Mean Effective Pressures and Fuel Consumption Characteristics of a Direct Fuel Injection Spark Ignition Two-Stroke Engine

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Abstract

The effects of direct fuel injection at 800 rpm were explored on a two-stroke single-cylinder engine using gaseous methane (CH_4). Methane was injected at 12 atm, with start timings ranging from immediately after port closure (homogeneous) to 65°BTDC (stratified). Two injection durations (25°) and (19°) were evaluated, and their output and efficiency compared those of premixed charges with fuel-air equivalence ratios ranging from the lean limit to the rich limit.

Direct fuel injection clearly showed an increase a decrease in specific fuel consumption and a corresponding increase in thermal efficiency. Pressure curves suggested that this was due to higher cylinder peak pressures for a given equivalence ratio. Direct fuel injection also seemed to advance the timing of peak pressures, so further work on mapping spark and injection timings would prove valuable.

This report speculates on the thermodynamics behind the increases in efficiency, and attributes the causes to higher volumetric efficiency and less short-circuited fuel. However, further tests must be done, and care must be taken regarding the validity of our data due to some possible errors in calibration.

Introduction

Gains in specific output, thermodynamic efficiency and power density are the goals of every engine designer. The two-stroke engine, with the ability to deliver a power stroke at twice the rate of that of a four-stroke engine, is the choice when a high power to weight ratio is of primary importance. However, the two-stroke engine is still unused in the passenger car market and other potentially lucrative industries because of its high hydrocarbon emissions.

Recently, powerplant, specifically car, manufacturers have accelerated advances in engine technology to conform with stricter emissions mandates. In fact, due to the sudden wealth of improvements in internal combustion engine technology, the US government has relaxed its zero emissions laws. One of these newer technologies is direct fuel injection, hereafter referred to as DFI. This report will introduce some of the characteristics and advantages of DFI, particularly with respect to a two-stroke engine.

Rationale

What Is Direct Injection

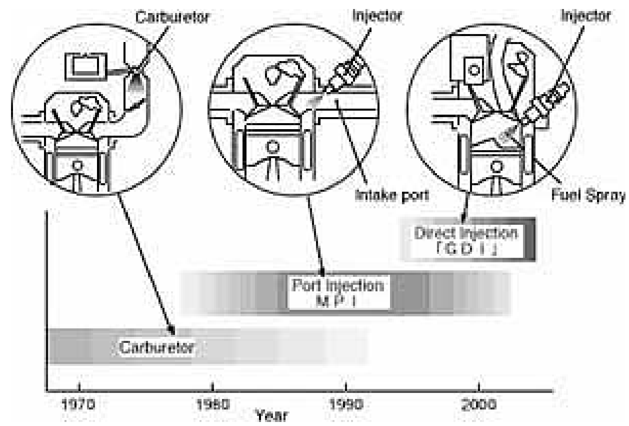


Figure 1. The age of direct fuel injection [1]

The greatest difference between DFI and regular port fuel injection (PFI) or premixed charges used in spark ignition gasoline engines is that, instead of a premixed mixture of air and fuel entering the combustion cylinder, only air is inducted in a DFI engine, while fuel is injected directly into the cylinder at a later time. (Figure 1). In this way, the DFI concept is similar to that of a compression ignition diesel engine, but has output characteristics similar to that of a gasoline engine. The advantages of DFI are numerous, and its gains over PFI have been compared to the gains found in PFI during the carburetor age.

Why Direct Injection

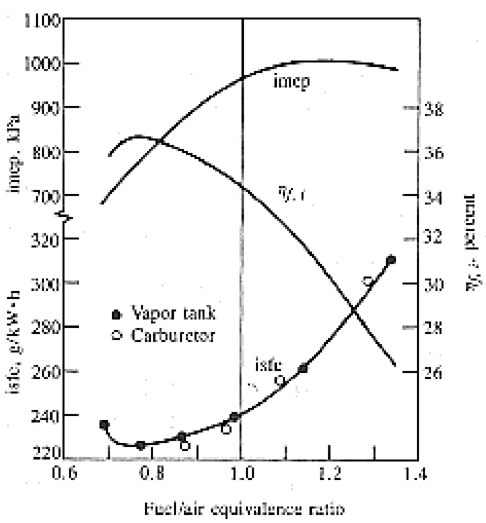


Figure 2. Efficiency vs. Stoichiometry. [2]

Lean burning, where the amount of air is greater than the stoichiometric amount, is desired for high efficiencies. Figure 2 shows the thermodynamic efficiency as a function of compression ratio and stoichiometry. However, lean mixtures are hard to ignite reliably with regular PFI and thus have not been widely implemented. But with DFI, the fuel-air mixture can be stratified such that the fuel near the spark is within the flammability limits while the rest of the charge is lean enough such that knock is unlikely [3]. This allows for the high efficiencies previously unattainable with PFI. On the other hand, regular homogeneous mixtures can also be emulated when high power is required by injecting relatively early and allowing the mixture to become uniform prior to ignition (Figure 6).

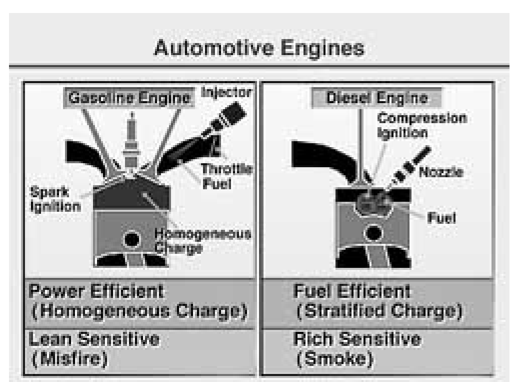


Figure 3. A hybrid between spark ignition gasoline and compression ignition diesel. [1]

In addition to the efficiency gains due to lean burning, DFI is like the diesel design and allows load and output control by determining the fuel quantity injected rather than the air inducted. This eliminates the need of a throttle, and pumping losses are significantly reduced [3]. Mitsubishi Motors estimates the efficiency increase from this factor alone to account for half of the gains attainable through the use of DFI in four-stroke engines [4]. But, unlike a diesel, the spark timing can also be adjusted. Therefore, because fuel is injected independently of the air, there exists the parameter of injection timing in addition to spark timing. Mitsubishi Motors feel that the most distinctive feature of DFI is the freedom of mixture preparation [5]. Furthermore, difficulties in controlling NOx emissions in lean burning port fuel injections is less severe as direct injection engines tend to be very tolerant to internal exhaust gas recirculation (EGR), which, in itself, also reduces pumping work [6].

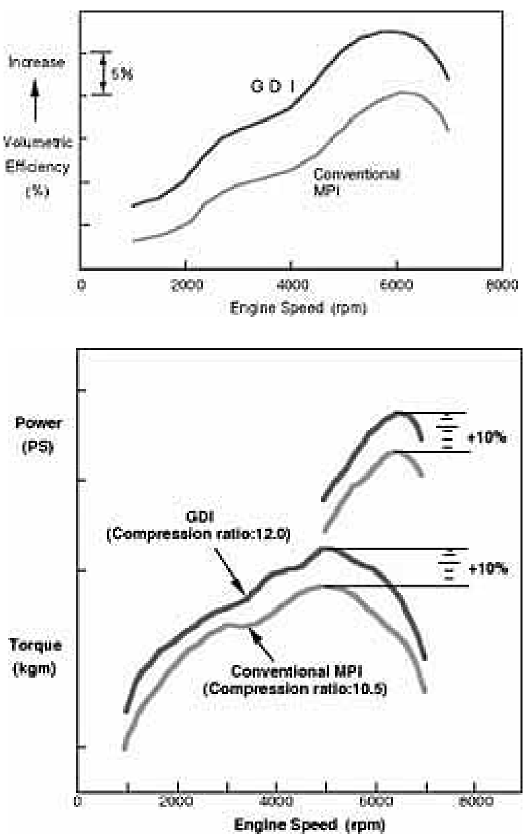


Figure 4. Attainable increases in volumetric efficiency and torque due to DFI [1].

