was not able to adjust to such early starting positions, and the first cycle injection was often missed. This problem was modified in the last set of tests (0°C) for which position A was set to 90° BTC Compression. A series of tests were conducted to determine the effects of the difference in position. While the initial position was off the norm by approximately 70°, results indicate that it was early enough in the cycle that it did not make a significant difference in the results at 90°C and 20°C.

- Position B: 75° ATC Compression to represent early injection, still well before IVO.
- Position C: 100° ABC Expansion to represent injection shortly before IVO. Injection was completed before the intact valve opened.
- Position D: TC Expansion - very early open valve injection.
- Position E: 95° BBC Compression - open valve injection. For both positions D and E, injection was completed before the intake valve closed.

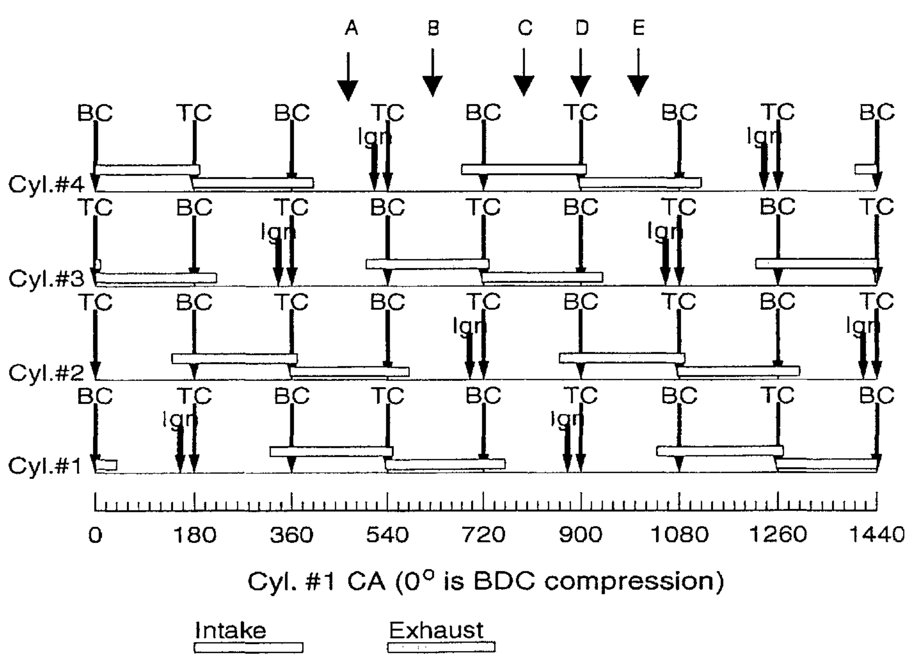


Figure 3.2: Firing diagram for Ford Zetec
Spark timing set at 10° BTC

The graph above represents the firing sequence for the Ford Zetec engine. As the arrows indicate, positions A, B, C, and E represent the typical stopping positions of a four cylinder engine. Position D was chosen and tested as a comparison to position E: both are open valve injection.

Because positions D and E are during IVO, there was concern regarding the timing of the injection relative to IVC. Figure 3.3 indicates that, at the given engine speeds, there is enough time (measured in crank angle degrees) for the fuel to be injected before IVC.

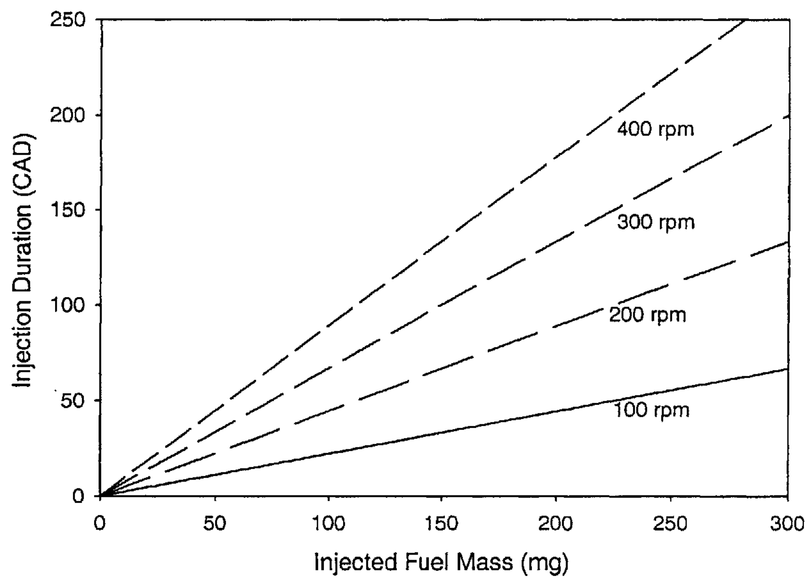


Figure 3.3: Injection duration

Chapter 4: OPERATING CONDITIONS AND TEST PROCEDURES

All experiments were conducted on a 2-liter four-cylinder PFI SI engine. As mentioned earlier, fueling for the first three cycles of cranking in cylinder 4 was controlled independently of cylinders 1, 2, and 3, which ran on the Ford ECU. The fueling strategies for the different tests are discussed below. The exhaust runner for cylinder 4 was also separated to enable accurate engine-out HC measurements to be recorded. The assumption was made that the events recorded in cylinder 4 are an accurate representation of what happens in the other cylinders, differing only in phasing.

Three different series of experiments were conducted in which the experimental preparations, operating conditions, and test procedures varied slightly.

4.1 HOT STARTS

The hot tests, in which the engine is maintained at an operating temperature of 90°C, were recorded in series. Tests were run at approximately five-minute intervals. In order to investigate the effects of residual fuel in the intake port, tests were run in which cylinder 4 was purged by turning off the injector during the last 10-15 seconds of operation. The engine was shut off when the in-cylinder HC measurements reached a low plateau (essentially zero), indicating that no more fuel was entering or remained in the cylinder. These results were compared with tests in which the engine was not purged, and the difference was negligible. During the motored cycles that the engine turned over before stopping after the controller was shut off, the in-cylinder signal also showed a minimal change in in-cylinder λ, thus indicating low residual fuel levels in the intake port.

The series of hot starts was very comprehensive. Initial tests, in which both 1st and 2nd cycle fueling was varied, were run to determine the amount of injected fuel needed in the 2nd cycle to provide a stoichiometric mixture in-cylinder following an optimized 1st cycle. This value was set at 26 mg. The bulk of the experiments consisted of maintaining constant 2nd cycle fuel pulse widths while varying the 1st cycle injection amount and timing. First cycle FPW were varied from 13 mg to 150 mg in 13 mg increments. This corresponds to 5 ms increments for the given injectors. For the majority of the tests, the spark timing was set at 10° BTC; a short set of tests was performed in which the 1st and 2nd cycle fueling was kept constant at 26mg while the spark timing was tested from 10° BTC to 15° ATC in 5° increments. These tests were only conducted for starting position E; the remainder of the tests were performed at every starting position. Results include an analysis of 1st cycle behavior, as well as an analysis of the effects of residual fuel on the 2nd and 3rd cycles.

4.2 AMBIENT STARTS

The ambient series took more time to complete due to the fact that it required cooling the engine to ambient temperature, approximately 22°C, before every test. The cranking and start-up data were collected at ambient temperature; the engine was then warmed up to operating temperature of 90°C. This was done to eliminate the effects residual fuel build-up in the intake port. After the tests were completed, the system was cooled in two stages. Initially, the engine was cooled with coolant, which was circulated through an external heat exchanger supplied with city water. Once the engine reached a low temperature, the cooling system was shut off and the engine returned to ambient temperatures, warmed by the air around it and with the help of an industrial fan.

At these temperatures, the HC sampling system did not need special preparations or modifications. To ensure this, tests were run in which the exhaust FID sampling probe was heated. The results were compared with those from tests in which the sampling probe was not heated. The results were comparable and negligible differences were observed. Methods to heat the probe will be discussed shortly, as the procedure was necessary for cold starts.

An approach similar to hot start testing was used to collect a comprehensive ambient temperature data set. Initial tests were performed to determine an appropriate 2^{nd} cycle injection to provide a stoichiometric mixture following an optimized 1^{st} cycle. This value was set at 26 mg and was kept constant for the remainder of the experiments. 1^{st} cycle FPW were varied from approximately 75 mg to 250 mg, which corresponds to 30 ms to 100 ms for the given injectors. For the experiments performed under these conditions, spark timing was set at 10° BTC. All tests were performed for the five different starting positions. However, due to the fact that starting the engine at position E resulted in misfires or severe partial burns, only a few tests were run at this position.

4.3 COLD STARTS

The final set consisted of cooling the system to 0°C and running a similar series of tests. Again, the cooling procedure was completed in two steps. The engine was first cooled using city water, which cooled the coolant as it passed through an external heat exchanger. This brought the engine temperature down to approximately 13°C. The engine and fuel, which were installed in an enclosed cold box, were then cooled with an industrial refrigerating unit to approximately 0°C. The process took approximately six to eight hours to complete.

Because of the low temperatures and the high summer humidity, some testing equipment had to be modified to ensure accurate results. Both the in-cylinder and exhaust FID sampling probes had to be heated for reasons explained in section 2.1.4. The exhaust probe was modified by first wrapping it with heat tape and covering it with insulation. The heat tape was then connected to a vari-ac, which enabled probe heating inside the cold box toward the end of the cooling cycle. It was not possible to modify the in-cylinder probe and cylinder head to allow for heating within the box. The probe was therefore heated in a lab oven prior to insertion into the engine. The results were closely examined and it was determined that this process was satisfactory to eliminate condensation in the tube and produce reliable results. Both FID heads were pre-heated outside the cold box during the last stages of the cooling process and connected to the probes immediately before collecting data.

The cold start tests were less comprehensive than both the hot starts and ambient starts. This was due to two factors: cooling the engine required some time, and large amounts of fuel were required to start the engine (otherwise, later positions would likely not start). Therefore, a comprehensive set of tests was run at starting position A, and these results were used to compare with the tests performed at other temperatures. The second cycle FPW was set based on the amount of fuel delivered by the Ford ECU; the amount was set to 70 mg and kept constant throughout the experiments. The first cycle fuel pulse width was varied from approximately 125 mg to 500 mg, or 50 ms to 200 ms. The spark timing was maintained at 10° BTC for all tests.

Chapter 5: RESULTS

The following section describes the findings of this project. It is divided into four main groups: hot starts, ambient starts, cold starts, and a comparison across the three temperatures.

5.1 HOT STARTS

5.1.1 First cycle

The first series of tests consisted of running the engine at operating temperature (90°C). The fueling strategy for the first two cycles cylinder 4 was evaluated, and the results were compared across the five starting positions. Unless otherwise noted, the spark timing was set at 10° BTC.

