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I am submitting herewith a thesis written by Harley Douglas Ferguson, III entitled "Development of a Single-Cylinder Common Rail Injected Diesel Test Engine." I have examined the final copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.


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**DEVELOPMENT OF A SINGLE-CYLINDER COMMON RAIL
INJECTED DIESEL TEST ENGINE**

A Thesis

Presented for the

Masters of Science Degree

The University of Tennessee, Knoxville

Harley Doug Ferguson, III

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This thesis is dedicated to....

*My father, a mechanical engineer himself, who taught the skills to become
an engineer and always believed in me.*

And

My mother, who quietly stood by and encouraged me.

And

Meghann

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ABSTRACT

Due to more stringent regulations set forth by the Environmental Protection Agency (EPA) to take in effect in the year 2007 [1], the automotive industry has been required to evaluate new diesel engine technologies to further reduce tailpipe emissions. Of major concern is the reduction of oxides of nitrogen (NO_x) and particulate matter (PM). These two emissions are linked to environmental pollution and health concerns respectively.

In recent years, the primary method employed to reduce these emissions has been to increase combustion efficiency by increasing injection pressure. These higher injection pressures increase the atomization the fuel by increasing the injection velocity of the fuel droplets into the combustion chamber. These higher injection pressures and velocities have been shown to decrease engine out emissions while increasing power output [2,3]. In addition to higher injection pressures, the electronic control of the fuel injection systems has become necessary to help meet these newer emissions standards. Before the use of electronic fuel injection, the maximum injection pressure was a function of engine speed as well as the camshaft profile that actuated the fuel pump and injectors. Since the introduction of such electronically controlled fuel injection systems, such as the unit injector, unit pump, and distributor pump, pressure generation has been de-coupled somewhat from these constraints. The Robert Bosch Corporation achieved complete de-coupling with the release of the high-pressure common rail injection system for European markets in 1997 and the American market in 2001 [4,5]. This system allows high injection pressure (up to 30,000psi) to be developed at idle engine speeds and maintained throughout the engine's operating range. In fact, optimum injection pressures are

available for any operating condition and any engine speed. This system also allows for highly flexible rate shaping of the fuel injection sequence as well as flexible injection timing.

The purpose of this study was to develop a test cell that incorporates a high-pressure common rail fuel (HPCR) injection system on a single-cylinder test engine. The intended use of this test cell would be to conduct research in the area of optimizing fuel injection timing, injection rate, fuel pressure generation, and injection length with respect to reducing fuel consumption, tailpipe emissions, reduction of combustion noise, and increasing power. This test cell uses an alternating current (AC) vector controlled motoring/absorbing dynamometer with a Hatz single cylinder direct injection (DI) diesel engine retrofitted with a Bosch HPCR fuel pump, fuel injector, an universal solenoid driver, and a personal computer (PC) based programmable engine control module (ECM).

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LIST OF SYMBOLS

EPA	Environmental Protection Agency
NO _x	Oxides of Nitrogen
PM	Particulate Matter
HPCR	High Pressure Common Rail
AC	Alternating Current
DI	Direct Injection
PC	Personal Computer
ECM	Engine Control Module
GVW	Gross Vehicle Weight
IDI	Indirect Injection
UI	Unit Injector
NO	Nitrogen Oxide
O ₂	Diatomic Oxygen
CO	Carbon Monoxide
HC	Hydrocarbon
H ₂	Diatomic Hydrogen
N ₂	Diatomic Nitrogen
IGBT	Insulated-Gate Bipolar Transistor
PWM	Pulse Width Modulation
RPM	Revolutions Per Minute
VDC	Volts Direct Current
HP	Horsepower
TDC	Top Dead Center
ADT	American Diesel Tube
UTK	University of Tennessee, Knoxville
NI	National Instruments
VI	Virtual Instrument
ORNL	Oak Ridge National Laboratory
BSFC	Brake Specific Fuel Consumption
TTL	Transistor-to-Transistor Logic
LED	Light Emitting Diode
BDC	Bottom Dead Center
CA	Crank Angles
BTDC	Before Top Dead Center
cc/inj	Centimeters per Injection

CHAPTER I – THE PURPOSE OF THE STUDY

Introduction

Diesel engine technology has become considerably more advanced in recent years and many of these advancements have resulted from new fuel injection techniques and developments. Direct injection and electronic fuel injection are becoming commonplace and HPCR fuel injection systems show great promise for increasing torque and reducing emissions. Direct injection is the direct spray of fuel into the combustion chamber. The main advantages of DI are better fuel consumption and higher efficiency. Other advantages include reduced combustion noise and better emissions. Additionally the use of electronic injection allows for the precise metering and injection of fuel. HPCR systems use a high-pressure pump to generate injection pressures up to 30,000 psi. This unique system stores that high-pressure fuel in an accumulator where it is equally available to each cylinder. Coupled with electronically controlled and solenoid operated fuel injectors, HPCR systems are capable of multiple injections during each combustion cycle. Together, DI and HPCR can have a great effect on fuel consumption and tailpipe emissions [4,5].

The future of diesel engine performance development and emission reduction will require a greater knowledge and understanding of how these new techniques can be used to influence engine operating conditions. The purpose for developing this test cell is to provide a test bed to investigate optimization of fuel injection timing, injection rate, fuel pressure generation, and injection length with respect to reducing fuel consumption, tailpipe emissions, reduction of combustion noise, and increasing power. Fundamental

research in these areas benefits greatly from a single-cylinder test bed where the uncertainties associated with multiple cylinder engines such as cycle-to-cycle distribution of air and fuel, manufacturing tolerances, and cylinder cooling are avoided.

Background

Diesel engine technology has become considerably more advanced in recent years and many of these advancements have resulted from new and improved fuel injection techniques and systems. Before the nineteen nineties, most light-duty trucks (GVW $\leq$ 6500 lbs.) and passenger cars equipped with diesel engines used indirect injection (IDI) fuel systems [6]. This fuel system utilizes a small prechamber attached to the combustion chamber via a small passage way. Located in this small chamber are the fuel injector and sometimes a glow plug unit, which is used as a cold start assist device. Towards the end of the compression stroke, the fuel is injected into a localized region of compressed air within the prechamber. Here the injected fuel has a suitable environment for the combustion to begin. As the diesel fuel diffuses and mixes with the compressed air it begins to combust and propagates out of the prechamber and into the cylinder at speeds close to sonic velocity. The pre-chamber design is used because it allows for more ideal conditions for the injected fuel quantity to mix with the oxidizing air and create a controlled diffusion flame process. In addition, the prechamber introduces the injected fuel and air from the chamber to the main cylinder in a turbulent airflow regime to increase mixing [7].

In recent years however, direct injection (DI) fuel systems have begun to replace IDI systems in engines for this class of vehicles. DI systems inject the fuel directly into the cylinder and usually operate at injection pressures much higher than IDI systems. Because of advances in fuel pressure generation, the fuel can be injected into the cylinder at much higher injection pressures. These higher injection pressures allow the fuel to mix more rapidly and combust more quickly, thus eliminating the need for a prechamber and the more complex valve/prechamber head geometry and manufacturing [7]. Additionally, DI engine designs reduce heat loss and improve thermal efficiency.

Additionally, electronic control of these injection systems has allowed for more precise control of the fuel injection event. Some DI fuel systems, such as the unit injector (UI), unit pump, and distributor pump, use electronic solenoid valves to precisely open the injector at the desired injection time or once the desired injection pressure is obtained. The combination of electronic controls with higher injection pressures has made diesel engines more fuel efficient and powerful [8].

The most recent introduction of a DI fuel system is the high-pressure common rail fuel system. It uses a crankshaft driven fuel pump to generate high fuel pressures of up to 30,000 psi and stores the fuel in a fuel accumulator or rail. This rail makes the pressurized fuel available to each cylinder at any time. This system is unique in that it can develop this high fuel pressure at idle speeds and can constantly supply this pressure at any operating condition, especially part load. Additionally, it can hold this pressure

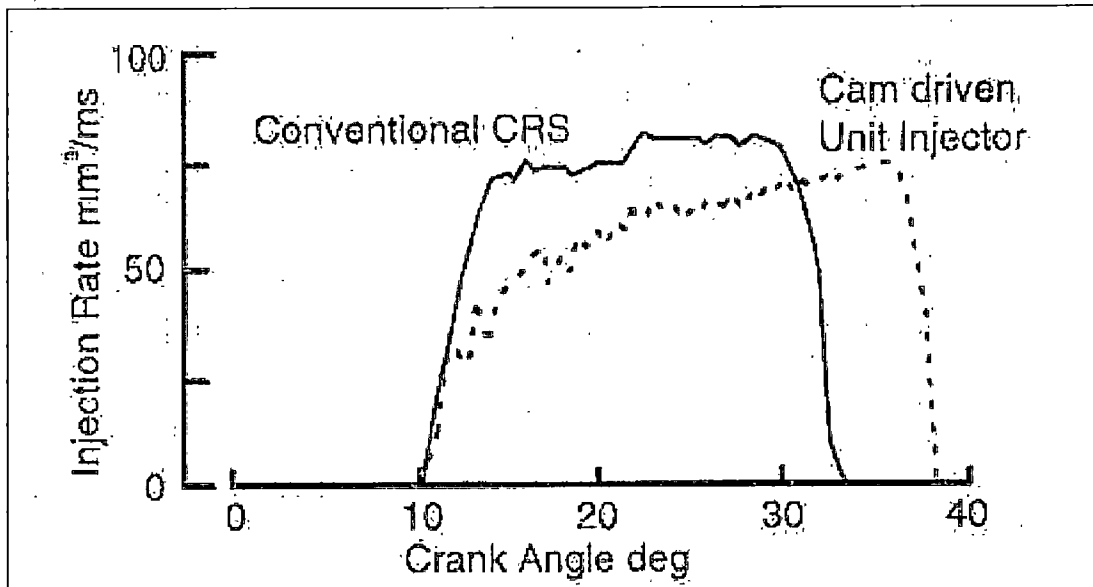


Figure 1. Injection Rate for CRS and Unit Injector

constant over the entire injection duration, unlike UI, which builds pressure linearly during the injection event. The injection rate of fuel for a typical HPCR and UI are shown in Figure 1. It can be seen that a conventional HPCR rail system injects fuel at a constant rate whereas a UI builds fuel pressure from the beginning of the fuel injection duration until the end [9]. The advantages and disadvantages of the rate shaping the fuel injection event will be discussed at length later.

With approaching emissions regulations in 2007, which will lower NO_x emissions to 0.2 g/HP-hr and PM emissions to 0.01 g/HP-hr for heavy-duty diesel engines [1], some DI fuel systems have been found to be limited in their ability to limit these pollutant emissions and lower fuel consumption. A study by DaimlerChrysler AG compared the UI, the distributor pump, and the HPCR fuel systems and found the HPCR to have the most potential to fulfill future emission requirements [9]. The result from the investigation showed significant benefits in reducing NO_x and smoke during part load

conditions. The researchers contributed these benefits to the HPCR flexibility, specifically the system's ability to vary injection timing in addition to its capability for multiple injections.

Recently, the use of multiple injections and shaping the rate of fuel injection has been a major area of study for automotive engineers. The injection of fuel before the main injection is referred to as pilot injection. The injection of fuel after the main injection is referred to as secondary, or post, injection. Pilot injection and increased injection pressure have been shown to reduce combustion noise [10,11]. The use of secondary injection is much newer than pilot and is believed that it may be a convenient source for the reducing agents used in exhaust after-treatment devices such as the de-NO_x catalyst and the NO_x adsorber [12].

Even though catalyst technology has been used in spark ignition engines since the nineteen seventies, an effective catalyst has yet to be developed and commercially produced for current production diesel engines. In the nineteen seventies, the oxidation catalyst was used in spark-ignition gasoline engines to oxidize carbon monoxide and hydrocarbons to carbon dioxide and water [7]. The problem with the oxidation catalyst was that it did not address the sufficient reduction of nitrogen oxide (NO). To solve the problem of NO emitted in the exhaust stream, the industry moved to introduce the three-way catalyst in the early nineteen eighties. The three-way catalyst effectively brings the exhaust gas composition to near-equilibrium state where enough reducing agents are present to reduce NO and enough diatomic oxygen (O₂) to oxidize to carbon monoxide (CO) and hydrocarbon (HC) [7].

Currently most diesel engines, both IDI and DI, do not use catalyst technology to reduce engine emissions after being emitted from the engine. Even though diesel engines can benefit from the use of oxidation catalysts to reduce CO and HC, most engines do not use them because CO and HC emissions are low enough without further reduction under current emissions regulations. Furthermore, diesel engines cannot use the three-way catalyst because their fuel-lean operation is not conducive to the near stoichiometric conditions, which are required for their operation [7].

Two systems currently being researched to reduce NO in diesel exhaust are the de-NO_x catalyst and the NO_x adsorber. The de-NO_x catalyst uses CO, HC, and diatomic hydrogen (H₂) in the exhaust to reduce the NO to diatomic nitrogen (N₂). The NO_x adsorber works to adsorb and temporarily store NO_x. Over time the absorbed NO_x found in the exhaust stream is stored as a base metal nitrate in the adsorber [13]. Before the adsorber becomes permeated, reduction agents are created by rich combustion and are used to reduce the stored NO_x to N₂ downstream over another catalyst material.

Two problems are presented with the use of these systems however: 1) diesel engines do not produce enough hydrocarbons during normal operation and 2) the sulfur found in diesel fuel and engine oil has been found to contaminate the catalyst material. To create sufficient hydrocarbons for the conversion process researchers are implementing two strategies. The first uses a secondary injector located upstream of the catalyst which usually requires a dedicated ECM to inject the necessary fuel [12]. This type of system is commonly used with the de-NO_x catalyst. The second type of system uses in-cylinder post or secondary injection, which uses partially burned fuel to supply the needed to

supply the needed reduction hydrocarbons. This system is commonly used with NO_x adsorber systems [13].

One particular problem that occurs with these systems is the contaminations of the catalyst material by the sulfur found in the diesel fuel and lube oil. To address the dilemma of sulfur contamination and the possibility of developing a NO_x reduction scheme, The Department of Energy funded a preliminary study to evaluate the issues and provide groundwork for future studies [14]. This study showed that the uses of these after-treatment devices and the use of post injection did indeed show promise in reducing both NO_x and PM while creating engine operation strategies to clean the catalyst material of collected sulfur. More importantly, this study shows the future possibility of using in-cylinder post injection as a way to introduce catalyst reduction agents.

CHAPTER II – THE PROBLEM

Statement of the Problem

Testing new fuel injection techniques that study possible noise suppression with pilot injection and NO_x reduction with secondary injection requires a specialized engine that allows a modification of the engine operation to test specific parameters. In addition, the test cell must be able to reduce run time and increase the degree of optimization. Without such a test engine, a production engine would have to be modified for each test to be run. In most cases, use of a production engine would require that both the camshaft profiles and/or the ECM program be changed. In both cases this can be a time consuming and costly task. In addition, exploring concepts such as decoupling fuel pressure from engine speed and the creation of NO_x reduction agents can be laborious on some engines, if not impossible. Thus the purpose of this thesis is to design and build such a test bed and to do some limited evaluation of its flexibility. Engine manufactures have had such test engines for quite some time and use them to test new ideas and to optimize current systems.

The advantages of a single-cylinder test engine with a flexible HPCR fuel system are:

1. Modifiable injection rate.
2. Multiple injection events.
3. Modifiable injection timing.
4. Variable injection pressure.
5. Reduced cylinder-to-cylinder variation in fuel delivery, airflow, and combustion inherent in multi-cylinder engines.

The primary objective of this study is to develop a single-cylinder test engine that uses a common rail fuel injection system and one that has specific operating parameters, which can be modified by the operator. The parameters that can be user modified are injection timing, injection rate, fuel pressure, and injection duration. This study will concentrate on injection timing, injection rate, and injection duration.

CHAPTER III - EQUIPMENT

Engine Dynamometer

The engine dynamometer incorporated into the test cell uses a direct-coupled Baldor VECTOR DRIVE MOTOR and can be seen in Figure 2. The motor is a 3 phase AC motor operated at a line voltage of 460 volts and a line frequency of 60 Hertz. All other motor data is listed in Table 1.

The vector drive motor is controlled by a Baldor LINE REGEN VECTOR DRIVE, which provides full motoring capability by using insulated-gate bipolar transistor (IGBT) and pulse width modulation (PWM) technology to give quiet operation and full rated torque to zero revolutions per minute (rpm). Vector control separates the motor current into its flux and torque producing components. These components are adjusted

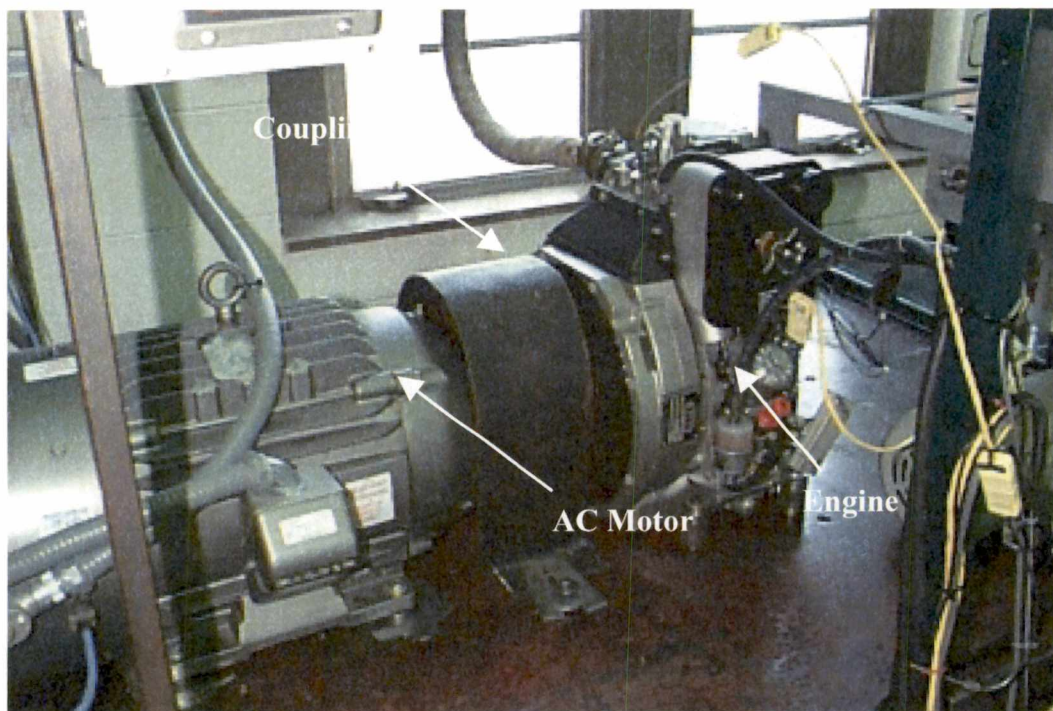


Figure 2. Engine Test Cell

Table 1. Baldor Motor Data

Baldor VECTOR Drive Motor	
Catalog Number:	ZDM2333T
Frame:	256TC
Rated HP:	15 TE
Volts:	230/460
MAG. Current:	17.8/8.9
RPM:	1725
MAX RPM:	4000
Hertz:	60
Slip:	1

independently and added vectorially to maintain a 90° angle between them [15]. The main advantage of vector control is the simplistic user interface and maximum torque at zero speed. Other advantages of vector control include reduced line harmonic distortion and unaffected output voltage due to line voltage transients.

The vector drive has two analog, 0-5 volts direct current (VDC), outputs which can be user programmed from 10% to 100% full scale of the output. A list of the possible outputs is listed in Figure 3.

Control of the dynamometer is through a small portable, hand held keypad with a digital read-out. All dynamometer control is accessible through the keypad, including the programming of the analog output channels as well as motor data calibration. For the purposes of this test cell, the keypad controller was mounted adjacent to the dynamometer and engine.

The operation of the dynamometer is similar to most motoring/absorbing dynamometers. The operator at the dynamometer control sets the engine speed and load is determined by the amount of fuel. The operator sets fuel delivery and likewise load by adjusting the throttle arm located on the engine. The load is a linear function

Two Analog 0-5VDC linear outputs may be configured to represent any of 19 conditions as follows:

ABS Speed - Represents the absolute motor speed where 0VDC = 0 RPM and +5VDC = MAX RPM.

ABS Torque - Represents the absolute value of torque where +5VDC = Torque at CURRENT LIMIT.

Speed Command - Represents the absolute value of commanded speed where +5VDC = MAX RPM.

PWM Voltage - Represents the amplitude of PWM voltage where +5VDC = MAX AC Voltage.

Flux Current - Represents the actual portion of total current used for excitation. 5VDC = MAX flux current.

CMD Flux CUR - Represents the calculated value for flux current. 5VDC = MAX commanded flux current.

Load Current - Represents the actual portion of total current used to produce torque (CW and CCW torque).
5V = Max. CW torque, 0V = Max. CCW torque.

CMD Load Current - Represents the calculated value of load current.
5V = Max. commanded load current.

Motor Current - Amplitude of continuous current including motor excitation current. 5VDC = Rated Current.

Load Component - Amplitude of load current not including the motor excitation current. 5VDC = Rated Current.

Quad Voltage - Load controller output. Used to diagnose control problems.

Direct Voltage - Flux controller output. Used to diagnose control problems.

AC Voltage - A scaled AC waveform that represents the AC line to line motor terminal voltage. 0V = Neg Peak PWM voltage. 2.5V centered.
5V = Pos Peak PWM voltage. At rated motor voltage, a full 0 to 5V sinusoidal waveform should be present. This waveform should be at or greater than the motor base frequency. (At half the motor base frequency, a 1.25V to 3.75 sine wave is present.)

