Proceedings of ASME
ICERT 2003
Erie PA, September 2003

ICEF2003-758

DESIGN OF A COMPRESSION PRESSURIZED AIR BLAST DIRECT INJECTION SYSTEM FOR SMALL DISPLACEMENT TWO-STROKE ENGINES

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ABSTRACT

In two-stroke engines replacement of carburetors with direct fuel injection systems greatly reduces engine emissions and fuel consumption by eliminating fuel short-circuiting. Air-blast direct fuel injection using a dedicated air pump has been successfully applied to both two- and four-stroke engines. In this study we re-examine the design of a low cost compression pressurized direct injection system. This system uses gases extracted from the combustion chamber during the compression stroke to supply pressure for the air blast injection, thus eliminating the air pump [1,2]. Gases, predominantly scavenging air, are transferred to a mixing cavity from the combustion chamber via a small (5mm diameter) solenoid poppet valve as the piston rises during the compression stroke.

Proper functioning of the system requires careful optimization of the mixing cavity size and the blast valve timing to ensure adequate mixing cavity pressure and fuel atomization. To assist in the optimization of these design parameters a one-dimensional fluid dynamics model has been developed. Parameter sensitivity studies were carried out using the model to determine the optimum cavity size, blast valve timing, and fuel injection duration. These parameters were optimized over a wide range of engine speeds and throttle settings.

Results show that a mixing cavity pressure of 500 kPa is attainable over the range of 1000 to 6000 rpm, from closed throttle to wide open throttle (WOT) without cavity pressurization encroaching into the ignition regime. Fuel maps and valve timings are presented and results are contrasted with the carbureted case, showing improved fuel efficiency and emissions for the direct injection system. These data will be used in the design of a physical demonstration engine.

INTRODUCTION

In the US two-stroke engines are used primarily in weight sensitive hand-held applications, some off-road motorcycles and small boats. In developing countries, however, two-stroke engines may make up the bulk of the engine population of the small transport sector. In some cities in India, for example, two-strokes account for up to 65% of the overall vehicle population [3]. This high vehicle population coupled with the two-stroke engines tendency to have high hydrocarbon (HC) emissions due to fuel short circuiting results in very high levels of HC air pollution in urban areas in many developing countries. It is estimated that up to 70% of all HC emissions in India are due to small-displacement two-stroke transport engines [3].

A popular solution to the two-stroke emissions problem is direct fuel injection (DI). In DI systems the fuel is introduced into the combustion chamber only after all of the ports have been closed, thus eliminating fuel short-circuiting. The fundamental problem of DI is achieving the proper level of fuel atomization in the short time available between injection and ignition. There are a number of ways to achieve appropriate atomization. High-pressure direct injection systems (HPDI), for example use very high fuel pressures (5MPa) to atomize the fuel during injection. Another approach is to use a "dual fluid" injection where a gas (typically compressed air) is injected into the combustion chamber together with the fuel. In this "Air-Blast" direct injection (ABDI) the fuel is finely atomized in the shear layer between the injected gases and the cylinder gas.

Several DI systems have been developed for both two and four-stroke engines and are commercially available. Mitsubishi has applied HPDI to automobiles, Yamaha has employed HPDI in marine engines [4], and the Orbital ABDI system has been used on a number of different engine systems. Several manufacturers including Aprilia [5], and Piaggio [6] have applied ABDI systems to small displacement two-stroke motor scooters.

Given the growing concern over the global impact of air pollution it is probable that legislation in many countries will either restrict sales of new two-stroke vehicles (as has happened in Malaysia and India) or require some form of engine modification, most likely DI or the addition of a reduction catalyst, to meet increasingly strict emissions requirements. This may not, however, make an immediate impact on air quality, as the existing fleet of two-stroke vehicles in developing countries is likely to remain a significant source of pollution of decades to come. ABDI has successfully been applied as a retrofit to existing two-stroke snowmobile engines with great success [7]. We have therefore focused our efforts on modifying the ABDI system for application to small displacement two-stroke engines in developing countries.

