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Electronic Direct Fuel Injection (EDFI) for Small Two-Stroke Engines

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BKM, Inc.

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ABSTRACT

The benefits of direct cylinder fuel injection to the fuel economy and exhaust emissions of small spark ignited two-stroke engines is well known. The selection of a commercially viable fuel injection solution continues to receive evaluation and scrutiny by the engine manufacturers.

This paper describes the development and demonstration of an EDFI solution which is applicable to low cost and high production volume engines in several industries. The system is based on the “accumulator” fuel injection operating principle, which involves pressurizing fuel within an injection nozzle and subsequently releasing the pressurized fuel into the combustion chamber on command. This concept provides very short injection duration throughout the dynamic operating range of the engine as well as high injection frequency capability. The addition of full authority electronic control to the direct fuel injection system provides control flexibility and the opportunity for speed and load dependent calibration of the fuel injection event. The combination of unique components, control schemes and combustion systems has resulted in a flexible EDFI solution which is applicable to low cost two-stroke engines. Components include a single plunger pressurizing pump, an accumulator fuel injector and a precise, magnetically latching 2-way solenoid valve. The control scheme involves both fuel quantity and timing control by means of solenoid valve timing strategy. Skip-injection strategy for improved part load or idle combustion efficiency is optional. Combustion systems which have been evaluated include hollow cone spray, conventional and piston bowl combustion chambers and spray impingement techniques.

This paper describes the EDFI hardware design and operation, the application of system simulation software for design iteration, specific application designs and test results. The design applications include a 46cc handheld utility engine and a 50cc 2-wheeler engine. The 46cc utility engine results include comparisons of power, fuel consumption and exhaust emissions. Emissions results include:

1. Data at maximum power and idle
2. Comparison to the California Air Resources Board (CARB) for the year 2000
3. Environmental Protection Agency (EPA) marine engine regulations for 2006

The 50cc 2-wheeler data includes comparisons of power and fuel consumption to a standard (carbureted) engine. Data is compared at Wide Open Throttle (WOT) and part load conditions. In addition to achieving desired fuel consumption and hydrocarbon emission reductions, test results included power increase, particularly in the range from peak torque speed to rated power speed. It is concluded that the two-stroke spark ignited engine has the potential to continue a dominant role in low cost, high power density applications while meeting regulated exhaust emission standards.

INTRODUCTION

Due to the high power density and simple construction of the two-stroke cycle gasoline engine, it has been instrumental in the development of the 2-wheeler transportation market, the outboard marine engine market and the handheld power equipment industry. However, the unburned hydrocarbon exhaust emissions from conventional two-stroke engines are very high due to fuel loss during the scavenging process.

The high level of exhaust emissions and poor fuel economy typical of small piston ported two-stroke spark ignited engines mandates the need for improved fuel efficiency over the operating range of the engine. Direct, in-cylinder fuel injection has been demonstrated to significantly reduce unburned hydrocarbon emissions by timing the injection of fuel in such a way as to prevent the escape of unburned fuel from the exhaust port during the scavenging process.

Figure 1 illustrates the typical relationship between exhaust emissions and the air/fuel ratio, defined by the excess air factor lambda (λ). Lambda is the ratio between actual air/fuel ratio and stoichiometric air/fuel ratio. Stoichiometric air/fuel ratio is the theoretically correct ratio for complete combustion. Lambda less than 1.0 is a rich mixture and lambda greater than 1.0 is a lean mixture.

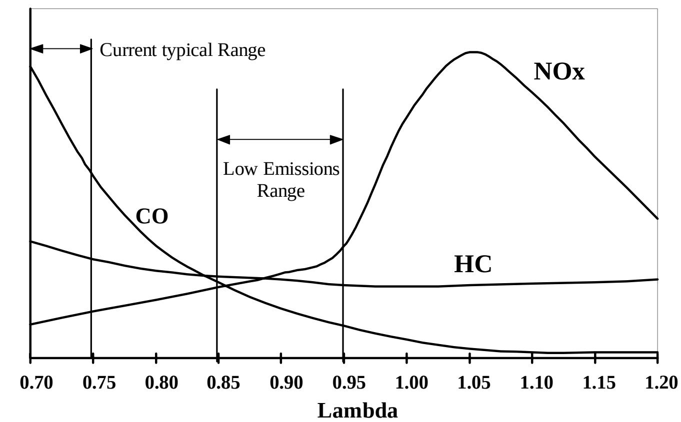

Figure 1. Influence of Excess Air Factor, Lambda, on Emissions

As shown in Figure 1, many contemporary two-stroke engines operate in the range of 0.70 to 0.75 lambda in order to optimize power and reduce combustion temperature. Unfortunately, this condition results in very high CO emissions as well as adding to the already high unburned HC emissions. The Oxides of Nitrogen (NOx) emissions however, are very low due to the low temperature of this rich combustion mixture.

Exhaust emissions can be minimized if lambda is very lean (greater than approximately 1.5). Such lean air/fuel ratios may be achievable using direct injection of fuel and charge stratification strategies. However, without additional air charge boosting, maximum engine power is reduced to an unacceptable level. Another strategy involves rich homogeneous mixture for full power, and lean or stratified mixture for part load. In this case, a smooth transition from stratified to homogeneous is required. In the range of lambda 0.85 to 0.95, emissions may be reduced without significant power loss. It has been demonstrated that the combination of in-cylinder fuel injection (reduced scavenge loss) and operation in this air/fuel ratio range (λ=0.85-0.95) results in significantly reduced emissions levels.

Compounding the basic two-stroke inefficiencies described above, it is normal for crankcase scavenged two-stroke engines to misfire at part load. Part load operation of conventional spark ignited engines involves reducing both the fuel flow and throttling the airflow through the engine in an attempt to maintain an ignitable air/fuel mixture. Misfire at part load in a two-stroke engine is caused by the presence of residual exhaust gas and degraded scavenge efficiency. This part load misfire contributes greatly to added unburned fuel emissions and increased fuel consumption. Direct in-cylinder injection alone does not solve this part load misfire problem.

Dynamic fueling range is another challenge for fuel injection equipment. The fuel injector must accommodate both the full load fueling rate, as well as the minimum fueling rate required to idle the engine. Therefore, the fuel system is required to provide precise well-atomized fuel sprays at these very small fuel deliveries, particularly as fuel consumption and emissions are reduced.

A hardware design and control method has been developed to operate low cost two-stroke spark ignited engines, which are fueled by electronically actuated accumulator type fuel injectors. This system is referred to as a Single Plunger System (SPS) to differentiate it from a related Common Rail System (CRS) previously development for multi-cylinder engines [1]. It is also now apparent that the SPS system may be desirable for multi-cylinder engines in order to achieve cylinder-to-cylinder fuel delivery trimming.

INJECTION SYSTEM DESIGN

INJECTOR DESIGN AND OPERATION – The electronic gasoline injector consists of three elements:

1. Solenoid valve
2. Accumulator
3. Nozzle tip

These elements are shown schematically in Figure 2. Note that in this context, the solenoid valve is considered to be part of the injector, but may by physically located near the pump. An overall scheme of the fuel system is illustrated in Figure 3.

Fuel pressure is provided to the injector by means of a single plunger pump, integrated into the engine design and timed to the engine cycle. Due to the compressibility of the fuel, the mass of fuel in the accumulator increases as the accumulator charge pressure increases. The accumulator is simply the internal volume of the injector body. When the accumulator pressure and inlet fuel pressure equalize at the maximum inlet pressure value, a check valve at the accumulator entrance closes, thereby trapping high pressure fuel within the injector. Fuel quantity is controlled by regulation of the charge pressure.

The solenoid valve is the interface between the injector and the electronic controller. The timing of solenoid closure, to allow pressure buildup, is referred to as “fill timing”. The fill pulse unlatches the solenoid, or closes it. The timing of the fill pulse determines the peak pressure developed by the pump, and therefore the quantity of fuel injected into the engine. By delaying the fill pulse, more of the pump stroke is vented back to the tank, so the injection pressure and quantity is less. Earlier fill times result in higher pressures and fuel flow.

Subsequent opening of the solenoid valve is referred to as “inject timing”, since valve opening results in needle lift. The vent pulse latches the solenoid open. The timing of the vent pulse determines the actual timing of injection into the engine. When venting, the hydraulic force on top of the injector needle is greatly reduced, while a check valve in the injector holds the peak pressure below the needle. When the pressure on top of the needle drops to a critical value, the higher pressure on the bottom of the needle provides adequate force to lift the needle. Injection stops when the pressure on the bottom of the needle is no-longer great enough to keep the needle open against the force of the injector spring.