Bus Voltage - Bus voltage scaled to 0-5VDC. 5V = 1000VDC.

Figure 3. Analog Input/Output for Baldor REGEN Controller

Torque - Bipolar torque output. 2.5V centered, 5V = Max Positive Torque, 0V = Max negative torque.

Power - Bipolar power output. 2.5V = Zero Power, 0V = negative rated peak power, +5V = Positive rated peak power.

Velocity - Represents motor speed scaled to 0V = negative max RPM, +2.5V = Zero Speed, +5V = positive max RPM.

Overload - (Accumulated current)² x (time), Overload occurs at +5V.

PH 2 Current - Sampled AC phase 2 motor current. 2.5V = zero amps, 0V = negative rated peak amps, +5V = positive rated peak amps.

PH 3 Current - Sampled AC phase 1 motor current. 2.5V = zero amps, 0V = negative rated peak amps, +5V = positive rated peak amps.

Process Feedback - Represents the selected Process Feedback signal. 2.5V centered, 5V = 100%, 0V = -100%.

Setpoint Command - Represents the selected Setpoint Command signal. 2.5V centered, 5V = 100%, 0V = -100%.

Position - Position within a single revolution. +5V = 1 complete revolution. The counter will reset to 0 every revolution.

Figure 3. Continued

of the position of where the throttle arm is located between the zero and full fuel limits.

These limits can be seen in Figures 4 and 5. The arm turned completely clockwise indicates the zero fuel limit and inversely, the arm turned completely turned counter-clockwise indicates the full fuel or load limit.

To set the load at a percentage of full load, the percentage must be calculated from full load torque or power values and set by adjusting the fuel/throttle arm until the desired torque or power is obtained. The torque and/or power can be observed by programming one or both outputs of the driver to display them on the keypad output or both can be connected to a remote data acquisition system. For the purpose of this test cell, both engine power and torque were monitored with a remote data acquisition system, which will be detailed later.

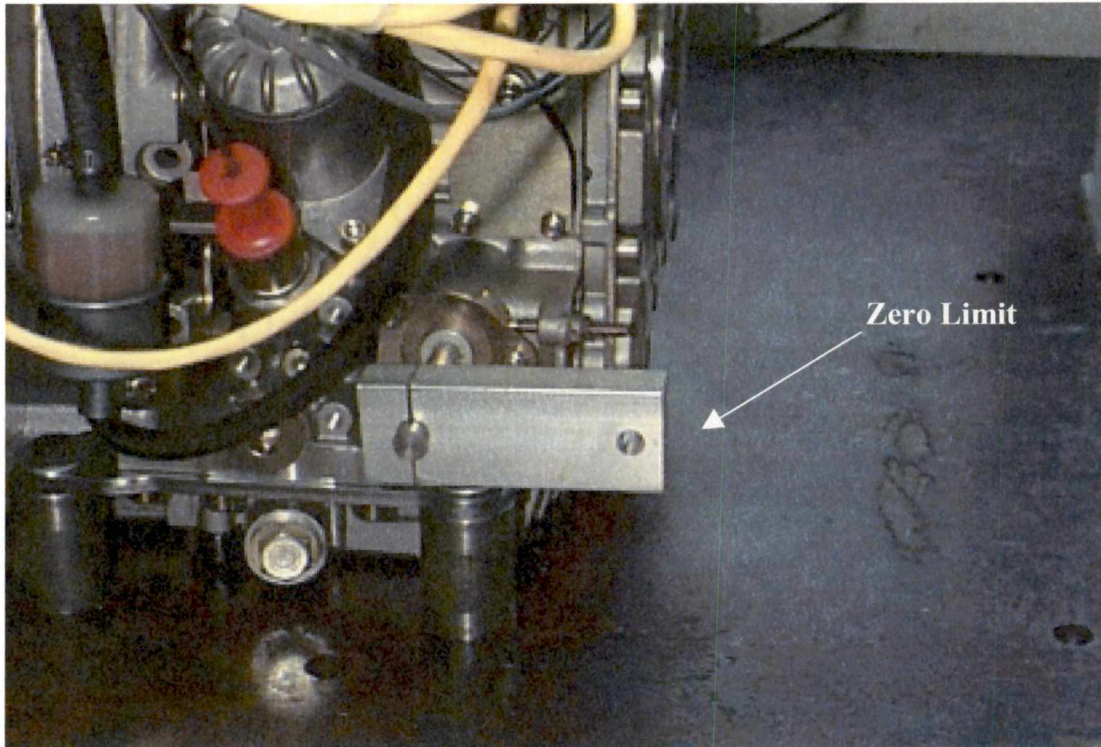


Figure 4. Throttle Arm at Zero Limit

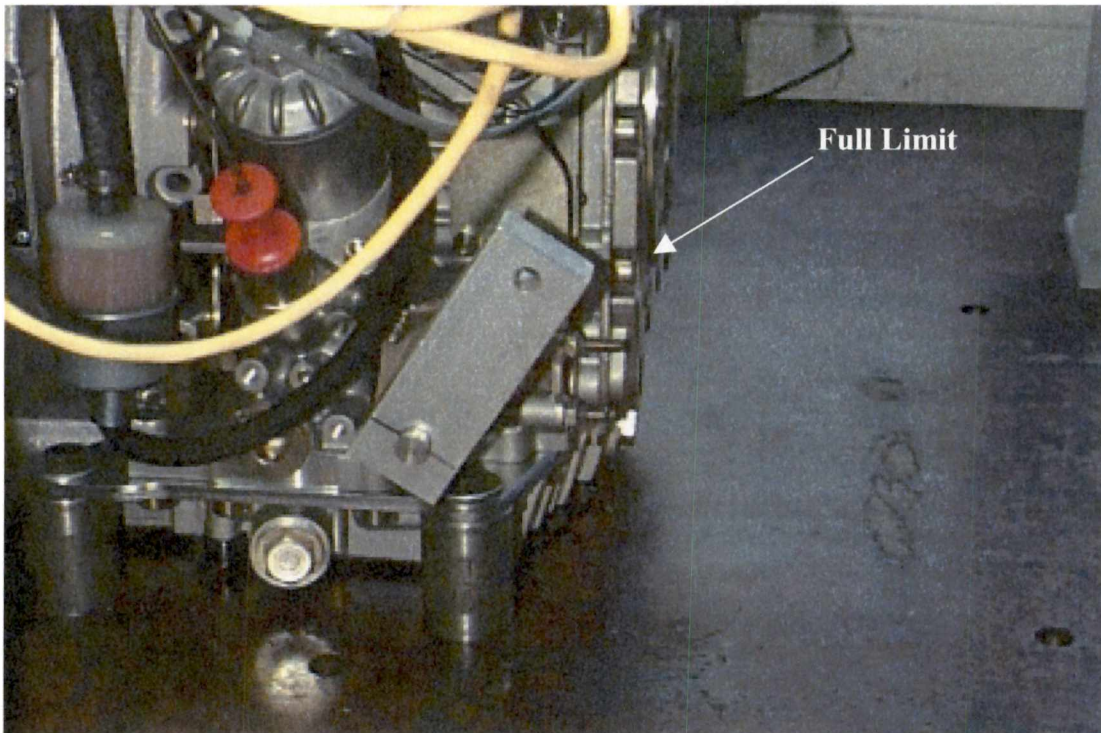


Figure 5. Throttle Arm at Full Limit

As indicated in Figure 3, the analog output signals for motor speed and power are 0-5 VDC. Where 0 VDC is zero motor speed and power and 5 VDC is maximum speed and power. For the current setup, the maximum engine speed is 4000 rpm and the maximum power is 15 horsepower (HP). To equate these analog voltage signals to data with useful units, a conversion scale was used for each signal. To convert VDC to rpm, a linear scale was developed that set 0 VDC to 0 rpm and 5 VDC to 4000 rpm. Using the equation for a line the scale for rpm can be found as follows:

$$y = m * x + b$$

where,

$$y = \text{rpm}$$

$$m = \text{slope} = \frac{4000\text{rpm}}{5\text{VDC}} = 800 \frac{\text{rpm}}{\text{VDC}}$$

$$b = 0$$

then,

$$\text{rpm} = 800 * \text{VDC}$$

The following equation is then used in the Measurement and Automation software managed by the data acquisition equipment to convert the raw voltage signal to useful engine speed data. The data acquisition equipment and the Measurement and Automation software are discussed in more detail later in the chapter.

The conversion of the analog voltage signal for power to horsepower is not a linear function and must use a conversion curve provided by the motor/driver manufacture. A copy of the conversion curve is shown in the appendix.

Test Engine

The test engine chosen for use in the test cell was an air-cooled, single cylinder Hatz 1D50Z DI diesel engine and can be seen in Figure 6. The engine has a bore of 3.819 in (97 mm) and a stroke of 2.756 in (70 mm) with a total displacement of 517 cc (0.517 liters). The engine has a mean piston speed of 23 ft/s (7 m/s) and a compression ratio of 20:1. These values are characteristic to small, high-speed diesel commonly used in automotive applications [7]. Other attributes to the engine include a die-cast light-alloy crankcase, gray cast iron cylinder, multi-hole fuel injector, pressure lubrication, and a flywheel fan. These attributes lead to such advantages as a long engine life, low fuel consumption, and reliability. Another reason for choosing the Hatz 1D50Z was the availability of parts and service in the United States in

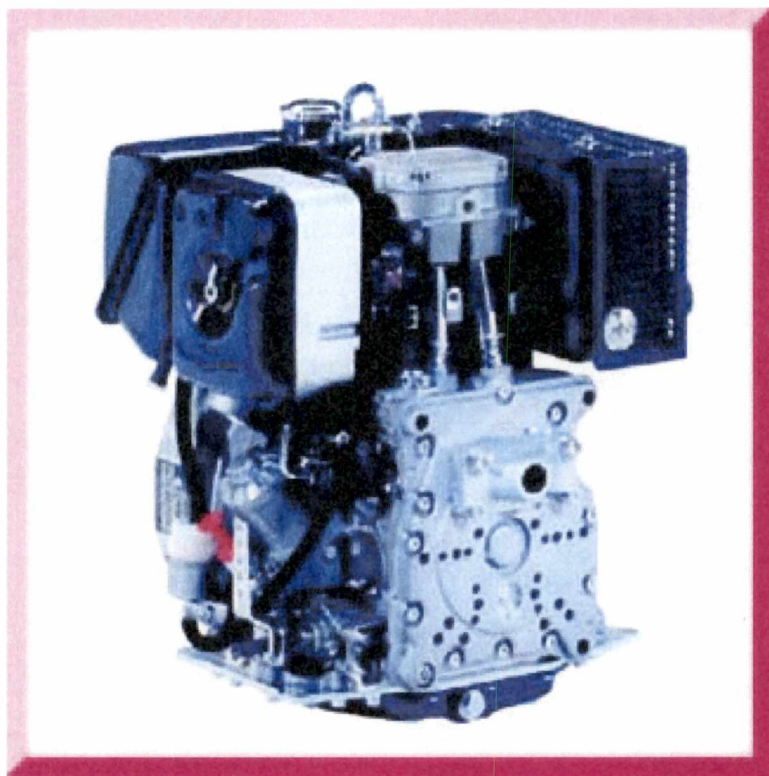


Figure 6. Hatz 1D50Z Diesel Engine

addition to local engineering help from Hatz's Wisconsin engineering facility. The engine is engineered to produce 11 HP at 3000 rpm and 21 ft-lbs. of torque at 2200 rpm [16]. At continuous load the engine power is reduced to 9 HP at 3000 rpm.

The engine uses a direct injection fuel system, which consists of a Bosch P-type single plunger fuel pump, injection fuel line, and a five-hole nozzle injector. The fuel pump is driven by the camshaft and turns at one half-engine speed. The injection timing is statically set by Hatz to begin injection at 20 degrees before top dead center (TDC) and end at TDC. The injection opening pressure is approximately 2900 psi. To begin injection, the camshaft rotates and compresses a set metered amount of fuel. The compressed fuel travels through the hydraulic fuel line to the injector. At the injector, the pressurized fuel acts to overcome the mechanical force of the nozzle spring. Once the hydraulic force of the fuel is greater than that of the spring, the nozzle opens and fuel is released into the combustion chamber [17]. The sprayed fuel is directed into a combustion bowl in top of the piston by the spray cone angle geometry. The combustion bowl helps add turbulence to the compressed air and injected fuel for better mixing. A picture of the combustion bowl can be seen in Figure 7.

The hydraulic flow of the injector was measured by Bosch and published at 332 cc for 30 seconds at a pressure difference of 1450 psi. A picture of the nozzle tip can be seen in Figure 8. The figure also shows the difference in the sizes of the intake and exhaust valves used in this engine. Additionally, the engine uses traditional poppet valves operated by push rods and rocker arms driven by the camshaft. A picture of the upper end of the valve train can be seen in Figure 9.