Figure 5.1 represents typical cranking behavior at the hot temperature for the FORD Zetec. Several interesting behaviors should be observed. An important behavior is the speed change over the first 0.7 seconds. The graph represents a start at position C, in which the piston in cylinder 4 is midway through the exhaust stroke. Following the speed trace, one can see the defined oscillations as the speed increases over the first several cycles. As cylinder 4 progresses through a first cycle, the speed trace shows a significant acceleration, in particular during combustion. These oscillations and acceleration are due to the events in the other three cylinders. Robust combustion in another cylinder significantly accelerates the engine during cranking. This phenomenon, which can also be seen in figure 5.2, shows the speed at ignition in cylinder 4 for the five different starting positions. There is a considerable difference in instantaneous speed for

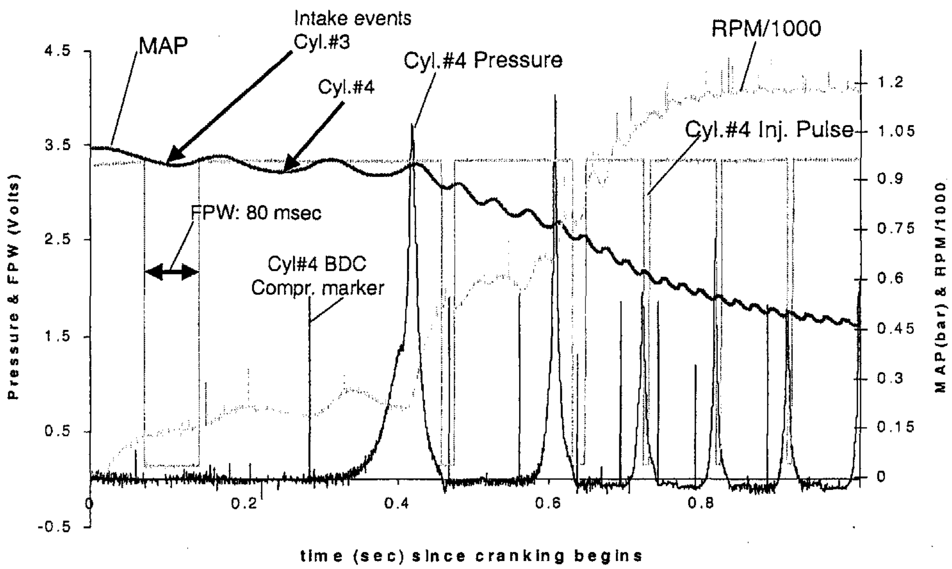


Figure 5.1: General cranking behavior

the different positions. This can be attributed to the events in the other three cylinders before cylinder 4 reaches ignition. When the engine starts at position A, cylinders 1 and 3 have a chance to fire before cylinder 4 reaches ignition. Therefore, there is considerable acceleration, and the instantaneous speed is roughly 650 rpm. When starting at position B, only cylinder 3 fires, resulting in an instantaneous speed of approximately 500 rpm. However, when the engine is started at positions C, D, and E, there is not enough time before ignition in cylinder 4 for any of the other cylinders to fire. Therefore, the engine speed is roughly that of the starter motor, about 200 rpm. The following results aim to determine and explain the effects of such differences on engine behavior.

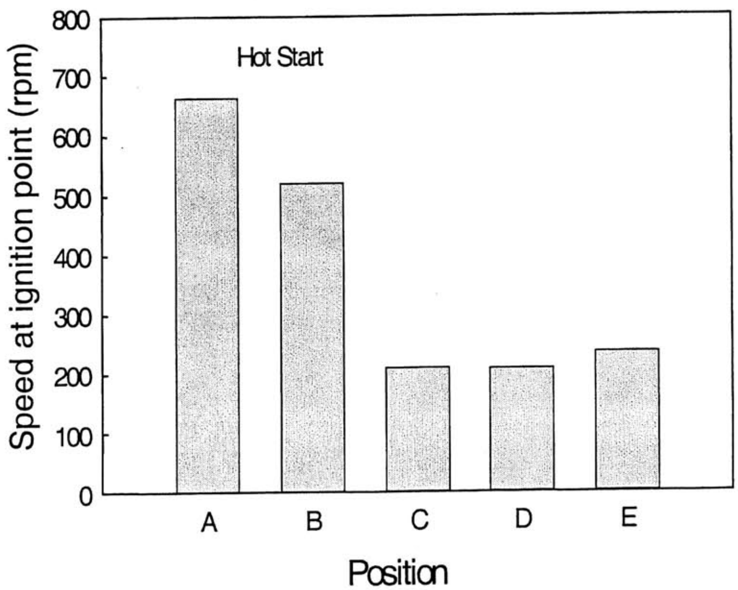


Figure 5.2: Instantaneous speed vs. engine starting position

First cycle in-cylinder λ as a function of injected fuel for the five different positions is shown in figure 5.3. The trends are similar across the different starting positions, with only slight differences between positions A and B, and positions C and D. This deviation can be attributed to the difference in induction flow velocity, due to the difference in speed during intake. Position E is considerably different than the other four positions. As is shown in figure 5.4, this is due to mixture non-uniformity caused by late valve injection. The figure, which represents in-cylinder FID measurements for positions A and E, shows a fair amount of variability in the FID measurement for position E. In-cylinder mixture non-uniformity affects the quality of the combustion.

One can also see in figure 5.3 the insensitivity of in-cylinder λ to increased fuel injections. With substantial increases in the amount of fuel injected, the in-cylinder λ initially decreases considerably, but seems to reach a saturation level around $\lambda = 0.5$.

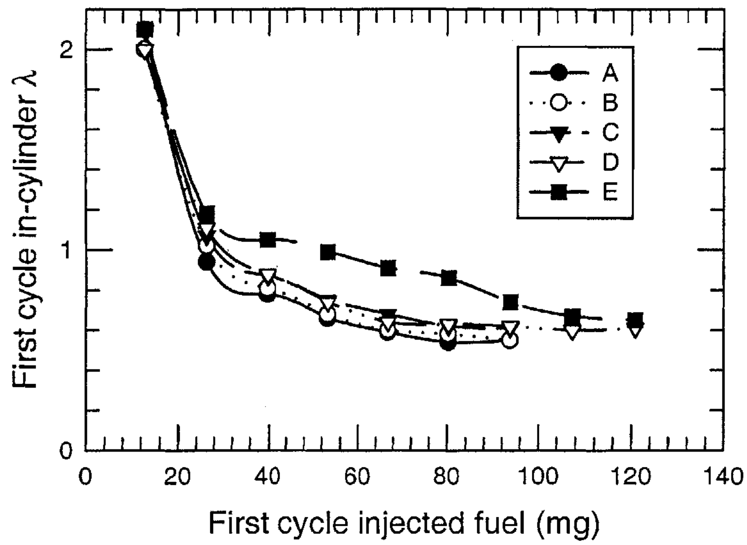


Figure 5.3: First cycle in-cylinder λ vs. injected fuel

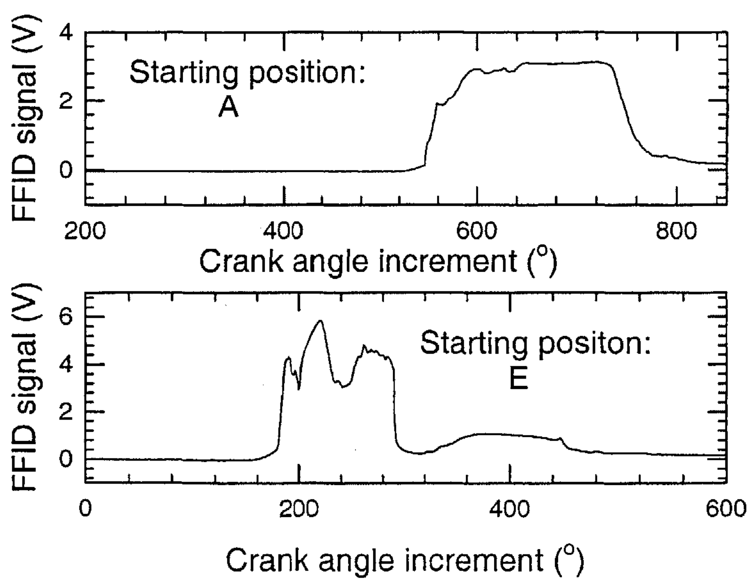


Figure 5.4: Positions A and E FID signal vs. CAD

The data presented in figure 5.5 supports this trend. The figure shows the fraction of injected mass in the charge, in vaporized form as measured by the in-cylinder FID, as a function of injected mass. The fraction decreases substantially as more fuel is injected, from about 90% to 40%, with a sensitivity of -0.1 for every 13mg of additional fuel injected, indicating a diminishing return from the increased amount of fuel injected. Once again, the question lies in where the remaining fuel is deposited[1].

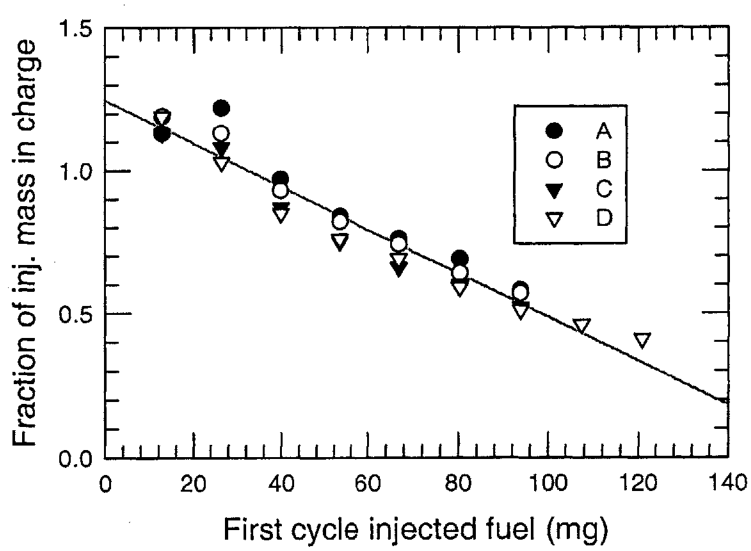


Figure 5.5: Fraction of mass in charge vs. injected mass

IMEPg behavior relative to injected mass and in-cylinder λ for the five different starting positions is shown in figures 5.6 and 5.7. Figure 5.7 shows that, for the different starting positions, an in-cylinder λ of approximately 1.1 is needed to obtain a good combustible mixture (high IMEPg) in-cylinder. However, it takes larger injections to reach this threshold at later starting positions than at earlier positions. Figure 5.7 also shows that IMEPg is fairly insensitive to in-cylinder λ in the stoichiometric to rich zone

[1] The discrepancy in the fraction of fuel for small injections is due to measurement error

(in-cylinder λ of 1.1 to 0.65). This phenomenon will be explained later in this section. As shown in figure 5.3, similarities are observed in overall trends in IMEPg behavior vs. both injected mass and in-cylinder λ, across the different starting positions. However, there is a substantial difference in maximum IMEPg between positions A and B, C and D, and E. As explained earlier, data from position E is not taken into account due to mixture non-uniformity and uncertainty in FID measurements. However, the discrepancy between A and B, and C and D needs to be expanded upon.

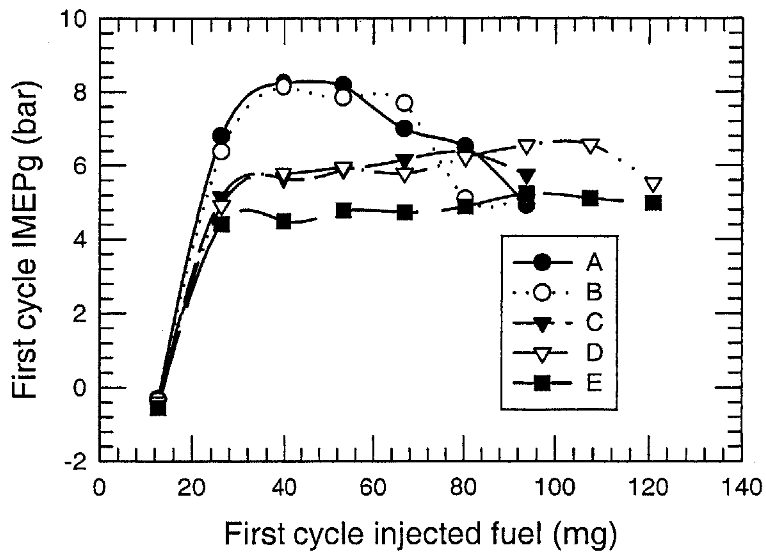
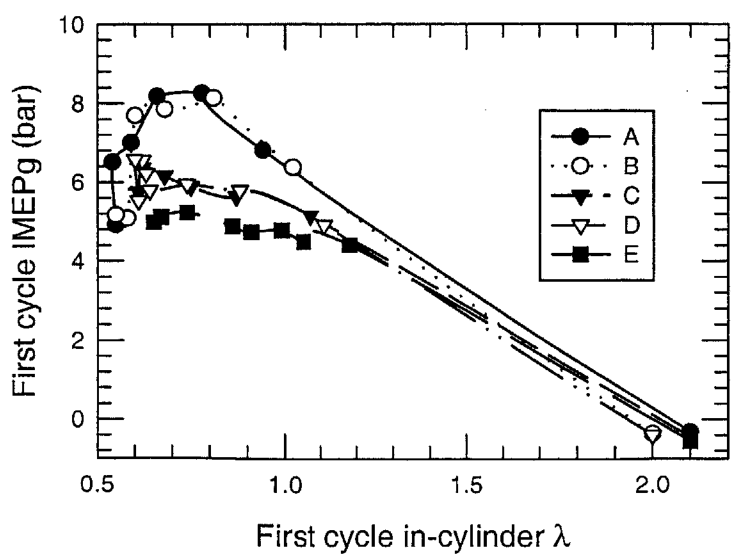