Injection takes place until the pressure in the accumulator drops to the needle valve closing pressure, preset by spring force. Injection duration depends only on total nozzle flow area and the pressure drop from peak pressure to needle closing pressure. The hydraulic system includes the single plunger pump providing pressurized fuel to the injector, and a return line from the solenoid valve to the low-pressure side of the system. The injector charging event and control by pressure regulation are illustrated in Figure 4.

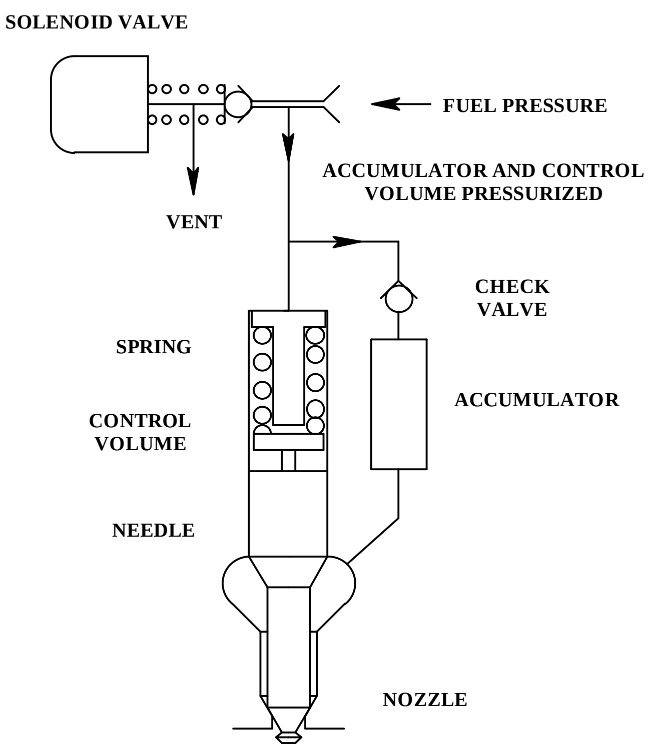

Figure 2. Injector Schematic

When configured for hollow cone spray, the SPS injector spray characteristic produces droplets on the order of 15 to 20 microns, which promotes good fuel and air mixing and consistent combustion. These small droplets are maintained by means of non-uniform velocity with respect to time, as illustrated in Figure 5. Due to the pressure decrease during injection, the first droplets leaving the nozzle have the fastest injection rate, followed by progressively slower rate droplets. This non-coalescing spray characteristic is referred to as "expanding cloud" fuel injection. The result is a finely atomized spray and good mixture preparation.

Advantages of this operating principle include the following:

- Simple, low cost hardware and design integration into the engine
- Good fuel delivery control resolution, since injector charge preparation occurs over a large portion of the engine cycle
- Cycle-by-Cycle control of fuel delivery and injection timing
- Intermittent cycle injection (skip-fire) provides part load combustion improvement if desired (not used in tests results presented in this report)
- Spray quality at starting and low engine speeds is maximized because injection pressure and duration are independent of engine speed
- Very short injection duration, less than 1 millisecond at full fuel delivery (dynamic range is independent of injection duration)
- A non-coalescing “expanding cloud” injection spray results from the decreasing injection rate during the injection event

SOLENOID VALVE – The unique solenoid valve developed for activating the fuel injection function uses a magnetic latching type solenoid. This valve has a two-pulse logic for activation and de-activation as compared to a conventional single pulse or pulse-width-modulated (PWM) logic. An initial pulse opens the valve. The valve remains open by means of residual magnetism. The valve is closed by spring force after de-magnetization initiated by a second pulse. The timing, separation, and individual pulse durations are independently variable. In the engine control logic, the initial pulse timing (fill timing) establishes fuel quantity by closing a fuel vent port during the desired portion of the fuel pump stroke. The second pulse timing (vent timing) determines the fuel injection timing by venting the fuel pressure from the top of the fuel injection needle as outlined above. It is not necessary to apply electrical current between these two events, due to the magnetic latching characteristic, thereby eliminating unnecessary electrical power consumption.

The solenoid valve complies with the following specification:

Type	-	2-way normally closed
Rated pressure	-	14 MPa (2,000 psi)
Minimum Flow Area	-	0.5 mm^2
Function	-	Accumulator injector vent
Liquid compatibility	-	Gasoline
Life expectancy	-	4.0 x 10^8 cycles
Timing precision	-	± 25 microseconds
Response time	-	Less than 2.0 milliseconds
Operating frequency	-	10 to 250 Hz
Voltage	-	12 VDC

Power requirement	-	Less than 6 watts at 5 % duty cycle
	-	Less than 30 watts at 30% duty cycle
Storage temperature	-	-50 to + 105 °C
Operating temperature	-	-40 to + 200 °C
Mechanical shock	-	100 g
Mechanical vibration	-	50 g at 5 to 200 Hz
Humidity	-	100 %

The solenoid valve uses spring force for closing, providing for an over-pressure safety relief function.

PUMP DESIGN – As illustrated in Figure 3, the pump assembly consists of a reciprocating plunger, driven by an eccentric cam, which is integrated with the engine crankshaft. Inlet and outlet check valves provide fuel supply and priming aid respectively. Fuel which is vented by the solenoid valve passes through a pressure relief valve designed to maintain sufficient system back pressure to prevent fuel percolation.

As shown, the solenoid may be placed in the pump housing as part of an integrated package. An alternate location, integrated with the fuel injector, has also been designed.

The spill port illustrated in Figure 3 provides for hydromechanical venting and initiation of injection, late in the engine cycle, if electricity is not available. This feature provides for “limp home” capability when electrical power generation or battery power is interrupted, or for engine starting if electrical power is not available during crank starting.