NOMENCLATURE

CO	Carbon monoxide
HC	Hydrocarbons
DI	Direct Fuel Injection
HPDI	High Pressure Direct Injection
ABDI	Air Blast Direct Injection
CPDI	Compression Pressurized Direct Injection
EPC	Exhaust Port Closed

AIR BLAST DIRECT INJECTION

A typical Air Blast Direct Injection system is shown schematically in figure 1. A crankshaft driven air pump is supplying air to an “air rail” at a pressure of 600 kPa, and a fuel pump is providing a fuel rail pressure of 650 kPa. When the fuel injector is opened fuel is sprayed at a differential pressure of 50kPa into a small cavity at the back of the “blast valve” which is exposed to the air rail pressure. The blast valve is an electrically actuated solenoid valve separating the combustion chamber from the air rail. The fuel remains in relative large droplets and forms a film on the walls of the cavity. The blast valve is opened to the combustion chamber just after the exhaust port is closed (EPC) when the combustion chamber is still at a relatively low pressure (120 kPa). The fuel is then transported by the rush of air from the air rail into the combustion chamber, creating a fine plume of fuel vapor in the combustion chamber.

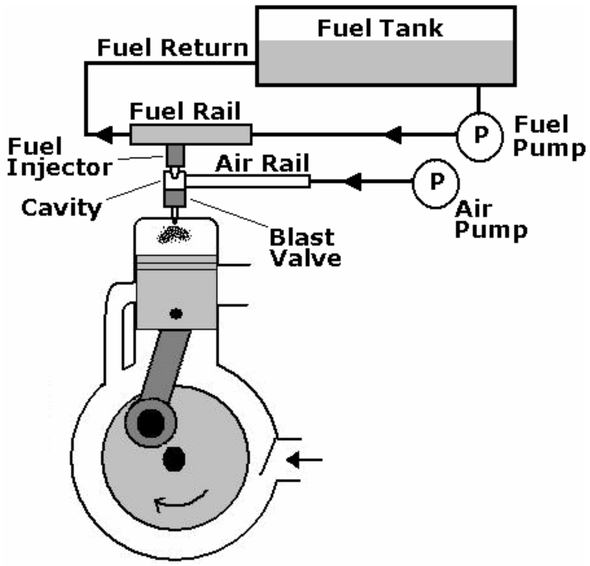

FIGURE 1 Schematic of an Air Blast Direct Injection System

Once the fuel enters the combustion chamber the blast valve is then closed before compression and ignition of the charge.

When included in the initial engine design the air pump may be incorporated into the overall engine design with relative ease. For a retrofit application, however, inclusion of such an air pump, with required pulleys, housing and pump mounting locations, might require significant modification to an existing small engine and interfere with vehicle operation. We would, therefore, like to find a less intrusive way to supply pressurized gas for the blast injection of the fuel to the combustion chamber.

COMPRESSION PRESSURIZED DIRECT INJECTION

If the blast valve is held open as the piston rises, pressure from the combustion chamber will eventually be greater then that of the air rail and gases will then flow from the combustion chamber back into the air rail. This technique is employed on some ABDI systems to help raise the air rail pressure rapidly during starting [8]. If the air rail is replaced by a small air pressure storage volume (the mixing cavity) we may rely solely on this compression pressurization of the mixing cavity thereby eliminating the need for the air pump. Such a Compression Pressurized direct injection (CPDI) system is shown schematically in figure 2.

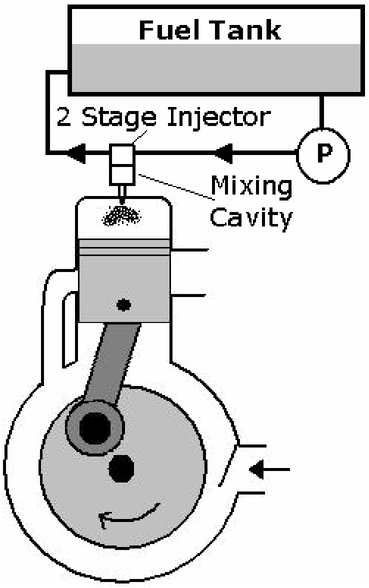

FIGURE 2 Schematic of a Compression Pressurized Direct Injection System

There are three phases to CPDI: Cavity pressurization, Fuel introduction, and Cavity discharge to the combustion chamber. The pressurization phase takes place during the compression stroke and is completed before ignition. The blast valve is held open, and the compressed gases from the combustion chamber are admitted into the mixing cavity, as in figure 3. Once the mixing cavity has attained the appropriate pressure, 500 kPa for example, the blast valve is closed, sealing the mixing cavity.