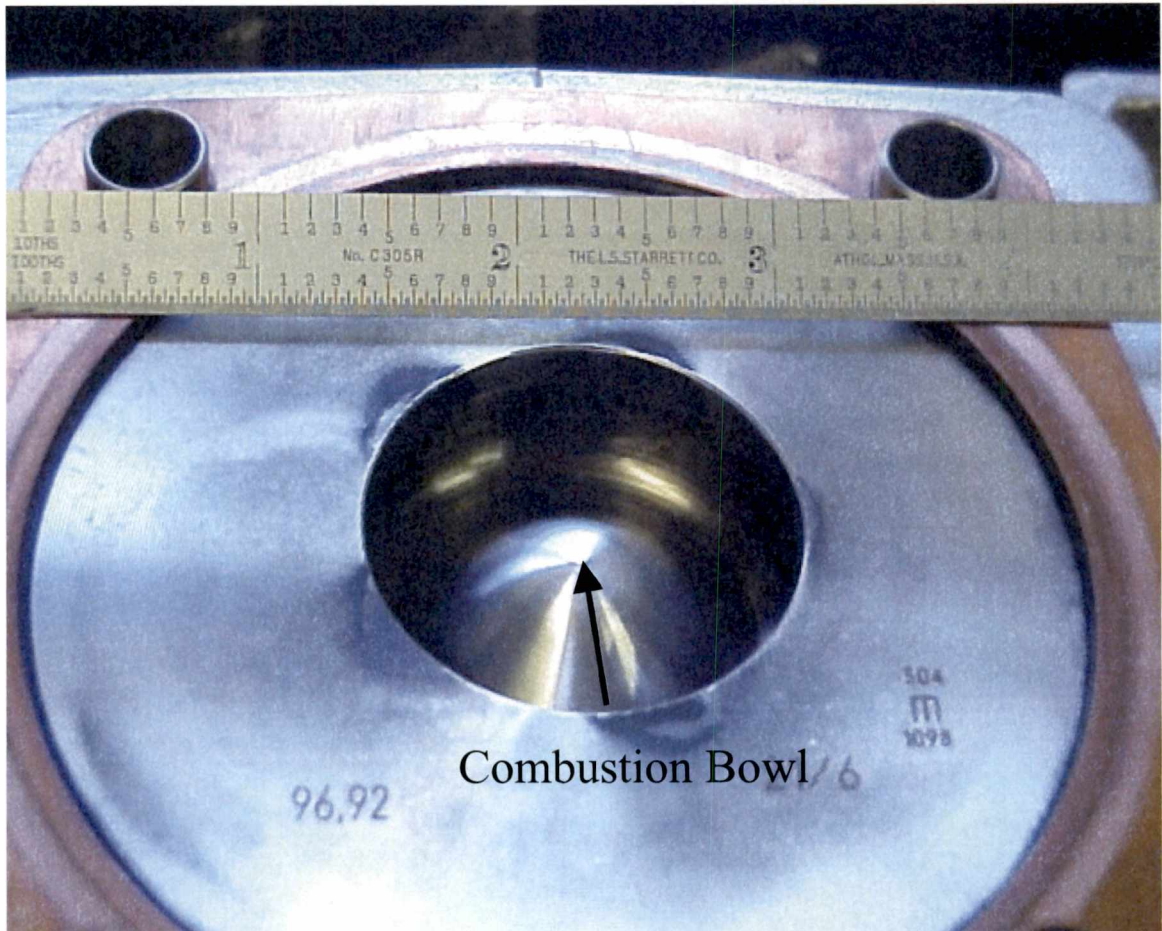


Figure 7. Hatz 1D50Z Piston with Combustion Bowl

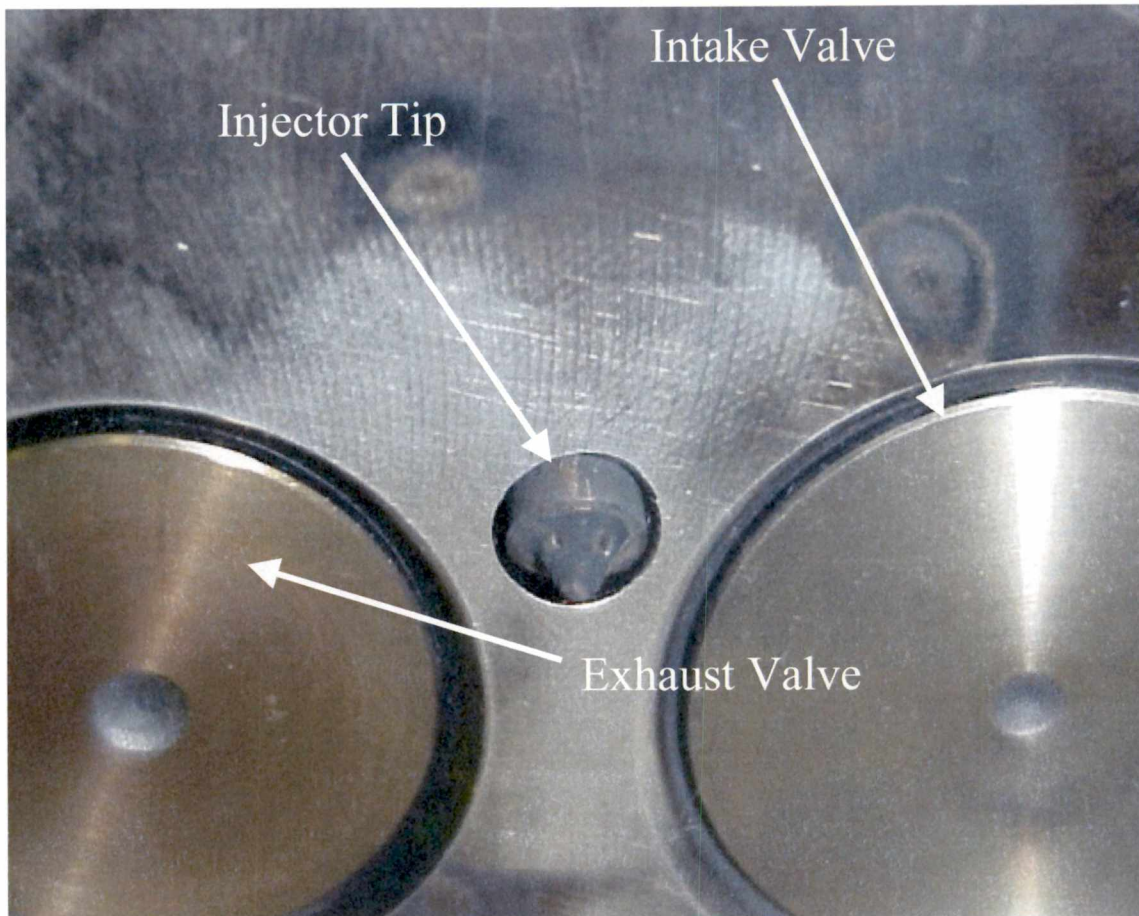


Figure 8. Hatz 1D50Z Injector Tip and Valves

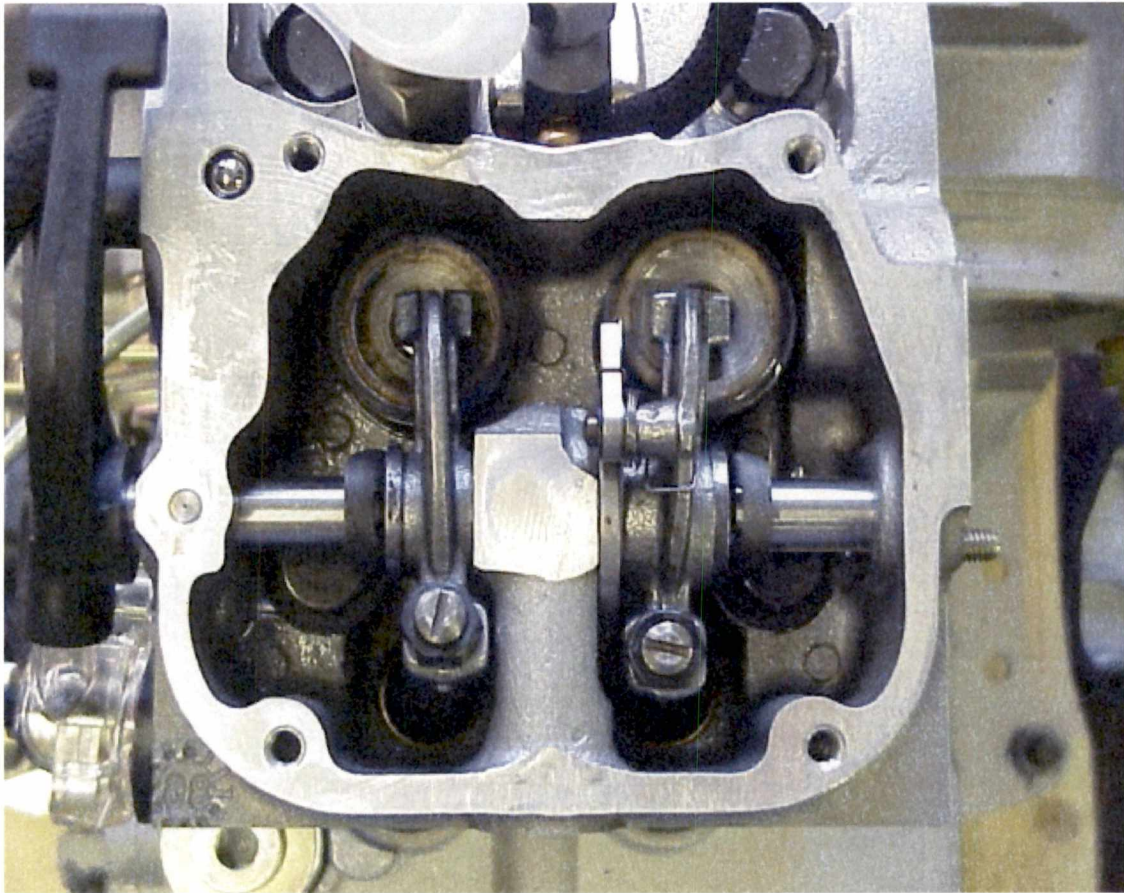


Figure 9. Hatz 1D50Z Valve Train

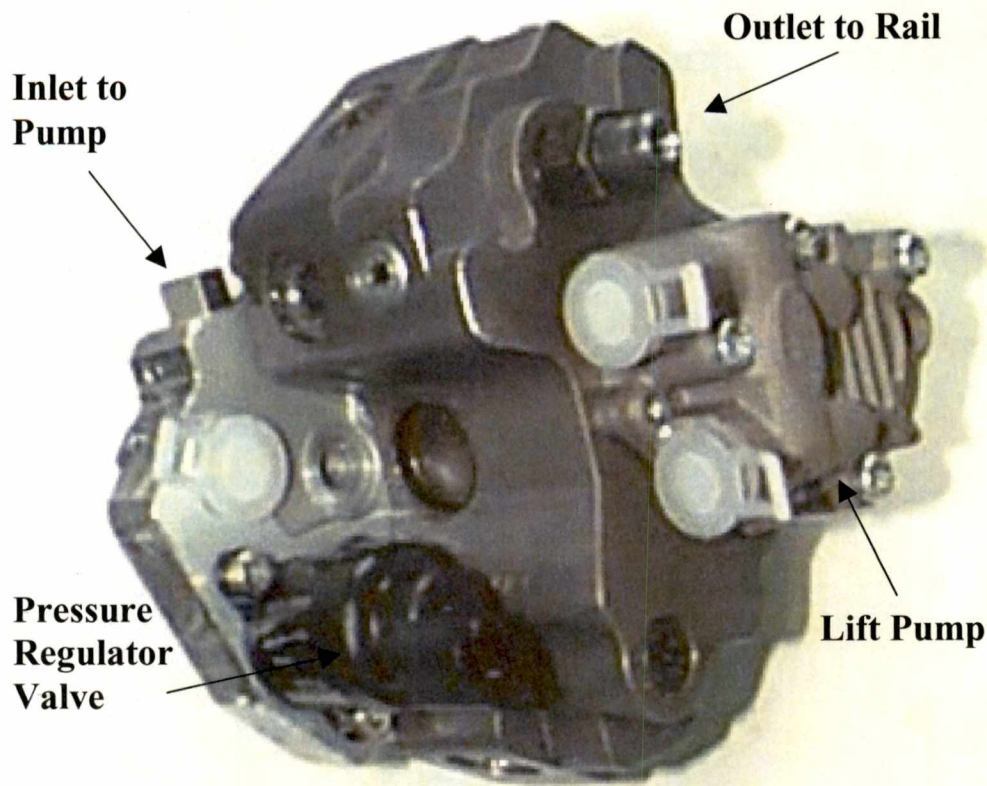


Figure 10. Bosch HPCR Fuel Pump

HPCR Fuel Injection Equipment

The HPCR fuel injection equipment used for this project consisted of a high-pressure fuel pump, fuel accumulator, high-pressure fuel lines, electronically controlled fuel injector, and a programmable ECM.

The high-pressure fuel pump is manufactured by Robert Bosch, GmbH and can generate fuel pressures up to 30,000 psi [17]. The pump consists of a shaft driven lift pump, eccentric pumping cam, three high-pressure pumping elements or lobes, and an electronic fuel pressure regulator. A picture of the pump can be seen above in Figure 10. The fuel is drawn to the pump by the shaft driven lift pump mounted on the rear of the

pump. The lift pump pushes the lifted fuel from the tank through a remotely mounted fuel filter then into the low-pressure cavity of the pump. The filter can be seen in Figure 11. Here the fuel is separated internally for two different purposes. One channel of the fuel is diverted to lubricate the internals of the pump during operation and the other channel enters the fuel pressure regulator cavity. Here the fuel flow is regulated to maintain the desired injection pressure. The pressure regulator valve is an electronic solenoid that is PWM controlled by the ECM to control the injection pressure. If the fuel accumulator or rail mounted pressure sensor indicates that the actual injection pressure is lower than the desired injection pressure, the fuel flow regulator pulses at a higher frequency to allow the flow of more fuel to the high-pressure stages of the pump.

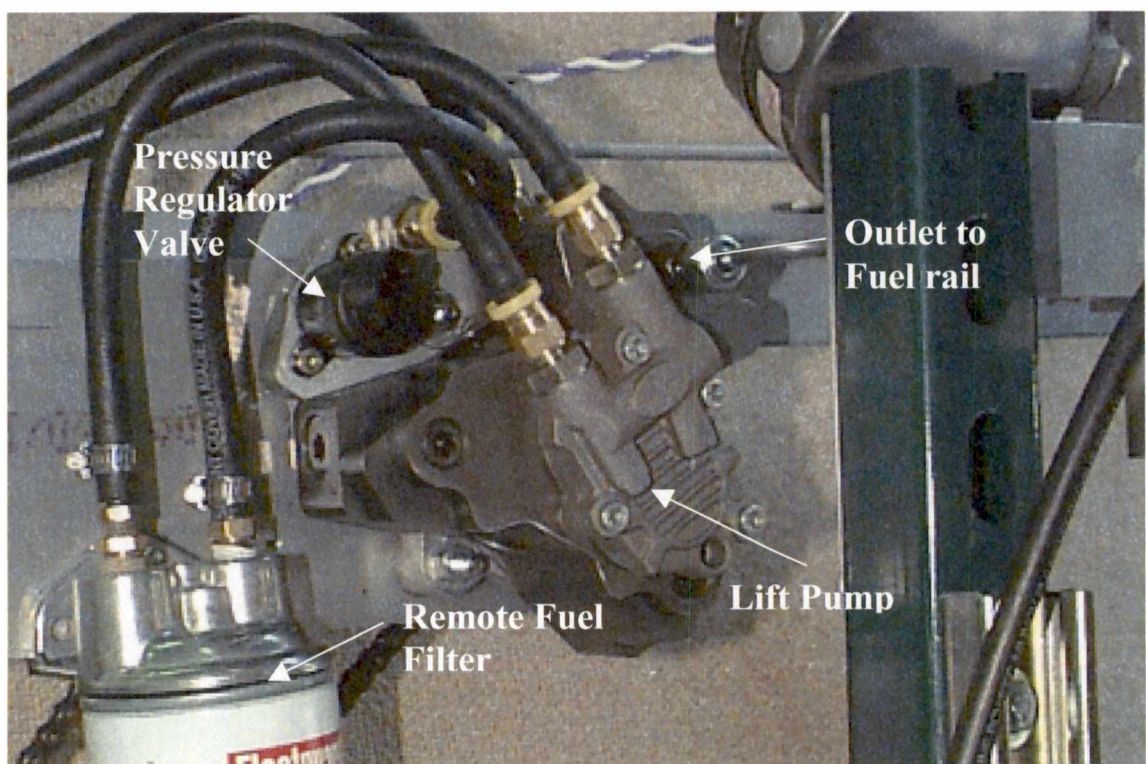


Figure 11. Bosch HPCR Fuel Pump and Remote Mounted Fuel Filter

If the pressure sensor senses a pressure higher than required in the rail, the regulator pulses at a lower frequency, thus allowing less fuel into the pump. The solenoid is designed such that maximum fuel flow is when the solenoid is not energized. For the purposes of this test cell, the pump mounted fuel pressure regulator was not used. Instead, a hand adjusted metering V-valve was used to regulate the injection pressure by bypassing excess fuel back to the tank. The metering valve effectively does the same job as the electronic regulator, however it meters the fuel in a steady-state fashion instead of a pulsed fashion. A complete schematic can be seen in the appendix.

At the onset of this project little was known about how the Bosch HPCR pump made fuel pressure and under what circumstances injection pressure was limited by engine speed. To eliminate these uncertainties, the pump was coupled to a one horsepower electric Baldor AC constant speed (1785 rpm) motor. This configuration allowed the pump to be driven at a constant speed set by the operator. The current setup with speed reduction via different sized pulleys drives the pump constantly at approximately 950 rpm. A picture of the drive system can be seen in Figure 12.

The fuel accumulator or rail is designed to hold a volume of fuel at desired injection pressure for each fuel injector used. Currently all production common rail systems are designed and manufactured for multi-cylinder engines, thus for this project a special rail was manufactured to accommodate only one fuel injector.

The high-pressure fuel from the pump enters the rail through a specially designed and built fuel line. American Diesel Tube (ADT) of Addison, IL was used to manufacture these lines.

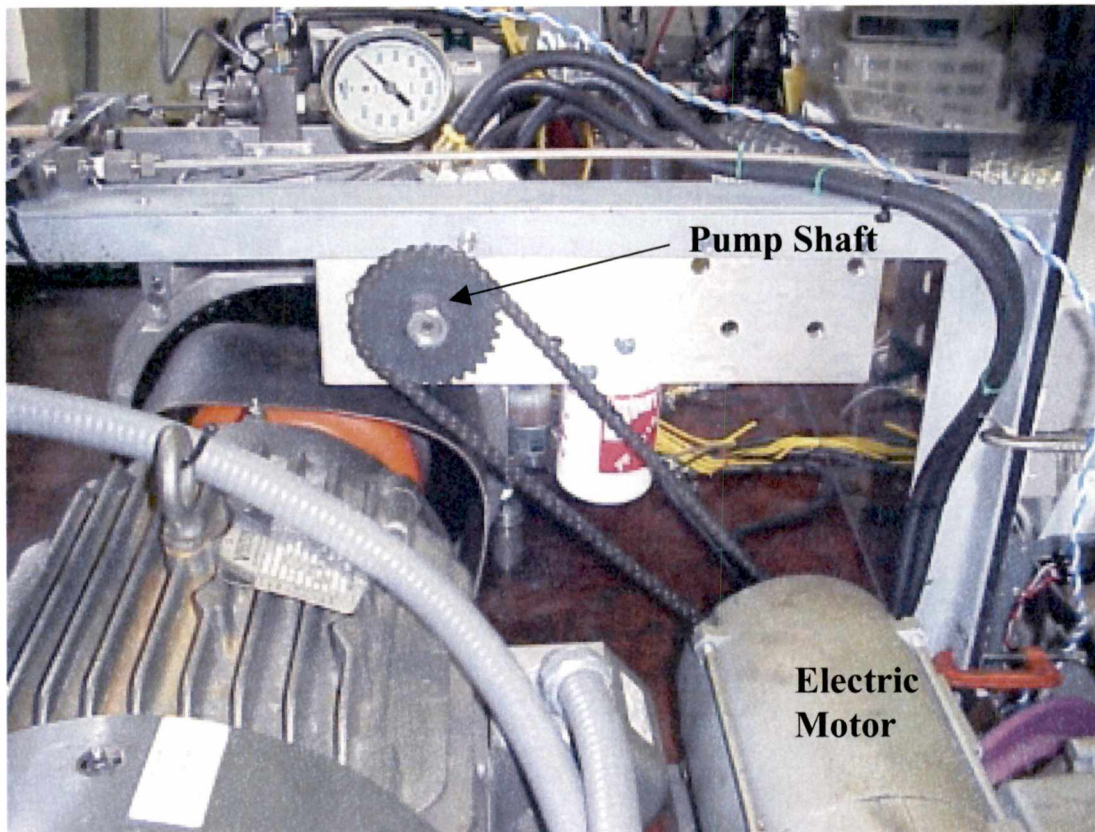


Figure 12. HPCR Drive System

Special care was taken such that each component was mated to the next component within the tolerances of the Bosch-type cone and nozzle connections required to contain the fuel at such high pressure. Three-dimensional templates were made at University of Tennessee, Knoxville (UTK) to match the test cell setup and then were sent to ADT for manufacturing. Before the final fuel lines were made, ADT sent sample lines to be compared with the actual test cell setup. After some small modifications, ADT manufactured two pump-to-rail and two rail-to-injector fuel lines capable of containing fuel at 30,000 psi. The routing of fuel to the hand-adjusted metering V-valve used off-

the-shelf high-pressure stainless steel tubing and connections from BuTech Pressure Systems of Erie, PA. Instead of using an electronic pressure sensor mounted in the rail, this test cell uses a stainless steel, liquid filled pressure gage that is mounted to read the pressure inside the rail. The gage used reads a maximum of 20,000 psi. With the use of the metering valve, the operator can adjust the rail pressure by visually reading the pressure from the gage as the valve is adjusted. Pictures of the metering valve and the pressure gage can be seen in Figures 13 and 14.

The fuel injector uses an electronic solenoid to control fuel flow. With the

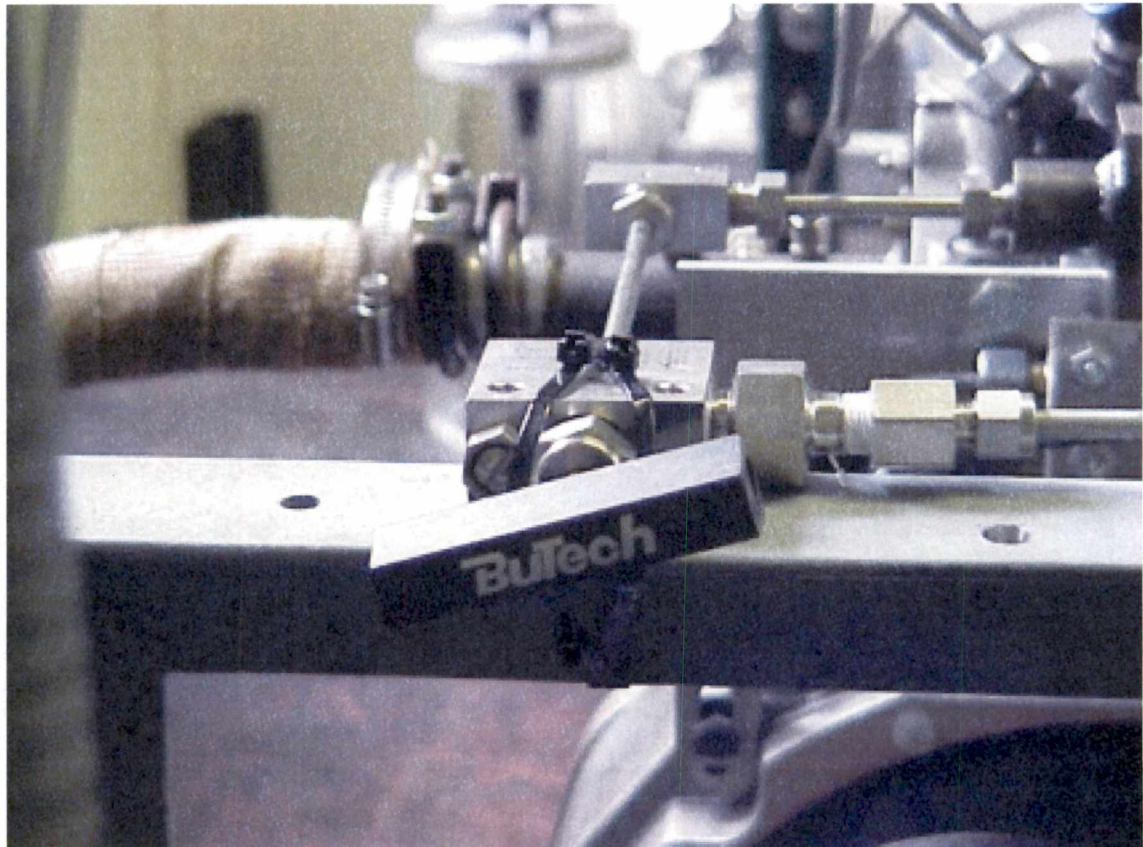


Figure 13. Hand Adjusted Metering V-Valve

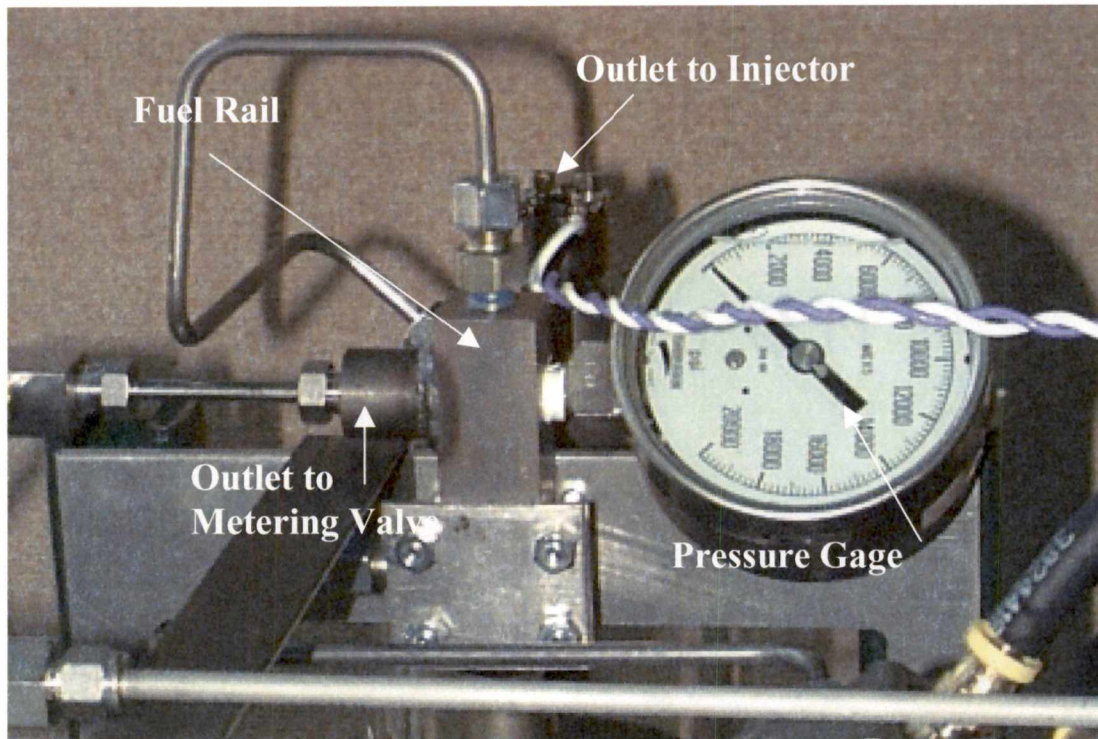


Figure 14. HPCR Fuel Rail and Pressure Gage

solenoid energized, the ball moves out of the way and allows fuel at high pressure to enter the injector. As the fuel travels down the feed passage into the nozzle, the fuel then enters a small pressure chamber in the tip of the nozzle. Here the pressurized fuel acts to overcome the mechanical force of the valve control plunger spring. When the plunger is completely recessed up in the injector body by fuel pressure, the fuel flows through the injector tip via small holes into the combustion chamber. To close to the injector, the solenoid closes and replaces the ball valve to its seat. By doing this fuel is no longer allowed to flow into the injector, allowing the fuel pressure and hydraulic force holding the plunger open to fall to a point less than the mechanical force of the plunger. When the plunger returns the closed position, fuel is no longer allowed to enter the combustion chamber. The fuel injector used in this project is a Bosch common rail

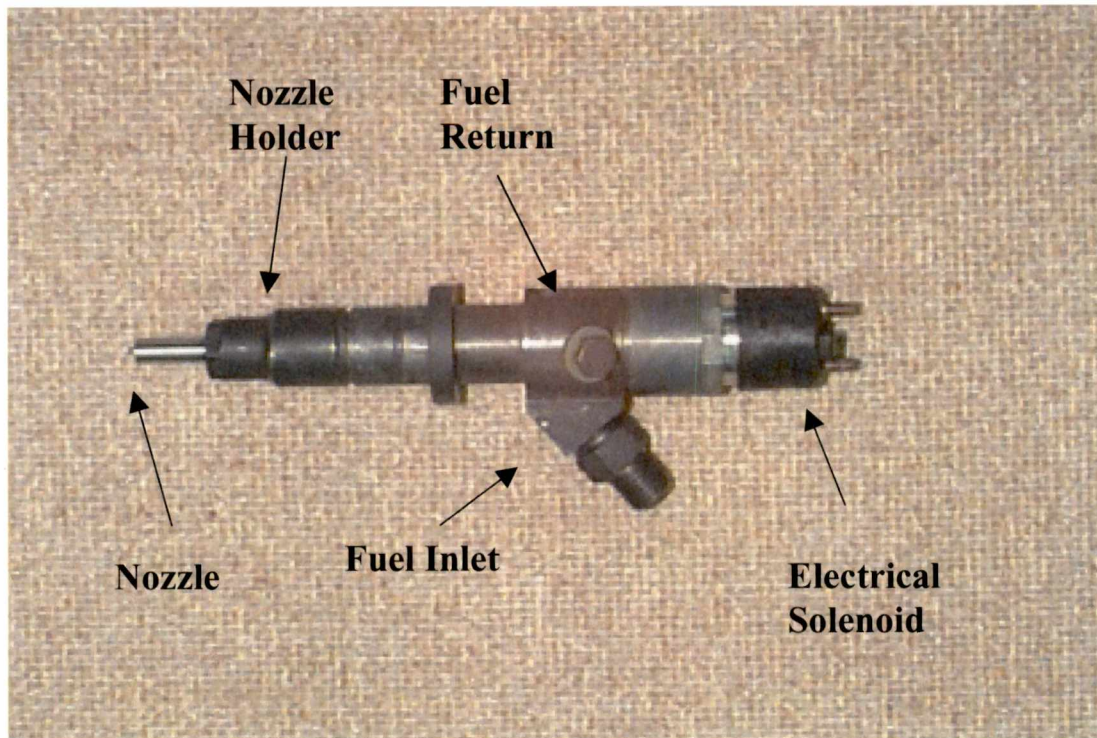


Figure 15. Bosch HPCR Fuel Injector

type and can be seen in Figure 15.