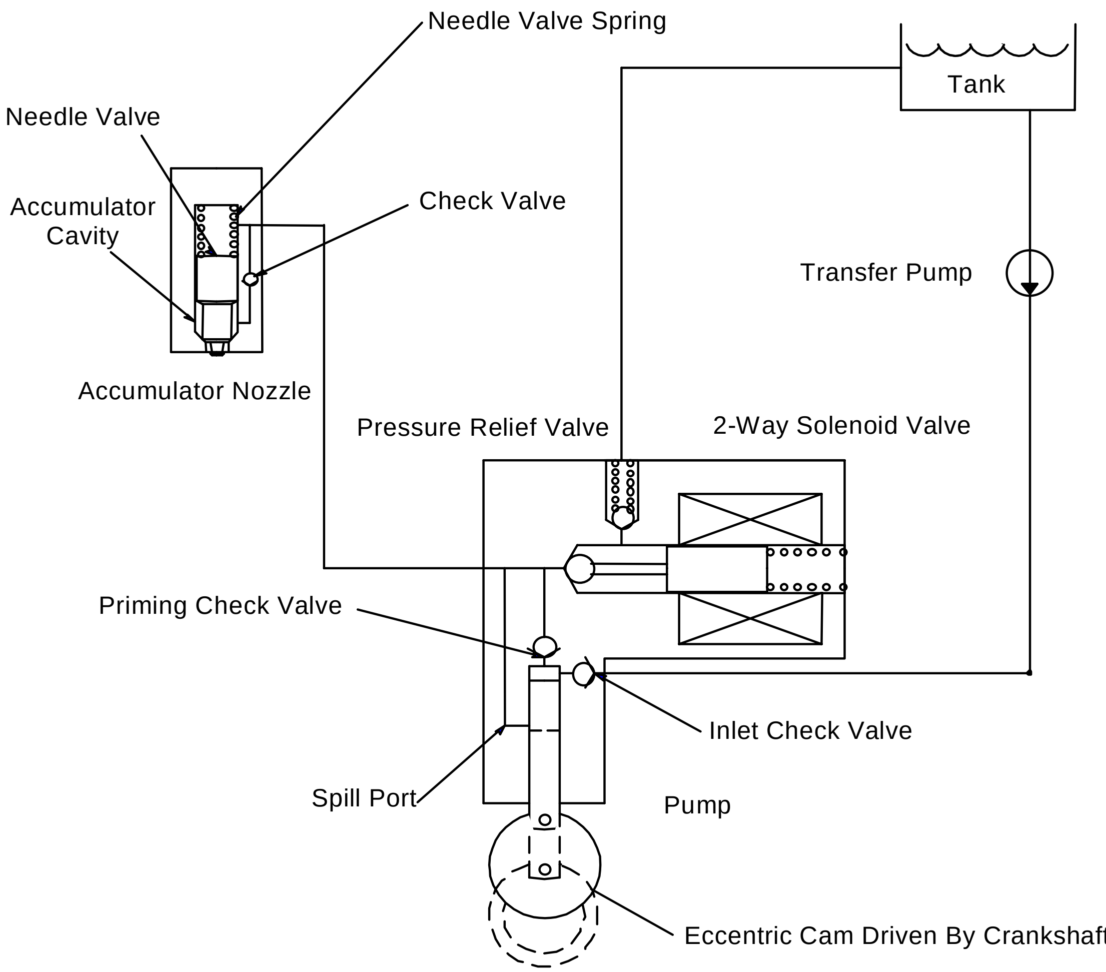

Figure 3. Single Plunger System (SPS) Configuration

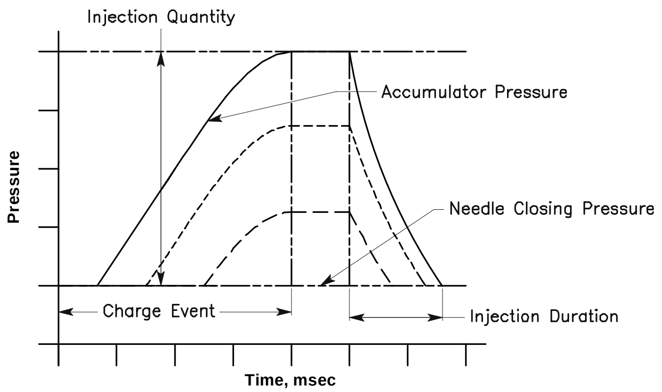

Figure 4. Injector Charging and Pressure Metering

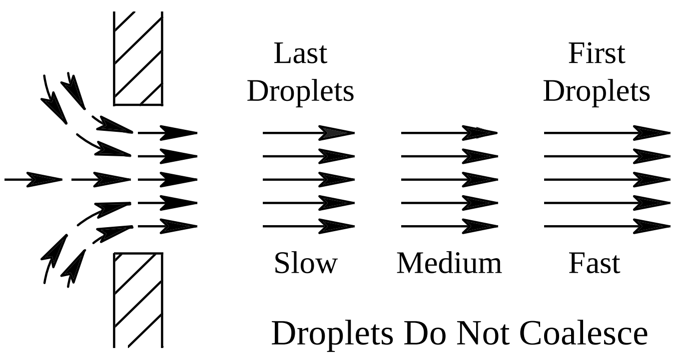

Figure 5. Droplet Velocity

ELECTRONIC CONTROL UNIT (ECU)

Prototype Fuel System Operation – In order to accelerate availability of a development controller and to concentrate on software calibration and validation, a decision was made to conduct initial testing with an existing, commercial multi-cylinder engine controller designed specifically for control flexibility and laboratory calibration. The unit selected for this activity and used for the test results included in this report is manufactured by MOTEC. In parallel with the calibration development using this large ECU, a prototype small engine ECU design program was initiated.

Another aspect of the overall ECU design was the driver circuit for the two-way latching solenoid valve developed for the fuel injector actuation. A prototype version of this driver circuit was developed for operation of the solenoid valve. It was also used:

a. for optimization of component sizing, such as a capacitor used to magnetize and de-magnetize the latching solenoid
b. to develop turn-on and turn-off dwell times for minimum energy consumption
c. to establish compatibility with “alternator only” variable voltage electrical supply.

ECU Design – The production intent ECU design addresses the following criteria and functional requirements. Figure 6 shows the overall control scheme for inputs and outputs.

Inputs

Engine Speed and Crank Position Sensor - Currently using an optical sensor reading a 36 minus 2 blade mask, which results in a series of 5 volt square waves. This provides both a high resolution position signal, and a once per revolution discontinuity using one mask and sensor. The total number of blades has not been optimized.

Throttle Position Sensor - Analog 0 to 5 volt sensor.

Temperature Sensor - Analog thermistor signal, best location to achieve both intake air and fuel correlation (engine specific).

Barometric Pressure Sensor - Analog signal, packaged internal to the ECU.

Supply Voltage – Input directly to ECU.

Outputs

Injection:

- Fill pulse duration = .7 ms at 13 volts
- Vent pulse duration = 2.0 ms at 13 volts
- Fill Timing - Leading edge of signal varies from 60 Degrees BTDC to 60 Degrees ATDC
- Vent Timing - Leading edge of signal varies from 120 Degrees BTDC to 240 Degrees BTDC

Ignition:

- Timing -Trailing edge of signal varies from 0 Degrees BTDC to 60 Degrees BTDC
- Dwell - Depends on ignition system and voltage.

Internal Logic – The ECU control of the fuel system is accomplished solely through the single solenoid valve. This one valve is used both as a pressure regulator and to trigger injection into the cylinder. The solenoid is a magnetically latching solenoid, so one pulse latches the valve open, then another pulse is used to de-latch, or close the valve. If the pulses to the solenoid are too short the valve will not latch or de-latch. If the pulses are too long electrical energy is wasted, or eventually components are overheated. Figure 7 illustrates an example of the system event timings, including solenoid function.