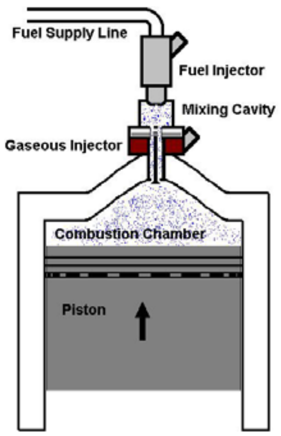

FIGURE 3 Pressurization of the mixing cavity

Since the blast valve is closed fuel may be injected into the mixing cavity while the engine is in the combustion, expansion or exhaust portion of the cycle. The fuel is injected directly at the back of the blast valve, as in figure 4. As the gases in the mixing cavity are still hot from compression, some fuel may vaporize, however much of it remains in droplets, or a liquid film on the back of the blast valve.

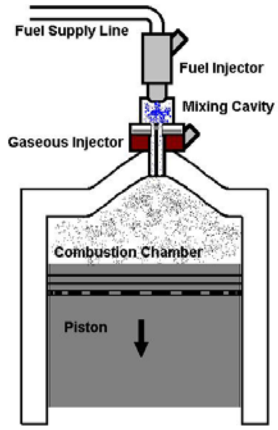

FIGURE 4 Injection of fuel into the mixing cavity

Once scavenging is complete and the exhaust port is closed the blast valve is again opened. As the pressure of the mixing cavity (approximately 500 kPa) is greater then the pressure of the combustion chamber (approximately 120 kPa) the contents of the mixing cavity are “blast” into the combustion chamber. The fuel droplets are finely atomized in the strong shear between the injected gases, and the combustion chamber gases, as in figure 5.

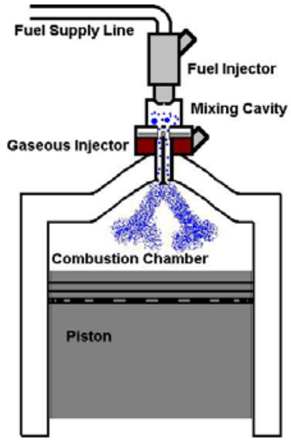

FIGURE 5 Blast injection of mixture

SYSTEM DESIGN

Several factors will be important to the success of a compression pressurized direct injection system: fuel injection timings will determine the air/fuel ratio, blast valve timing will determine the pressure of the mixing cavity, and the volume of the mixing cavity will have an effect on the mixture atomization, ultimate cavity pressure, and transient response of the engine. To facilitate in the design of our CPDI system a model was created using Ricardo's WAVE engine modeling software [9]. The basic parameters of the engine are shown in figure 6.

Engine Type	SI, 2-stroke, 1 cylinder
Bore	50.4 mm
Stroke	63 mm
Displacement	125 cc
Compression Ratio	12:1
Exhaust Port timings	+/- 85 CAD
Transfer Port timings	+/- 115 CAD
Fuel Injector	Orbital PFI
Blast Injector	Orbital Air Blast Valve

FIGURE 6 Basic engine parameters

As WAVE is a one-dimensional fluid dynamics model, all of the gaseous passageways are represented as ducts of the appropriate length, curvature and cross-sectional area, with special attention given to valves and flow discontinuities. WAVE also has the ability to calculating pressure rises from crank case compression, cylinder compression and has a combustion model which allows calculation of combustion pressures, gas composition and fuel consumption. The schematic diagram of our engine model is shown in figure 7. The large round icon in the center of the schematic represents the combustion chamber. Air flows into the engine from the left, passing a variable restriction representing the throttle labeled "Thrtl". The skeleton keyhole shaped icon in the center represents the crankcase.

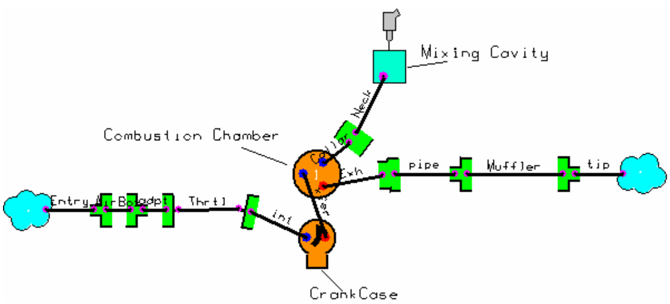

FIGURE 7 Schematic of 125cc 2-stroke CPDI engine model

The entry to the crankcase is modeled as a reed valve, and the exit is the transfer tube connecting it to the combustion chamber. The transfer port and the exhaust port are both modeled to give the appropriate opening area as a function of crank angle. A nominal "expansion chamber" muffler is added in the piping on the right which leads to the atmosphere at the far right. The square icon at the top indicates the mixing cavity which is connected to the combustion chamber via the blast valve. Protruding from the mixing cavity is the fuel injector. Both the fuel injector and the blast valve are modeled as electronically controlled solenoid valves, with appropriate flow profiles.