Cummins Engine Company of Columbus, Indiana donated the injector used. The injector was originally used in development of a HPCR fueled 5.9 L six-cylinder turbocharged engine. The configuration of the development two-valve Cummins engine was such that the injector inclination angle and injector body diameter matched that of the Hatz engine. The inclination angle is the angle at which the injector enters the combustion chamber with respect to the vertical axis of the cylinder. The inclination angle of the two engines is 20° and the injector body diameter is 19 mm.

The original nozzle of the Cummins has a hydraulic flow of 450 cc for 30 seconds at a pressure drop of 1450 psi. Compared to the original Hatz injector nozzle, the Cummins nozzle had an increase of 118 cc over the same amount of time and pressure. Even

though the Hatz and Cummins combustion chambers are comparable, the reason for the increase in flow is due to the Cummins being turbocharged and thus receiving more air charge per cycle. The effect of the difference in the two nozzle flows is discussed in detail later.

Data Acquisition Equipment

The acquisition of test data for the UTK test cell is handled by a National Instrument (NI) SCXI signal-conditioning chassis using two 1303 thirty-two-channel multiplexers. The signal conditioning chassis communicates digitally through a PCI expansion slot to a Micron Electronics, Inc. personal computer with a 400 MHz Intel Pentium III processor. The PC uses a LabView Virtual Instrument to collect the digital input and store it as delimited text file. The LabView VI also serves as a visual interface of current test cell data.

The SCXI signal conditioning chassis and 1303 multiplexers can interpret raw analog data as single-ended, ground referenced, and differential signals. The UTK test cell chose to use the differential mode for the ease of sensor application. The use of this mode allows for multiple ground references, which can greatly simplify grounding problems. One 1303 multiplexer was used for input of traditional analog input and the second was used for thermocouple input. Each individual input channel was calibrated using NI's Measurement and Automation Software. The software has the ability to initialize and calibrate each device used with the SCXI chassis. In addition, the software can initialize

and scale each input channel separately. This allows channels to be modified without modifying the VI.

The VI used for the UTK test cell was created specifically for its use. A picture of the VI interface can be seen in Figure 16. The VI visually displays and records the following test cell parameters: engine speed, engine power, ambient/inlet temperature, crankcase oil temperature, exhaust temperature, and fuel flow. The VI also has the capability to record emissions information, exhaust pressure, inlet pressure, and humidity.

The engine speed is determined by using a fiberoptic light sensor in conjunction with a frequency-to-voltage converter. The engine power is collected from dynamometer output. All temperatures are determined by using Omega K-type thermocouples with software built in cold junction compensation. The fuel flow is recorded as an analog voltage output from the fuel flow system.

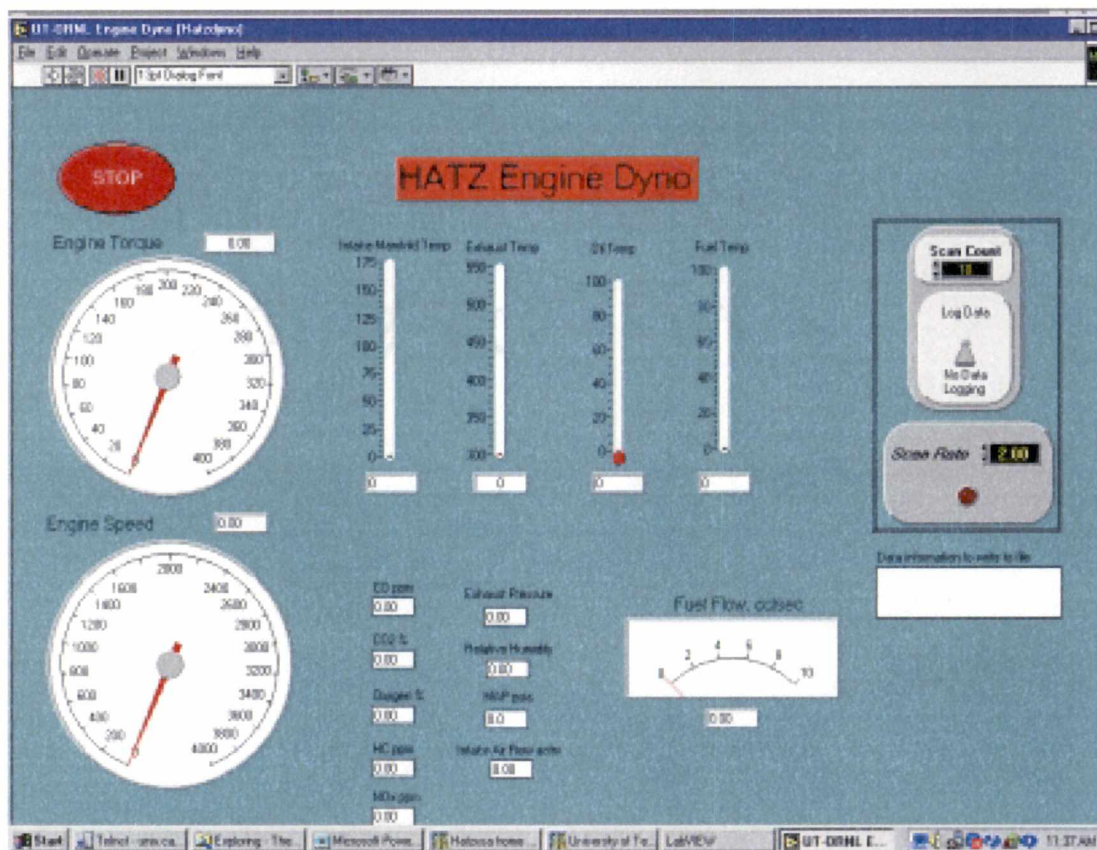


Figure 16. LabVIEW Hatz Engine Virtual Instrument

CHAPTER IV - PROCEDURE

Baseline Testing

The baseline testing of the engine presented an unexpected challenge and delay. Unlike a traditional dynamometer, which uses either a shaft mounted torque meter or a torque arm mounted load cell, the Baldor LINE REGEN VECTOR DRIVE uses stored motor data along with real time motor feedback to calculate the measured torque [15]. During early testing of the engine, the torque signal appeared to be highly irregular and plagued with line noise. Upon closer inspection and experimentation, it was determined that what appeared to be noise was the actual torque signal. When measuring the torque, the motor controller would experience the acceleration and deceleration of the piston as it approached TDC and BDC. In addition, the torque signal showed a large pulsation spike at maximum cylinder pressure. Experiments conducted by Oak Ridge National Laboratory (ORNL) have shown that engine shaft speed varies within one piston cycle of a 1.0 liter two cylinder Ruggerini MD191 DI diesel engine at 2100 rpm [18]. A plot of shaft speed during 720° of crank revolution is shown in Figure 17. This plot is important because since torque and engine shaft speeds are related it can show how torque can vary during one cycle. In ORNL's experiment, conducted by Howard Haynes, a magnetic speed sensor was placed such that it read each tooth of the engine's flywheel while the engine was operated at 2100 rpm under full rated load. The plot shows how the shaft speeds up during the power stroke of cylinder #1 until it begins to slow down from the compression of the air of cylinder #2. The shaft then speeds up again during the power stroke of cylinder #2. These two events, power and expansion, which occur while the

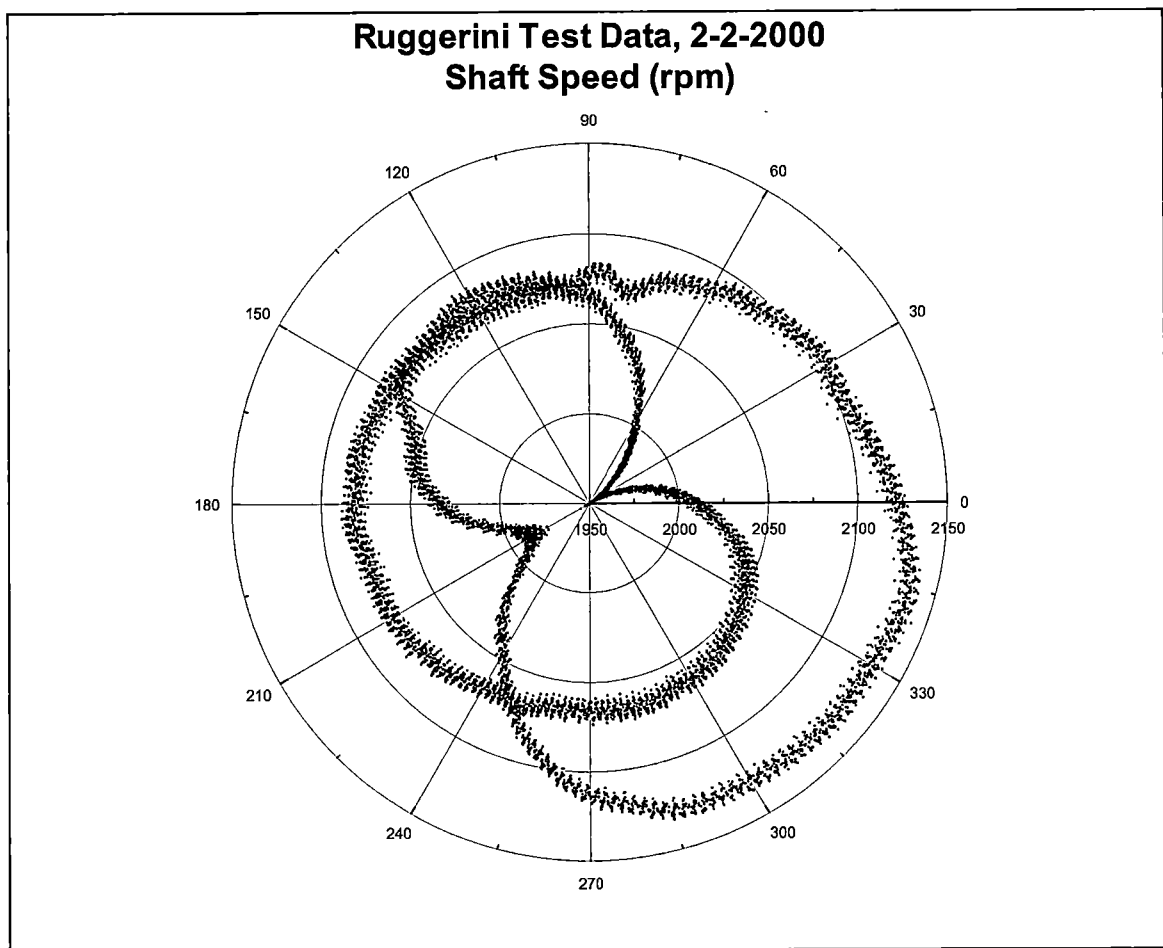


Figure 17. Shaft Speed of a Ruggerini 2-Cylinder DI Engine

intake and exhaust valves are closed, are the two most important events which play a role in the variation of the shaft speed. Even though this plot is complicated somewhat by two cylinders, it does illustrate the general trend. This trend causes the engine speed and effectively, the engine torque to vary. Most commonly, torque measurements are sampled at a frequency and averaged to nullify this variation. However this system, which uses the motor controller to calculate torque, was programmed to sample within the frequency range that shows these variations for a diesel engine. Discussions with software engineers at Baldor proved this theory and that the only solutions involved reprogramming the software, internal to the driver, or simply sampling enough and averaging data in the test cell's data acquisition system. A strategy of recording 200 samples at 50 per second was found to give reliable and repeatable data.

Once shielding had reduced line noise and the proper sampling frequency had been chosen, a baseline test of the stock Hatz engine power, torque and fuel consumption was completed. Figure 18 shows baseline engine power collected from the Hatz 1D50Z engine.

The plots show that measured power was acceptably close to the manufacture's published data. It is interesting to note at this point that the developed power decreased by approximately 30% at 2900 rpm and completely decreased by 100% at 3000 rpm. This suggests that the fuel governor was slightly misadjusted when delivered from the factory. Figure 19 shows fuel flow versus engine speed and supports the fact that the fuel pump governor decreased fuel flow to the engine at an engine speed of 2900 rpm.

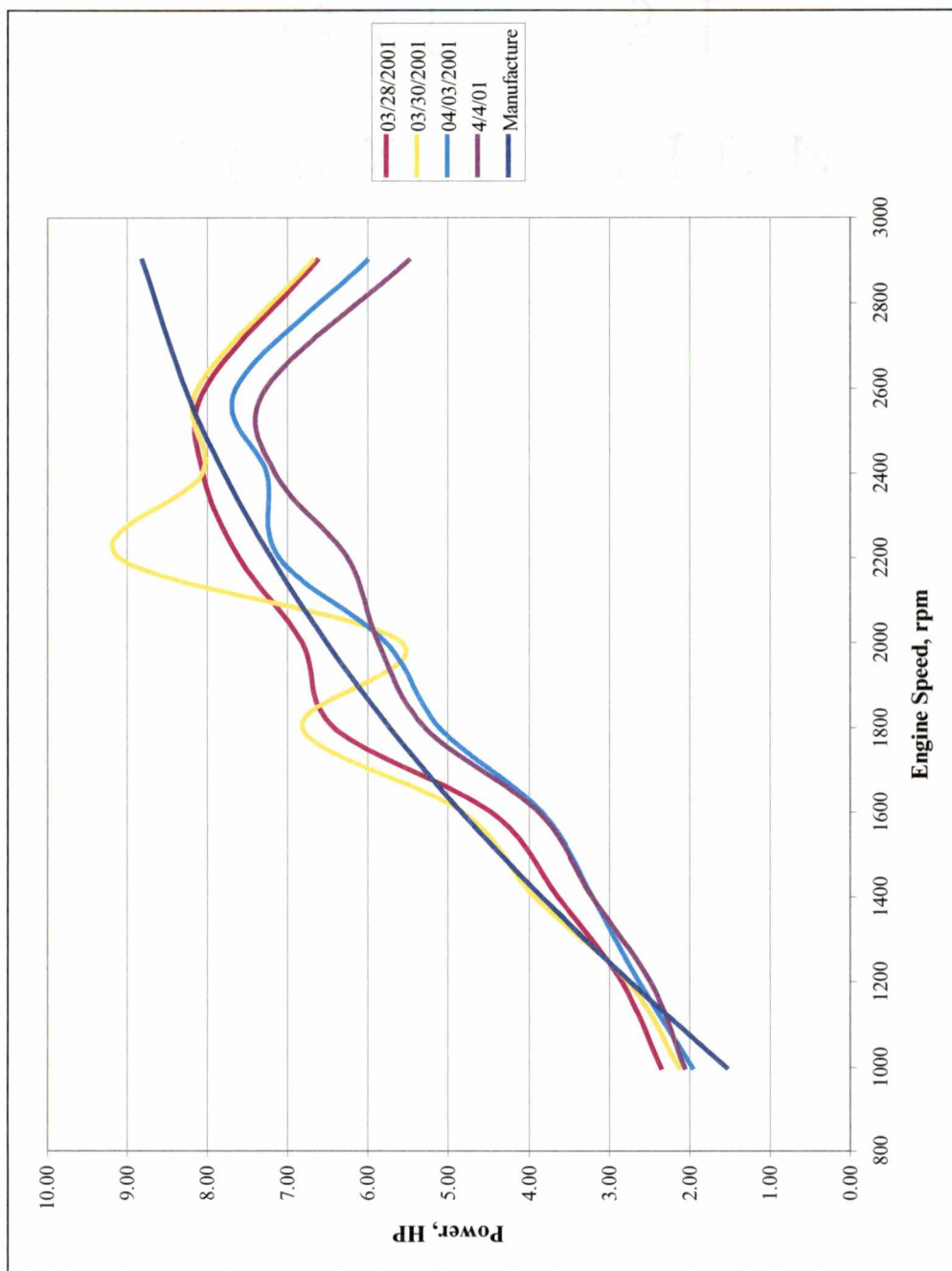


Figure 18. Power Comparison

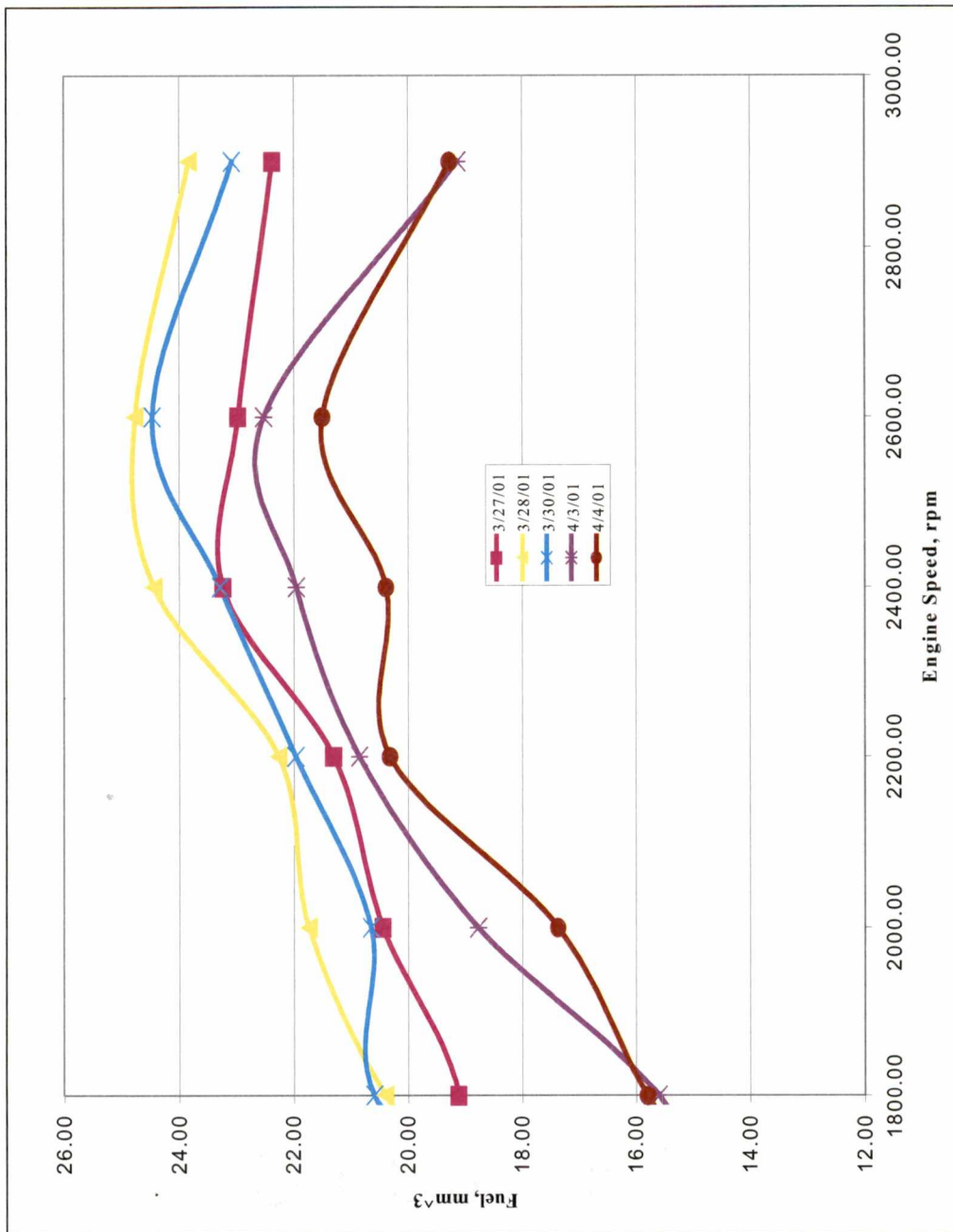


Figure 19. Fuel Quantity per Injection

Due to erratic torque pulsation and the difficulty in measurement, torque values at full load were calculated from measured power data using the following equation:

$$Torque = \frac{Power}{EngineSpeed}$$

The calculated torque plots are shown in Figure 20 below.