Figure 5.6: IMEPg vs. injected mass

Figure 5.7: IMEPg vs. in-cylinder λ



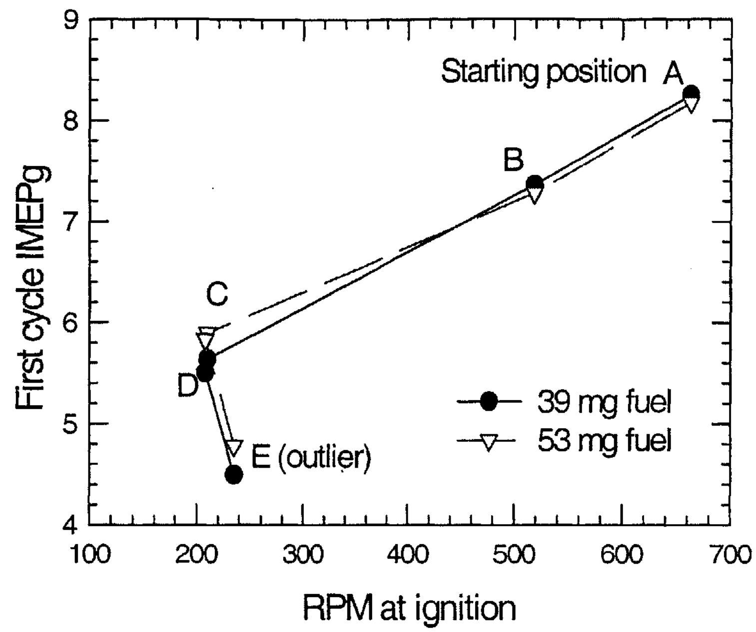


Figure 5.8: RPM effect on IMEPg at different positions

Figure 5.8 supports this trend by showing the relationship between speed and IMEPg at the different positions. There is a clear correlation between starting position, and therefore speed, and IMEPg. The following plots expand on this and explain it in more detail.

In order to gain a better understanding of this discrepancy, initial tests were conducted to determine whether the nominal spark timing is appropriate during the first few cycles of operation. Figure 5.9 represents the results of this experiment, in which the spark timing was swept from 10° BTC to 15° ATC in 5° increments at a given starting position and fueling strategy (optimized for stoichiometric at nominal spark). The instantaneous speed during these tests was 200 rpm. As the graph indicates, the nominal spark timing of 10° BTC is too advanced for the first cycle of operation. The data indicates that retarding the spark has the effect increasing engine output, most likely through improved combustion phasing. However, the IMEPg levels are still too low (63%) relative to positions A and B, in which the instantaneous speed is over 500 rpm. Therefore, more tests were needed to explain this discrepancy.

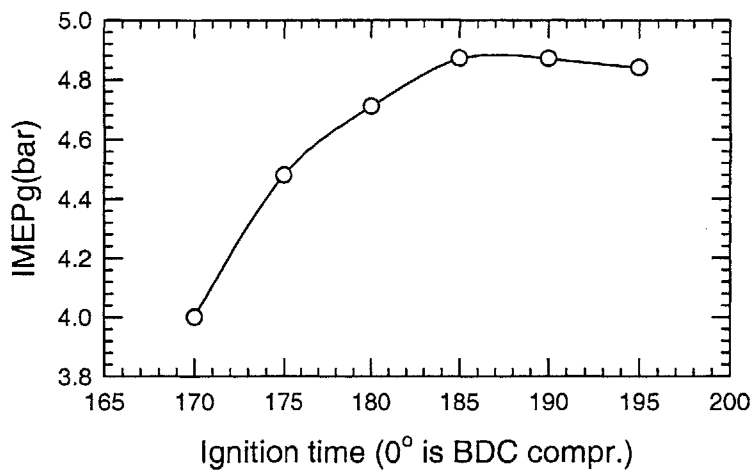


Figure 5.9: Spark sweep

The next step involved performing a heat release code, from which the results are represented in figure 5.10 and 5.11. These represent heat release schedules as a function of crank angle degree for positions A and C, respectively. This is a comparison of tests in which the fueling and spark timing were the same; the only variable was the starting position, which affects the instantaneous speed at ignition. The graphs show the pressure phasing within the cylinder, as well as the energy released, heat transfer to the wall, and indicated work. In comparing the two figures, one can see that the pressure trace in case C is more advanced and acute than it is in case A. In other words, at the lower speeds, the heat release occurs in a much smaller fraction of the cycle (fewer CAD) than at the higher speeds. Thus, combustion occurs in the much earlier phase of the expansion process, resulting in a higher pressure and a more favorable work transfer schedule.

Furthermore, there is a significant difference in heat transfer between the two cases. At about 200 rpm, there is much more time for heat transfer to the wall and combustion occurs in a more confined space; therefore, there is a greater opportunity for losses than when the engine is rotating at about 650 rpm. This results in more work being available at higher speeds.

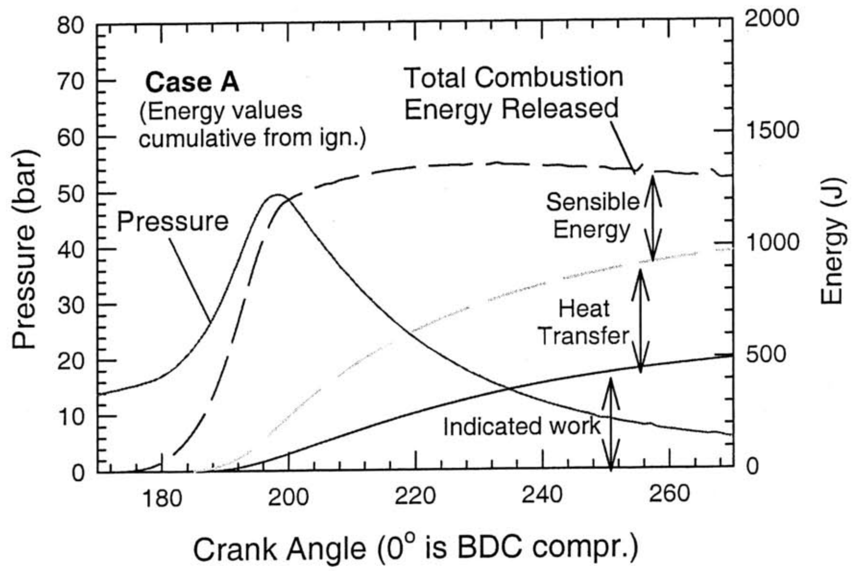


Figure 5.10: Heat release code for case A

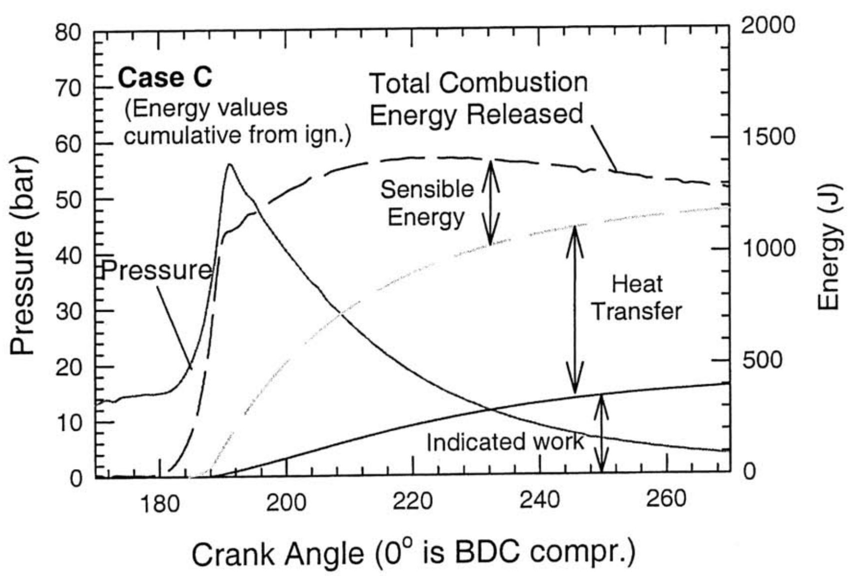


Figure 5.11: Heat release code for case C

From the graphs, one can obtain an explanation for the discrepancy in IMEPg between positions A and B, and positions C and D. Higher engine speed retards combustion, and this affects IMEPg in two ways. First, it creates a more favorable heat release schedule, relative to piston motion: faster engines result in heat release occurring over a larger crankshaft rotation, therefore creating a more favorable pressure distribution. In addition, since combustion occurs later in the expansion stroke of an engine that is rotating faster, heat transfer to the walls is decreased: for engines that are moving faster, there is less time, relative to piston motion, for heat transfer to the walls, resulting in lower overall heat transfer.

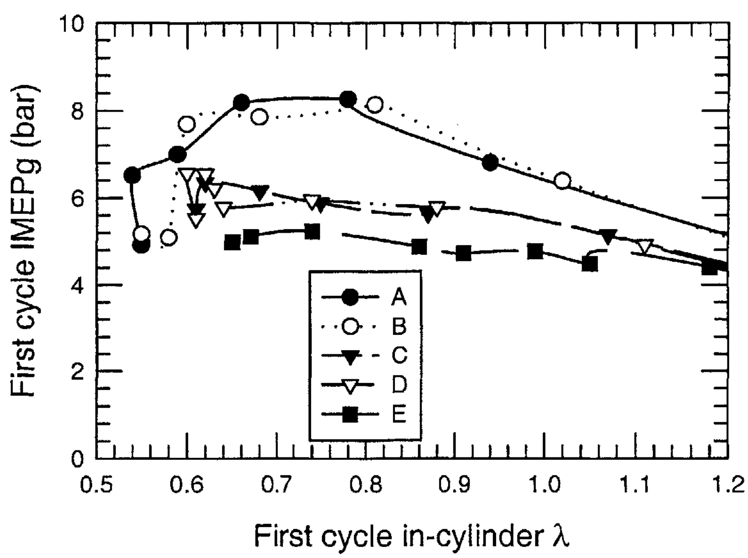


Figure 5.12: 1st cycle IMEPg vs. in-cylinder λ

As seen in figure 5.7, figure 5.12 shows first cycle IMEPg insensitivity to in-cylinder λ. In the range of λ ~ 1.1 to λ ~ 0.65, especially in the rich regions (and at lower speeds), IMEPg does not vary significantly. The explanation can be found in figure 5.13.

As more fuel is injected and the mixture becomes richer, the combustion efficiency decreases. However, since more fuel is burned, the energy released only decreases modestly. In addition, the moderate loss is compensated for by a slower burn, which improves the heat release phasing. Therefore, while over-fueling a cycle has negative consequences on other engine characteristics, there is a large margin of acceptable in-cylinder λ to obtain a strong IMEPg.