In terms of drivability, the most significant gains in utilizing DFI include higher specific torque and a lower knocking tendency, both of which result from charge cooling [3], when liquid fuels are used. Whereas in PFI, the fuel vaporizes by absorbing heat from the port wall and valves, the heat of vaporization in a DFI engine is supplied by the air itself, thereby reducing the average charge temperature and allows for higher compression ratio and higher thermodynamic (in particular, volumetric) efficiency [3]. For example, the heat of vaporization for gasoline is approximately 350 kJ/kg. This charge cooling effect also means less heat is lost through to the coolant through the cylinder walls [3]. Mitsubishi estimates a reduction charge temperatures of about 15K, and a decrease of about 30K when fully compressed for a 4-stroke multivalve engine [7]. Also, aggressive spark timings can be utilized without running into the autoignition limit, and compression ratios can be increased for even higher efficiency.

Although the fuel economy benefits of DFI running at stoichiometric is only a few percent compared to PFI (dependent on the degree of EGR, the reductions in fuel economy can be as high as 12% since the DFI can guarantee proper mixture ignition at gasoline AFR (air-fuel ratio) beyond 20:1 [3]. Other estimates are put at 20% when compared with PFI running at stoichiometric [8].

The two stroke engine, although with a high power to weight has been deemed environmentally unfriendly because it can be considered as on open system, where the fuel-laden intake charge used to exhaust the combusted mixture (scavenging) often short-circuits and escapes, unburned, out of the exit port [9], resulting in extremely high HC emissions and unused fuel. However, when DFI is utilized, injection timing can be adjusted such that fuel enters the cylinder only when the exhaust port is closed, so the earlier problems of short-circuiting are not a major concern. This means a significant reduction in wasted fuel, and thus even higher gains in efficiency are possible when DFI is used in conjunction with two-stroke designs.

Despite the multiple benefits of direct injection, it has not been widely implemented because the difficulty in co-ordinating the fuel, air and spark. With improvements in electronic control technology, however, the feasibility of DFI has increased dramatically.

Injection Timing and Injector Design

Initial designs of DFI had the spark plug and fuel injector close together to achieve a spray-guided distribution. (Figure 5).However, this technique shortens the lifetime of the plug due to the impingement of liquid fuel on the electrode and thus was deemed unreliable. A more reliable solution is obtained via a wall-guided technique, such as that used by Mitsubishi Motors in their DFI engine, where a cavity at the top of the piston guides the charge flow. An air-guided technique also exists, where the mixture is directed by a well defined in-cylinder flow [8]. Orbital and other companies are investigating air assisted injection which allows the fuel injector to spray air as well as fuel, at predetermined times to widen the flammability range further [6]. As aforementioned, in addition to spark timing, injection timing is of importance in a spark ignition DFI engine. In order to extract the highest volumetric efficiency from charge cooling, piston wetting is to be avoided, so later injection timings tend to be best, as long as it is not too late such that insufficient time remains for vaporization. Figure 6 shows the different injector patterns for different conditions.

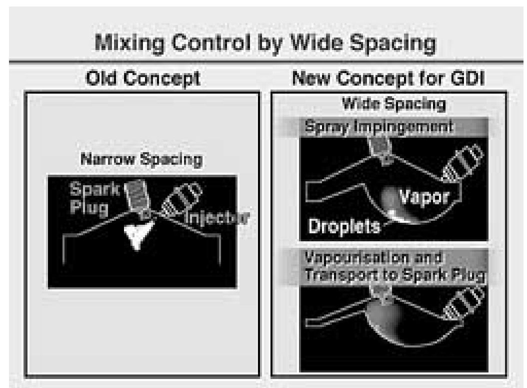


Figure 5. Injector location [1]

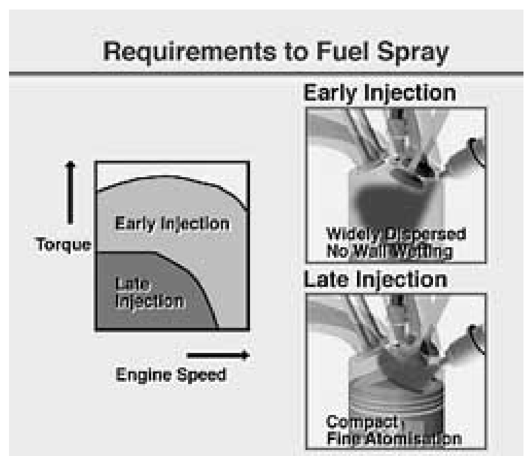


Figure 6. Different injection patterns for different conditions [1]

Definitions and Concepts

The realistic definition of thermal efficiency is given as the ratio between rate at which the engine produces work (power) and the rate of heat input (Equation 1). As aforementioned, part of the advantages of DFI can be attributed to increases in volumetric efficiency, so it is appropriate to define the term here. The volumetric efficiency is a measure of the effectiveness of an engine's ability to pump air (Equation 2). Using the ideal gas relations, the expression for volumetric efficiency becomes that shown in Equation 3, as a function of molecular weights, pressures, temperatures, the fuel-air ratio, the compression ratio and the ratio of specific heats for the mixture.

$$\eta = \frac{P}{\dot{m}_{fuel} Q_{combustion}}$$

Equation 1. Thermal efficiency

$$\eta_v = \frac{m_{air}}{\rho_{air} V_{dispent}}$$

Equation 2. Volumetric efficiency [2]

$$\eta_v = \left(\frac{M_{mix}}{M_{air}}\right)\left(\frac{p_{\text{int}\,ake}}{p_{air}}\right)\left(\frac{T_{air}}{T_{\text{int}\,ake}}\right)\frac{1}{\left[1+\left(FAR\right)\right]}\left\{\frac{r_c}{r_c-1}-\frac{1}{\gamma\left(r_c-1\right)}\left[\left(\frac{p_{exhaust}}{p_{\text{int}\,ake}}\right)+\left(\gamma-1\right)\right]\right\}$$

Equation 3. Volumetric efficiency re-expressed [2]

Since the intake pressure is the sum of the partial pressures of the air, fuel and water, and. the gaseous fuel exists within the intake system for a PFI engine, the partial pressure of air is lower than that of the entire mixture, and thus a PFI engine has a lower volumetric efficiency.