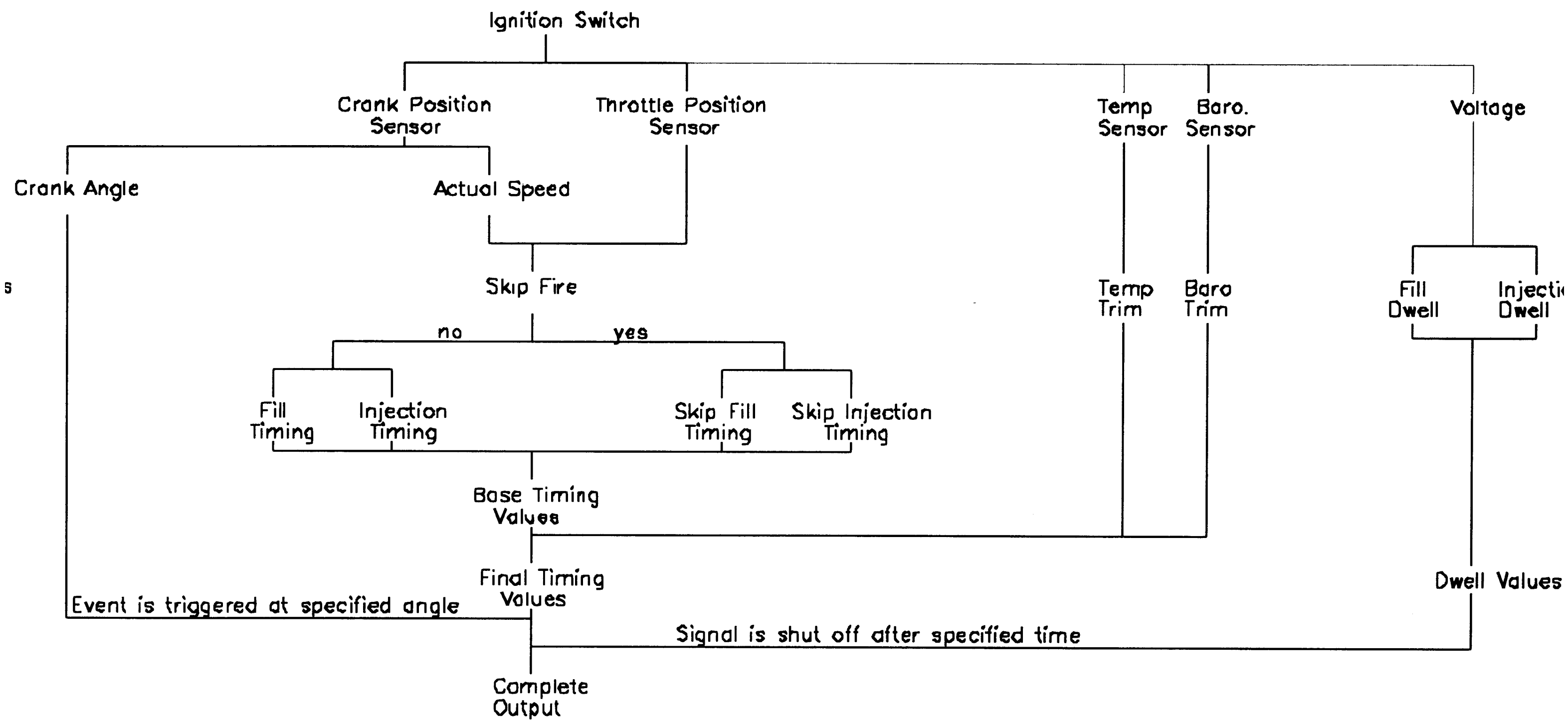

Figure 6. Control Logic

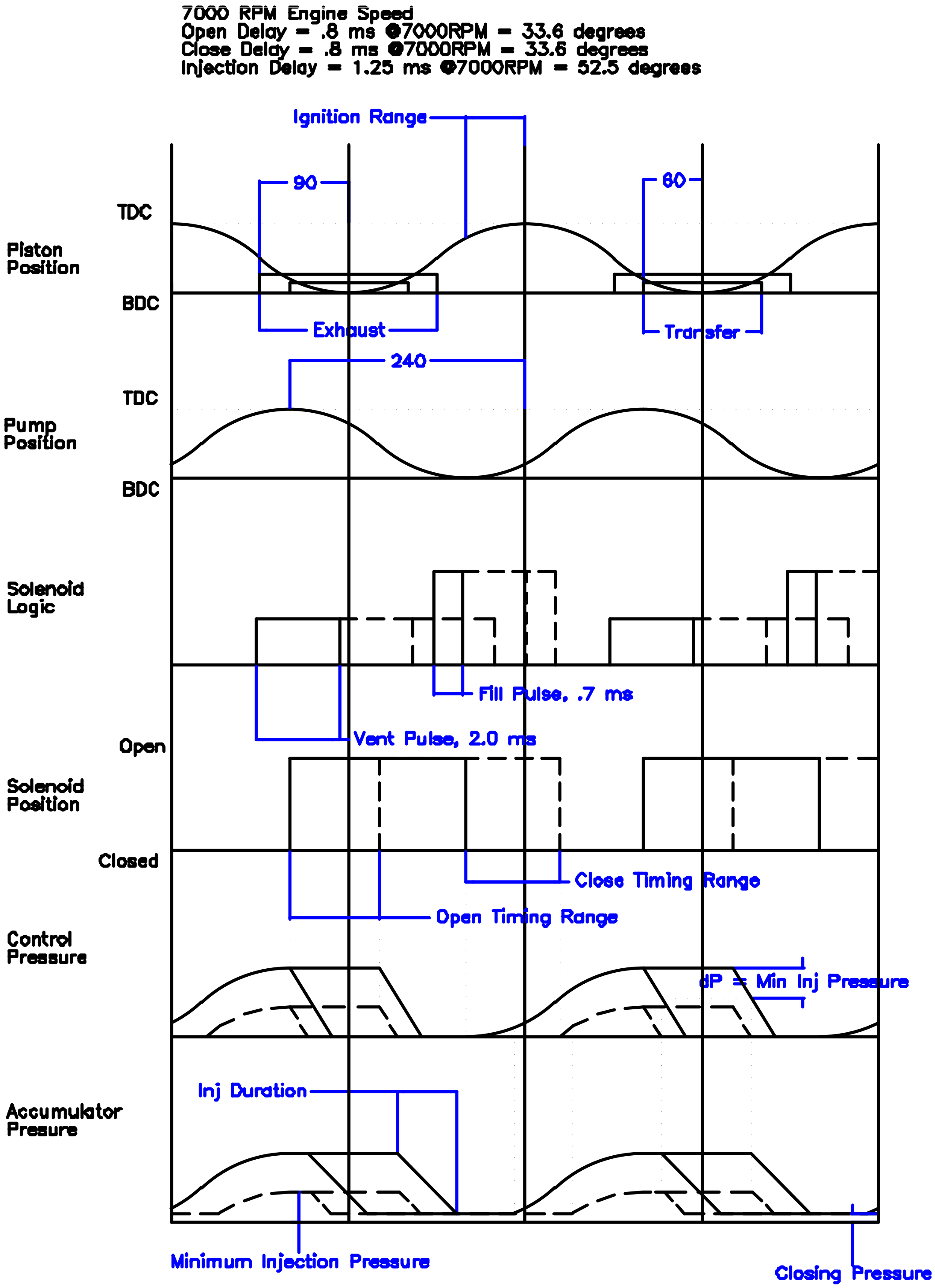

Figure 7. Timing Example

The ECU calculates the solenoid logic and optional ignition logic based on the engine rpm and throttle position. A calibration map or table is used for fill timing, vent timing, and ignition timing versus throttle position and rpm. Fill timing is corrected for barometric pressure and temperature to maintain a desired air/fuel ratio. The outputs between map sites are calculated by linear regression.

The logic pulse widths are dependent only on supply voltage. Otherwise, they are held to fixed values for each rpm and throttle position.

Skip Fire – Injection on intermittent cycles, or “skip fire”, may be used to reduce idle or part load misfire. Skip fire is achieved by omitting fill pulses for the cycles to be skipped. This causes the solenoid to never de-latch, or close, so all of the pump action is vented to the tank. The vent pulse can be left untouched. This ensures that the solenoid stays latched and open, but does not trigger injection because there is insufficient pressure to allow an injection to take place. For the skip fire operation there is another set of maps, fill, vent, and ignition timing that the ECU uses to calculate the outputs. These additional maps are required due to the improved scavenging efficiency during skip-fire operation. An additional map with throttle position and engine rpm is used to determine when the engine will operate as a normal two-stroke and when it will operate on skip fire.

APPLICATION DESIGN

The prototype fuel injection and ancillary systems design was completed for a 46cc utility engine. Key views extracted from the design layout are shown in Figures 8 thru 10. An additional adaptation was created for a 50cc 2-wheeler engine. The rationale for this additional engine was the need to validate the system performance for applications requiring more complex transient operation and other issues relative to drivability. Views from the 50cc engine design layout are shown in Figure 11.