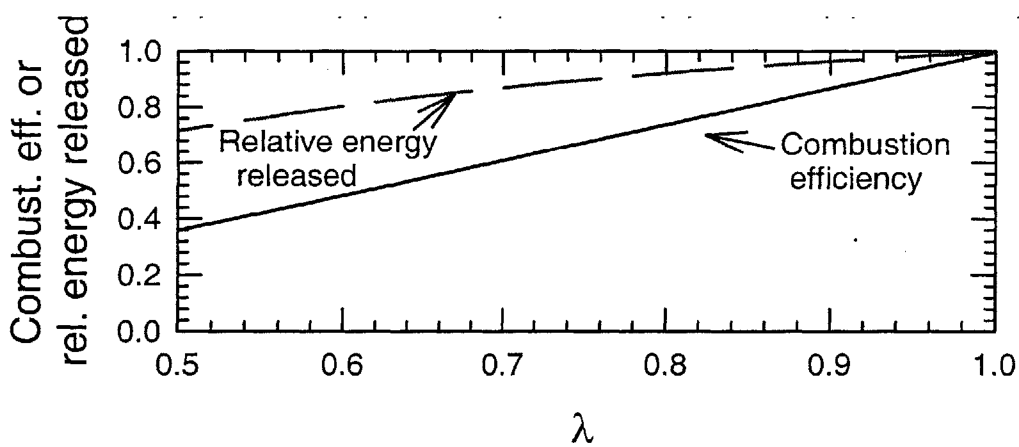


Figure 5.13: Combustion efficiency and energy released vs. in-cylinder λ

EOHC emissions behavior is represented in figure 5.14. Once again, due to mixture non-uniformity in the open valve injection case and uncertainty in the FID signal, data for starting position E are shown but are not considered in drawing conclusions. Similar trends are observed across the remaining starting positions. Lean mixtures, for which the data is not included in this graph, result in high HC levels due to misfires and partial burns. Similarly, as the cylinder is over-fueled and the mixture becomes increasingly rich, HC levels rise sharply due to partial burns. In the middle phase, for stoichiometric to rich mixtures ($1.1 > \lambda > 0.65$), HC levels remain fairly constant and are relatively insensitive to λ. Since IMEPg and HC levels remain relatively insensitive to in-cylinder λ in the mid-range, one would assume that there is some flexibility in optimized start-up strategies. This is not the case, however, for reasons illustrated in figure 5.15, which shows CO, CO_2, and H_2 levels for increasingly rich mixtures. While an in-cylinder λ of 1.1 to 0.65 was seemingly appropriate for low HC emissions, one can see from the graph that these rich mixtures result in high levels CO emissions, which get absorbed in the catalyst and render it ineffective for HC reduction for a substantial time. Therefore, determining an optimized strategy is more complex than it seems.

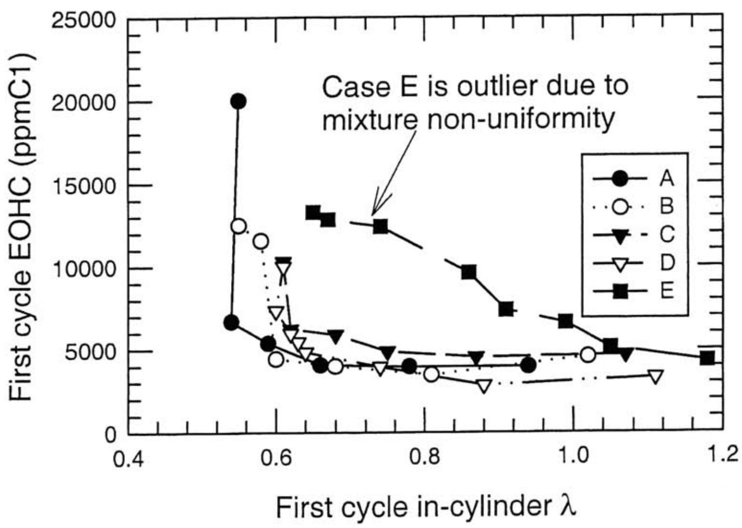


Figure 5.14: EOHC vs. in-cylinder λ

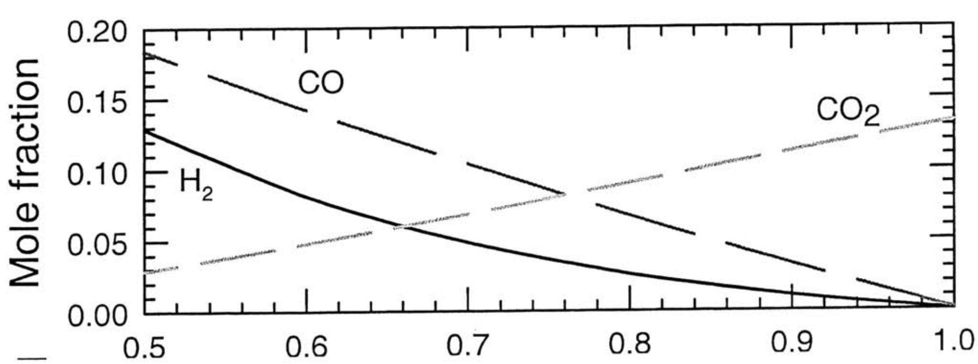


Figure 5.15: Mole fraction (CO, CO_2, HC) vs. in-cylinder λ

5.1.2 Hot start: subsequent cycle behavior

The following results focus on subsequent cycle behavior for hot start experiments. Instantaneous speed is not of consideration in this discussion, as the engine reached approximately 800 rpm for all cases. MAP for the second cycle ranges from 0.7 to 0.9 bar, depending on the starting position.

As described earlier, the fueling strategy for the second cycle was also specified by the operator. For the results presented below, the 2nd cycle fueling was set at 26 mg, which supplied a roughly stoichiometric mixture following an optimized 1st cycle at the typical MAP levels. For the third and subsequent cycles, the fueling was set to provide a stoichiometric charge, based on a speed-density calibration. For the first cycle, injection began 28° after start. For the second and subsequent cycles, start of injection was set to BDC expansion.

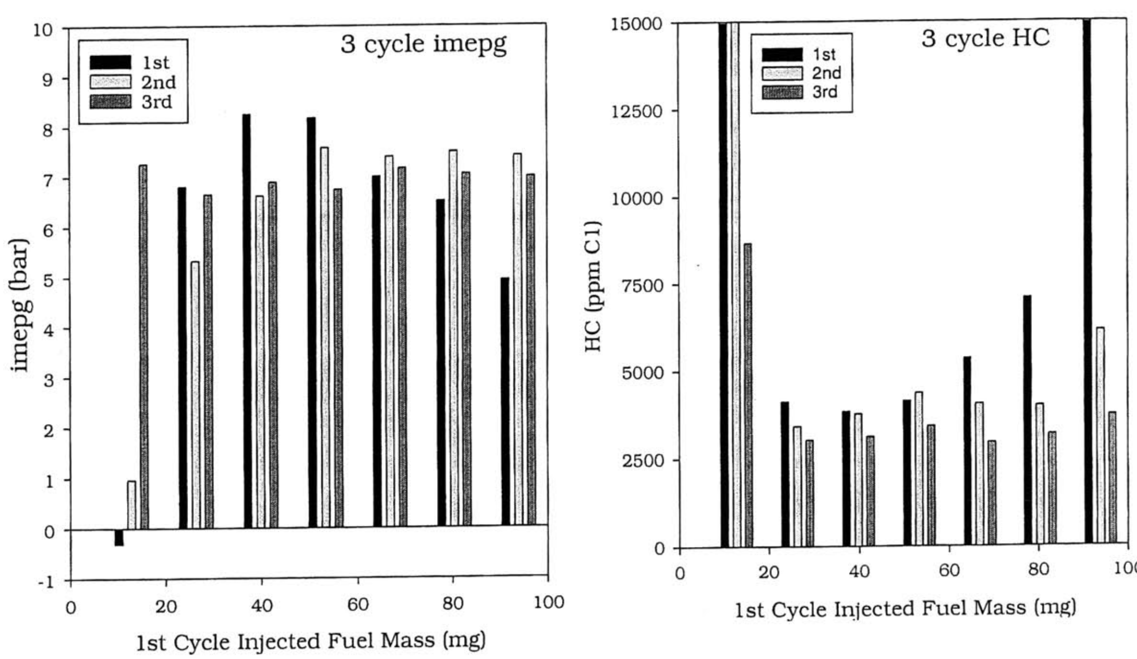


Figure 5.16: Residual fuel effects on IMEPg and EOHC

Figure 5.16 shows the effect of 1st cycle residual fuel on 2nd and 3rd cycle IMEPg and EOHC. There is little evidence, aside from position A at 13 mg of fuel, of strong residual effects on 2nd cycle IMEPg, and even less on 3rd cycle behavior. Strong IMEPg were obtained in all cases. The only case in which there is a strong effect is that in which the 1st cycle did not fire due to an under-fueled mixture. Similar trends can be observed in EOHC behavior. Over-fueling the 1st cycle has little effect on the 2nd cycle, and even less on the 3rd. There is a slight increase in HC with increasing 1st cycle injections, but it is relatively negligible relative to the 1st cycle increase in HC.

Figure 5.17 helps to gain a better understanding of the effects of 1st cycle residual fuel. It represents the change in 2nd cycle in-cylinder λ as a function of 1st cycle residual fuel. Across the five different starting positions, there is a slight effect from residual fuel: in-cylinder λ drops with a sensitivity of 0.1 λ for every 22 mg increase in fuel for the 2nd cycle. While this sensitivity is less than for the first cycle, it is significant enough at higher fuel injection amounts that it needs to be taken into account when calibrating 2nd cycle FPW.

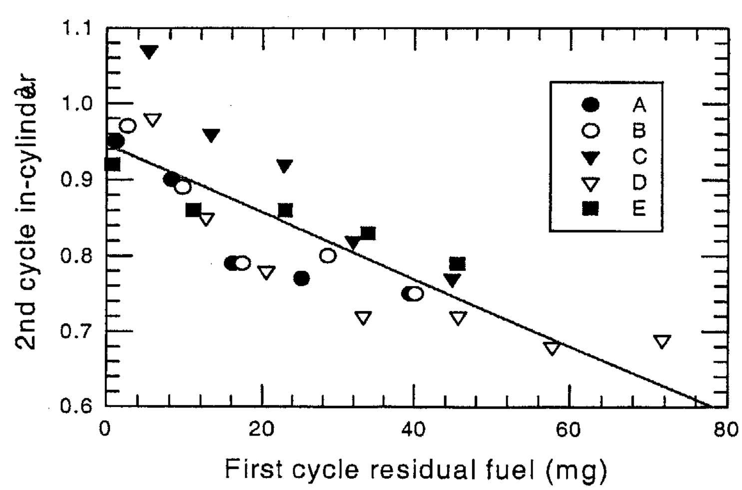


Figure 5.17: 2nd cycle in-cylinder λ vs. 1st cycle residual fuel

Figures 5.18 and 5.19 show 2^{nd} cycle IMEPg and EOHC behavior with decreasing in-cylinder λ. Both graphs indicate that 2^{nd} cycle behavior is fairly close to normal engine behavior under these conditions. IMEPg values range from 6 to 8, with the poorer performance resulting from lean mixtures, in which the amount of fuel is too low to obtain a uniformly combustible mixture. EOHC emissions are roughly 2000 to 3000 ppm C1 for stoichiometric mixtures. This is similar to normal operating conditions. In the range of in-cylinder λ from 1.0 to 0.65, HC levels seem to rise from about 2000 to 6000 ppm C1, as was noted for the 1^{st} cycle. Therefore, one would expect similar HC mechanisms for the two cases (partial burns due to over-fueling and mixtures that are too rich to burn completely).