To measure the fuel consumption of the engine, the fuel flow was measured by using MAX's Fuel Flow Measurement System. The system uses a method of keeping track of the amount of fuel necessary to maintain a certain level in a small tank located inside the flow system enclosure. As the level of fuel decreases inside the tank, due to consumption, the flow system supplies more fuel as needed to maintain the correct level. The flow meter transmitter keeps track of the amount of fuel supplied to the tank and records this value as the amount of fuel consumed by the engine. The measured fuel flow is measured and displayed as cubic centimeters per second or cc/sec. To equation used to calculate the fuel consumption of the engine is as follows:

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

In English units, BSFC is typically presented in units of pounds of fuel per horsepower-hour or lbs/HP-hr. The specific gravity of the diesel fuel used in the project was 843.4 kg/m³. A copy of the certification sheet for the diesel fuel used is attached in the appendix. Plots of the baseline fuel consumption are shown in Figure 21.

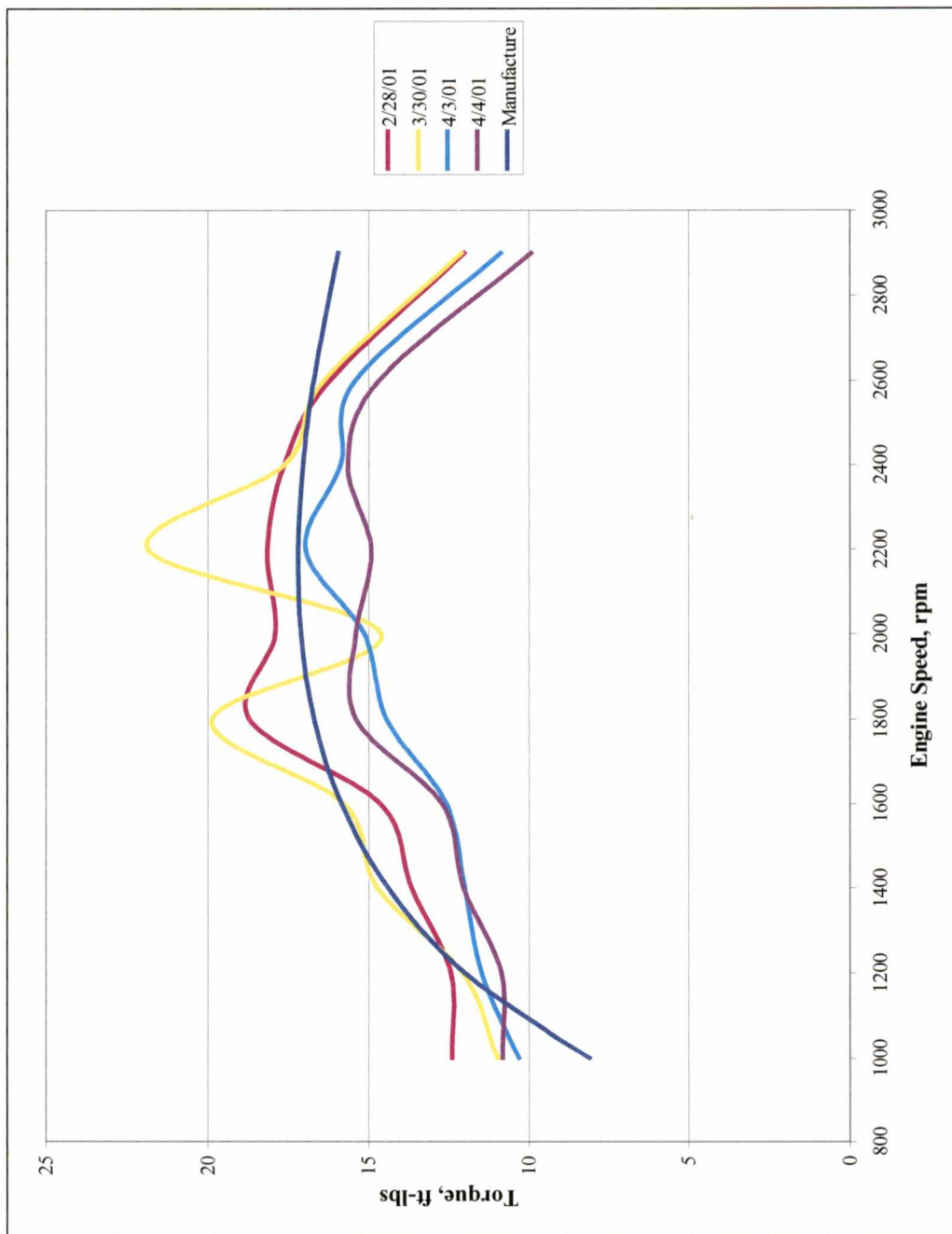


Figure 20. Torque Comparison

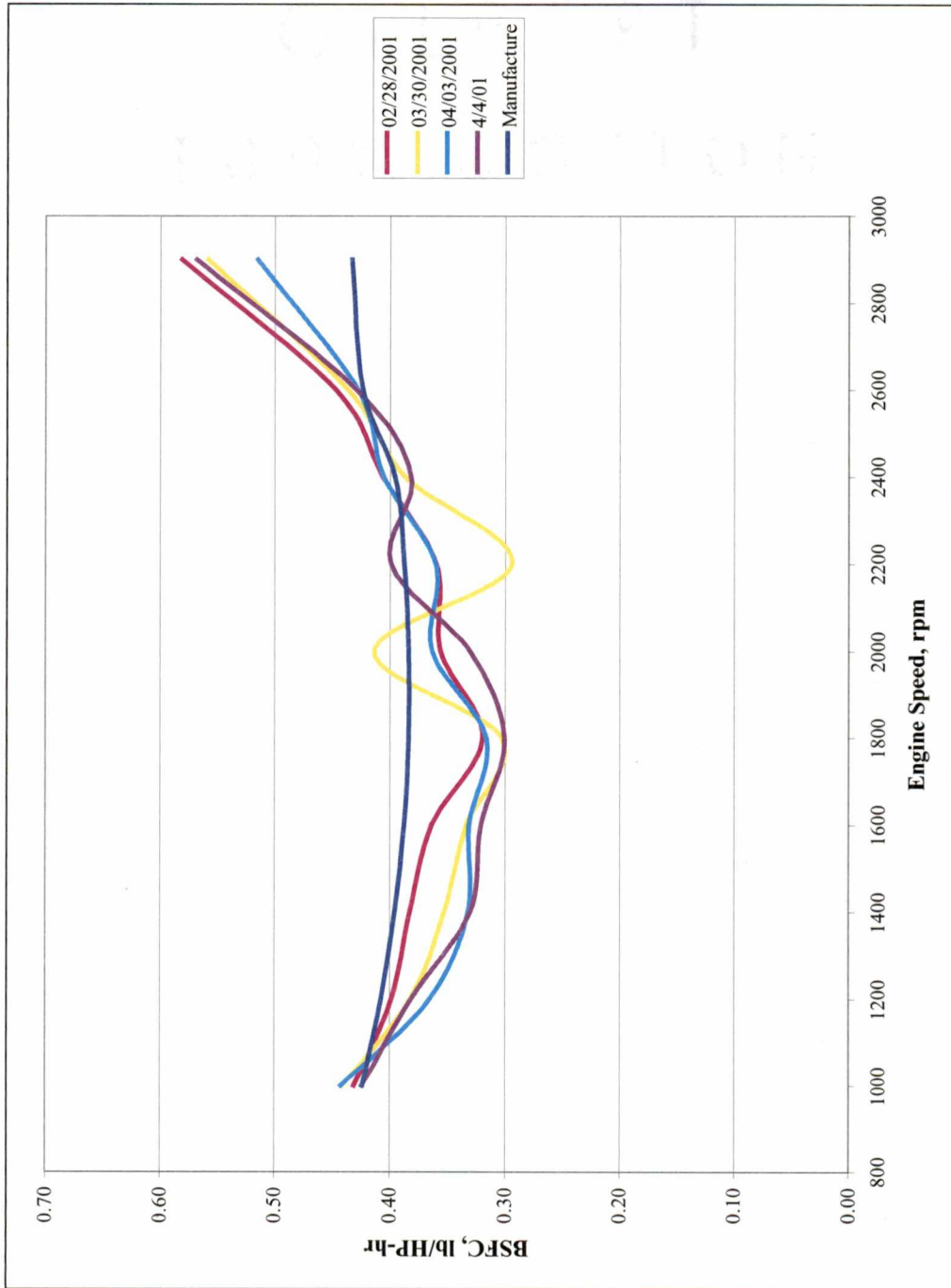


Figure 21. Brake Specific Fuel Consumption

These plots are consistent with traditional diesel engine fuel consumption plots. However it was surprising that the measured fuel consumption was lower than the published data from the manufacture. The difference might be due to the configuration of the engine, the lack of an exhaust muffler, and ambient and initial conditions. Furthermore, the fuel flow measured for this engine is in the low range of the MAX Fuel Flow System. However, multiple test runs did show repeatable data. Thus for the purposes of this test cell, this data can be viewed as accurate.

Development of Programmable ECM

To enable the test cell to be capable of testing a wide range of fuel injection characteristics, a programmable ECM was developed. The ECM allows for testing multiple injections and varying injection timing and lengths. In production vehicle systems, the ECM is programmed by the engine manufacture to match desired requirements such a high fuel economy, enhanced power, and lower emissions. Once the vehicle's ECM has been programmed, the ability to modify fuel injection characteristics is limited to special ECM programming modules. Since the Hatz originally was without electronic control, the test cell required the development of one that suited the needs of future researchers. The ECM would require two main components. The first was a solenoid driver that matched the requirements of the Bosch/Cummins injector solenoid. The driver would need to be able to supply the needed opening current and holding current to operate the solenoid at the desired frequency of injection. For use in the UTK test cell, a universal solenoid driver was obtained that used an 80 VDC-boost power

supply in addition with a 12 VDC-car battery to operate the injector at the required 93 VDC and current. The universal driver has the ability to adjust the opening current, the hold current, and the solenoid opening time. The use of such a universal driver allows researchers to use different injectors in the future that might require different operating parameters. The driver works by converting a simple transistor-to-transistor (TTL) signal to the required current signals to operate the injector [19].

The second component of the ECM would need to read the engine speed and determine when to fire to the fuel injector. Furthermore, the ECM would need to determine the number of injections and the timing sequence of those multiple injections. Most commonly an arbitrary waveform generator would be used to accomplish these tasks. However due to budget, a much cheaper solution was developed. An IBM 486DX personal computer, which operates under DOS, was chosen for its ability to manage its own clock speed. Operation under an operating system like Windows by Microsoft might have caused unnecessary drift due to time management, which might have changed clock speed to optimize other systems that might be running concurrently.

An added bonus is that the parallel port on PCs uses TTL logic signals to communicate with parallel port operated printers. By using a programming language, which can control the motherboard, a programmer could use the processor to manipulate the parallel port to communicate with the universal solenoid driver. Most commonly, C and BASIC are the languages used, which can control the input and output (I/O) of the motherboard. The UTK test cell uses the QBASIC version of BASIC because of its simplicity in programming and availability.

The TTL logic uses a square wave operating between the limits of zero and five VDC to convey either a one (1) or zero (0) in binary. Where zero VDC corresponds to a signal of 0 and five VDC corresponds to a signal of 1. Using the hexadecimal addresses of the parallel port output, QBASIC commands are used to develop the required TTL signals. By using the command "OUT &H3BC, 0", the computer would generate a zero (0) out the parallel port. Additionally the command "OUT &H3BC, 1" would have the computer generate a signal of one (1) out the parallel port. The use of these commands was verified by looking at the signals generated by the parallel port with an oscilloscope. The next task at hand was enabling the computer to generate the entire injection sequence, which included all the information needed to start and end the injection(s) in time with the engine.

To construct an injection sequence consisting of pilot, main and post injections, the ECM must know when to start the injection(s) and the required duration of each event. To simplify the timing control of the ECM, the operator is required to input the engine speed manually. The reason for manual input of the engine speed is because of the simplicity of the input and output of the parallel port. To calculate the timing of the injections the following information must be entered by the operator before each test run:

1. Engine speed [rpm]
2. Start of pilot injection [crank degrees after BDC on the compression stroke]
3. Duration of pilot injection [crank degrees]
4. Start of main injection [crank degrees after BDC on the compression stroke]
5. Duration of main injection [crank degrees]
6. Start of post injection [crank degrees after TDC of compression stroke]
7. Duration of post injection [crank degrees]

From the above information, the injection sequence can be calculated. A copy of the QBASIC code is attached in the appendix that shows the complete calculations. To match the clock speed of the computer to actual engine time a correlation factor was determined. The correlation factor was determined by comparing the time it takes for the computer to complete a task with the time it should have taken on an oscilloscope. An appropriate correlation factor was then multiplied by the task to equate the two.

For the computer to differentiate between the piston approaching TDC on the compression stroke or the exhaust stroke, a ferromagnetic trigger wheel was placed on the camshaft. The trigger wheel consisted of 120 teeth with two gaps. If positioned correctly, these two gaps would correspond to each TDC of the engine cycle (TDC on compression and TDC on exhaust). A small magnetic sensor is placed in close proximity to the wheel such that as the teeth pass by the sensor, the magnetic flux within the sensor changes and creates a small sinusoidal voltage that corresponds to the camshaft speed. By using the input of the parallel port of the computer, the computer could determine the starting trigger of the injection sequence by distinguishing between 0 and 5 VDC, with a corresponding 0 or 1. For instance, the computer could wait until it received an input of 5 VDC before it sent the injection sequence in TTL form. The command "N = INP(&H3BD)" would have the computer store the value recorded from the input pin as N. If the value of N were a 1 or high, then the computer would start the sequence. If the value of N was 0 or low, then the computer would continue to wait until the value of N was a 1 or high. Originally, the output of the trigger wheel was going to be used to indicate TDC on compression. However two problems prevented its use. The first was

that the sinusoidal voltage varied between 0 and 1VDC. For the computer to distinguish this as a high or low signal, it would have to be boosted with an OPAMP such that the peak of the sine wave would be 5 volts. The second problem was that two TDC positions were shown in the sine wave during one revolution of the camshaft. So for the computer to be properly synchronized with the engine, some other sensor would have to be used to differentiate between TDC on compression and TDC on exhaust. To simplify both problems, a fiber optic light sensor would be placed on the face of the camshaft trigger wheel and used in such a manner as to calculate engine speed and trigger the beginning of the injection sequence. The fiber optic light sensor uses a red light emitting diode (LED) light source to emit light upon a dark, unreflective surface. The fiber optic light sensor and the trigger wheel can be seen in Figure 22. When the LED passes over a reflective surface, the reflection grounds the 5VDC source excitation to ground. The grounded signal can be registered as a pulse, which can be counted with a frequency-to-voltage counter and converted to engine speed. In addition, if the reflective tape is placed correctly it could be used, as the external trigger needed to start the injection sequence. To minimize drift and magnification of any timing error, the light sensor was placed at bottom dead center (BDC) before the start of compression. This minimized the time needed to wait after the injection sequence had begun before the first injection.

Using the equation:

$$Time[ms] = \frac{CA * 60}{360 * RPM} * 1000$$

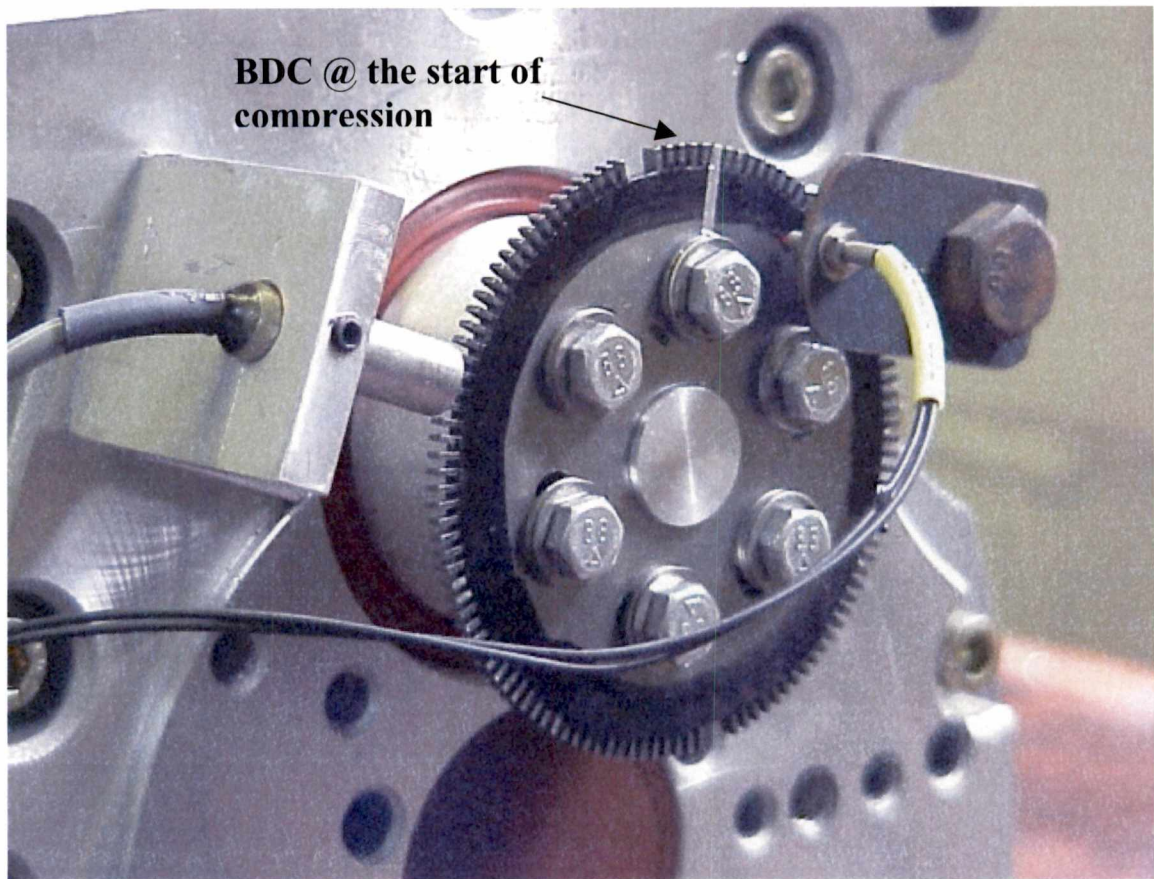


Figure 22. Fiber Optic Light Sensor

the time can be calculated for a determined number of crank angles (CA) to occur at a specific engine speed in revolutions per minute. The factor of 60 converts minutes to seconds, the factor of 360 converts revolutions to degrees, and the factor of 1000 converts seconds to milliseconds. Comparisons of the time the computer calculates using the above equation and the actual time it takes to receive the trigger and execute high and low commands showed acceptable tolerances. Figure 23 shows the comparison of the time it takes to rotate 140 degrees at 800rpm, 29.2ms, to the actual time the computer requires to receive the BDC input, count 140 degrees, and begin the pilot injection. The time required by the computer is 29.6 ms, which is 0.4 ms more than actual. At 800 rpm

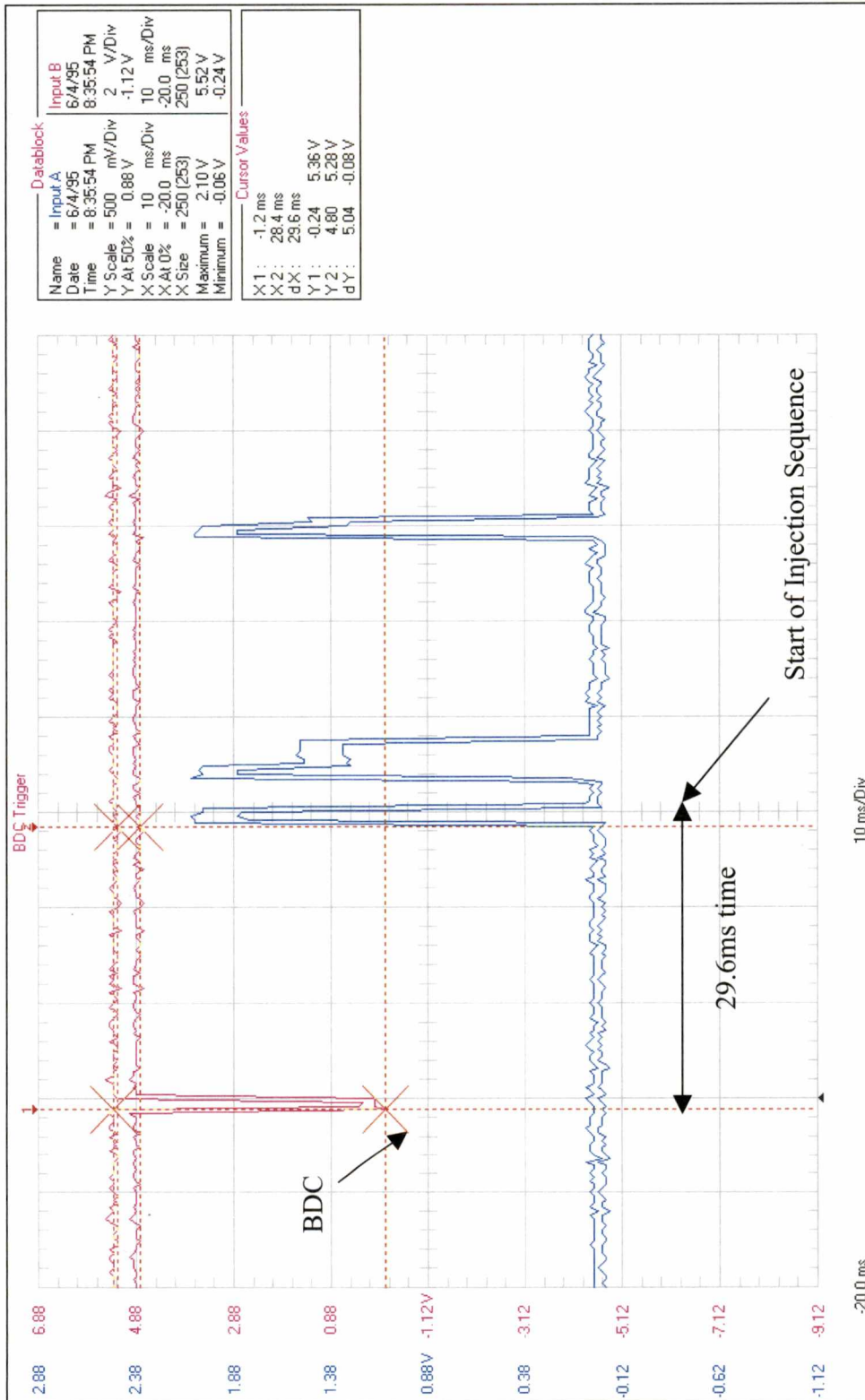


Figure 23. Computer and Engine Time Event Comparison

this is an error of almost 2 crank degrees, which is within the error of the placement and thickness of the reflective piece of tape. At the early stages of the development of this test cell this is viewed as an acceptable error.