An indicator of the output of an engine is the indicated work, W (the common unit is J, the joule). Please note that we have decided to use the gross value of indicated so as to eliminate slight changes in pumping work. Graphically, W is the area between the traces of the pressure-volume curves, excluding the loop at the lower right hand corner. The gross power, P (normally in kW), of an engine is then the product of the W and the engine speed, S (in revolutions per minute or revolutions per second), divided by the number of revolutions per power stroke per cylinder (2 for four-stroke cycles and 1 for two-stroke cycles), as shown in Equation 5. Therefore, one can see that, at a given speed and indicated work, a two-stroke engine will produce double the power, and it is for this reason that two-stroke designs are acclaimed for their high power density.

$$W = \int p dV$$

Equation 4. Work [2]

$$P = \frac{W \cdot S}{N}$$

Equation 5. Power

A more useful indicator is almost non-dimensional parameter for describing engine output is the mean effective pressure (MEP, usually in kPa or MPa). Although various definitions exist, this report will use the standard definition for gross indicated mean effective pressure (IMEP). IMEP is defined as work output per cycle divided by the engine displacement. (Equation 6).

$$IMEP = \frac{P}{V_{displacemn\,t} S}$$

Equation 6. IMEP [10]

Although comparisons in IMEP can give the displacement efficiency of the engine, the fuel consumption is not taken into account. Thus, the indicator of indicated specific

$$ISFC = \frac{\dot{m}_f}{P}$$

Equation 7. ISFC [2]

fuel consumption (*ISFC*) is more meaningful thermodynamically (Equation 7). It is defined as the mass flow rate of fuel divided by the power output, and thus has the units of g/kW h. The ISFC is the most commonly used indicator for efficiency, in lieu of Equation 1. There are also the fuel-air ratio (AFR) and fuel-air ratio (FAR) by mass (Equation 8 a, b). The inverse of this ratio is sometimes used too. The indicator ϕ, for equivalence ratio, is used to describe the quotient of the actual FAR (fuel-air ratio) over stoichiometric. Thus, ϕ greater than one signifies a rich mixture, and ϕ less than one signifies a lean mixture.

$$AFR = \frac{\dot{m}_{air}}{\dot{m}_{fuel}}$$

$$FAR = \frac{\dot{m}_{fuel}}{\dot{m}_{air}}$$

Equation 8a, b. AFR and FAR

$$\phi = \frac{FAR}{(FAR)_{stoichiometric}}$$

Equation 9. ϕ

Since the two-stroke engine is an open cycle where fresh charge displaces combusted gases, several more definitions are required. The scavenge ratio, SR, is the amount ratio of the mass of air supplied during the intake/scavenging period to the actual mass that could fill the entire cylinder volume (Equation 10). Several other definitions such as delivery ratio (DR),scavenging efficiency (SE), trapping efficiency (TR) and charging efficiency (CE) and charge purity (Π) describe similar ideas.

$$SR = \frac{m_{air}}{\rho_{air} V_{cylinder}}$$

Equation 10. Scavenge ratio [10]

We can also analyze how charge cooling reduces cylinder temperatures by considering an energy balance for a control volume. However, since that will not be the focus of the report, I will just give the equation for the temperature difference due to vaporization, assuming a constant specific heat ratio (Cp), Equation 11 shows that if the heat of vaporization can be supplied from the air alone such the charge temperatures will drop proportionally to the amount of fuel. This is the primary effect of charge cooling, which allows for higher compression ratios.

$$\Delta T = \frac{m_{fuel} Q_{vaporization}}{M_{air} C_p}$$

Equation 11. Reduction in charge temperature due to vaporization by air only [3]

Experiment

Idea and Goal

Because the two-stroke engine is favorable due to its high power density, we have chosen, in this experiment, to determine the effects of DFI on IMEP and ISFC in a two-stroke single-cylinder engine. It is hoped that, with the implementation of DFI, a noticeable decrease in ISFC can be observed. Although charge cooling is a major benefit of DFI since the practical fuels are in the liquid phase, we have chosen to use gaseous methane for feasibility reasons. Furthermore, this allowed us to estimate more accurately the increase in efficiency due to the elimination of fuel short-circuiting and volumetric efficiency due to an air-only induction.

Design

Table 1. Engine specifications with some data from [11].

Specifications	*Value*	*Units*
Bore	82.60	mm
Stroke	114.30	mm
Connecting Rod Length	254.00	mm
Intake Port Timing	±54	°BDC
Exhaust Port Timing	±67	°BDC
Squish Area	75	%
Clearance Height	1.5	mm
Cup Length & Width	36.60	mm
Cup Depth	26.40	mm
Cup Volume	35.36	cm^3
Displacement	611.75	cm^3
Total Volume	655.14	cm^3
Compression Ratio (Geometric)	14.6	
Compression Ratio (Exhaust Port Closure)	10.8	
Spark Plug	J Type	
Ignition	12V, GM LX-301	
Fuel Injector	Air Forced, Ford	
Injection Back Pressure	12	atm

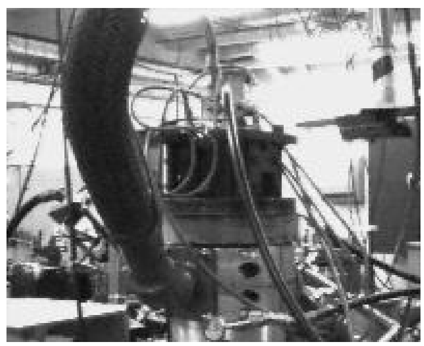


Figure 7. Engine and exhaust

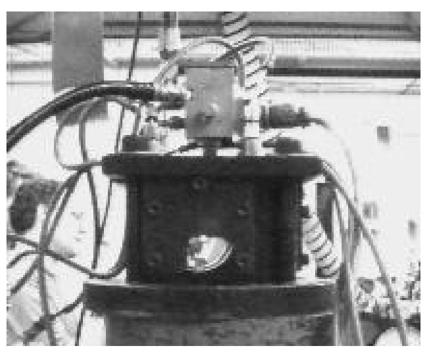
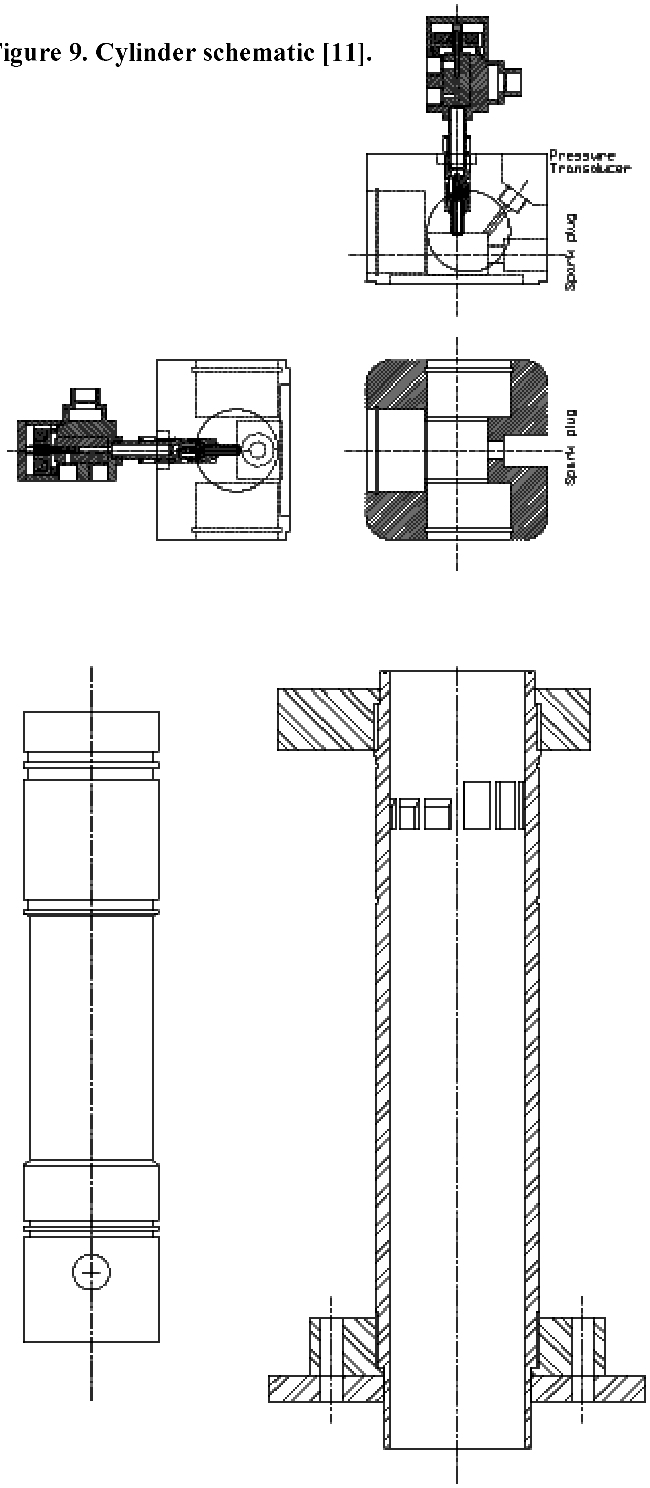