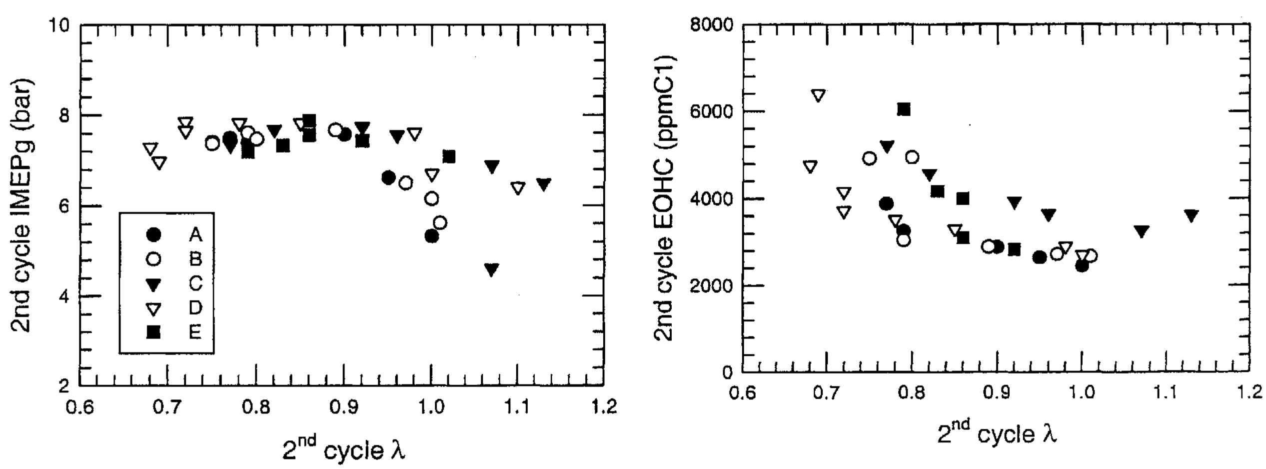


Figure 5.18 and 5.19: 2^{nd} cycle IMEPg and EOHC vs. 2^{nd} cycle in-cylinder λ

As was mentioned earlier, there is little effect of 1^{st} or 2^{nd} cycle residual fuel on 3^{rd} cycle behavior. Figure 5.20 shows 3^{rd} cycle IMEPg, in-cylinder λ, and EOHC as a function of 1^{st} cycle residual fuel. IMEPg values range from 6 to 8, with lower values resulting from lean mixtures. HC levels are within 2000 to 4000 ppm C1, which is very close to normal operating conditions. Once again, HC levels rise slightly with over-fueling, as was seen in both 1st and 2nd cycle results, and can be explained by the same HC mechanisms. In-cylinder λ decreases slightly with increasing residual fuel, but the sensitivity is much lower than for either 1st or 2nd cycles: a decrease of 0.1 λ for every 42 mg of increased fuel in the 3rd cycle.

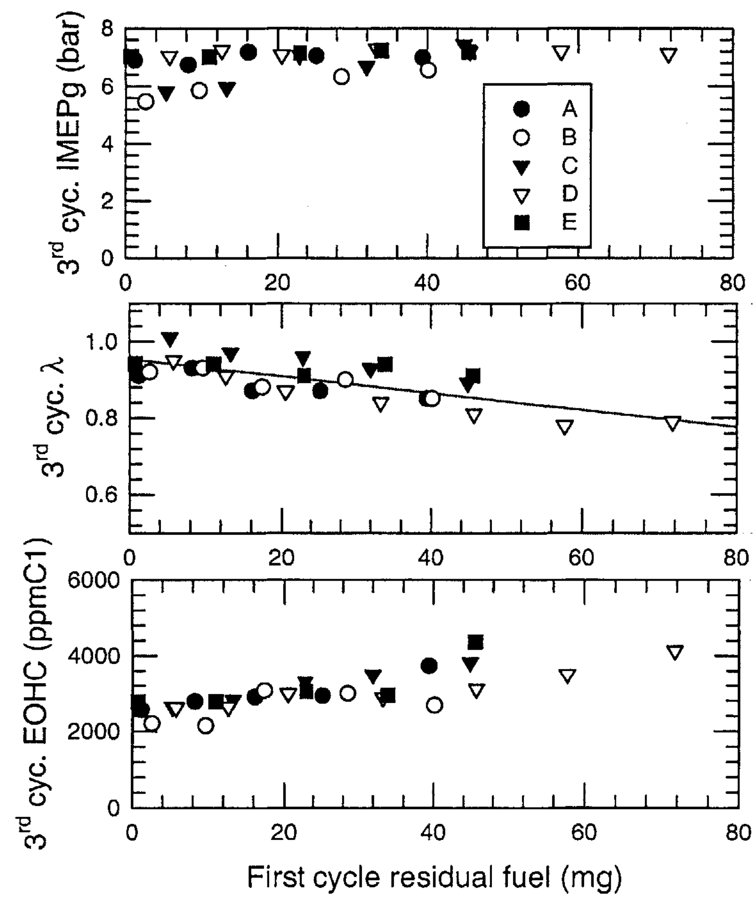


Figure 5.20: 3rd cycle IMEPg, in-cylinder λ, and EOHC vs. 1st cycle residual fuel

5.2 AMBIENT STARTS

A similar set of tests was conducted at ambient temperatures (20°C). Tests consisted of varying the 1st cycle fuel injection and comparing the results across the different starting temperatures. The effects of 1st cycle residual fuel on subsequent cycles were also evaluated by maintaining a constant 2nd cycle injection, set to provide a stoichiometric charge under optimal conditions (stoichiometric 1st cycle charge).

Unlike the hot starts, the Ford ECU did not consistently provide enough enrichment to the other three cylinders under these conditions. Therefore, the instantaneous speed at ignition in cylinder 4 varied greatly from start to start. In order to obtain a comparable data set, the only tests used were those in which cylinder 4 was the first to fire; the instantaneous speed was consequently constantly 200 rpm. In addition, at ambient temperatures, the engine did not fire when started at position E (late open valve injection) regardless of the amount of fuel injected. This is attributed to either lack of enough delivered fuel to provide a combustible mixture, or more plausibly, fuel wetting of the spark plug.

These first graphs, figures 5.21 and 5.22, represent 1st cycle IMEPg behavior as a function of both injected fuel and in-cylinder λ. Trends are similar to those noted for hot starts: low IMEPg levels for small injections or lean mixtures, with increasing values as in-cylinder λ decreases with larger injections. Unlike in the hot starts, however, IMEPg values are fairly close to being the same across the starting positions. This can be attributed to the fact that the effects of speed were not taken into account, as all tests were run at 200 rpm. There is not a noticeable effect on IMEPg from over-fueling the 1st cycle. Under these conditions, the firing threshold is around $\lambda \sim 1.1$; it takes approximately 80 mg to reach this threshold.

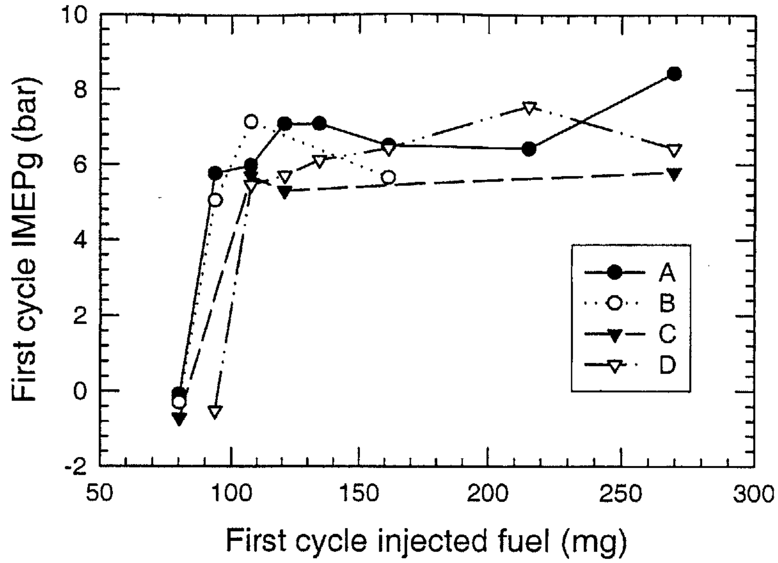


Figure 5.21: 1st cycle IMEPg vs. injected fuel

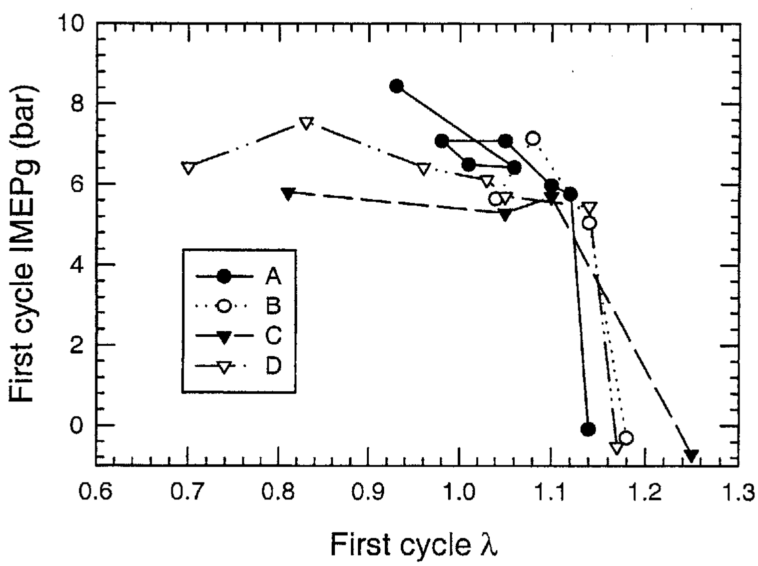


Figure 5.22: 1st cycle IMEPg vs. in-cylinder λ

The IMEPg results support the in-cylinder λ results, as can be seen in figure 5.23. The graph shows the decrease in in-cylinder λ with increasing injected fuel. The mixture becomes richer with increased injection, but seems to reach a plateau around $\lambda \sim 0.9$. One would expect the in-cylinder λ to decrease more than it does. This behavior is supported by data shown in figure 5.24, which represents the fraction of fuel injected that enters the cylinder in vaporized form. The fraction, which is substantially lower than for hot starts, decreases with increasingly large injections.

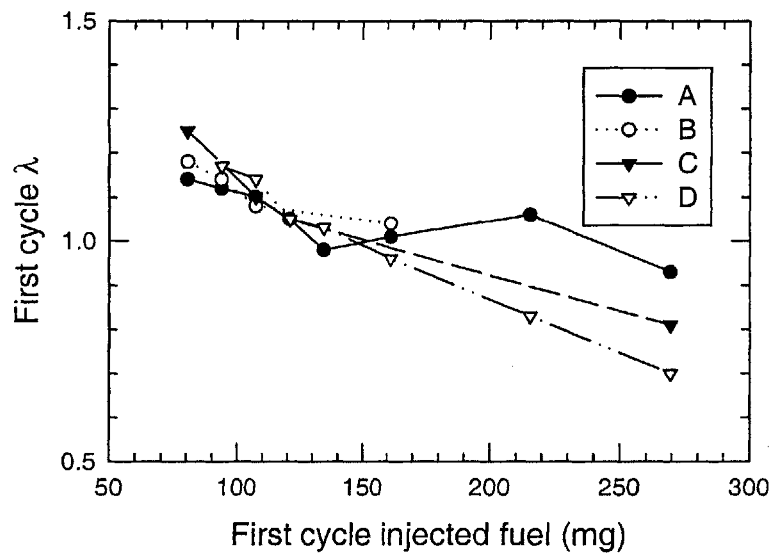


Figure 5.23: 1st cycle in-cylinder λ vs. injected fuel

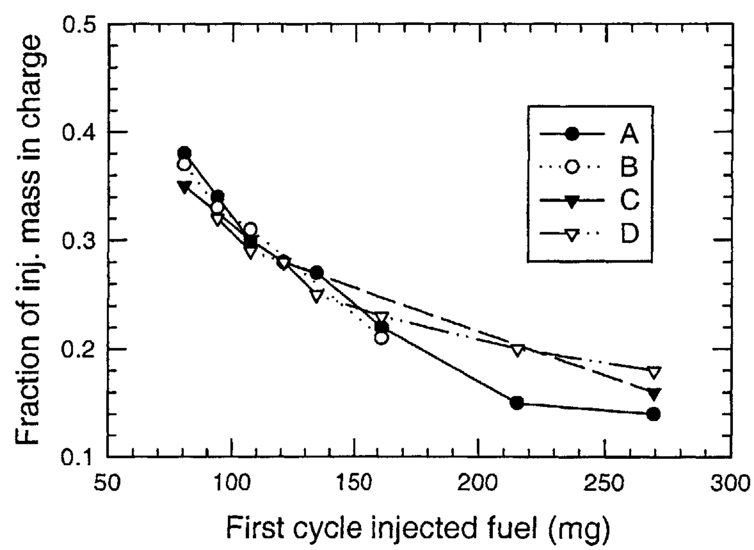


Figure 5.24: Fraction of fuel in charge vs. injected fuel

Figure 5.25 shows 1st cycle EOHC behavior as a function of in-cylinder λ. For lean mixtures in which the charge is not combustible, misfires occur, resulting in high HC levels (greater than 10,000 ppm C1). Once the firing threshold has been reached, and for slightly richer mixtures, HC levels decrease significantly. For most cases, HC levels remain relatively low (~5000 ppm) with increasing fuel injections. However, for the case in which the engine is started at position D, HC levels rise considerably with richer mixtures. This is attributed to open valve injection, which causes non-uniform mixtures and substantial liquid fueling entering the cylinder, resulting in potential partial burns.