Even though the current ECM program only allows for three injections, the software can easily be modified to allow for more. The only limitations for future research would be the limited accuracy of the triggering and timing mechanism and the minimum injector open time to flow fuel.

Fuel Injection Testing

Testing of the fuel injection equipment including the programmable ECM, began with the verification that the ECM could repeatedly and reliably create and send the injection sequence on time with the engine.

To begin this, the ECM was trigger by the engine rotation to send the injection sequence once each revolution of the camshaft or every other crankshaft revolution. This was completed for engine speeds ranging from 800 (idle) to 2800 rpm for every two hundred rpm. Initially the ECM failed at speeds greater than 1000 rpm. It was determined that the last command, which required the ECM to wait a calculated amount of time until the end of the sequence, was causing the ECM to miss the next engine cycle trigger. Removal of this line of code, which allows the injection sequence to end after the end of the post injection, proved to be successful for all engine speeds.

The next test of the fuel injection equipment tested the pump's ability to create high injection pressures. For this, the pump was coupled to an electric motor, which rotated the pump at a constant 950 rpm. Initially, the pump was driven by the use of a timing

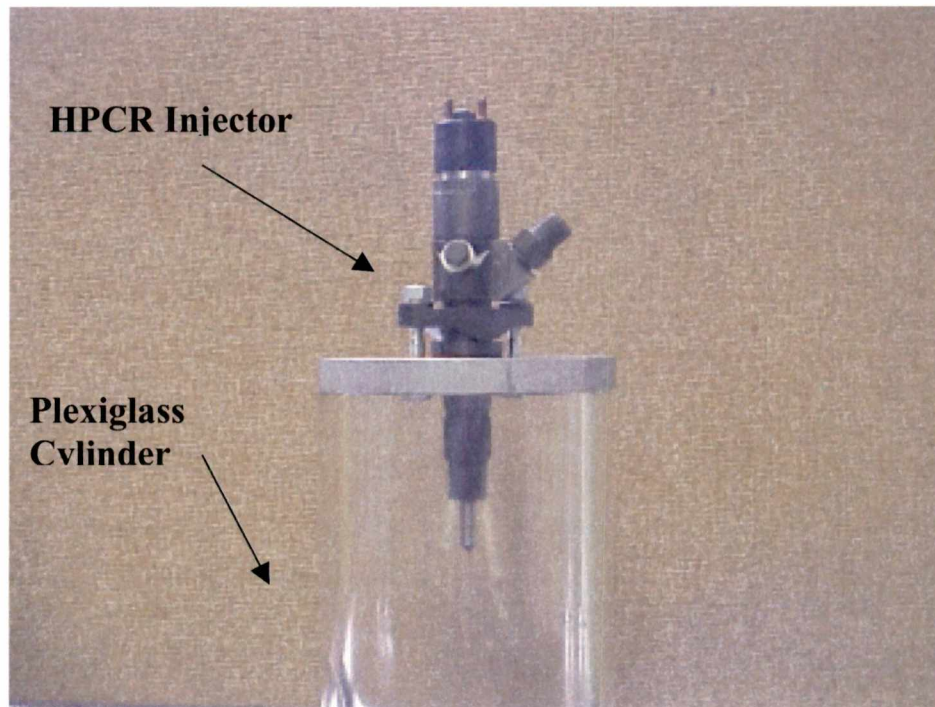


Figure 24. Injector Installed in Plexiglass Cylinder

belt and pulleys. However, after some testing, the belt and pulleys had to be replaced with sprockets and a roller chain assembly to prevent slippage. When the pump was delivered to UTK, no fuel flow schematic was enclosed. Therefore to determine the proper flow of the fuel, a trial and error method was employed. After the correct arrangement of the fuel lines was determined, the pump created positive pressure in the fuel rail. To test the entire fuel injection system as one unit, a temporary test stand was developed that placed the fuel injector in a plexiglass cylinder instead of the engine's cylinder. In addition to safety and fuel spray collection, the use of the plexiglass cylinder allowed for visual conformation of the injection event. A picture of the plexiglass cylinder can be seen in Figure 24 and a picture of the entire test stand setup can be seen in Figure 25. The test stand used the engine's camshaft mounted fiber optic light sensor to trigger the beginning

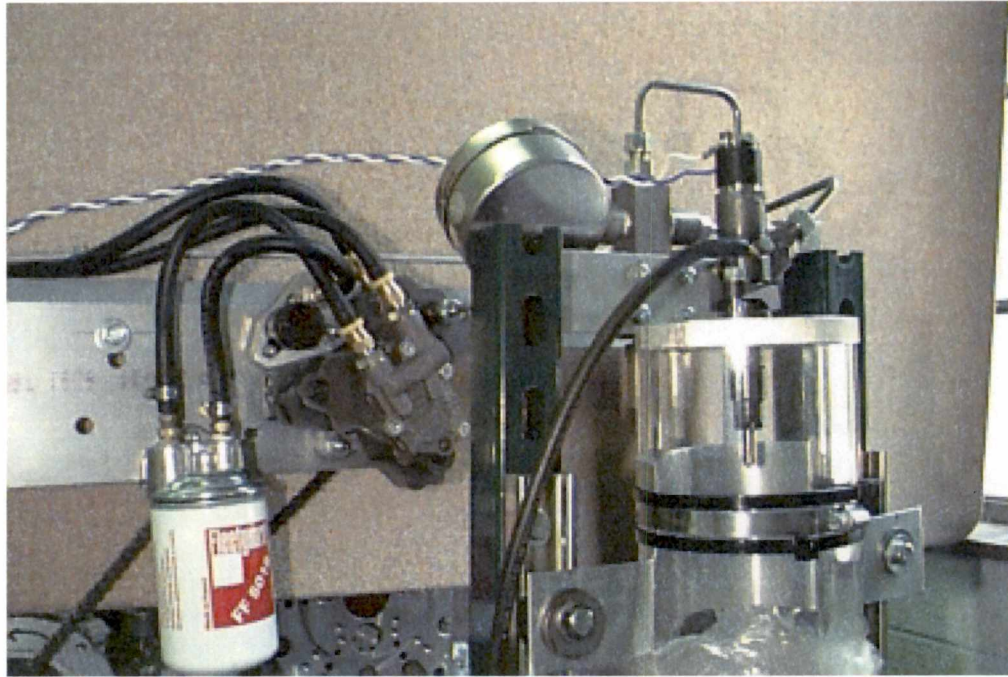


Figure 25. HPCR Test Stand

of the injection and the electric motor to drive the pump. Early tests were conducted to evaluate the system's robustness, repeatability, and flexibility.

To verify the system's ability to inject multiple injections during one engine cycle, the current to the injector was monitored with an oscilloscope and recorded to a PC based viewing software. Figures 26, 27, and 28 show the current waveforms for one, two and three injections for the engine operating at 800 rpm. The external trigger, BDC based on crank revolution, is shown on the top, shown in pink, and the subsequent injection sequence is shown on the bottom of the plot in blue. The viewing software for the Fluke Oscilloscope used in this project plots a high and low value of the waveform. The actual value is the average between the two semi-parallel plots. The trigger plot is set at 5VDC until grounded to 0VDC by the passing of BDC at the beginning of the compression stroke.

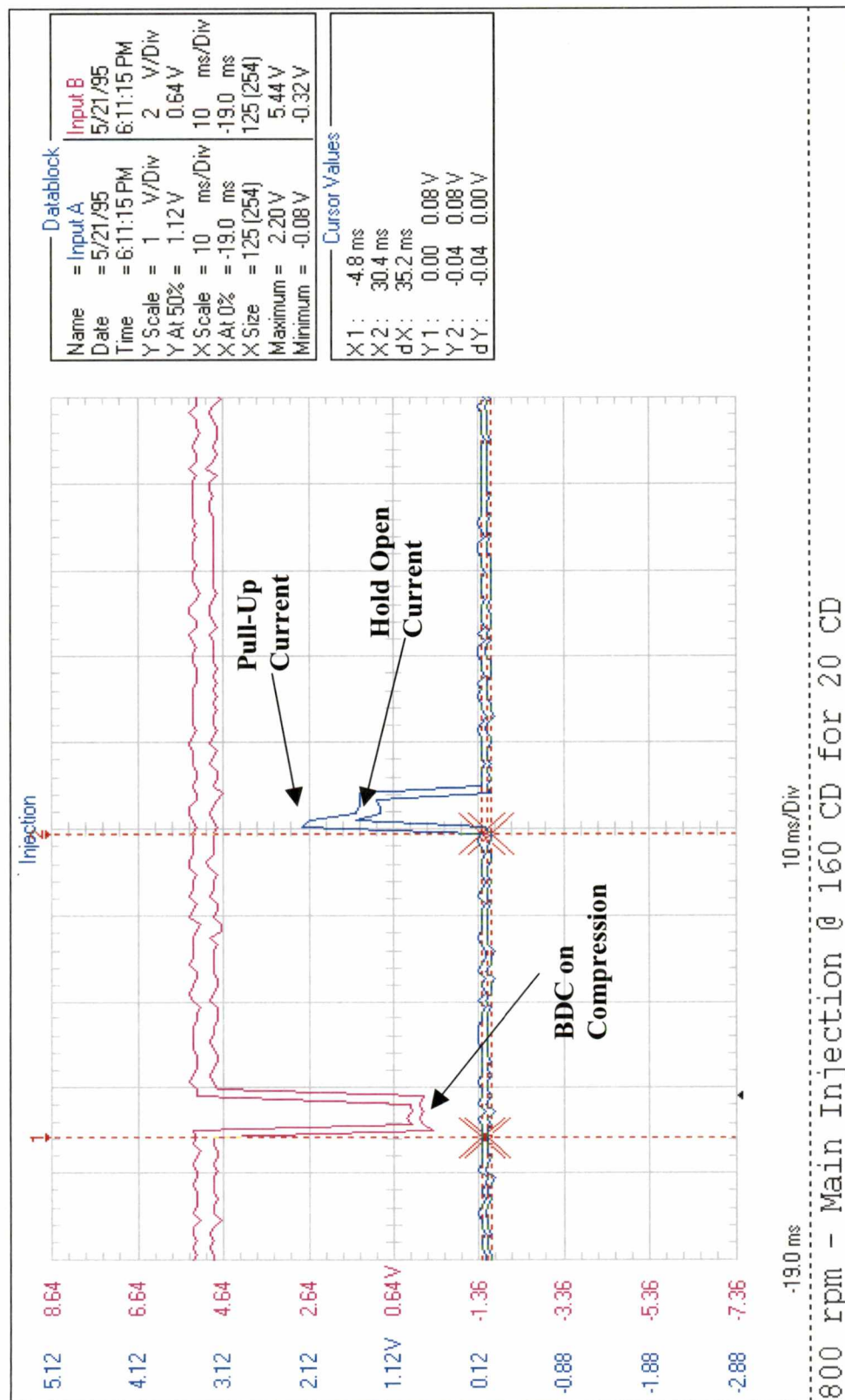


Figure 26. Injection Event

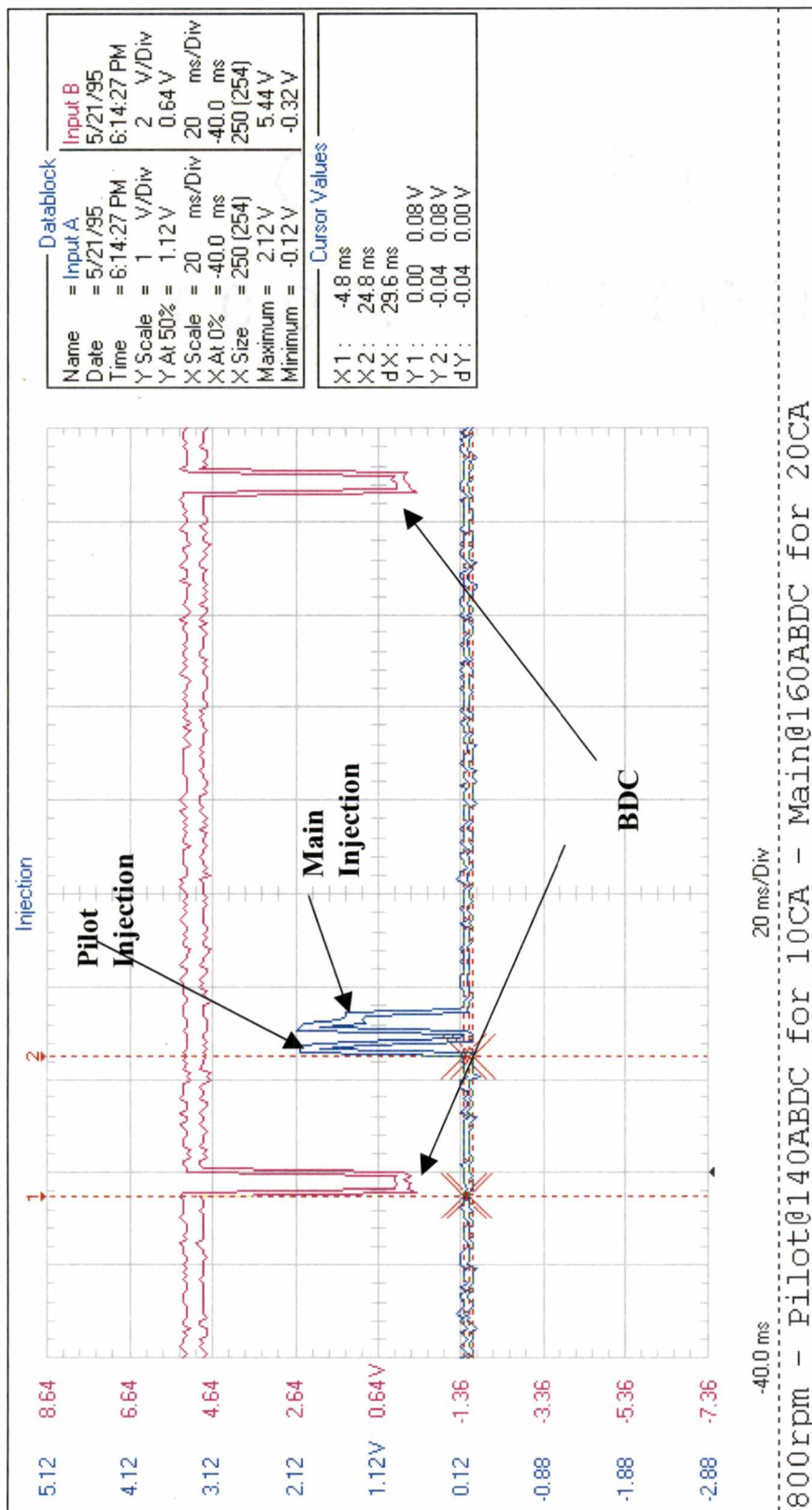


Figure 27. Injection Sequence of Two Injections

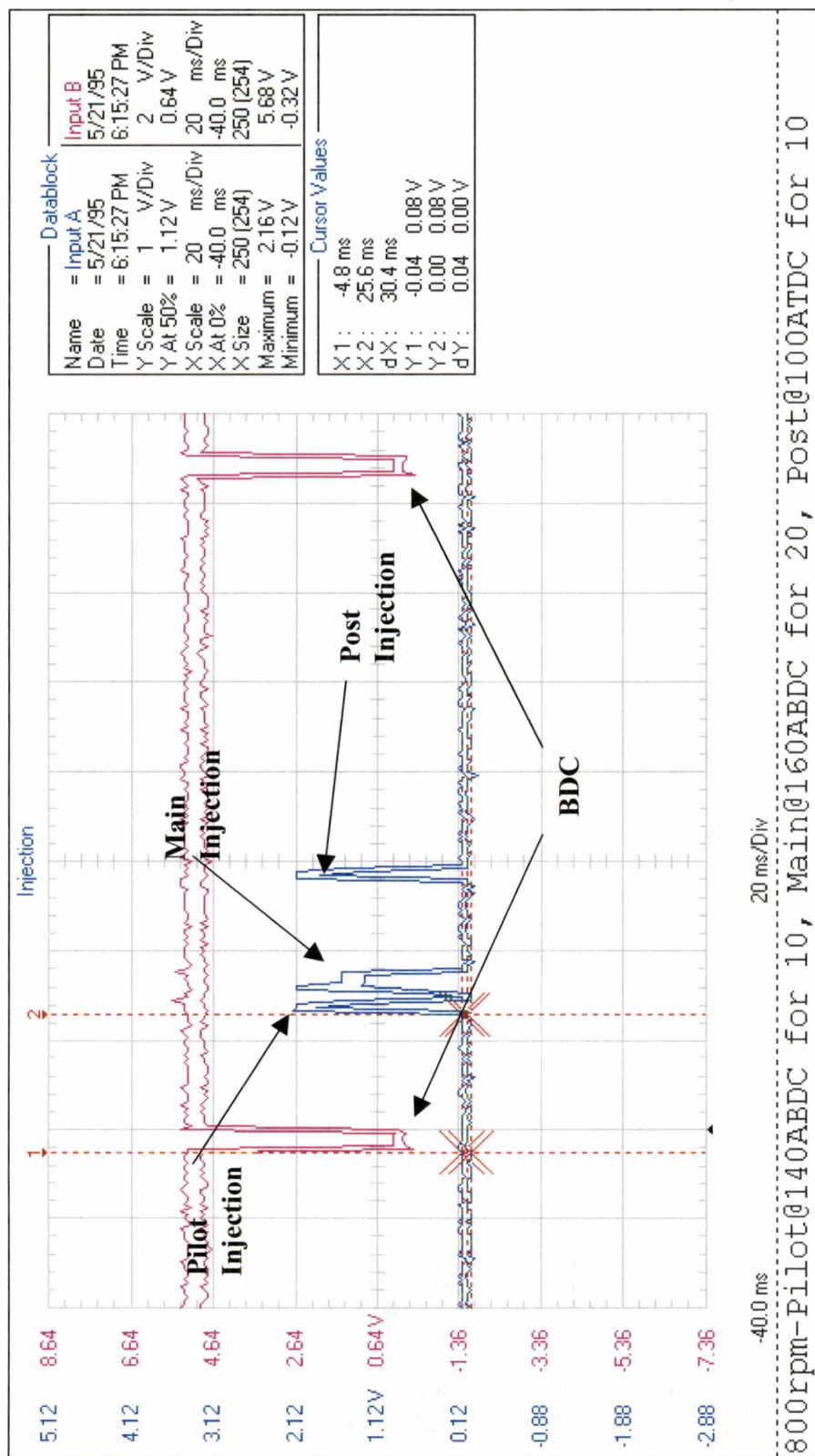


Figure 28. Injection Sequence of Three Injections

Once the trigger signal has been grounded to 0VDC, the computer begins the injection sequence. The actual injection current waveform consists of a bi-level square wave. The first part of the square wave, which is also the highest portion of the waveform, indicates the pull-in or opening current. The second portion of the injection waveform is the hold open current. This represents the current required to hold the injector open. For the purposes of this thesis it can be assumed that fuel begins to flow as soon as the injector solenoid is energized. Even though this is probably not true, to actually determine the instant that fuel begins to flow would require a much more complex test that is beyond the scope of this project.

Information provided by Cummins Engine Company stated that the HPCR injectors being used for this project would operate best with a pull-in current of 18.5 A and a hold-current of 13.5 A.