Figure 8. Cylinder head with glass windows

The tests were conducted on a two-stroke single cylinder engine, with design specifications listed in Table 1. The engine and cylinder head are shown as Figures 7 and 8, and its schematic is shown as Figure 9. In summary, the undersquare 611.75 cm^3 engine is of the cross-scavenged with 6 intake and 6 exhaust ports lying along the cylinder's circumference, with timings of 54° and 67° from BDC, respectively. There is also a 35.4 cm^3 cup in the shape of a rectangular prism at the top of cylinder head, with glass windows to facilitate combustion visualization. These dimensions result in a geometric compression ratio of 14.6, and a compression ratio of 10.8 after exhaust port closure. Most of these data are provided by De Risi [11], who used this engine for a SAE paper last year. Ignition was triggered by using a typical J-gap spark plug mounted

Figure 9. Cylinder schematic [11].



$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.52N_2$$

Equation 12. Chemical balance for methane in air (oxygen and nitrogen)

transverse to the cylinder axis and powered by a 12 VDC power supply and triggered by a GM LX-301 ignition module. The fuel injector for DFI was situated at the top of the cylinder head, parallel to the cylinder axis. A compressor supplied the air through a mass flow meter and a 32.4 L surge tank to steady out pressure disturbances. A vacuum pump prevented oil buildup in the combustion chamber.

Parameters for Premixed & DFI

Tests were run at 800 rpm and light load to simulate idling and low demand conditions. However, the spark was only triggered once every sixth cycle to reduce cylinder temperatures and to allow for good scavenging. The air flow was kept constant at 18.5 SCFM (standard cubic feet per meter indicated by a Hastings flow meter) or 655 cm^3/s, which is proper amount for a scavenge ratio of 1.0, assuming ideal scavenging. In reality, higher scavenge ratios are usually required to allow proper combustion due to inefficiencies. However, since 5 unfired, motored cycles preceded each fired cycle, we assumed that we could obtain entirely fresh charge before each ignition.

Spark timing was kept at 14° BTDC, which appeared to be the optimum for stoichiometric combustion at our engine speed. Because the ignition module had a built-in time delay, care was taken to determine its value using a strobe and an oscilloscope. At 800 rpm, the time delay corresponded to 24°. Determination of the theoretical stoichiometric AFR by mass was necessary; the balanced chemical equation is shown as Equation 12. Further calculations involving molecular weights gave an AFR of 17.19 by mass. However, we used a tabulated value of 17.23 [2], since Equation 11 does not include the trace gases found in air.

Premixed

Because PFI could not be obtained with the laboratory setup, methane from a storage cylinder, delivered at about 14-15 atm was regulated by a control valve and premixed about 3 m upstream from the cylinder. A Hastings flow meter indicated the rate of fuel flow. Since the flow meter was calibrated for air, multiplication by a published conversion factor of 0.69 was necessary to obtain the correct mass flow rate. We attempted to determine stoichiometry by an emissions tester. However, due to the fact that we were sparking only a sixth of the time, the emissions tester's resolution was deemed inadequate for our purposes. Instead, stoichiometric burning had to be determined by watching the flame color. Seven AFRs were recorded, ranging from the lean to the rich limit.

DFI

A Ford air forced injector was used in the DFI tests. The computer signaled a timed controlled voltage generator. The output voltage was then inverted and sent to a current generator to activate the injector's poppet. Any uninjected fuel would be routed through an overflow tube, and this excess methane was burned. We based our injection start timing and duration parameters on De Risi's [11] SAE paper. For safety and injector durability concerns, the methane backpressure was set at 12 atm. It should be noted that the amount of fuel injected was not just a function of injection duration. Rather, it was also a function of injector start timing since the amount of fuel is proportional to the difference between the methane and cylinder pressures. Thus, with an earlier injector start timing, the cylinder pressure was lower, and more methane could be injected.

Due to the requirement of a 12 atm injection pressure, the Hastings flow meter used for the premixed charges could not be used for DFI. Instead, two Matheson FM-1050 mass flow meters were used, one for the main flow, and one of the overflow. Since these were uncalibrated, a Singer volume recorder was used to determine the relation between meter divisions and actual mass flow. For the main flow, expansion valves were necessary since the volume recorder could not withstand pressures significantly higher than 1 atm. Since the overflow would be vented and burned at atmospheric pressure, the use of expansion valves was unnecessary for the overflow meter calibration. A calibration chart for both meters are shown as Figure 18 in the appendix. The mass flow rate through the main flow meter was read, from which 5/6 of the value indicated by the overflow meter was subtracted. The 5/6 factor was due to the fact that we were firing every 6^{th} stroke. Some errors were inherent with our measurements because we assumed a steady state flow, which was obviously not the case with DFI; the flow meters, especially the overflow meter, also showed significant fluctuation at times. However, the mass flow rates were repeatable as long as we gave a minute or two to allow for the methane pressure to reach a pseudo-equilibrium in the tubes. Still, some error is involved due to the assumption of the 1 atm overflow pressure. When we ran the DFI engine unfired, the overflow reading did not correspond to the main flow reading; a difference of about 0.05 g/s was observed.

Two injection durations were used: 5.42 ms or 26°, and 4.00 ms or 19°. For each duration, tests were done with four injection start timings, 115°, 100°, 90° and 60° (all ABDC). De Risi [11].suggested a start timing of 125°ABDC. However, with our 12atm backpressure, reliable firing could not be obtained.