In addition to the CPDI version, a carbureted version of the same basic engine was also created for comparison purposes. The schematic, shown in figure 8, is identical except for the exclusion of the mixing cavity, and the placement of the fuel injector, which is located just before the crankcase. In this case the injector is modeled as a continuously flowing fuel supply, maintaining a consistent air to fuel ratio in the air stream.

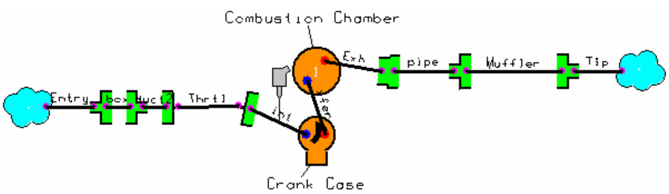

FIGURE 8 Carbureted 125cc two-stroke engine

SIMULATION RESULTS

Nominal conditions of wide-open throttle (WOT) were simulated at 4000 rpm for an equivalence ratio (Φ) of 1 and a mixing cavity volume (V_{cav}) of 1 cm^3. Figure 9 shows pressure traces from the combustion chamber and the mixing cavity as a function of crank angle over one cycle of the engine. Top dead center is located at the center of the diagram. The compression and combustion pressure rise in the cylinder are clearly evident, resulting in a maximum cylinder pressure of 3850 kPa.

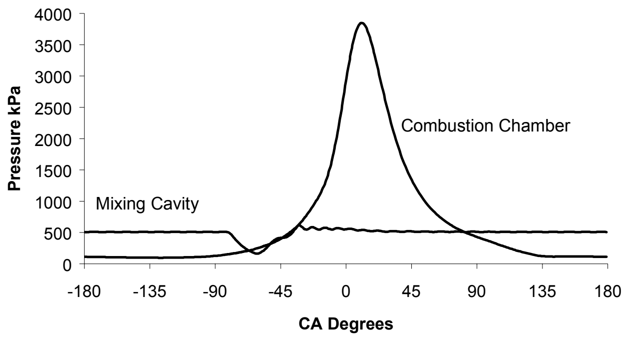

FIGURE 9 Combustion chamber and mixing cavity pressure versus crank angle, 4000 rpm, WOT, $\Phi = 1$

The blast valve is opened immediately after EPC (85 degrees BTDC), and held open continuously until BVC (blast valve closed) at 35 degrees BTDC. Initially after opening the pressure in the cavity drops as the contents are blast into the combustion chamber. After about 18 degrees, the pressure of the cavity and the combustion chamber are equal, but the pressure of the cavity continues to drop as momentum of the flow continues to carry fluid from the cavity. Eventually the flow reverses, as shown in figure 10, and gases from the combustion chamber pass back through the blast valve into the mixing cavity resulting in a cavity pressure of 560 kPa at BVC. Fuel is injected into the cavity at TDC well after the blast valve has been closed.

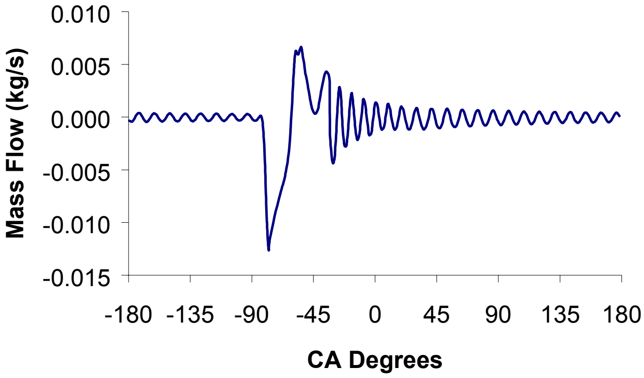

FIGURE 10 Mass flow from the combustion chamber into the mixing cavity versus crank angle, 4000 rpm, WOT, $\Phi = 1$

The total duration of the injection event is approximately 34 degrees (1.4 ms). The pressure of the cavity drops slightly after the fuel is added. This is due to cooling of the cavity gases by the fuel, figure 11, which is cooler upon injection (starting at TDC , or 0 degrees CA) and continues to evaporate during its residence in the cavity. When the mixture is finally injected on the subsequent cycle the mixing cavity pressure will have dropped to 510 kPa.