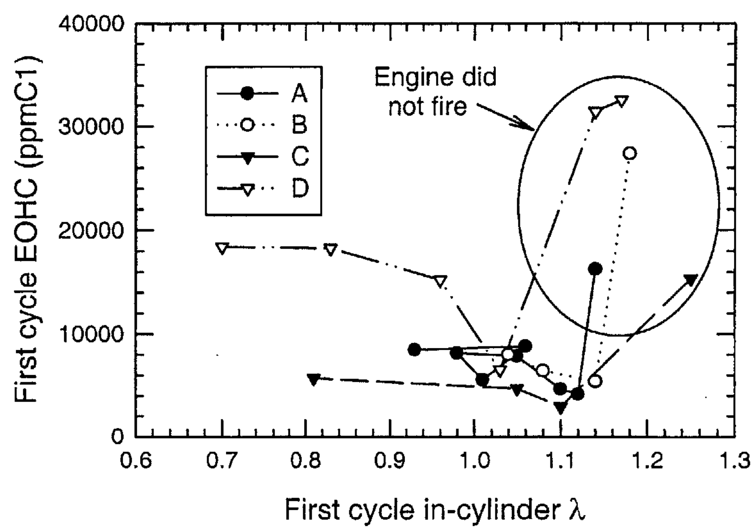


Figure 5.25: 1st cycle EOHC vs. in-cylinder λ

As in the hot start cases, results were evaluated to elucidate the effect of 1st cycle residual fuel on 2nd cycle behavior. Figure 5.26 shows the relationship between 1st cycle residual fuel and 2nd cycle in-cylinder λ. As for the hot starts, there is a decreasing effect on in-cylinder λ with increasing fuel. This effect is significantly smaller than for hot starts: a decreasing sensitivity of 0.1 for a 74 mg increase in fuel. However, this effect is still important, as significantly over-fueling the first cycle can have detrimental effects on 2nd cycle characteristics, as seen in previous graphs.

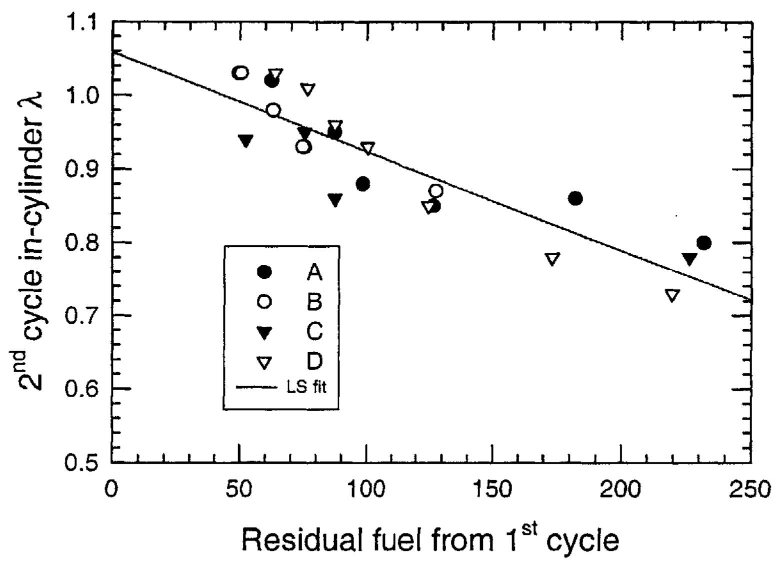


Figure 5.26: 2nd cycle in-cylinder λ vs. 1st cycle residual fuel

Due to the variation in firing of the remaining cylinders (1, 2, and 3), 2nd cycle IMEPg data was found to have significant scatter. It is therefore not presented in this paper.

5.3 COLD STARTS

As was explained earlier, these tests were conducted at 0°C. The engine was cooled in two phases: the coolant was initially run through an external heat exchanger in which it interacted with city water. This first stage cooled the engine to approximately 20°C. The whole system was then cooled with an industrial refrigeration unit in an enclosed test box including the fuel. Both the in-cylinder and exhaust probes were heated to eliminate the problems associated with cold and liquid samples.

The Ford production ECU, which controlled the spark timing and fueling strategy for cylinders 1, 2, and 3, did not provide sufficient enrichment to those cylinders in the first cycle of operation. Therefore, acceleration of the engine before and during combustion in cylinder 4 is not of consideration, and the data presented represents tests in which the engine was running at 200 rpm, approximately the starter motor speed, at ignition.

Due to time and ambient temperature limitations, cold tests were only run at starting position A when cylinder 4 is midway through the compression stroke.

Figure 5.27 represents the first cycle injected fuel vs. the first cycle in-cylinder λ. As would be expected, in-cylinder λ decreases with increasingly large injections. However, while the mixture is initially significantly enriched, the in-cylinder λ levels off around $\lambda \sim 0.9$.

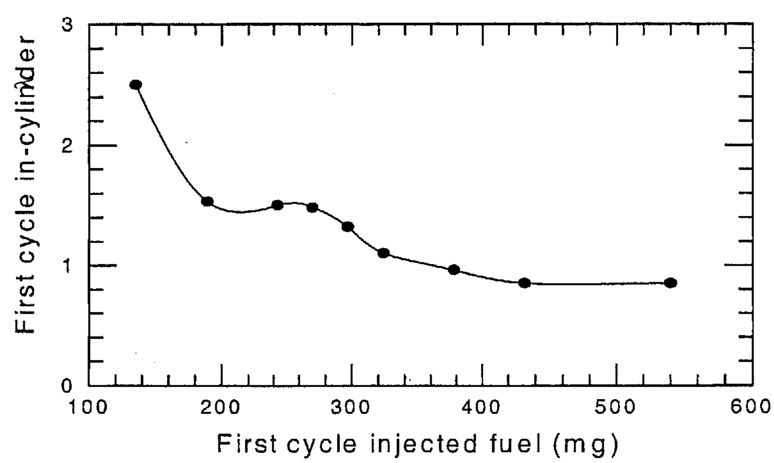


Figure 5.27: 1st cycle injected fuel vs. in-cylinder λ

This notion is supported by the data presented in figure 5.28, which shows the fraction of the injected mass that makes it into the charge, in vaporized form, as measured by the FID, as a function of injected fuel. Under these cold starting conditions, less than 10% of the fuel actually makes it into the cylinder in vaporized form. As more fuel is injected, a smaller fraction makes it into the cylinder: a 0.01 decrease in fraction for every 160 mg increase in fuel injected. This results in a diminishing return: the more fuel that is injected, the smaller the return from the injection. The negative slope of this line follows the results presented above (in-cylinder λ saturation).

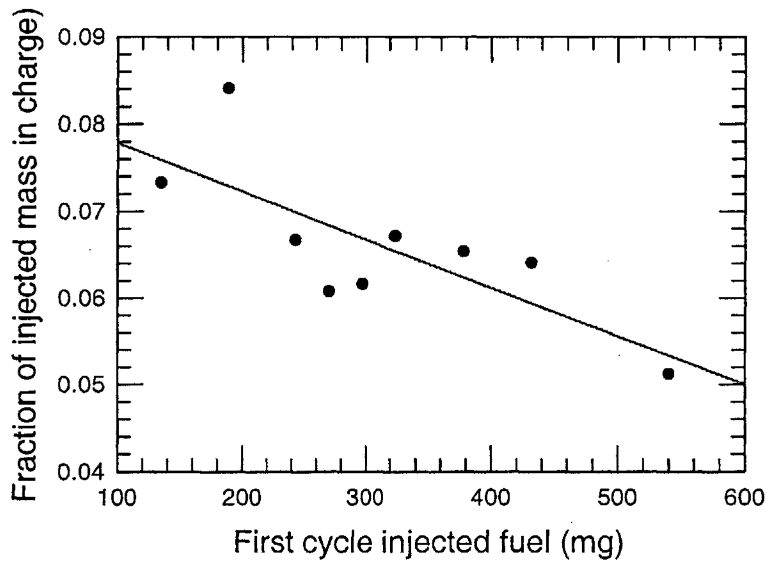


Figure 5.28: Fraction of mass in charge vs. 1st cycle injected fuel

Figures 5.29 and 5.30 represent the IMEPg behavior under these cold starting conditions. Similar trends can be observed under these conditions as observed for hot and ambient starts. For lean mixtures, in which fuel injections are too small to produce a combustible mixture, IMEPg values are negative. As in-cylinder λ drops with increasing fuel injections, IMEPg levels increase significantly, indicating strong burns in-cylinder. The threshold for achieving such burns is at approximately $\lambda = 1.1$. For 0°C starts, this mixture condition is reached with an injection of 326 mg.

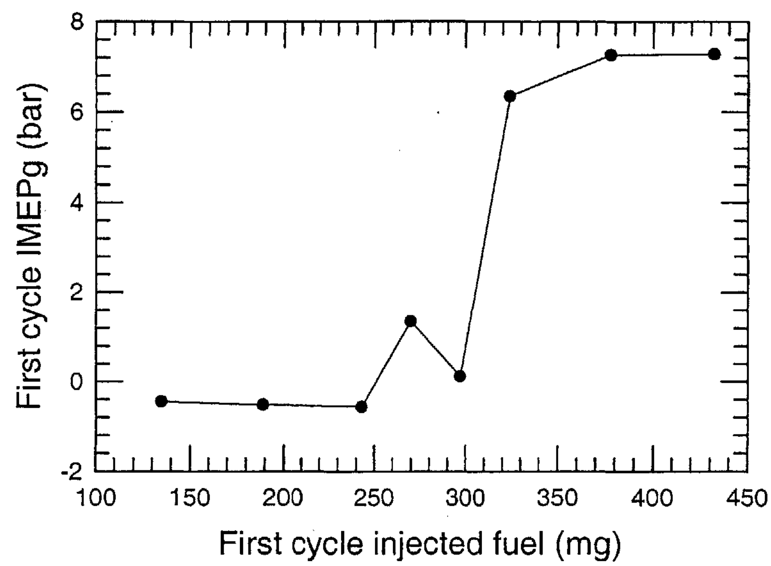


Figure 5.29: 1st cycle IMEPg vs. injected fuel

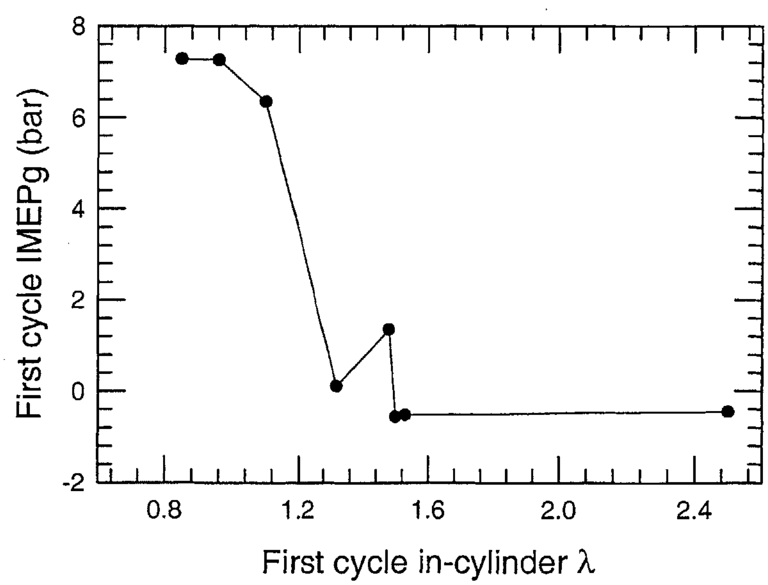


Figure 5.30: 1st cycle IMEPg vs. in-cylinder λ

HC emissions behavior is represented in Figure 5.31. Once again, the trends seen below are similar to those observed under hot and ambient starting conditions. For approximately stoichiometric mixtures, HC levels are minimized around 10,000 ppm C1. For lean mixtures, HC levels increase significantly and reach levels of approximately 60,000 ppm C1. The graph also shows in-cylinder HC levels as a function of in-cylinder λ. For extremely lean mixtures, which result in misfires and partial burns, most of the fuel injected exits the cylinder as unburned HC. As the mixture is enriched, resulting in partial burns, a smaller fraction of the fuel exits as unburned HC. Once the firing threshold has been crossed, and for richer mixtures, less than 10% of the in-cylinder mixture exits as unburned HC. At the cold temperatures, EOHC levels for cycles which produced a strong burn are higher relative to ambient and hot results as well as normal operating levels.