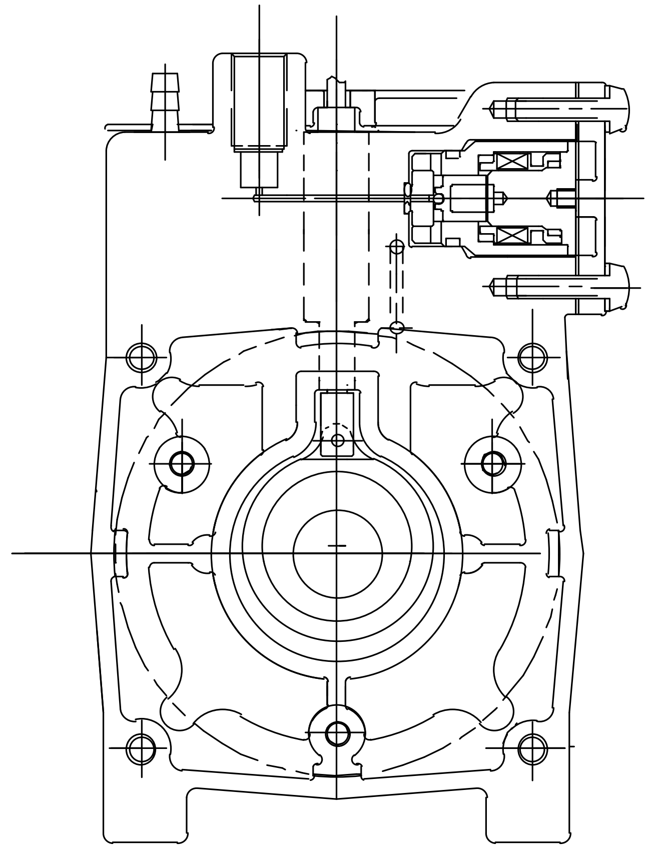

Figure 8. Utility Engine, Cross-section Through Cam and Solenoid Valve

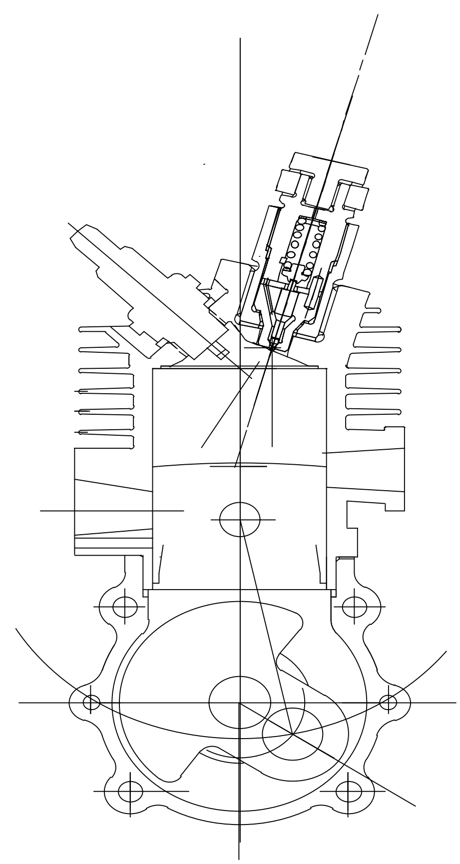

Figure 9. Utility Engine, Cross-Section Through Engine and Injector

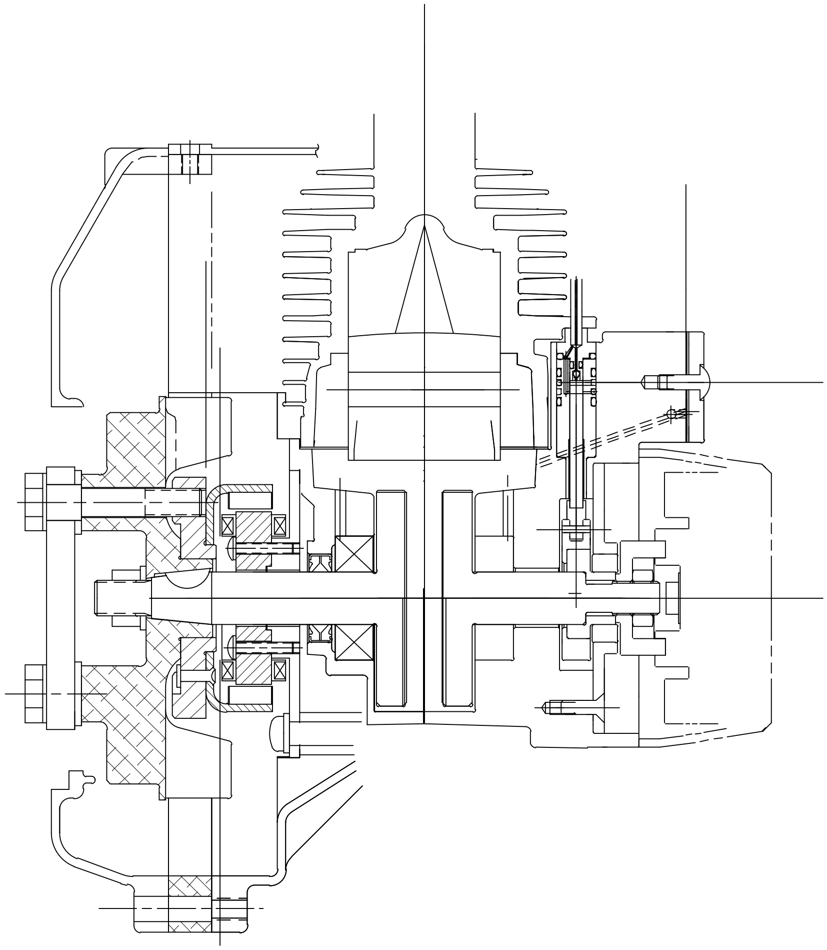

Figure 10. Utility Engine, Cross Section Through Crankcase, Flywheel, Alternator and Pump