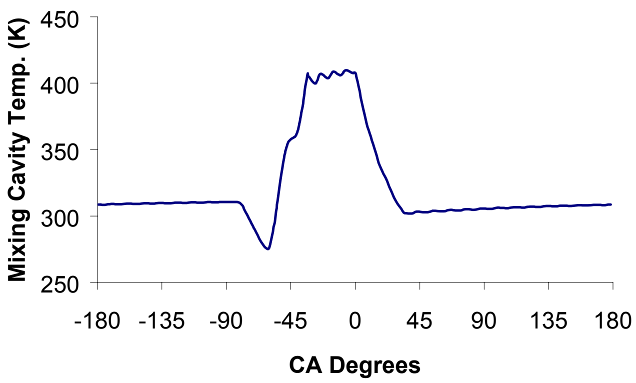

FIGURE 11 Mixing cavity temperature changes during blast, recharging and fuel injection

Blast valve and transfer and exhaust port timings and areas for this cycle are shown in figure 12. The blast valve has a much smaller effective area then either of the ports, and is seen to open just as the exhaust port is closed. Fuel injection (into the mixing cavity) is schematically indicated, the actual area of the fuel injector has been exaggerated for clarity.

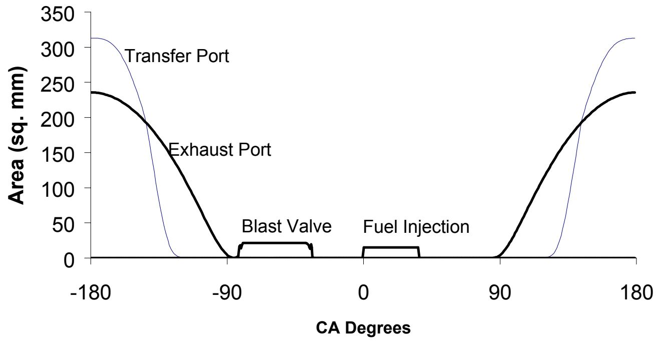

FIGURE 12 Port and Blast Valve timings and areas

In general the blast valve will be opened as soon as possible to inject the mixture with the largest pressure differential and to provide the fuel the maximum residency time for evaporation. The timing of the valve closing may be adjusted to vary the mixing cavity pressure. Figure 13 is a plot of the mixing cavity pressure immediately before injection versus BVC.

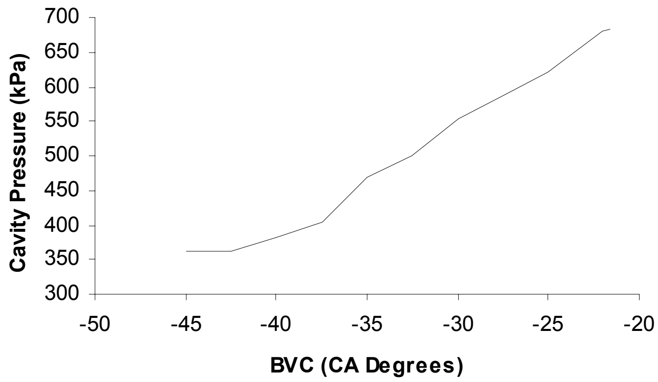

FIGURE 13 Mixing cavity pressure vs. BVC, 4000 rpm, 20% Throttle, $\Phi = 1$, $V_{cav} = 1\ cm^3$

For the maximum pressure we would like to keep the blast valve open as long as possible without encroaching on combustion. The blast valve is not designed to withstand the high temperatures associated with combustion. For this simulation we modeled combustion via a Wiebe function which resulted in a constant ignition timing of 21.1 degrees BTDC. The model requires all valves to be closed before the initiation of combustion for calculation purposes, resulting in a minimum BVC of 22 degrees BTDC, and limiting the maximum mixing cavity pressure to approximately 700 kPa. In reality the blast valve may actually be held open even after ignition as long as there is a sufficient ignition and flame propagation delay before the combustion gases reach the blast valve. This may allow us to attain even higher mixing cavity pressures. In our experience with the conversion of snowmobile engines to ABDI we have found air rail pressures of 460 kPa to be sufficient for good atomization of the fuel [7].