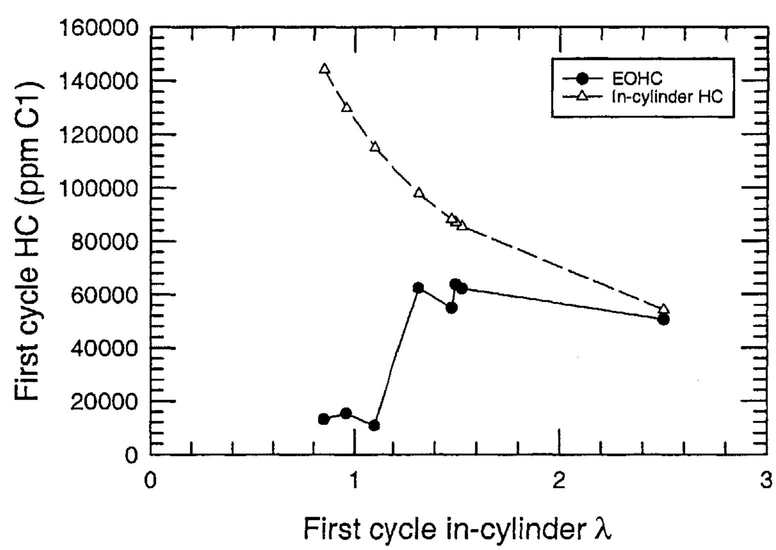


Figure 5.31: 1st cycle in-cylinder λ vs. EOHC

5.4 COMPARISON BETWEEN 90°C, 20°C, AND 0°C STARTS

Once the experiments were completed, the results from the three different starting temperatures were compared. The graphs represent data from engines running at 200 rpm at ignition in cylinder 4 with constant spark timing set at 10° BTC.

The figure below (figure 5.32) represents the fraction of injected fuel that makes it into the cylinder in vaporized form (as measured by the Fast FID) as a function of first cycle injected fuel. As the graph shows, the fraction decreases with increasing injected fuel and with decreasing starting temperature. The fuel transport sensitivity decreases with decreasing temperature as well. In other words, the decreasing slope under hot conditions is much steeper than under ambient conditions (-0.1λ/26 mg vs. -0.1λ/74 mg). The ambient condition slope is also much steeper than under cold conditions (-0.1λ/74 mg vs. -0.01λ/160 mg). Under optimal hot starts, more than 90% of the fuel injected makes it into the charge as vaporized fuel. As the graph insert indicates, less than 10% of the fuel injected makes it into the cylinder under cold conditions.

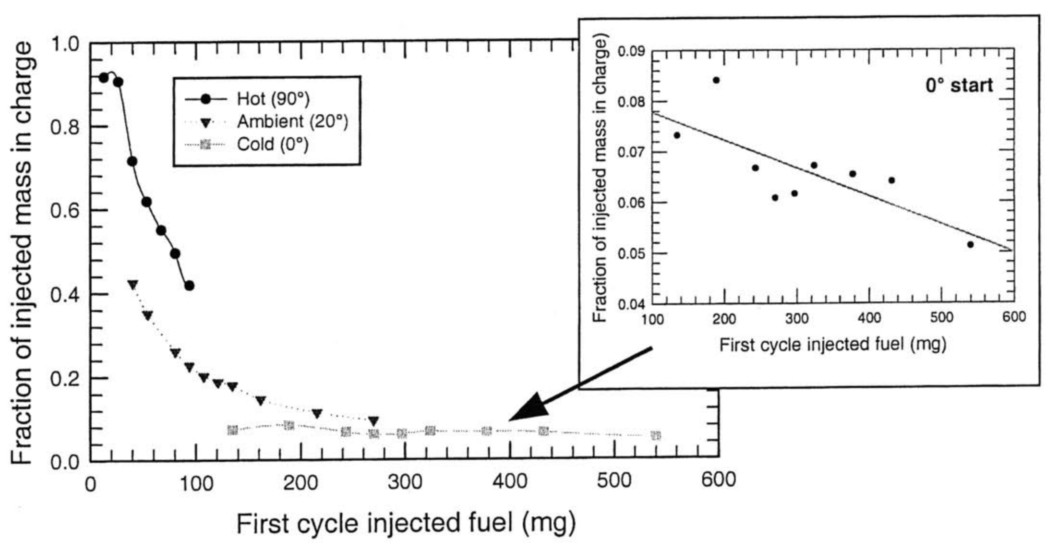


Figure 5.32: Fraction of mass in charge vs. 1st cycle injected fuel

Interesting trends can also be observed in figure 5.33, which shows in-cylinder λ as a function of 1st cycle injected fuel in the first cycle. Tests run at the three different starting temperatures reveal similar trends: small injections (relative to different temperatures) result in extremely lean mixtures. As more fuel is injected, in-cylinder λ decreases and eventually levels off with large amounts of fuel injected. In-cylinder λ seems to reach a saturation point, which differs at the different temperatures: $\lambda \sim 0.5$ for hot starts, and $\lambda \sim 0.9$ for ambient and cold starts.

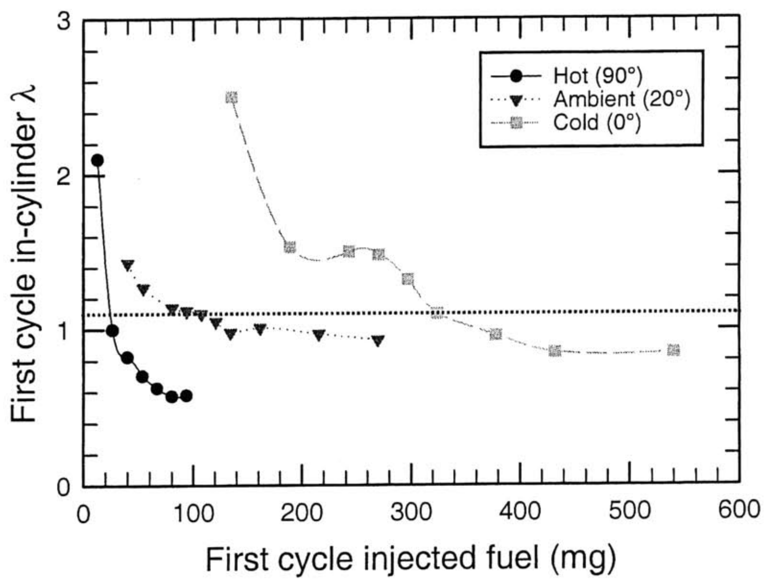


Figure 5.33: 1st cycle injected fuel vs. in-cylinder λ

IMEPg behavior is represented in the graphs below as a function of injected fuel and in-cylinder λ. As figure 5.34 shows, lower starting temperatures require larger first cycle injections to produce a robust combustion. Interestingly, at the same λ, lower starting temperatures have higher IMEPg (7.5 vs. 6.35 and 5.5). This can be attributed to the different air densities at different temperatures. More fuel can be burned with the air in-cylinder under cold conditions than under hot conditions. The graph also shows how similar the trends are across the different temperatures. With lean mixtures, IMEPg values are negative. With increasingly rich mixtures, IMEPg values increase significantly as a result of robust combustion during the cycle. The firing threshold, $\lambda \sim 1.1$, is consistent across the temperatures.

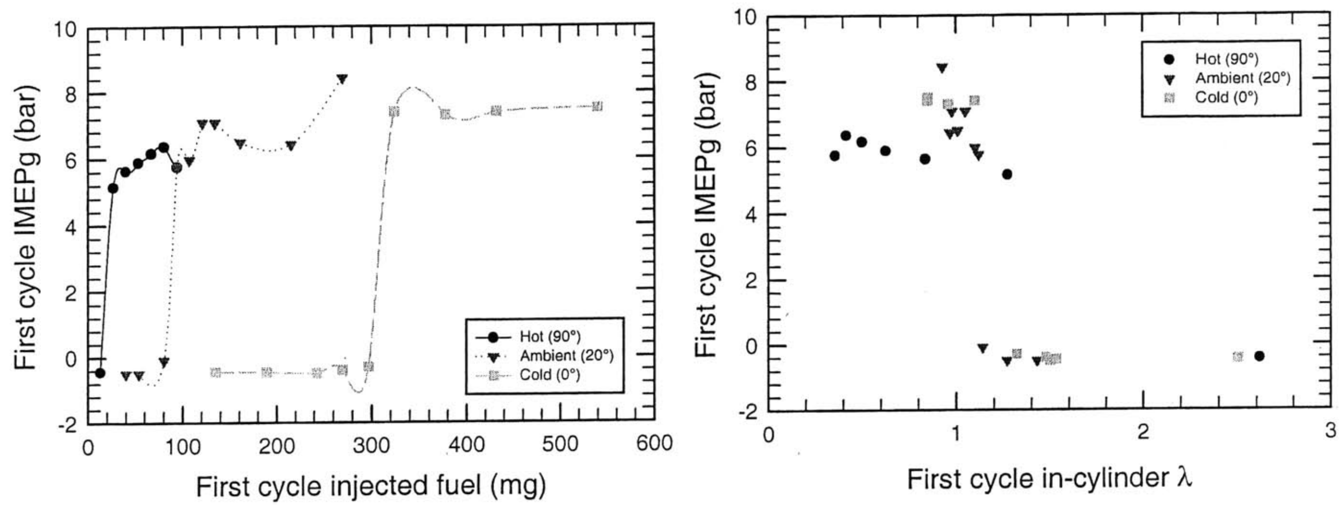


Figure 5.34: IMEPg vs. 1st cycle injected fuel and in-cylinder λ

Finally, EOHC emissions behavior is displayed in figures 5.35 and 5.36. Once again, similar trends can be observed across the different temperatures: for lean mixtures, EOHC levels are extremely high as there is not enough fuel to provide a combustible mixture. Once the firing threshold is reached, HC levels drop significantly. There is a slight difference in HC levels under different firing conditions: these levels increase with decreasing temperature. Under ambient and cold conditions, EOHC levels do not increase significantly with large amount of fueling. This is in contrast with hot starts, in which the effects of over-fueling can be seen with increasing HC levels. The lack of sensitivity at the colder temperatures relates back to the observation that much of the injected fuel did not contribute to the combustible gaseous mixture, as discussed earlier (see figures 5.32 and 5.33). A substantial amount of the injected fuel neither burns nor escapes to the exhaust. This fuel may remain in the port, on the cylinder walls, or in the lubrication oil.