<u>Data Acquistion</u>

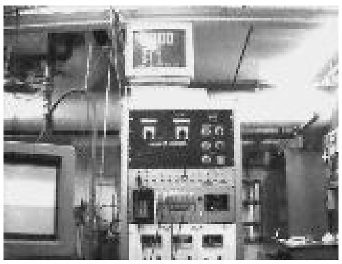


Figure 10. Data acquisition computers

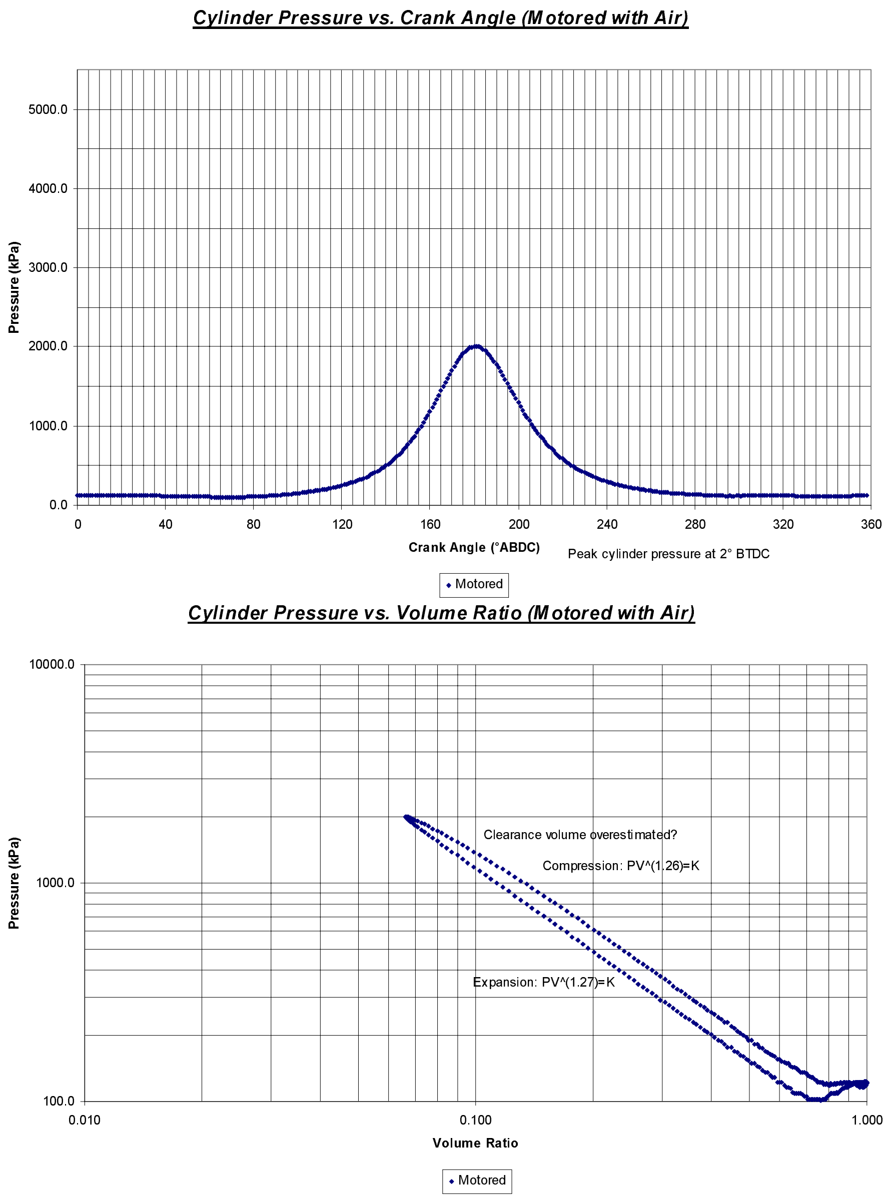


Figure 11. Pressure curves for motored test

The cylinder head was equipped with a pressure transducer, and a transducer was also installed at the injector to verify that the injector's poppet was operational. The amplified signals from the transducers were used as inputs for a personal computer. Live engine speed data were displayed on a second personal computer, which also served as the trigger for spark and injection timings. Figure10 shows the data acquisition computers. An air-only motored run was recorded to determine the integrity of the measurement procedure, and is shown in Figure 11. Using Lancaster's paper [12] as a guideline, it appears that our cylinder clearance volume might have been overestimated since there was a curve at the upper end of the compression stroke of the pressure vs. volume diagram (Figure 11b).

We believe that this was perhaps due to the volume taken up by the spark plug and injector tip, which were unaccounted for in our clearance volume approximation. Also, although the ratio of specific heats as determined by the slopes of the pressure and volume curves (Figure 11b) were about 1.26, lower than the ideal gas model's 1.4. However, this is acceptable since the compression and expansion processes are not really adiabatic. Also, although the reference pressure was set at atmospheric, the acquired data appeared to show pressures of about 15-16 psi, suggesting an error in the C computer program. Nevertheless, the general shape of the curves were representative of the actual process. The pressure data acquired were all 1° early because the computer could not send a trigger exactly at TDC, so they had to be shifted before any calculations were performed. Data were averaged over 20 fired cycles. IMEP was calculated in a way similar to the Riemann sum method. The change in pressure at each volume was determined and multiplied by the volume increment. For example, the volume at 45°BTDC and 45°ATDC were identical due to symmetry, and so the pressure difference at these two volumes were determined, and multiplied by a volume increment that extended both earlier and later in crank angle. However, this did introduce some error since the volume increment per degree is larger at the middle of the strokes, when the connecting rod was not straight, and the piston had higher speed.

Test Results & Discussion:

Results are given in tabular form in Table 2, and in graphical form, starting from Figure 12. Please note that an error must have occurred with the flow meter calibrations for the premixed case. A stoichiometric flame was obtained, but the flow meter readings gave a ϕ of 1.09 (slightly rich). Furthermore, the lean limit was reached at a ϕ of 1.07! This was not surprising as it proved

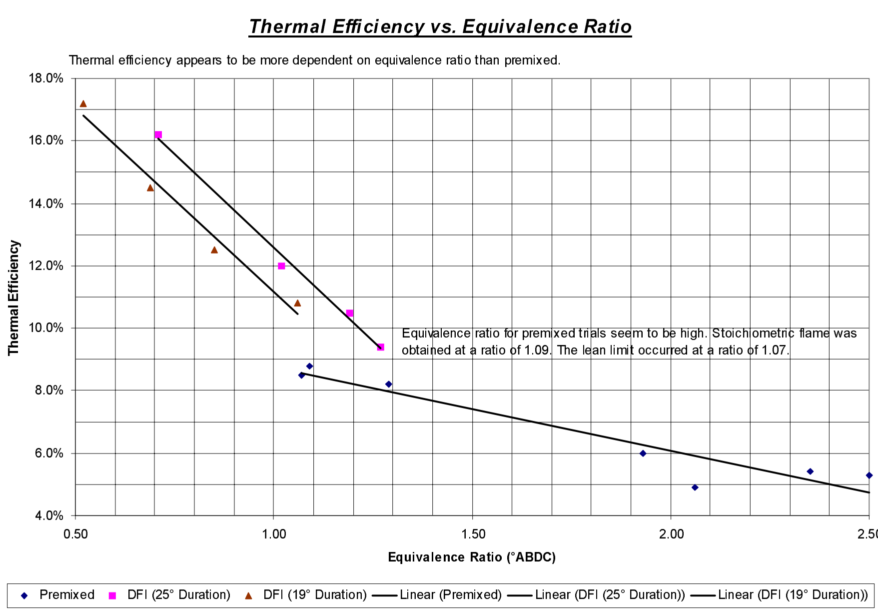


Figure 12. Thermal efficiency vs. ϕ

difficult to calibrate the Hastings fuel flow meter; there was significant drift, even when all methane pressure was released. This error meant that our ISFCs for the premixed trials were higher than in reality, and the thermal efficiencies were lower. Still, one can see in Figure 12 the decrease in efficiency with an increasing equivalence ratio. The first observation is that the thermal efficiencies were not only significantly better for DFI, but they also appeared to be a lot more sensitive with DFI than with premixed charges (Figure 12). Possible reasons will be discussed by analyzing the IMEP and ISFC graphs.