The affect of mixing cavity volume can be seen in figure 14. In general as the mixing cavity volume is increased, the mixing cavity pressure drops at constant BVC timing. This is due to the effective decrease in compression ratio when the blast valve is open. When the blast valve is open, the mixing cavity volume counts as additional clearance volume. The deviation of some of the points near V_{cav} = 1cm^3 is related with the acoustic resonance characteristics of the mixing cavity, discussed below (figure 18). Mixing cavity volume will also strongly affect the degree of fuel hang-up in the cavity, and transient response. Transient responses are not modeled here, as WAVE is a steady state simulation.

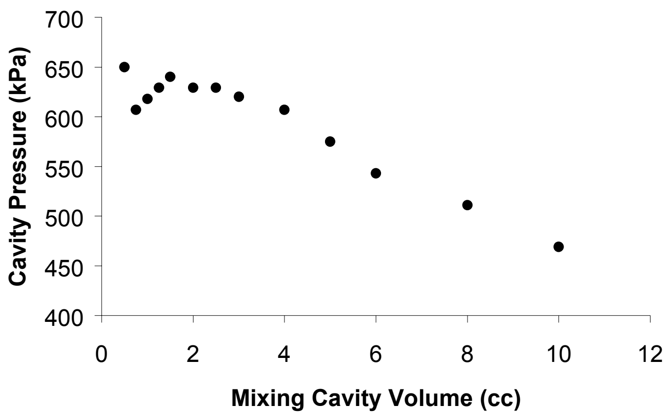

FIGURE 14 Mixing cavity pressure versus mixing cavity volume, 4000 rpm, 20% Throttle, Φ = 1, BVC = 25 deg. BTDC

The ratio of the mass of air injected to the mass of fuel injected is also important for atomization. Too little air will not provide sufficient flow for atomizing the fuel. In general systems are operated such that the injected air/fuel mass ratio is 0.2 or above [10]. Figure 15 is a plot of the A/F injected mass ratio as a function of mixing cavity volume. Under the conditions shown the minimum volume is on the order of 0.5 cm^3 and provides an injected air/fuel mass ratio of 1.8 well over the 0.2 limit.

As we have the capacity to inject significantly more air than is generally needed, this will allow us some flexibility in the injection design and valve timings. For example the air blast valve can be opened briefly to inject the fuel with an air/fuel mass ratio of 0.2. Later in the compression stroke when the combustion chamber pressure has exceeded the mixing cavity pressure the valve may be reopened to charge the mixing chamber. Among other things this will reduce the resistive heating of the solenoid in the injector.

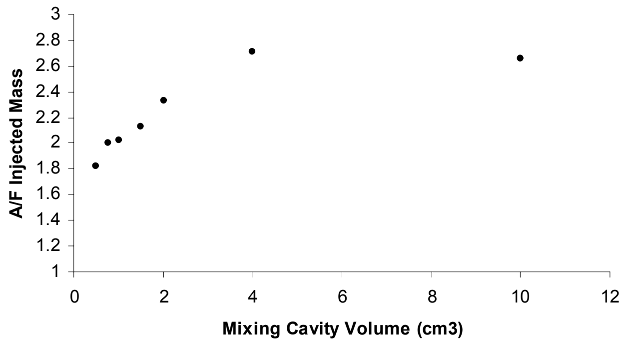

FIGURE 15 Injected A/F mass ratio versus mixing cavity volume, 4000 rpm, 20% Throttle, Φ = 1, BVC = 25 deg. BTDC

A mixing cavity of 1 cm^3 was chosen as a reasonable compromise between power and atomization quality based on injected air pressure. Given the modeling constraints on BVC timing, the mixing cavity pressure (immediately before blast injection) was targeted for 510 kPa. With these two parameters fixed, it was then possible to determine the appropriate injector timings required for a stoichiometric mixture and 510 kPa blast injection. Figure 16 shows the resulting fuel injection duration as a function of engine speed for various throttle settings.