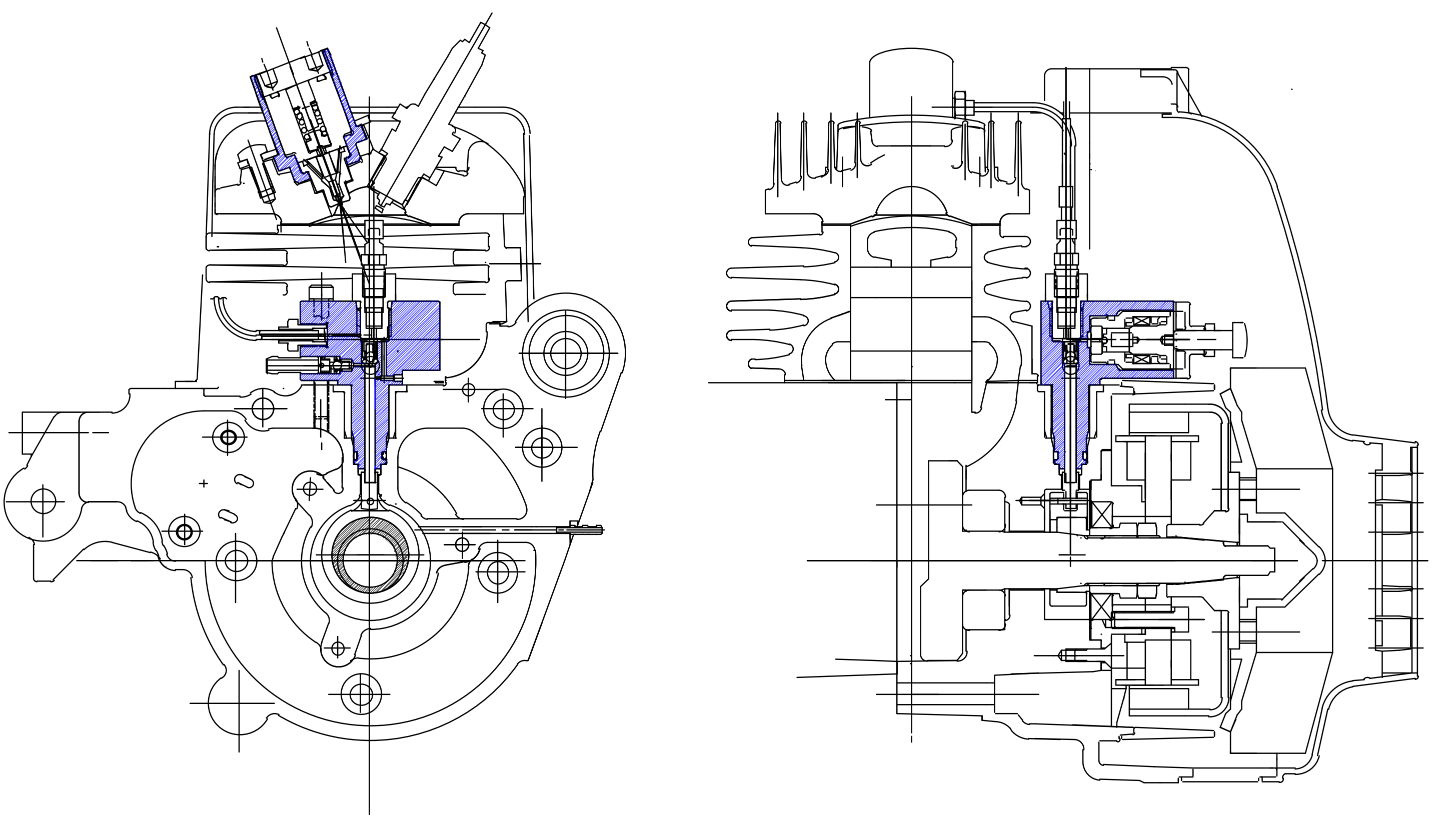

Figure 11. 2-Wheeler Engine Design

Parametric Study: Injection Timing and Fuel Metering

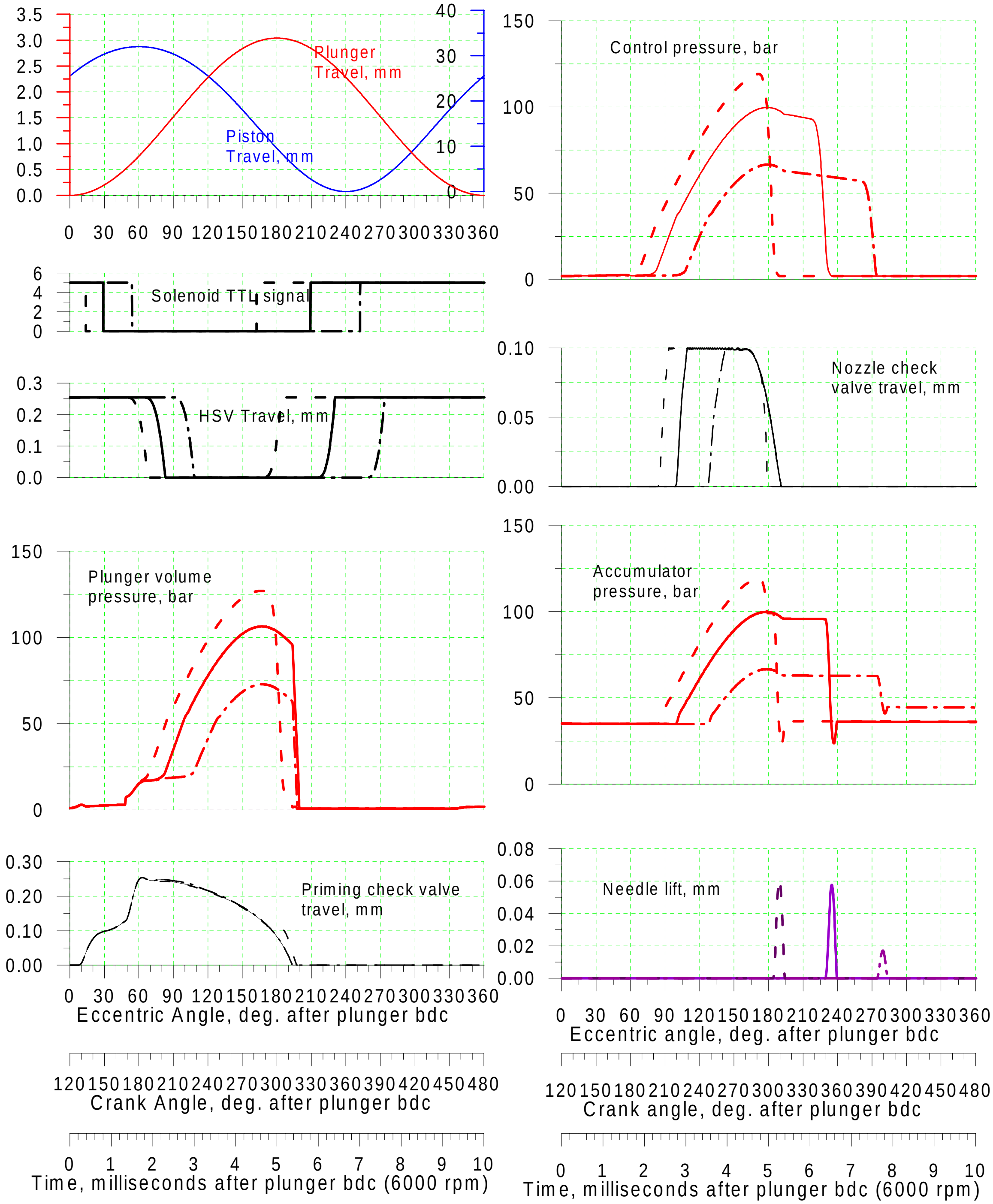

Figure 12. Parametric Study: Injection Timing and Fuel Metering

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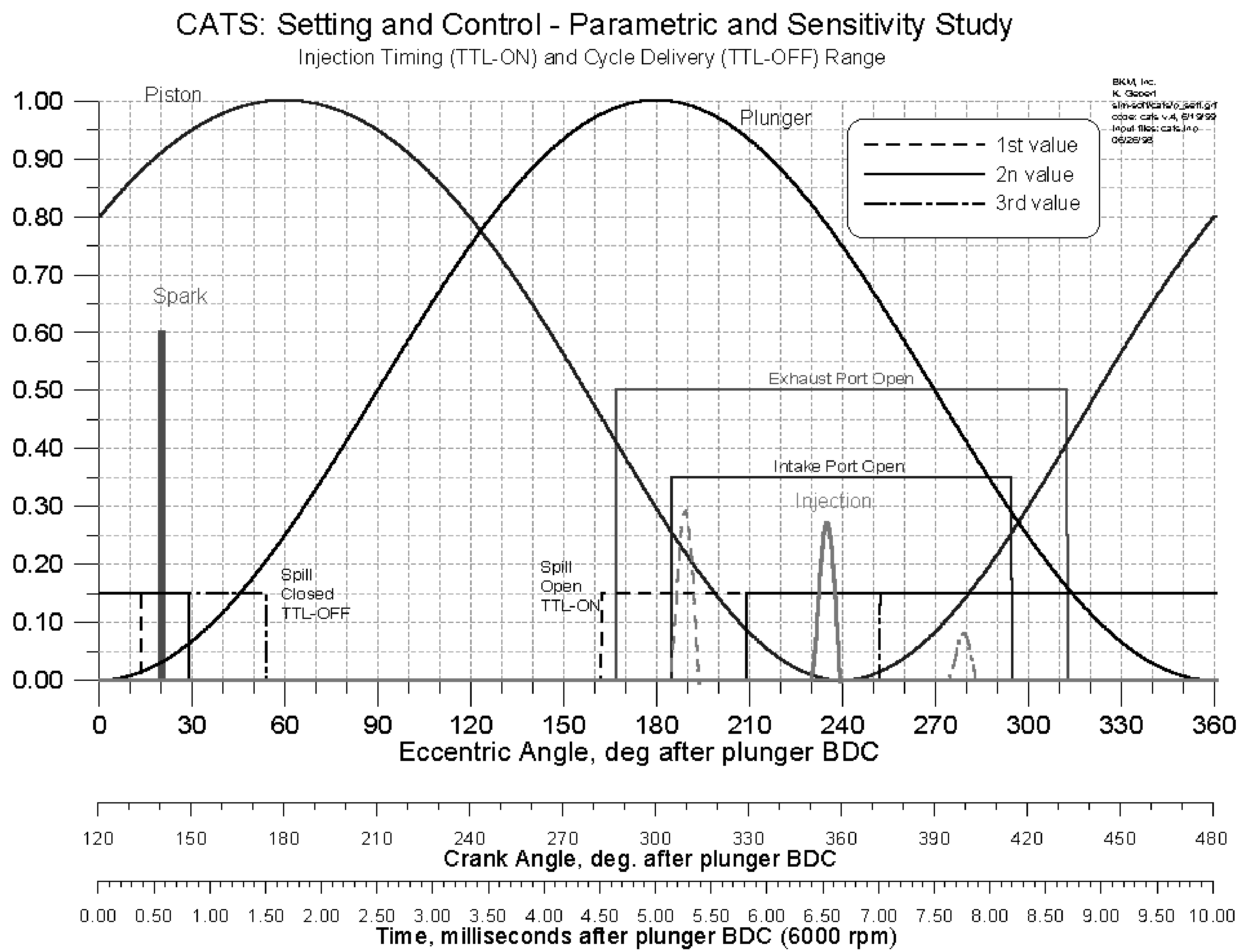

Figure 13. Setting and Control: Parametric and Sensitivity Study

A dynamic simulation program was created to aid in the determination of design arrangement and dimensions for the specific engine to be tested. This FORTRAN based program provides time based graphical results of pressures, motions and flow rates for critical areas within the system. This tool may be used for predicting injector performance and for identifying potential imperfections in system design or operation prior to fabrication and testing. Sample outputs from this program are shown in Figures 12 and 13 for a fuel delivery and injection timing sensitivity study.

This one-dimensional mathematical model describes the flow inside the high-pressure lines and passages inside the injector. The flow is considered transient, isentropic and compressible. Two source equations are applied: conservation of momentum and conservation of mass. This approach distinguishes between forward and reflected pressure waves and their tracking along the line. Pressure loss in the line is modeled as Darcy-Weisbach hydraulic friction or, optionally, is based on an experimentally obtained correlation. Pressure histories inside concentrated volumes are obtained from mass conservation. Short high pressure lines are considered as concentrated volumes.