To gain a further understanding of how the fuel flow rate injected varies with injection pressure a series of tests was run to collect pertinent data. Figure 29 shows fuel flow rates for injection duration of 20 crank degrees for various increasing injection pressures at an engine speed of 1000 rpm. As can be seen from the plot, fuel flow rate increases linearly as injection pressure increases from 4000 psi to 16,000 psi. It is interesting to note that during this test, it was determined that the minimum injection pressure required to open the injector nozzle was 4000 psi. It is also important to note that the fuel flow rate for 20 crank degrees at 1000 rpm for an injection pressure of 4000 psi is almost four times the amount required by the manufacturer at the same crank duration for the same engine speed. Without changing the nozzle, the only way to match the same fuel

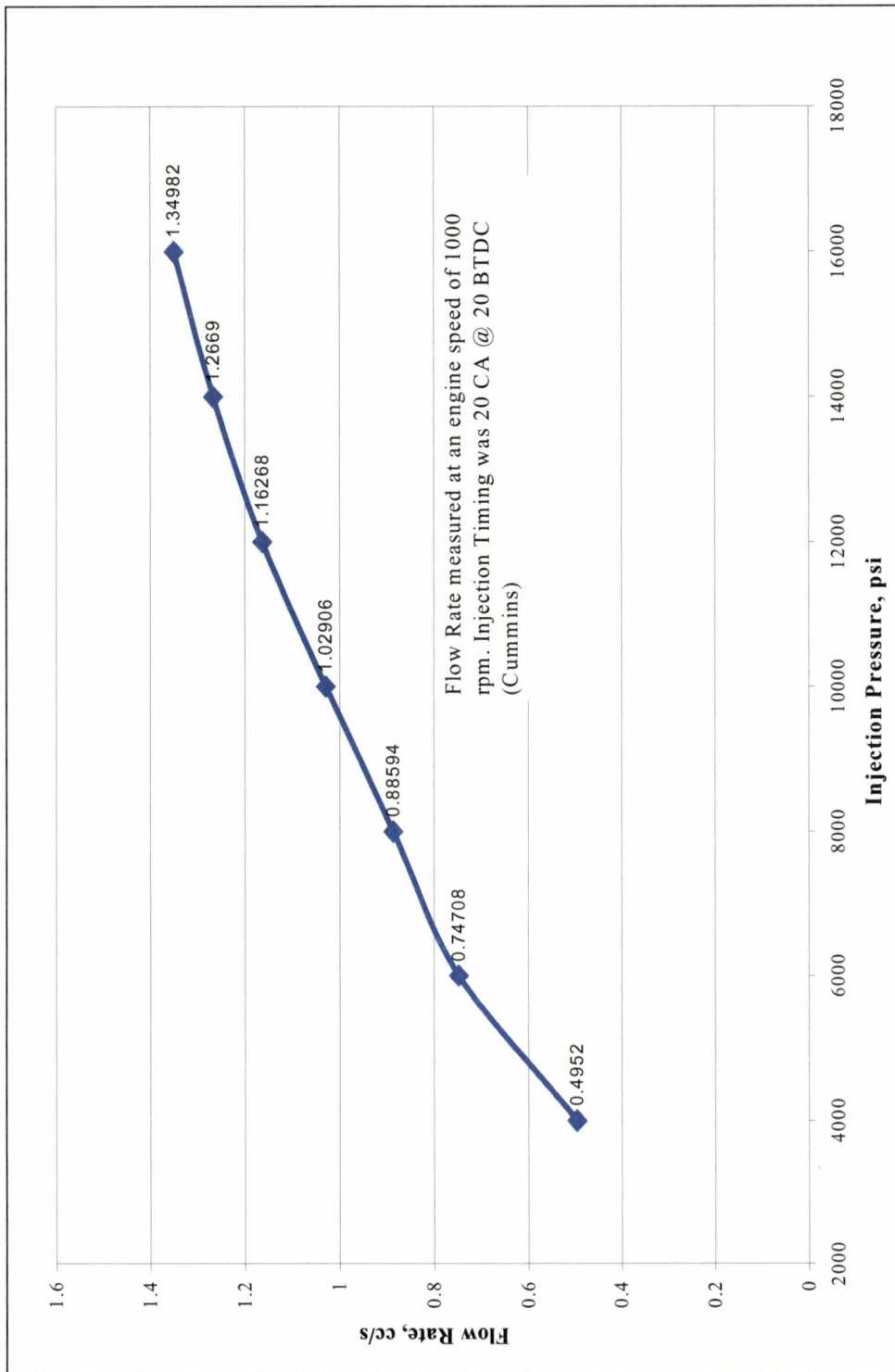


Figure 29. HPCR Injector Flow Rate vs. Pressure

flow rate for a given engine speed was to possibly shorten the time the injector was open. Data collected during follow-up tests is shown in Figure 30. This series of tests collected data for injection pressures of 4000, 6000, and 14,000 psi at various crank angle duration for an engine speed of 1000 rpm. In addition to the Cummins nozzle, a Peugeot HPCR nozzle was used to collect data at 4000 and 6000 psi. The information contained in this plot shows that the Peugeot nozzle has a lower hydraulic flow rate than that of the Cummins nozzle. However, information on the exact hydraulic flow rate of the Peugeot nozzle is not available. These tests concluded that obtaining the original fuel flow rate of the Hatz engine would be difficult for low engine speeds. Additional tests, at an engine speed of 2000rpm, were conducted to investigate the possibility of matching original fuel flow rates.

Plots in Figure 31 show flow rates for the Peugeot nozzle at 6000 psi for 1000 and 2000 rpm. To plot both graphs together, the x-axis is normalized to time instead of crank angles. To match the original flow rate of the Hatz at 1000 rpm, the HPCR system would have to inject fuel at a rate of 0.13 cc/sec. With the current nozzles available to the UTK test cell, this would be impossible. However to match the original flow rate at 2000 rpm (for rated speed and load), the HPCR system would need to inject fuel at a rate of 0.3 cc/sec. Figure 31 shows that the HPCR system operating at 6000 psi could obtain a flow rate of 0.3 cc/sec, if the fuel was injected for a duration of approximately 0.75 msec.

To be able to consistently predict the needed crank duration of injection, a method was developed to determine the time to inject fuel for each engine cycle. The method started by plotting the number of cubic centimeters per injection (cc/inj) at 1000 rpm

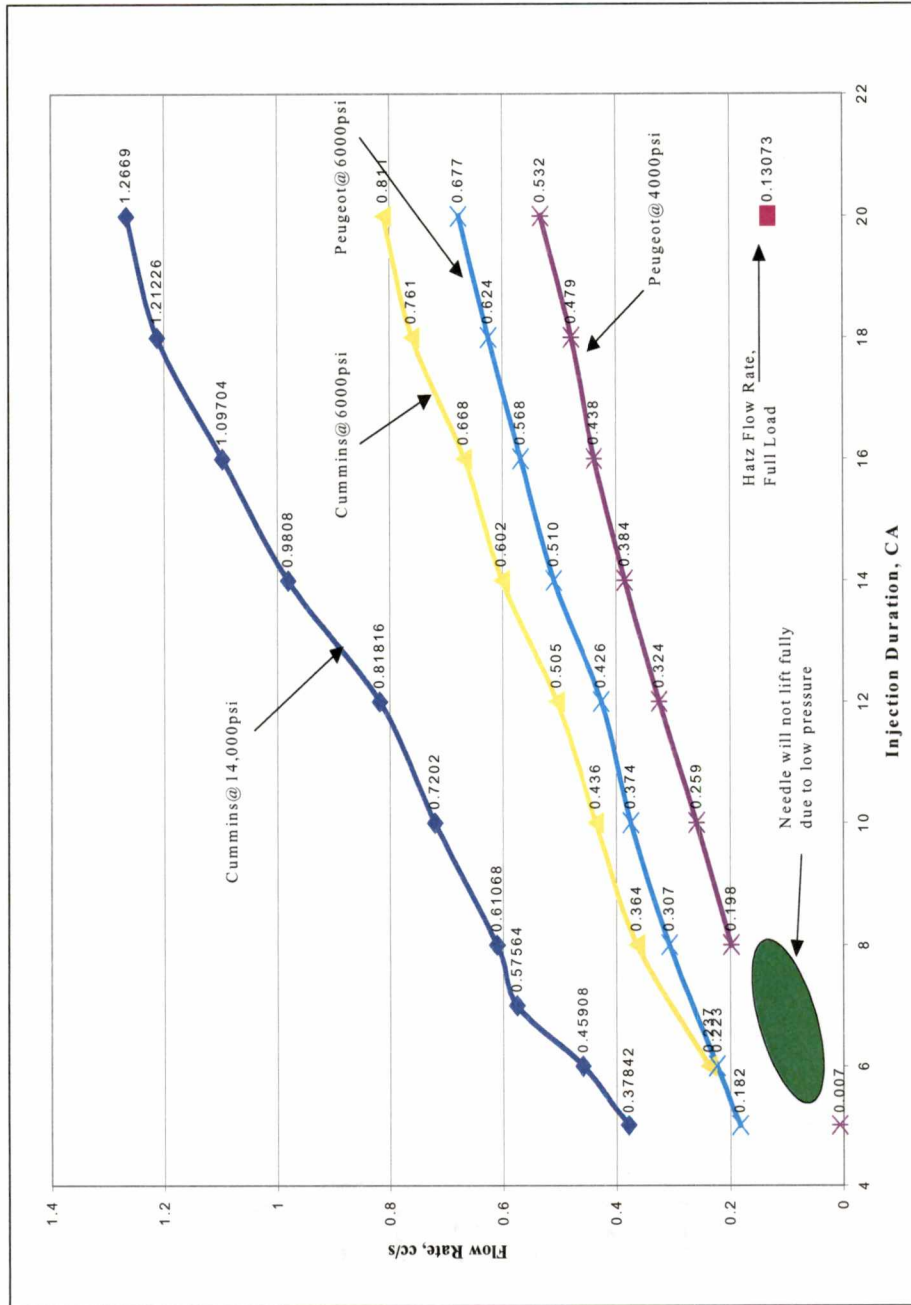


Figure 30. Injector Flow Rate vs. CA for 4000, 6000, 14,000 psi at 1000 rpm

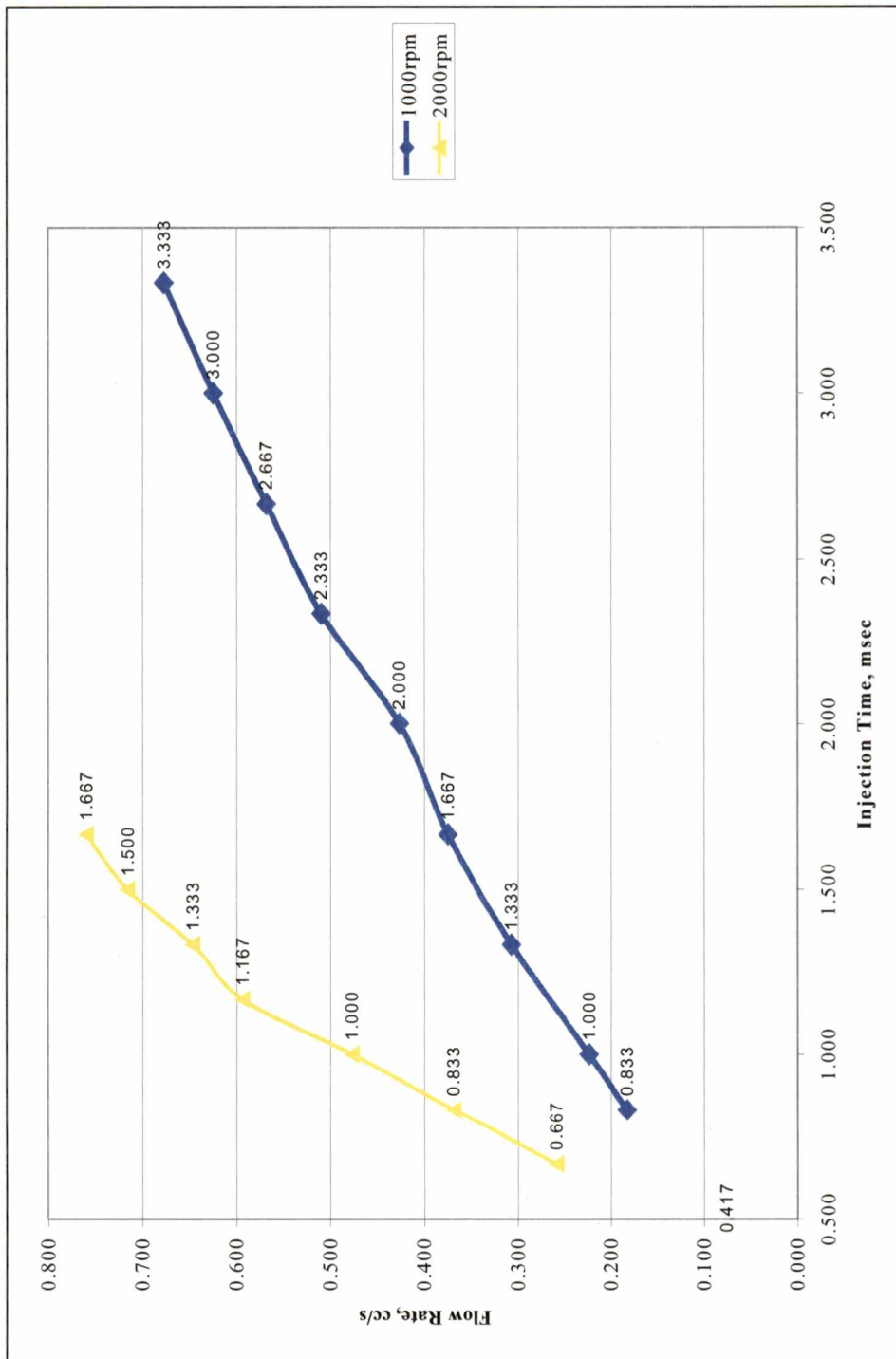


Figure 31. Injection Flow Rate vs. Injection Time for 6000 psi at 2000 rpm

versus the time required for each injection for crank duration varying from 5 CA to 20 CA. Again the x-axis was normalized to time in milliseconds. The cc/inj for 1000 rpm were determined by dividing the various flow rates (cc/sec) by the number of injections per second. The number of injections per second can be determined by knowing that fuel is injected once every other revolution, at 1000 rpm there is 8.33 injections per second. This information is plotted in blue in Figure 32. To predict the time for injection at 2000 rpm, the normalized flow rate for 1000 rpm is divided by number of injections per second for 2000 rpm. This quantity determines the amount of fuel needed during each injection at 2000 rpm. Plotted against the normalized time from the 1000 rpm injection events, this yields information, which will determine injector on time for 2000 rpm. The key to make this method work is to assume that the injector flows the same amount of fuel for a given injection duration, no matter the engine speed. The essence of HPCR makes this statement true because injection pressure generation is independent of engine speed. This information is plotted in Figure 32 as "2000 rpm(C)".

When compared to the actual measured data at 2000 rpm, the predicted is reasonably acceptable. To further test this method of predicting the injector on time per injection for a known cubic centimeter per injection, a test was run which compared predicted data for 2200 rpm to measured data for 2200 rpm. The plot of the comparison can be seen in Figure 33. Again the prediction is reasonably acceptable.

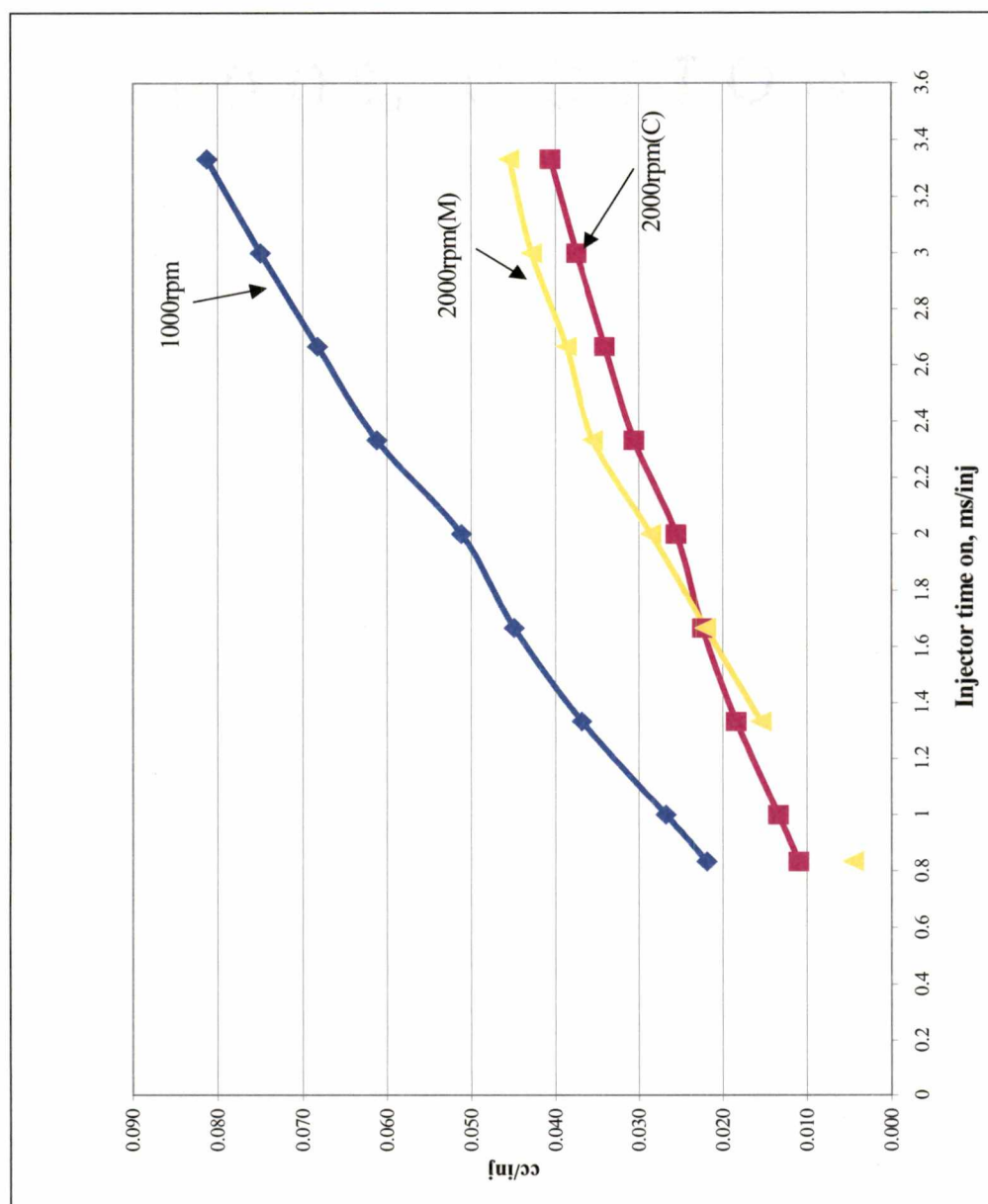


Figure 32. Injection Duration Period

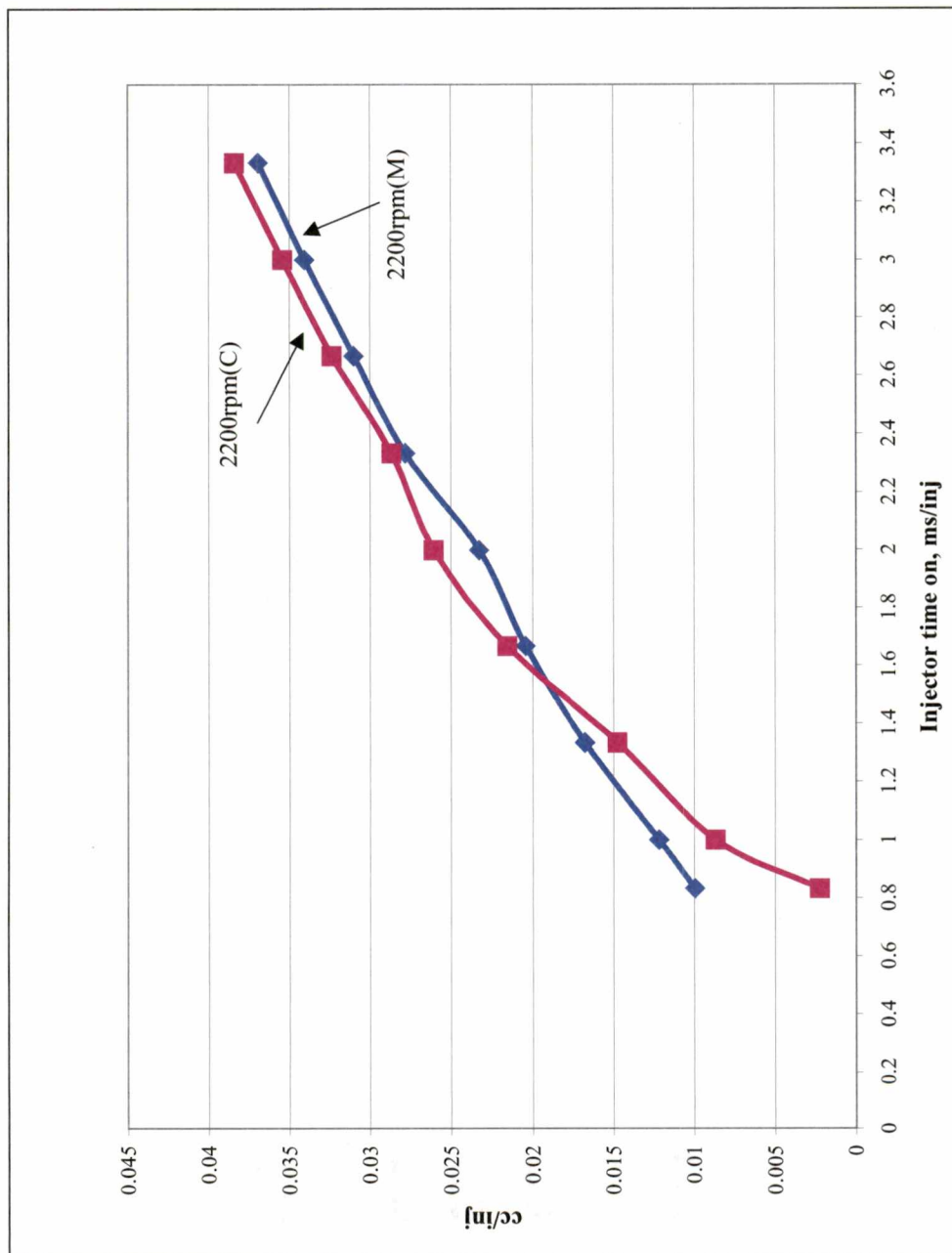


Figure 33. Injection Duration Prediction

To use the method to determine the number of crank angles to open the injector, the injector time on found from the above method must be substituted into the following equation:

$$CA[degrees] = \frac{t * 360 * RPM}{60 * 1000}$$

Since the above equation bases the injection event on once every camshaft revolution or twice every other crank revolution, the CA found in the above equation must be divided by two to make it equate to a crank revolution event.