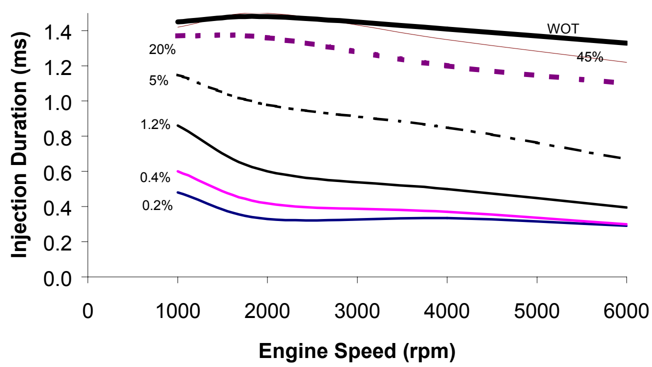

FIGURE 16 Fuel injection duration versus engine speed for various throttle settings, Φ = 1, P_{cav} = 510 kPa

In general the amount of fuel required at a given throttle position drops off as a function of rpm. This is directly related with the decrease in the volumetric efficiency of the engine with increasing rpm. At nearly closed throttle positions and high speeds the flow becomes choked at the throttle. As engine speed is further increased there is no more increase in the airflow therefore the fueling curves tend to go flat at higher engine speeds for small throttle settings.

Blast valve timings for the same conditions are presented in figure 17. The somewhat chaotic appearance is related to the wave dynamics in the mixing cavity.

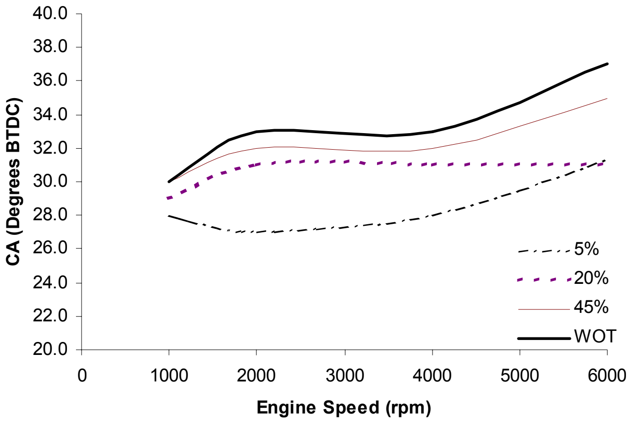

FIGURE 17 BVC versus engine speed for various throttle settings, $\Phi = 1$, $P_{cav} = 510$ kPa

The cavity will act as a resonance tube with pressure waves reflecting off the open end of the blast valve, and the closed back end of the mixing cavity. The frequency of the waves will depend on the density of the gases in the cavity. Figure 18 shows oscillations in the mixing cavity pressure caused by these cavity resonance waves. The phase of these waves around BVC can either increase or decrease the cavity pressure, and affect the required timing of BVC.

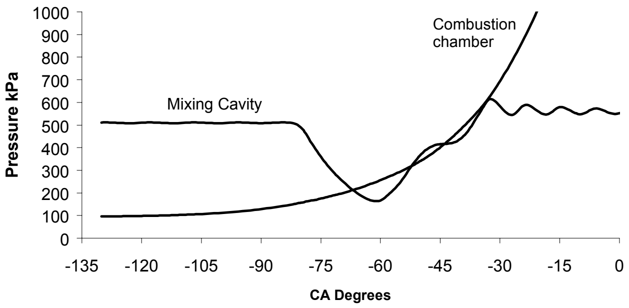

FIGURE 18 Mixing cavity pressure and Combustion cavity pressure versus crank angle near BVC

These data will form the basis of a preliminary fuel map and blast valve-timing map in the development of a physical model of the CPDI injection system.

The main motivation for this work is to reduce the hydrocarbon emissions by avoiding fuel short-circuiting. The magnitude of the problem is clearly visible in the results gained from the carbureted version of the same engine. Figure 19 shows model results for brake specific hydrocarbon emissions on the carbureted engine as a function of engine speed at various throttle positions. As the model assumes essentially complete combustion, virtually all of the HC emissions are from fuel short-circuiting.

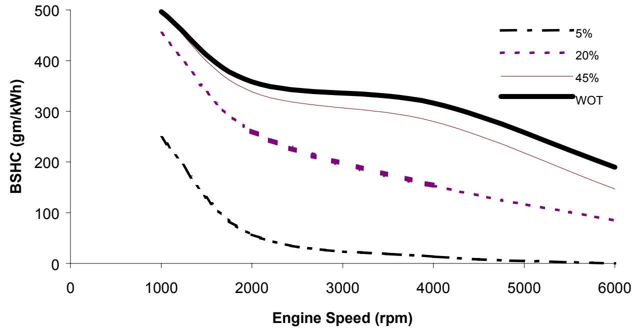

FIGURE 19 Brake specific hydrocarbon emissions from the short-circuiting in the carbureted model versus engine speed at various throttle settings

As up to 35% of the fuel in a carbureted engine is lost unburned, another advantage of DI systems is a significant reduction in the fuel consumption. Figure 20 shows the brake specific fuel consumption of the two models as a function of engine speed for various throttle settings. According to the model the CPDI system requires approximately 2/3 of the fuel used by the carbureted engine at WOT.