Fuel leakage throughout the fuel system is an essential phenomena which effects system performance. Leakage through narrow annular gaps such as at the pump plunger, from control volume back to plunger volume and from plunger volume to crankcase, has been modeled. Leakage across the needle stem is also included. The dynamics of moving parts are obtained by solving the equations based on the equilibrium of forces acting on each part. In general, the forces include: inertia, friction, spring forces, pressure forces, and magnetic forces. Friction is modeled as hydraulic friction and springs are considered as linear elements. Density, bulk modulus, viscosity, sonic velocity, and surface tension of the fuel are considered as pressure and temperature dependent. The actual values are updated at each computational time step. The onset of fuel cavitation inside the plunger volume is calculated and depends on fuel temperature and supply valve opening pressure. Fluid boundaries are considered to be infinitely rigid. Motion of mechanical components is assumed non-oscillatory except for the needle valve, where an option for oscillatory contact was provided. The nozzle flow discharge coefficient is assumed to be needle lift dependent. A lumped parameter model, in which the magnetic field is represented by an equivalent magnetic circuit, used to calculate the solenoid magnetic force at the control valve. The electric circuit (control valve driver) is not modeled. The driver circuit current history is instead obtained experimentally and approximated as linear segments, in this case where residual magnetism is used as a latching force. The equations are solved using modified 4^{th} order Runge-Kutta procedures with variable time step.

DEVELOPMENT TESTING

Several fuel spray and combustion bowl arrangements were evaluated in order to achieve best fuel/air mixing and fuel distribution within the combustion chamber. In addition, a unique combustion system arrangement developed by Shigeru Onishi, President, Nippon Clean Engine Laboratory Co., was tried with very good results. This system is called Impinging Fuel Jet Diffusion (OSKA) and is found in the literature regarding alternative fuel experiments as well as diesel engines [2]. This OSKA system provides fast mixture preparation, particularly when combined with strong squish flow, and using a single hole nozzle. Application of this unique system to small two-stroke gasoline engines is new. Using this technique combined with the injection rate characteristics of the accumulator type injector appears desirable. A comparison of the impinging spray and hollow cone designs are shown in Figures 14 and 15.

Hardware iterations evaluated during this development test period include the following variables:

- Injector spray configuration
  - 30° included angle hollow cone
  - Pencil stream (impinging spray, OSKA)
- Injector accumulator volume
- Injector needle closing pressure (minimum spray pressure)
- Ignition
  - Standard CDI ignition system and timing
  - High Energy Inductive (HEI) ignition system with variable timing
- Piston
  - Standard
  - Inserted bowl
  - Target for OSKA spray stream (various designs)
- Combustion Chamber
  - Standard
  - Central hemispherical
- Injector location
  - Central
  - Offset (various positions)
- Spark plug location
  - Central
  - Offset (various positions)

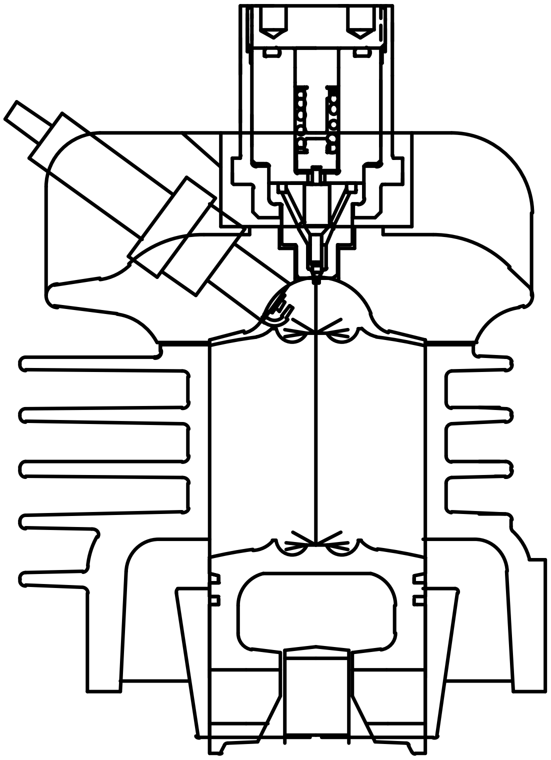

Figure 14. Impinging Spray (OSKA)

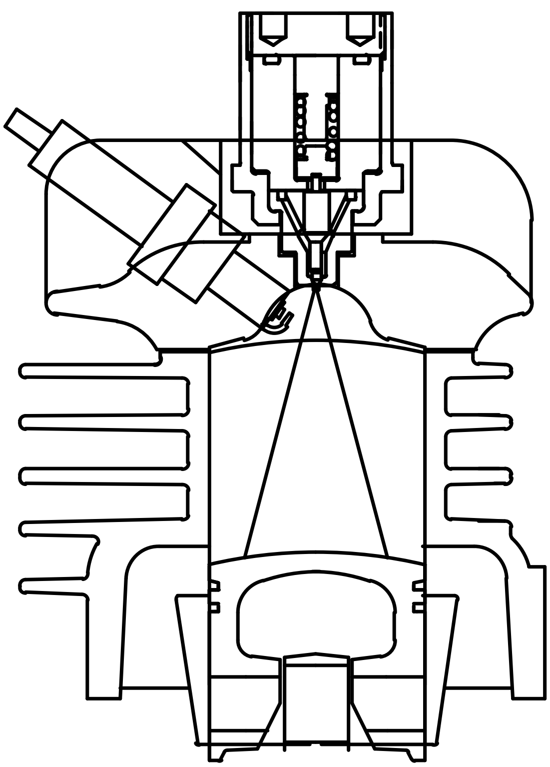

Figure 15. Hollow Cone Spray

TEST RESULTS – The 46cc utility engine results include comparisons of power, fuel consumption and exhaust emissions. The 46cc engine emissions data was recorded by University of California, Riverside, College of Engineering, Center for Environmental Research and Technology (CE-CERT). Hydrocarbon data was recorded using a Flame Ionization Detector (FID). Emission results include data at maximum power, comparison to the California Air Resources Board (CARB) Tier II regulations for handheld utility engines and comparison to marine engine regulations set by the U.S. Environmental Protection Agency (EPA) for the year 2006. The results for the utility engine and fuel system configuration listed below are illustrated in Figures 16 through 21.

The configuration of the utility engine during these tests was as follows:

- Injector spray configuration: Pencil stream
- Injector accumulator volume: 208 cu. mm.
- Injector needle closing pressure: 100 bar (est.)
- Ignition: HEI with optimized timing
- Piston: Inserted bowl
- Combustion Chamber: Central hemispherical
- Injector location: Central
- Spark plug location: Angled, intake port side

During the utility engine test program, oscilloscope recordings were made of electronic signals to the solenoid valve driver circuit and the pressure history between the pump and injector. These dynamic events were compared to the calculated dynamic simulation with very good correlation. The recorded events and simulation were compared at three different operating points (idle, part load, full load).

The 50cc 2-wheeler data shown in Figures 20 through 25 includes comparisons of performance and fuel consumption with a standard (carbureted) engine. Data is compared at Wide Open Throttle (WOT) and part load conditions.

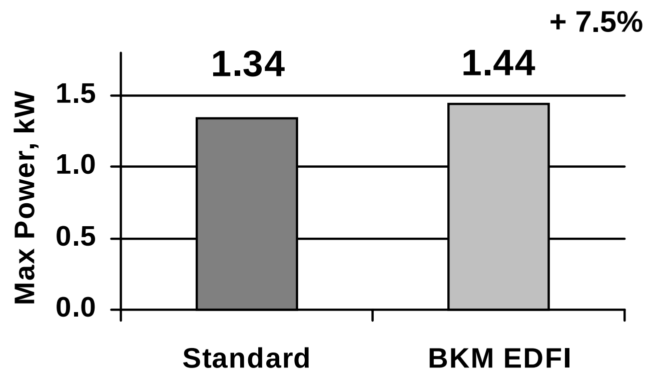

Figure 16. 46cc Utility Engine Maximum Power Comparison at 7,000 RPM

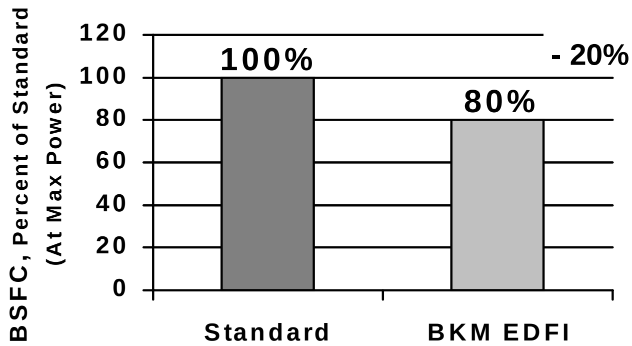

Figure 17. 46cc Utility Engine Brake Specific Fuel Consumption (BSFC) Comparison At Maximum Power, 7,000 RPM