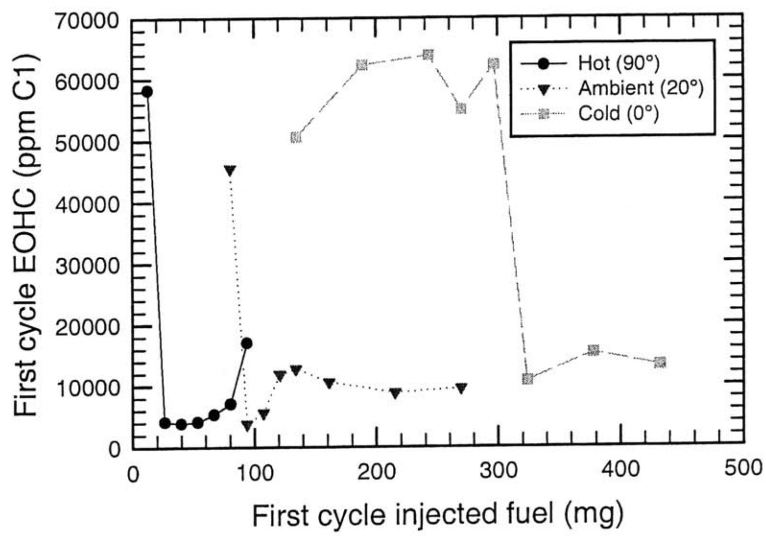


Figure 5.35: 1st cycle injected fuel vs. EOHC

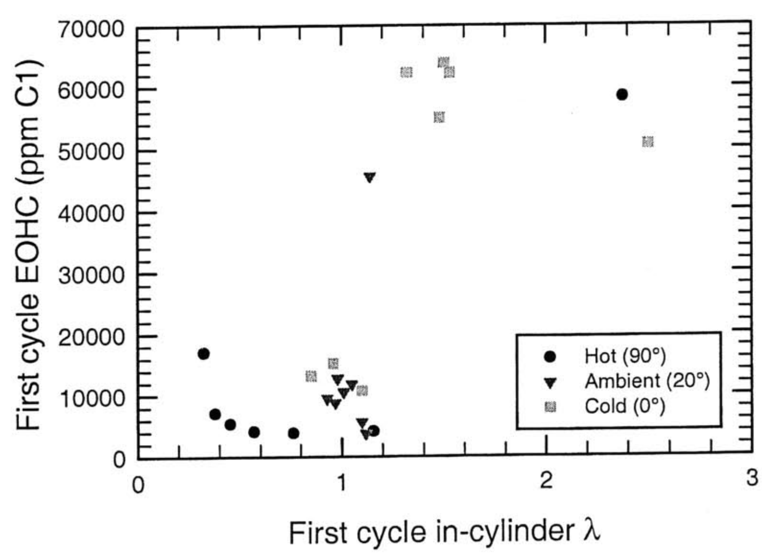


Figure 5.36: 1st cycle in-cylinder λ vs. EOHC

Chapter 6: CONCLUSIONS AND DISCUSSION

6.1 CONCLUSIONS

6.1.1 General Cranking

- The position at which an engine stops is a function of the overall gas load balance from all the cylinders. An engine tends to stop at the positive to negative transition of gas load power over the engine. Therefore, a four cylinder engine will typically stop with one of the pistons at approximately 88° ABC compression, or roughly midway through the compression stroke in one of the cylinders. A six-cylinder engine will tend to stop at approximately 118° ABC compression, or roughly 2/3 of the way through the compression stroke (1/6 of the cycle).
- The most important effect of this stopping, and subsequently starting, position of the engine is on the instantaneous engine speed. The RPM values at the time of ignition in a cylinder depend on the events that take place within the other cylinders before reaching that point, given the appropriate enrichment. Referring to the configurations in this experiment (see figure 3.2):
  - ♦ For starting position A, cylinders 1 and 3 fire before cylinder 4 reaches ignition. The engine is therefore significantly accelerated, and is running at approximately 650 rpm at ignition.
  - ♦ When the engine is started at position B, only cylinder 3 fires before cylinder 4 reaches ignition. The engine is running at approximately 500 rpm at ignition.

- For positions C, D, and E, none of the other cylinders is able to fire before cylinder 4 reaches ignition. The engine is therefore running at approximately the starter motor speed at ignition, which is approximately 200 rpm.

- Open valve injection results in poor combustion or complete misfire. This creates significant charge non-uniformity, directly deposits liquid fuel in-cylinder and consequently increases EOHC emissions, and plausibly wets the spark plug causing misfires.

6.1.2 Hot Start (90°C)

- Late open valve injection delivers much of the fuel as liquid directly into the cylinder, which results in significant charge non-uniformity. This leads to poor engine combustion and uncertainty in in-cylinder FID measurements. This behavior was not observed for early open valve injections.

- The mixture does not get enriched in corresponding proportions with substantial amounts of fuel injection. As more fuel is injected, a smaller fraction actually makes it into the cylinder in vaporized form. The net effect is a diminishing return for an increase in the amount injected.

- First cycle in-cylinder λ is relatively insensitive to starting position at a fixed amount of fuel injection; the slight difference is attributed to differences in induction flow velocities. However, there are considerable differences in IMEPg values between positions A, B and C, D. This is attributed to the difference in speed, which greatly affects combustion phasing, as compared with normal operating conditions.

- Tests showed that the nominal spark timing of 10° BTC is too advanced during cranking, mostly due to the slow engine speeds. Engine performance can be improved by optimizing the spark timing, in particular for the 1st cycle at slower speeds.
- The heat release schedule is greatly affected by the engine speed. At low cranking speeds of approximately 200 rpm, the combustion event is fast in terms of crank angles. Therefore, a nominal spark timing of 10° BTC is substantially too advanced. At this fixed timing, when the speed increases, the IMEPg increases significantly as the combustion event is effectively slowed down in terms of crank angles. Thus, the combustion phasing is more favorable. The heat loss is also lower because combustion occurs later in the expansion process when the change in temperature is smaller. The fact that the heat transfer time is shorter at the higher speeds is also beneficial, but this effect is relatively small.

- The firing threshold is $\lambda \sim 1.1$, reached with an injection of approximately 26 mg.
- For a given starting position, IMEPg is relatively insensitive to in-cylinder λ in the stoichiometric to slightly rich range ($1.1 > \lambda > 0.65$). This is due to the combination of decreased combustion efficiency and improved heat release phasing. While the combustion efficiency decreases with richer mixtures, the relative energy released only decreases modestly since more fuel is burned. This is compensated for by a more favorable heat release schedule, generated by a slower burn. The net effect is that the IMEPg is relatively insensitive to in-cylinder λ in this range.
- EOHC levels are high for lean mixtures due to misfires. Once the firing threshold is reached, HC levels drop significantly, reaching levels within the normal operating

range. As the cylinder is over-fueled and the charge becomes increasingly rich ($\lambda <$ 0.65), HC levels begin to rise again due to partial burns.

- Residual fuel from the 1st cycle enriches the 2nd cycle mixture by about 0.1 λ for every 22 mg increase in fuel. However, these effects are not very noticeable in IMEPg and HC behavior. IMEPg values range from 6 to 8 bar, and HC levels from 2000 to 6000 ppm C1. Both are within the range of and follow trends typical of normal operating conditions.
- There is also no clear evidence indicating any effect from 1st cycle residual fuel on 3rd cycle behavior. Values fall within the normal operating range.

6.1.3 Ambient starts (20°C)

- If insufficient cranking enrichment is provided at lower temperatures, the sporadic firing of the cylinders in the cranking process causes large variability in instantaneous engine speed, which consequently affects engine behavior.
- The firing threshold is at approximately in-cylinder $\lambda \sim 1.1$. In comparison with hot starts, considerably larger amounts of fuel are needed to obtain a combustible mixture (80 mg). In addition, a substantially lower fraction of the fuel injected enters the cylinder to comprise the combustible mixture, and this fraction decreases with increasing amount of injected fuel.
- EOHC levels are similar to the hot start case, both in trends and level, except for open valve injection, which results in higher HC levels. Especially in over-fueling, this final observation is attributed to substantial amounts of liquid fuel entering the cylinder.

- 2^{nd} cycle behavior is also affected by the 1^{st} cycle residual fuel with a decreasing sensitivity of 0.1 λ for a 74mg increase in fuel.

6.1.4 Cold start

- As the amount of injected fuel increases, the in-cylinder λ initially decreases, but levels out around $\lambda \sim 0.9$. It follows that the fraction of injected mass that constitutes the combustible mixture decreases with increasingly large fuel injections.
- The threshold for robust firing is at an in-cylinder λ of 1.1. This is the same value as for the ambient and hot starts.
- Substantially larger amounts of fuel are needed at the cold temperatures to arrive at the above value of λ. The amount, approximately 324 mg, is more than seven times the amount that is required to make a stoichiometric mixture out of the air that is occupying the displacement volume at ambient conditions. It is also considerably larger than the values at hot starts.

6.1.5 Comparison across the temperatures

- For robust firing, the lean limit for the in-cylinder λ is approximately 1.1, irrespective of the engine temperature.
- Lower starting temperatures require larger first cycle fuel injections to produce robust combustion. The fuel injection threshold for firing was 26 mg, 80 mg, and 324 mg at 90°C, 20°C, and 0°C, respectively.

- Similar trends in IMEPg as a function of in-cylinder λ are observed at the different starting temperatures. The IMEPg is relatively insensitive to in-cylinder λ in the range of 0.65 to 1.1. However, the IMEPg values were higher at colder temperatures; this observation is attributed to the change in air density and the capacity to burn more fuel at the same λ.
- The trends in in-cylinder λ as a function of the injected amount of fuel are similar across the different temperatures: significant initial decrease in λ with increasing injections, followed by a leveling out of the λ value.
- The fraction of injected fuel that makes it into the combustible mixture decreases with decreasing temperature. Comparing the values at the same in-cylinder stoichiometric charge, close to 90% of the fuel enters the cylinder in vaporized form under hot conditions; this fraction decreases to ~40% at ambient temperatures, and less than 10% at 0°C.
- EOHC emissions show similar trends across the different temperatures when compared at the same in-cylinder λ: very high levels when the mixture is too lean for robust combustion, low levels around stoichiometric λ, and rapidly increasing levels when λ is very rich due to partial burns. As compared to hot starts, ambient and cold starts show little sensitivity to over-fueling. Minimum EOHC values for ambient and cold starts are higher than for hot starts and are above the normal operating range.

6.2 CLOSURE

This project has provided understanding of the cranking phenomenon. Results suggest that it is in fact feasible to reduce HC emissions and improve engine stability with appropriate control of the fueling strategy during cranking. Specific recommendations include the following:

- Sufficient injected fuel is required for robust combustion in the first cycle. The required amount of fuel increases substantially with decreasing temperatures.
- Engine combustion at the low cranking speeds is substantially different from normal operation. Therefore, appropriate spark timing using the instantaneous engine speed of the particular cycle is recommended. An alternative strategy would be to crank the engine to a higher speed (approximately 700 rpm) before fueling.
- Open valve injection is to be avoided for good start-up.

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APPENDIX

Type	Ford Zetec
Number of cylinders	4
Bore x Stroke (mm)	85 x 88
Displacement	2.0 L
Compression ratio	9.5:1
Valve mechanism	DOHC
Valves/cylinder	4
Fueling	Port injection
Fuel injection	SEFI Return / pc
Fuel	California Phase II
Firing order	1-3-4-2
IVO	10° BTC
IVC	50° ABC
EVO	50° BBC
EVC	10° ATC

Figure A.1: Engine specifications

What is success?
To laugh often and much;
To win the respect of intelligent people
and the affection of children;
To earn the appreciation of honest critics
and endure the betrayal of false friends;
To appreciate beauty;
To find the best in others;
To leave the world a bit better,
whether by a healthy child, a garden
patch, or a redeemed social condition;
To know even one life has breathed
easier because you have lived;
That is to have succeeded.

Ralph Waldo Emerson