With the spark timing fixed at 14 °BTDC, Figure 13 shows that premixed IMEPs increased with richening of the mixture (from 343 kPa at the lean limit to 492 kPa at the rich limit), understandably as more fuel meant more released hit. ISFC increased from 844 g/kWh to 1371 g/kWh as a result of the added fuel (Figure 14). The greater rate of ISFC increase than IMEP increase caused efficiencies to drop from 8.5% at the lean limit to 5.3% at the rich limit. The only exception was that the lowest ISFC and highest efficiency were obtained at stoichiometric, probably an error caused by the testing of the lean burn cycle on a different day. FI test results depended on injection duration. For the 26° (5.42 ms) duration DFI trials, ϕ ranged from 0.71 to 1.27. Leaner charges corresponded to later start timings due to the decrease in the cylinder and injector pressure gradient. For

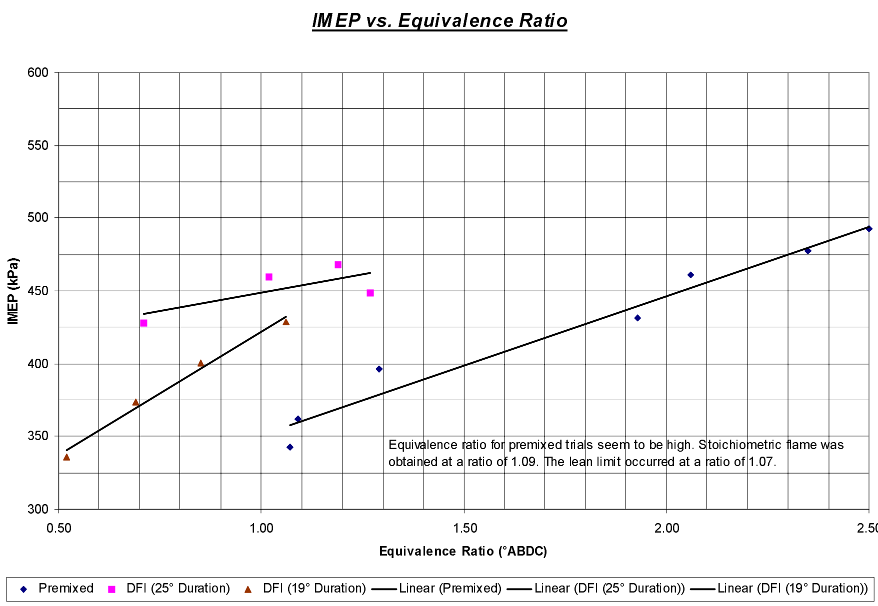


Figure 13. IMEP vs. ϕ

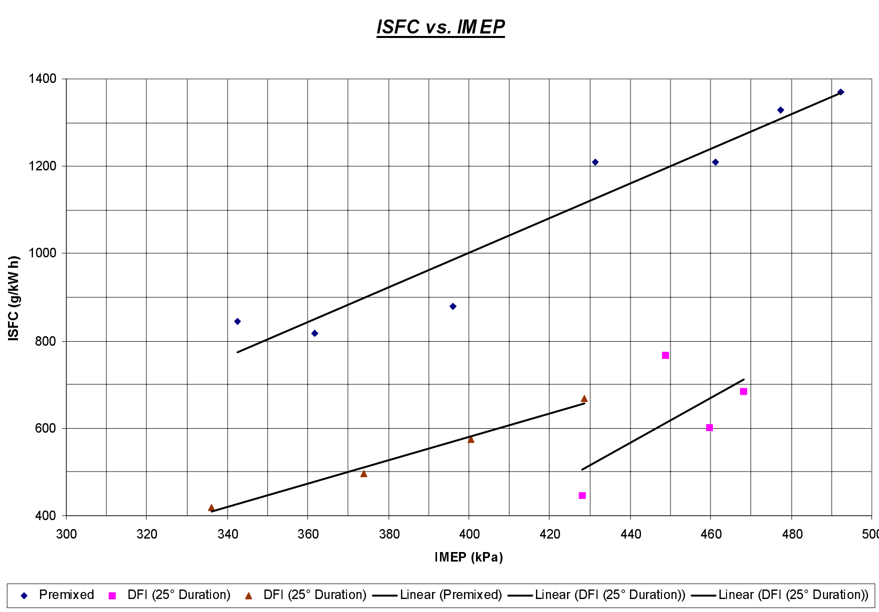


Figure 14. ISFC vs. ϕ

Premixed				DFI (26° Duration)					DFI (19° Duration)				
ϕ	IMEP (kPa)	ISFC (g/kWh)	η (%)	Injection Start (°ABDC)	ϕ	IMEP (kPa)	ISFC (g/kWh)	η (%)	Injection Start (°ABDC)	ϕ	IMEP (kPa)	ISFC (g/kWh)	η (%)
1.07	343	844	8.5%	115	0.71	428	446	16.2%	115	0.52	336	418	17.2%
1.09	362	817	8.8%	100	1.02	460	602	12.0%	100	0.69	374	497	14.5%
1.29	396	878	8.2%	90	1.19	468	685	10.5%	90	0.85	401	574	12.5%
1.93	431	1209	6.0%	67	1.27	449	767	9.4%	67	1.06	429	668	10.8%
2.06	461	1211	4.9%										
2.35	477	1330	5.4%										
2.50	492	1371	5.3%										

Table 2. Table of results

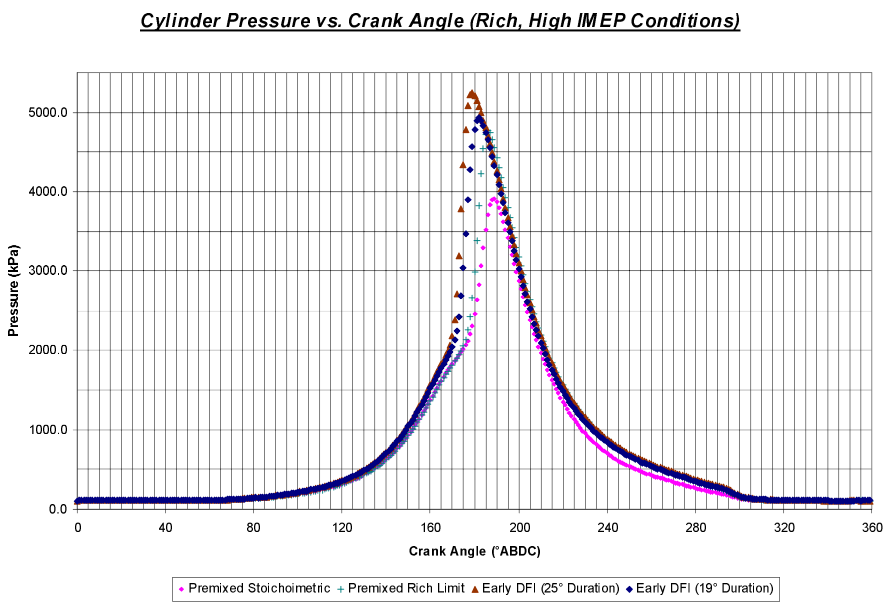
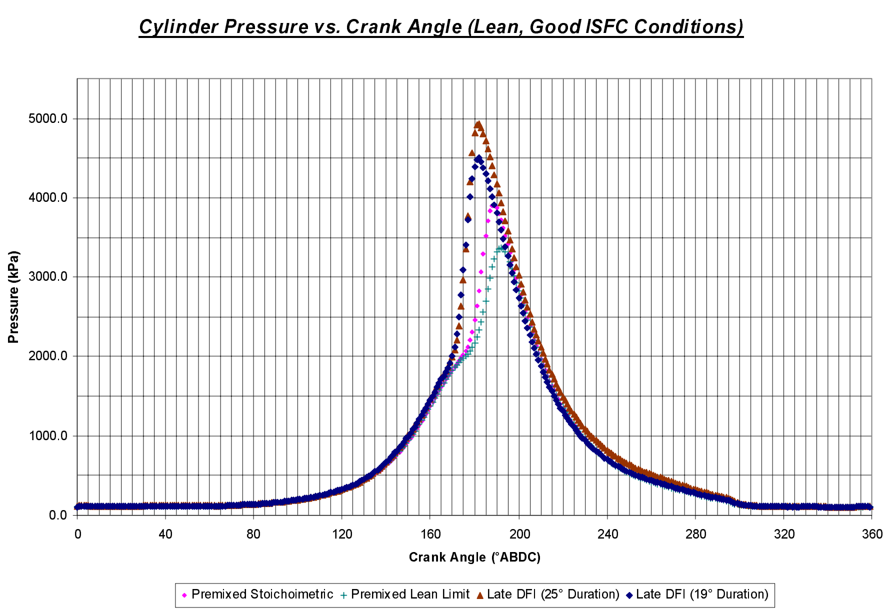


Figure 15a, b. Cylinder pressure versus crank angle.



the 16° duration (4.00 ms) trials, ϕ ranged from a very lean 0.52 to a slightly rich 1.06.