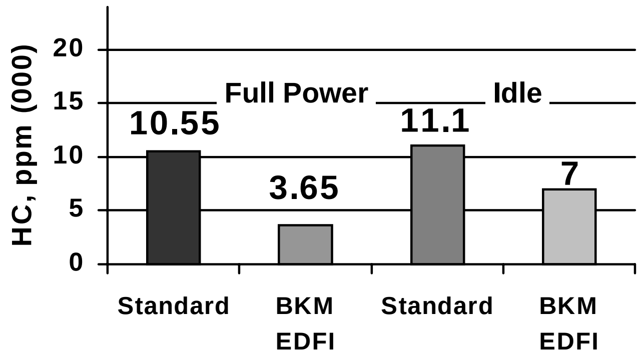

Figure 20. 46cc Utility Engine Parts Per Million Hydrocarbon Emissions Comparison At Full Power And Idle

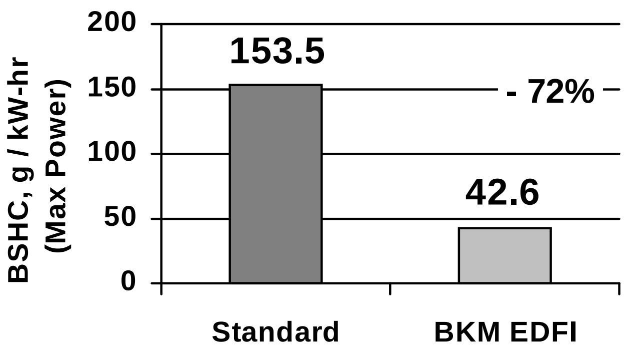

Figure 18. 46cc Utility Engine Brake Specific Hydrocarbon Emissions (BSHC) Comparison At Maximum Power, 7,000 RPM

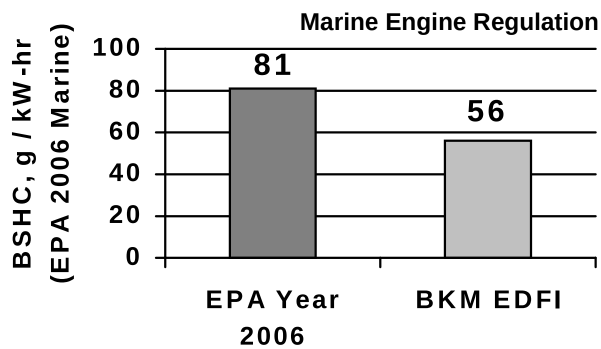

Figure 21. 46cc Utility Engine Brake Specific Hydrocarbon Emissions (BSHC), US EPA 2006 Marine Engine Regulation Comparison

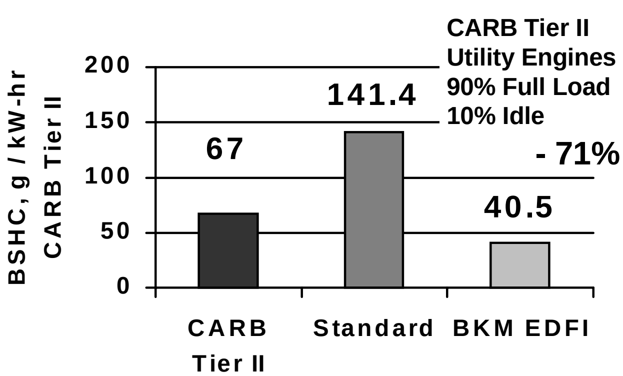

Figure 19. 46cc Utility Engine Brake Specific Hydrocarbon Emissions (BSHC), CARB Tier II Comparison

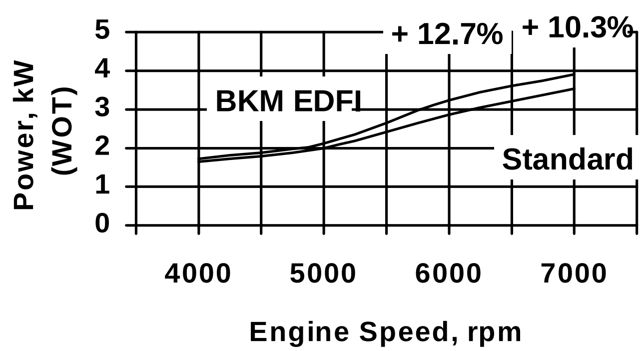

Figure 22. 50cc 2-Wheeler Wide Open Throttle (WOT) Power Comparison

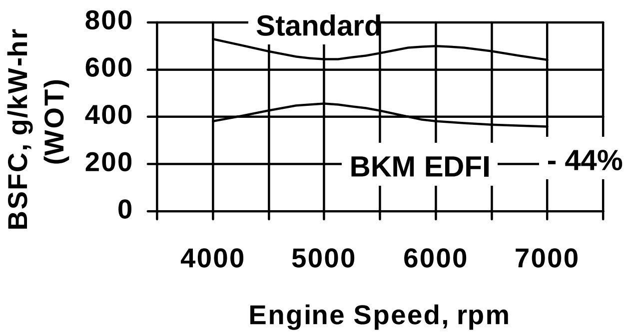

Figure 23. 50cc 2-Wheeler Wide Open Throttle (WOT) Brake Specific Fuel Consumption Comparison

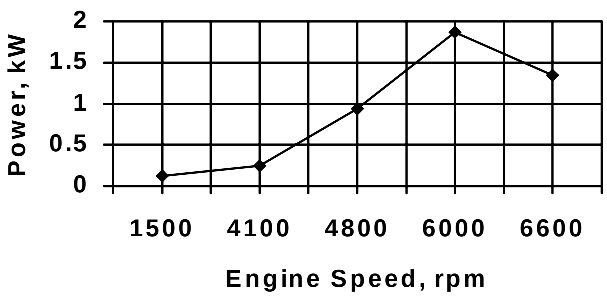

Figure 24. 50cc 2-Wheeler Part Load Test Cycle

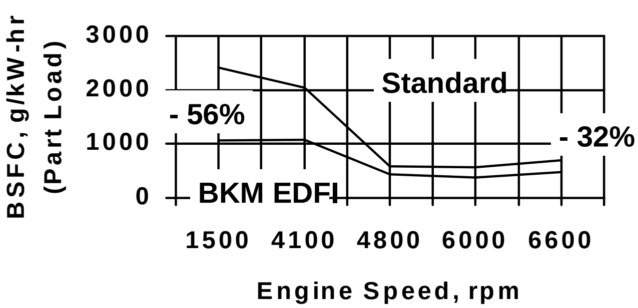

Figure 25. 50cc 2-Wheeler Road Load Fuel Consumption Comparison

CONCLUSION

A detailed discussion of an EDFI technology for two-stroke engines has been provided. This technology provides manufacturers of 2-stroke engine powered utility engines, motorbikes, marine engines and other applications a means to significantly reduce exhaust emissions.

In addition to reducing exhaust hydrocarbon emissions and exceeding baseline engine power, the unique spray impingement combustion system prevents engine component mechanical or thermal problems resulting from deviation from conventional evaporative cooling effects. This result occurs regardless whether the OSKA spray impingement or hollow cone spray is used, since impingement occurs with either method.

Two-strokes dominate applications requiring a lightweight high power engine. The fuel injection technology outline in this paper is universally applicable to current two-stroke powered products. The technology applies equally well to applications such as two-stroke utility engines, mopeds, scooters, motorcycles and outboard engines. In more sophisticated and demanding applications, the addition of catalytic converters may also be considered.

This technology focuses on pollution prevention by reducing emissions at the source. Redesigning the engine and its systems will dramatically reduce emissions from small two-stroke engines. In particular, emissions of HC, NOx and CO should comply with future exhaust emissions regulations.

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1. Johnson, D., Wong, H.C. “Electronic Direct Fuel Injection System Applied to an 1100cc Two-Stroke Personal Watercraft Engine”, SAE Pare 980756, 1998.
2. Kato, S., Onishi, S., “New Mixture Formation Technology of Direct Fuel Injection Stratified Combustion SI Engine (OSKA)”, SAE Paper 871689, 1987.