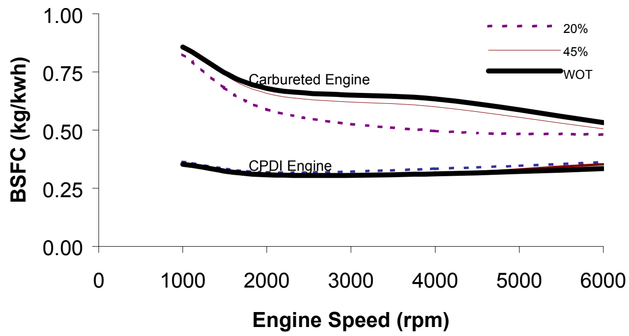

FIGURE 20 Brake specific fuel consumption for the carbureted engine (top) and the CPDI engine (bottom) versus engine speed for various throttle settings

Finally we must compare the power output from our modified engine with the carbureted base engine. Sequestering some of the compressed gases in the mixing cavity essentially reduces the effective compression ratio, reducing the engines power output. Figure 21 shows brake horsepower versus engine speed at various throttle settings. While the curves have very similar trends, it is notable that the CPDI engine is generating approximately 4% less power at 6000 rpm, WOT than the carbureted engine.

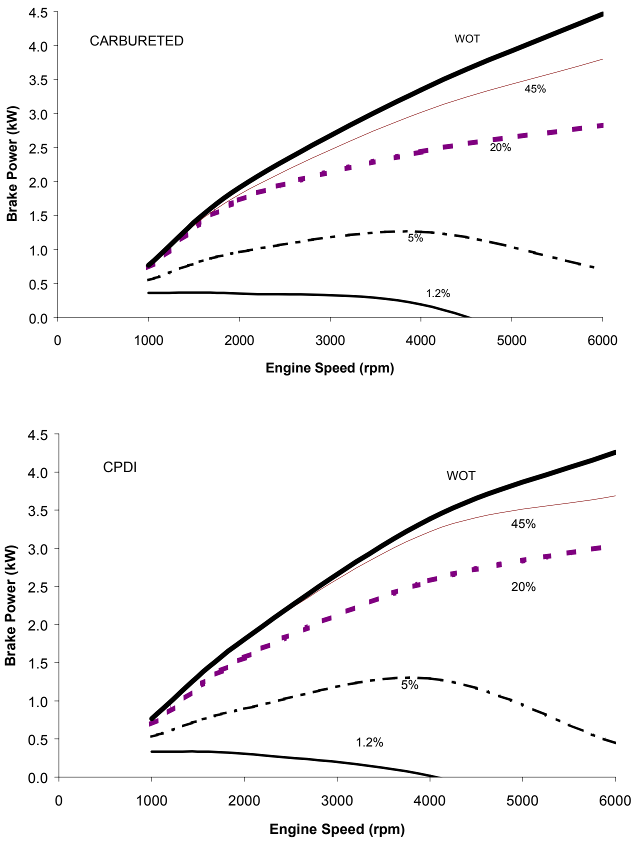

FIGURE 21 Engine power for the carbureted engine (top) and the CPDI engine (bottom) versus engine speed for various throttle settings

CONCLUSIONS

Two-stroke powered vehicles will continue to be a source of atmospheric pollution for years to come. While existing direct injection technologies may be applied to improve the emissions of new two-stroke vehicles they may be difficult to apply in retrofit applications to existing two-stroke engines. To reduce this difficulty we have designed a compression pressurized direct injection system which eliminates the need for a crank-driven air compressor, improving the feasibility of application for retrofit purposes. A one-dimensional fluid dynamics model was used in the design of our system, and results indicate viability of the concept over the range of interest. Data generated by the model will form the basis of electronic fuel injection timing maps. Our future efforts will be focused on production and analysis of a demonstration engine using the compression pressurized direct injection technique.

ACKNOWLEDGMENTS

I would like to thank Dr. Rudolf Stanglmaier for his wisdom and guidance in this area and Tim Bauer and Nathan Lorenz for their helpful input.

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