As with premixed charges IMEPs and ISFCs rose with richening of the overall mixture. However, the IMEP of the 19° duration case appeared to be more sensitive to changes in AFR (Figure 13), whereas the IMEP of the 25° case was not as sensitive. Although Heywood (Figure 2) suggests that IMEP is largely a function of ϕ only, the discrepancy was perhaps due to the fact that the AFRs were not changed only by varying duration, but also by injection start timing. Therefore, IMEPs of the DFI tests were not a direct function of ϕ, and the IMEP sensitivity differences cannot be attributed to duration of ϕ alone.

Also, for a given equivalence ratio, the 25° duration appeared to give very slightly lower ISFC values. Perhaps with fuel injection lasting 6° later into the cycle allowed for a better stoichiometry around the spark plug. In summary, IMEPs for the direct injection tests ranged from 336 kPa to 449 kPa, roughly in the same range as those obtained with premixed charge. However, these IMEPs were obtained with significantly lower ISFCs, from a low of 418 g/kW h to a high of 767 g/kW h. Note that even the highest ISFC for DFI trials was lower than the lowest of the premixed tests.

Although some gains were expected in the DFI trials, improvements of this magnitude were not expected. One of the sources of error was the fuel flow meters. As aforementioned, the calculated equivalence ratios based on the flow meter readings appeared to be very high for the premixed case, and inaccuracies in calibrating the flow meters for DFI could have widened the gap between DFI and premixed charge. In fact, since the ISFC slopes of all the trials were similar, it was more likely that all the were conducted within the same ϕ range, because if the premixed trials were really as rich as indicated, ISFC slopes would be much higher, as depicted in Figure 2.

The cylinder pressure vs. crank angle diagrams (Figure 15) also reveal some interesting characteristics. It appears that, for both premixed and DFI, richer mixtures not only resulted in higher peak pressures due to the increase mass of fuel burned but also advanced the timing of the peak pressure. In fact, under very

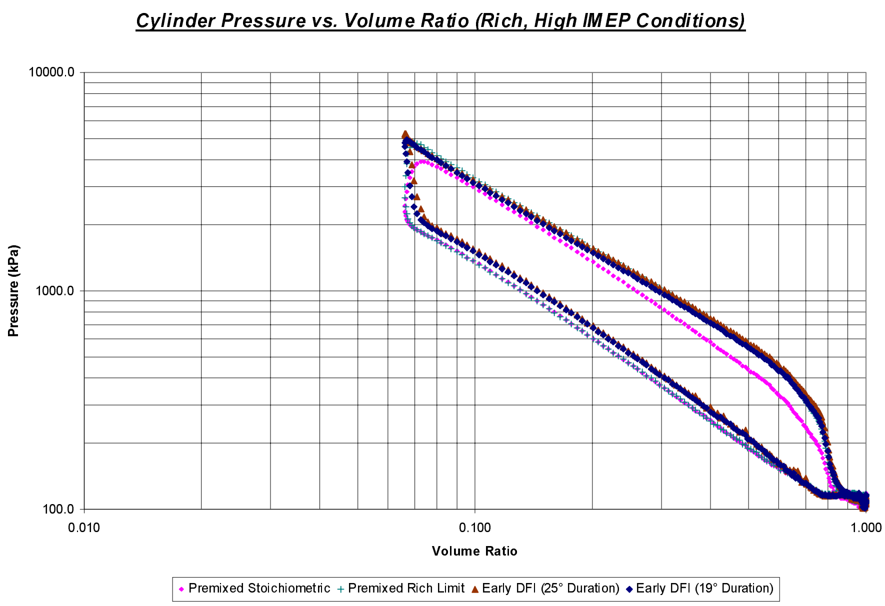
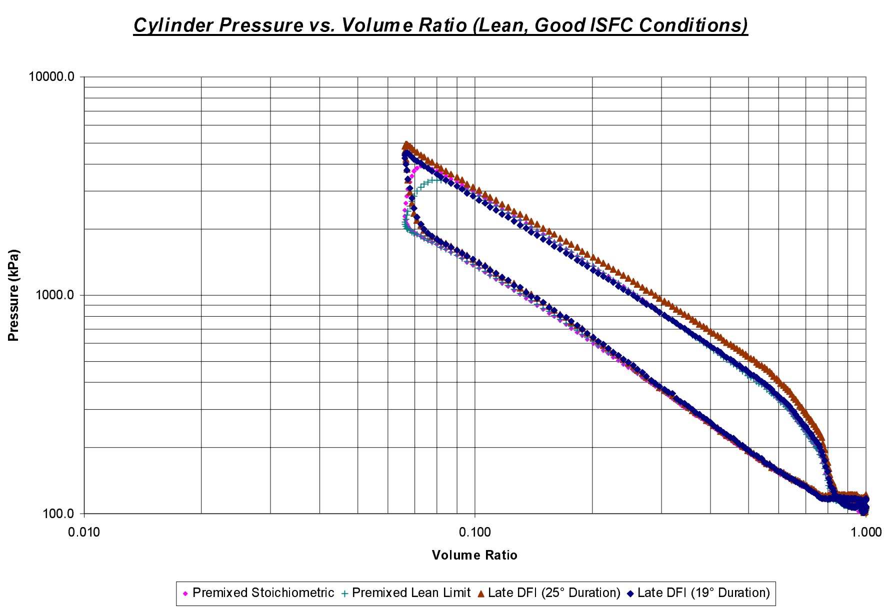


Figure 16 a, b. Cylinder pressures vs. volume ratio



rich conditions, the peak pressure was sometimes prior to TDC, meaning work was lost because of the high pressures during compression. Perhaps this could be explained by a faster rise in temperatures due to the increased fuel content. The opposite was true for lean mixtures; peak pressures were lower and retarded, probably because less heat was released, and at a slower rate. These results are presented in a different format in the cylinder pressure versus volume ratio plots.

One characteristics of DFI as seen from the pressure plots is that DFI appeared to advance peak pressures in general, characterized by a very sharp bend around TDC on the pressure versus volume ratio curves (Figure 16). These figures also exhibit another characteristic of DFI: higher pressures throughout the expansion stroke. In some cases, the DFI also showed higher pressures in the early parts of the compression stroke. These peculiarities of higher and earlier peak pressures suggest higher overall cylinder temperatures, which probably resulted from the fact that the engine had been running for a long time when DFI trials were performed.