To test the above prediction method, the crank duration in degrees was predicted for full load at 2000 rpm. To match the original Hatz fuel flow of 0.3 cc/sec, the injector would have to inject 0.018 cc/inj or 18 mm³/inj. Using Figure XX, the time required to inject 0.018cc would be approximately 1.4 milliseconds. Using the above equation, 1.4 milliseconds are equal to 8 CA based on crank revolutions. This is verified by Figure 34, which shows that for an injection event of approximately 8 CA at 2000 rpm, the Peugeot HPCR injector flows fuel at a rate of 0.307 cc.

Complete System Testing

Once the HPCR fuel system had been tested and verified, as to its ability to inject fuel in time with the engine, the system was moved and installed into the Hatz engine. The installation of the equipment was followed by a complete system test, which included running of the engine under the management of the HPCR programmable ECM.

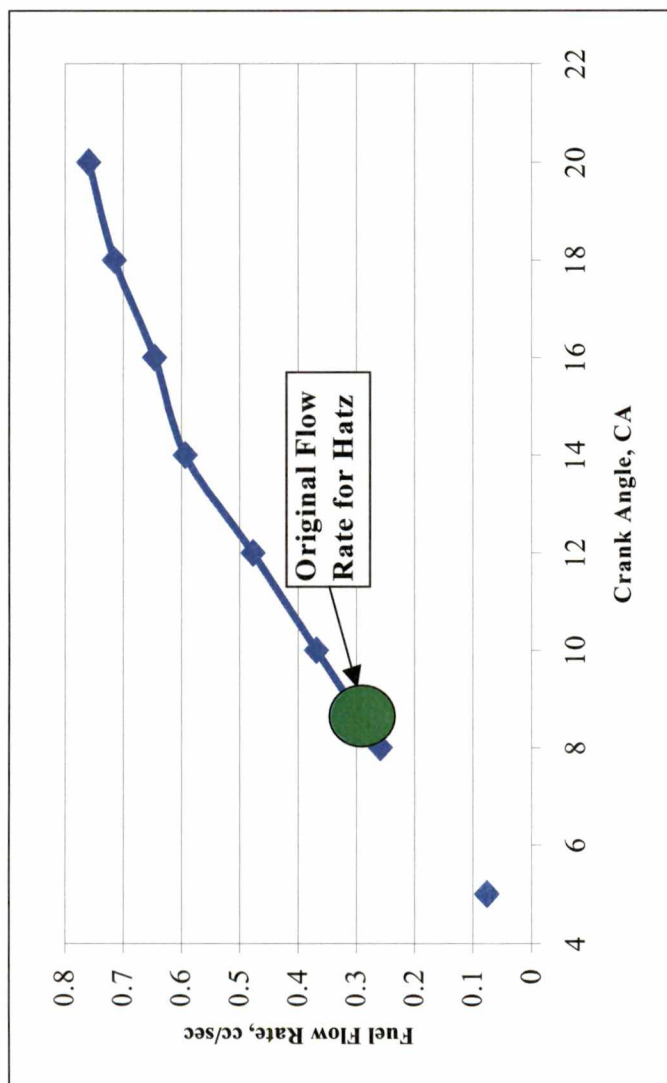


Figure 34. Fuel Flow Rate for the Peugeot HPCR Nozzle

To install the new fuel injector into the Hatz cylinder head, little modification was required to the original injector mounting. Because the Cummins injector and the Hatz injector both utilized a 19mm Bosch injector body and a 20° inclination angle, the injector port in the cylinder head did not require any modifications. The only modification that was required was the fabrication of a new clamping yoke. The clamping yoke is the fixture that holds the injector in place in the cylinder head. The yoke slides into position above two grooves on the injector and once in place, a single bolt is used to hold the injector firmly in place. The original Cummins injector used a clamping yoke that used was affixed to the injector and was similar to a flange, which used two bolts to hold the injector to the cylinder head. To mount the Cummins injector into the Hatz engine, the two-bolt flange was removed and a yoke, similar to the original yoke, was designed and fabricated such that the original bolting system could be used.

The fuel rail was mounted to a bracket, which is affixed to the crankcase of the engine. The rail is mounted in position such that the same fuel lines used in test stand could be reused and finally the mount, which holds the HPCR fuel pump and remote fuel filter, was relocated such that it was reconnected to the fuel rail.

To verify the injection system's ability to operate the engine under its own management, a test was conducted, which would operate the engine at full rated power at 1200 rpm. The original Hatz fuel management system injected fuel at a rate of approximately 0.14 cc/sec. Note, the original Hatz fuel management injected fuel at 20° before top dead center (BTDC) for 20 CA. To match the same fuel flow rate of 14 cc/sec, using the HPCR system with the Peugeot nozzle, the new system would have to only

inject fuel for 5 CA. To maintain safe operating conditions, the fuel injected for 5 CA was injected at 5° BTDC. This was initiated to keep the peak of the pressure curve on the power stroke side of TDC. Data was collected for 4 individual runs at 1200 rpm using the HPCR system. Exhaust temperatures and power information recorded showed that the HPCR system was indeed creating positive power and burning fuel. The data was collected using the same method used in all data collect, i.e. 200 samples, sampled at 100 samples per second. The summary of the data is shown in Table 2 below.

The information shows that for the same fuel flow rate, the power increased by 19% and the fuel consumption is likewise improved by 18%. The results from this test will be discussed in later.

Table 2. HPCR Injection Management Comparison

Test Run	Engine Speed, rpm	Power, HP	Fuel Flow, cc/sec	Fuel Consumption, lb/HP-hr
1.00	1203.18	3.20	0.13	0.27
2.00	1202.43	3.00	0.13	0.30
3.00	1203.53	3.30	0.15	0.30
4.00	1203.30	3.23	0.15	0.30
Average	1203.11	3.18	0.14	0.29
Original Configuration	1200.00	2.68	0.14	0.35
Difference	-	19%	-	18%

CHAPTER V – CONCLUSIONS

Results and Discussion

As outlined in the Chapter I, the purpose of this thesis was to create a test engine that used a HPCR fuel system and a programmable ECM to modify the injection time, length, and the injection rate of the injection event. The purpose of this test cell, including the engine, data acquisition equipment, and the engine dynamometer, would be determined by future researchers, however research into fuel injection optimization and emissions research are within the scope of the test cell's ability.

The procurement and installation of the HPCR pump, rail, and injector completed the task of converting a mechanically fuel injected diesel engine to one that has a complete HPCR fuel system. The development and testing of the programmable ECM proved that indeed the HPCR system could be managed in such a manner as to modify the fuel system's injection time, length, and injection rate. Data collected from a digital oscilloscope showed that the ECM could modify these injection parameters and can be seen in Figures 26, 27, and 28 of Chapter IV.

The final testing of the entire HPCR fuel system, including the ECM, proved successful by operating the test engine under its own fuel management. In fact, the new system operated the engine with an efficiency range not achieved by the original mechanical system. Tests showed an increase in horsepower and a decline in brake specific fuel consumption. Observers should keep in mind that the injection event recorded for this test was not comparable to the original injection event. Timing and injection pressure were modified in such a manner to suit these benefits. However, the

ability to modify these events to optimize engine-operating conditions over the entire operating range is exactly the use and future of this test cell.

Recommendations

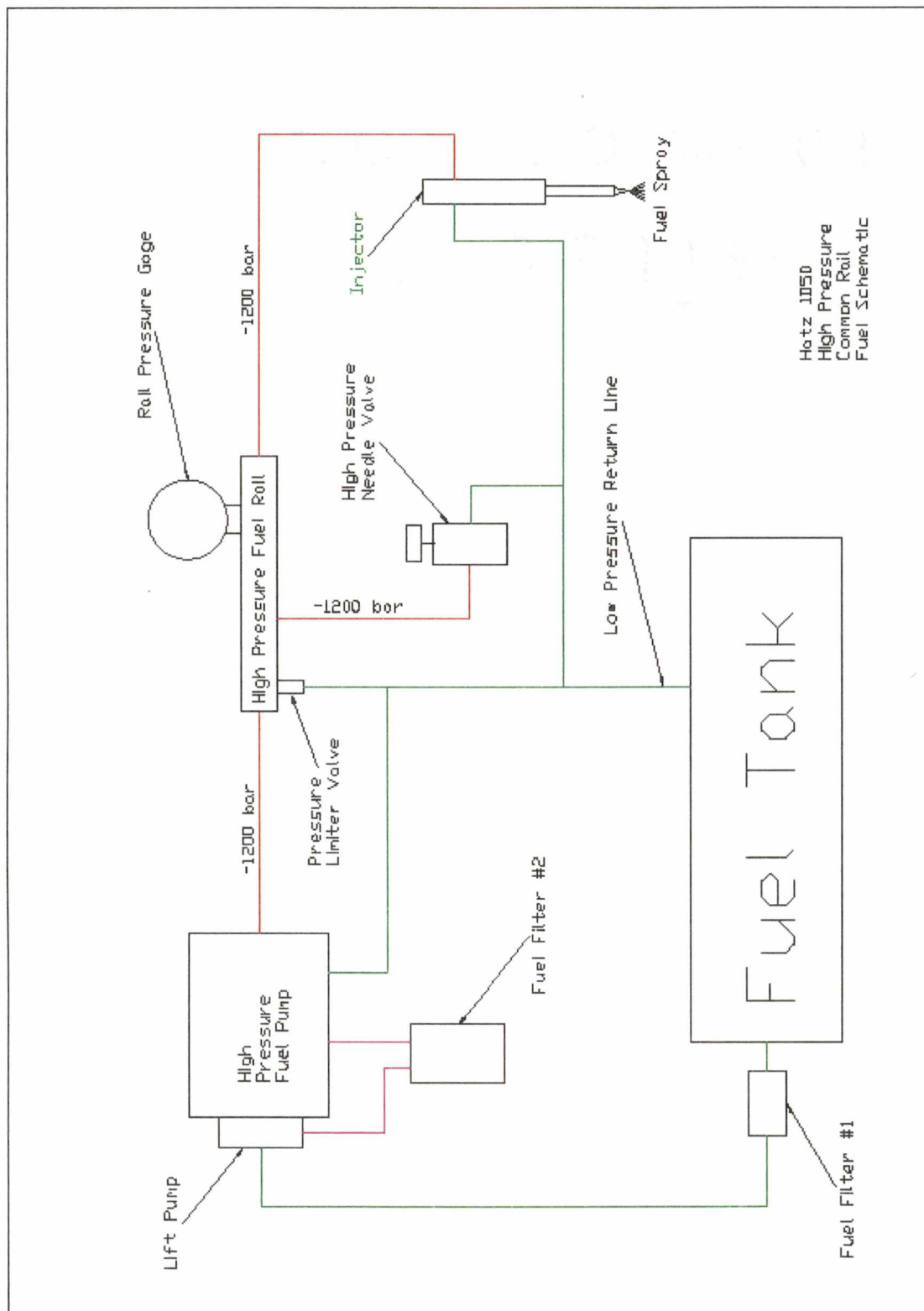
The following recommendations can be made to improve the test cell's ability to conduct meaningful and timely research:

1. The installation of a shaft mounted torque meter specified for the range of the test engine.
2. Variable speed electric motor to operate the HPCR fuel pump at varying rotational speeds.
3. Graphical user interface for programmable ECM control.
4. Injector nozzle to match original hydraulic flow rate.
5. A more accurate device used for injection timing.

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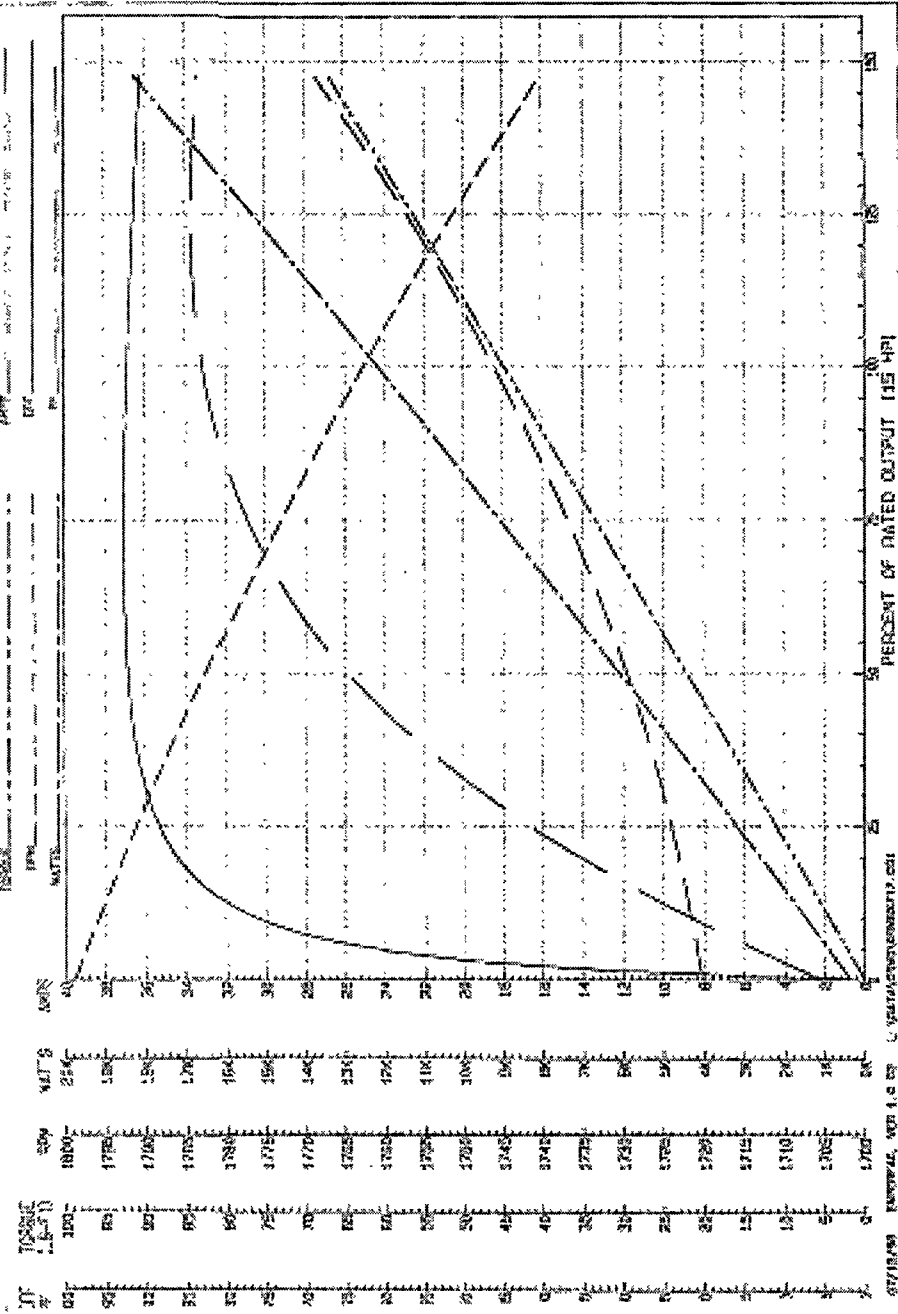
APPENDIX



WINDING # D96GX717

15 HP 3 PH 60 HZ 1725 RPM 480 V 3PH
TERMINALS (LB-71). PO-155 PU-68.2 LB-79.1 LPA-122.9

BALDOR ELECTRIC COMPANY





Certificate of Analysis

PHILLIPS CHEMICAL COMPANY

A DIVISION OF PHILLIPS PETROLEUM COMPANY

SPECIALTY CHEMICALS

P.O. BOX 988

BIRMINGHAM, TX 75003-0988

DATE OF SHIPMENT
09-22-03

CUSTOMER ORDER NO.
62XLU308

INV/REC'D NO.
484484

CONTAINER NO.
6 X 54 GALLON DRUMS

DIESEL OIL SULFUR TYPE 2 LOT D-434

TESTS	RESULTS	SPECIFICATIONS	METHOD
Specific Gravity, 60/60	0.8434	Report	ASTM D-4052
API Gravity	58.28	32 - 57	ASTM D-4233
Conc'n, 50°C, 3 hrs.	5A	3.0 Max	ASTM D-4130
Sulfur, Wt%	0.3	0.03 - 0.05	ASTM D-2622
Flash Point, 1P, Pm	158	150 Min.	ASTM D-57
Pour Point, 1P	-3	Report	ASTM D-57
Cleat Point, 1P	-0.3	Report	ASTM D-2500
Viscosity, cs-40C	2.5	2.2 - 3.4	ASTM D-445
Carbon, wt%	88.8	Report	Calculated
Hydrogen, wt%	13.4	Report	Calculated
Calorific Density (grammation)	2758	2750 - 2800	Calculated
Net Heat of Combustion, BTU/lb.	18462	Report	ASTM D-3336
Particulate Matter (mg/l)	3.9	18 Max.	ASTM D-3279
Cetane Index	47.7	+0 - +6	ASTM D-578
Cetane Number	48.0	43 - 49	ASTM D-813
DISTILLATION, 1P			
IBP	353.0	340-400	ASTM D-38
5%	437.3		
10%	422.1	450-480	
15%	433.6		
20%	445.0		
30%	453.7		
40%	453.2		
50%	458.7	470-540	
55%	513.6		
70%	630.0		
80%	651.6		
90%	663.7	650-630	
95%	610.1		
EP	646.4	610-690	
Loss	0.4		
Residue	1.0		
HYDROCARBON TYPE, VOL%			
Aromatics	22.2	27 Max.	ASTM D-1318
Olefins	2.0		
Saturates	65.6		
Aromatics	33.0		ASTM D-5483

RT/ps
052248
RFJTCO

```

*****
*Computer code for universal solenoid driver
*Doug Ferguson - Master's Thesis Project - Spring 2001
*Development of a Single-Cylinder Common Rail Fueled Diesel
*Test Engine
*
*This code is to create a TTL (0-5VDC) signal to determine the
*Timing and injection length of the three (pilot, main, and post)
*injections.
*The parallel port will be used for the TTL signal out and trigger
*signal in. The address for the parallel port output on IBM
*433DX *machine is 3BC in hexadecimal. ZERO (0) is TTL low
*(0VDC) and ONE is TTL high (5VDC).
*The address for the input is 3BD in hexadecimal. A code of "127"
* is high 'and "255" is low.
*****

```

CLS

INPUT "Enter engine speed in rpm...", RPM

INPUT "Enter the start of pilot injection in degrees ABDC...", PLTSRT

INPUT "Enter the injection duration for the pilot injection in crank angle degrees...",
PLTDUR

INPUT "Enter the start of main injection in degrees ABDC...", MNSRT

INPUT "Enter injection duration for the main injection in crank angle degrees...",
MNDUR

INPUT "Enter the start of post injection in degrees ATDC...", PSTSRT

INPUT "Enter the injection duration for the post injection in crank angle degrees...",
PSTDUR

PLTTIME = (PLTDUR * 60 * 1000) / (RPM * 360)' in milliseconds

MNTIME = (MNDUR * 60 * 1000) / (RPM * 360)' in milliseconds

PSTTIME = (PSTDUR * 60 * 1000) / (RPM * 360)' in milliseconds

A = ((PLTSRT * 60 * 1000) / (RPM * 360)) / .145

B = PLTTIME / .145

C = (((MNSRT - PLTSRT - PLTDUR) * 60 * 1000) / (RPM * 360)) / .145

D = MNTIME / .145

E = ((PSTSRT * 60 * 1000) / (RPM * 360)) / .145

F = PSTTIME / .145

G = (((360 - PSTSRT - PSTDUR) * 60 * 1000) / (RPM * 360)) / .145

100

N = INP(&H3BD)

```
IF N = 255 THEN

FOR i = 1 TO A
    OUT &H3BC, 0
NEXT
FOR j = 1 TO B
    OUT &H3BC, 1
NEXT
FOR k = 1 TO C
    OUT &H3BC, 0
NEXT
FOR l = 1 TO D
    OUT &H3BC, 1
NEXT
FOR m = 1 TO E
    OUT &H3BC, 0
NEXT
FOR N = 1 TO F
    OUT &H3BC, 1
NEXT
OUT &H3BC, 0

GOTO 100
ELSEIF N = 127 THEN

GOTO 100

END IF

END
```

VITA

Harley Douglas Ferguson, III was born in Knoxville, TN on May 24th, 1973. Doug attended the Farragut school system, in Farragut, TN where he graduated from Farragut High School in the spring of 1991. He enrolled in The University of Tennessee, Knoxville the following fall to study mechanical engineering. In the fall of 1993, he was academically dismissed from The University and continued his education at Mississippi State Technical Community College until he was readmitted to UTK in the fall of 1996. With a newfound and earned discipline he finished his Bachelor of Science degree in Mechanical Engineering in the summer of 1999. As an undergraduate student he accepted a position the summer of his junior year to assist researchers from Oak Ridge National Labs, who were conducting research in the area of diesel engine emissions. The research and subject area interested him, which prompted him to accept a graduate research assistantship to conduct research for the Army concerning portable diesel fueled generator units and work towards a Master of Science degree in Mechanical Engineering.

Toward the conclusion of his work at UTK and ORNL he got engaged to Meghann Shaw of Knoxville and accepted a job with Southwest Research Institute in San Antonio, TX as a Research Engineer conducting research in the area diesel engine emissions.