Future Work

This section serves to give a glimpse of the abundance of future research topics in the emerging field of DFI. This experiment evaluated the gains of DFI injection due mainly to increases in volumetric efficiency and reductions in fuel short-circuiting. However, Houston explains that "[i]t is the real world simultaneous achievement of low engine-out emissions of NOx and hydrocarbons without detriment to the fuel economy capability, which is important to the introduction of DFI technologies." [5] Much more work needs to be done on the gains of DFI that are more widely applicable. That is, since most practical fuels are in the liquid phase, charge cooling plays a major role. Furthermore, it cannot be forgotten that another very important side-effect of DFI is the dramatic reduction in throttling losses because output can be controlled by the amount of injected fuel rather than the volume rate of air inducted. Of course, low emissions have to be verified before the certification of a new design is possible

Although we kept spark timing constant at 14°BTDC, we expect changes in spark timing to have a very significant impact on the potential of DFI. Furthermore, because DFI eliminates the problem of fuel short-circuiting, much higher scavenge ratios should be tested, and the increased flow of air are expected to have an impact due to higher mass flow and flow velocities.

Future Considerations: Why not DFI in North America?

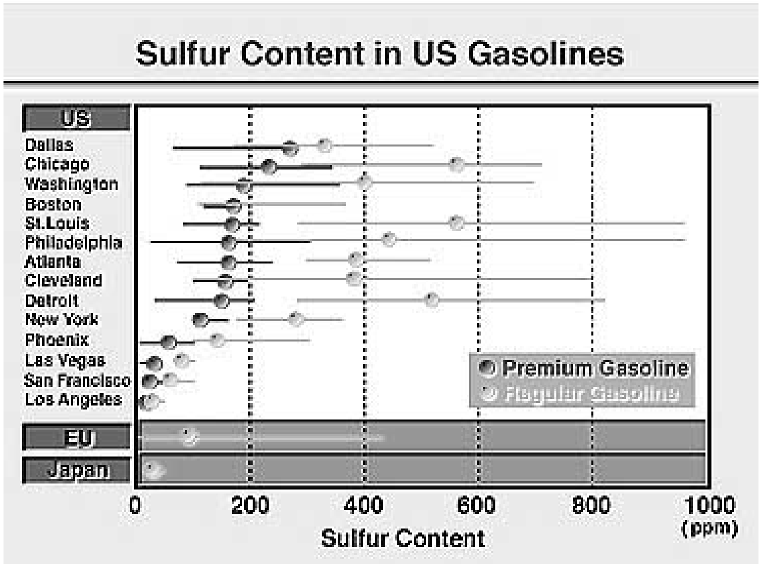


Figure 17. Sulfur content in US gasolines

The North American market has been slow to catch onto DFI technology because, with low fuel prices, emissions considerations far outweigh those of specific fuel consumption. Although recent developments in DFI technology have reduced the HC and soot emissions of DFI engines significantly, NOx emissions continue to be troubling because the overall lean mixture requires a catalytic converter, like that on Mitsubishi cars, with specific HC/NOx feedgas restrictions that do not conform with the North American 3-way closed-loop catalyst system [5]. Furthermore, the iridium based catalyst for lean burn cycles is highly sensitive to sulfur in fuel, and sulfur content in parts of the United States remains unacceptably high, as shown in Figure 17. However, Mitsubishi Motors has recently promised that their DFI engine should make it into the US market by 2000 with a newly unveiled catalyst [13]. Perhaps, one day, Orbital will be able to market its two-stroke versions here too, if the public can be convinced that previous problems with two-strokes have largely disappeared with DFI technology.

Conclusion

From a thermodynamics point of view, DFI certainly showed appreciable gains in reduced specific fuel consumption and increased efficiencies with negligible difference in mean effective pressures. These results can mainly be attributed to the reduction in fuel short-circuiting and increased volumetric efficiency. The reduction in wasted fuel makes DFI extremely beneficial for two-stroke engines. However, other potential gains such as charge cooling and the reduction of throttling losses need to be analyzed. Other parameters such as spark timing also need to be re-optimized with DFI as DFI appeared to advance peak pressures slightly. Unfortunately, the actual values reported may not be correct due to calibration errors, but the general favorable characteristics of DFI can be clearly seen.

Appendix

Acronyms, Abbreviations, Symbols, Nomenclature…

ϕ	fuel-air equivalence ratio
Π	charge purity
ρ	density
AFR	air-fuel ratio
BDC	bottom dead center
CE	charging efficiency
Cp	specific heat at constant pressure
DFI	direct fuel injection
DR	delivery ratio
EGR	exhaust gas recirculation
FAR	fuel-air ratio
GDI	gasoline direct injection
HC	hydrocarbons
IMEP	indicated mean effective pressure
ISFC	indicated specific fuel consumption
m	mass
NOx	oxides of nitrogen
P	power (gross)
rc	compression ratio
SR	scavenge ratio
TDC	top dead center
TE	trapping efficiency
V	volume
W	indicated work (gross)

Table 3. Table of acronyms, abbreviations, symbols, nomenclature…

Calibration Data for DFI Fuel Flow Meters

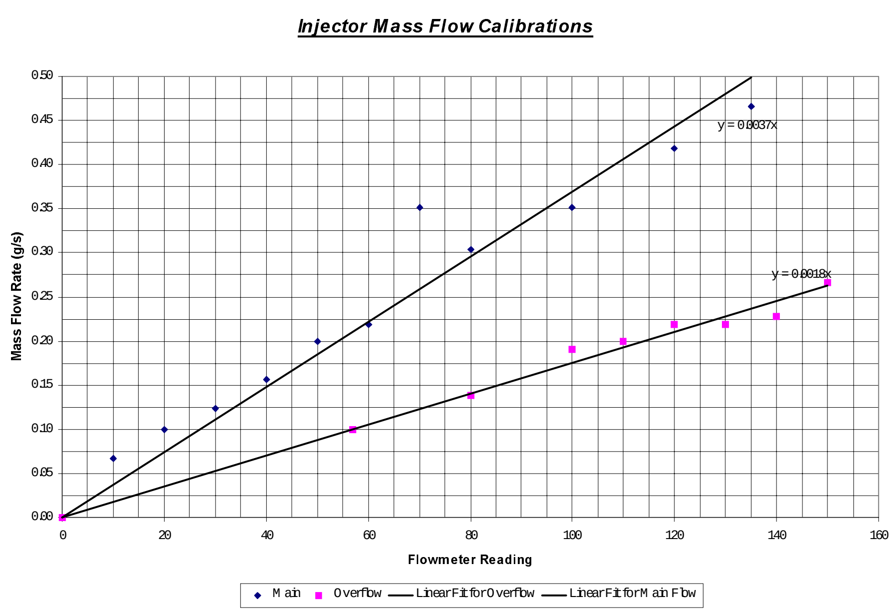
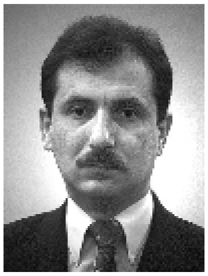


Figure 18. DFI fuel flow meter calibration chart

Acknowledgements



Boguslaw (Boggie) Gajdeczko, for being our lab mentor and for his help in troubleshooting whenever things went wrong (i.e., very often).

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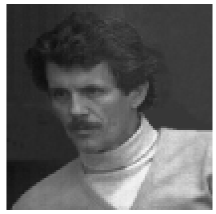
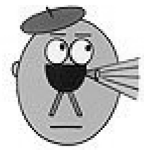


Professor Phil Felton, for sparking and injecting our interest in DFI and for explaining the key concepts of DFI.



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