University of Southern Queensland

Faculty of Engineering and Surveying

DESIGN OF ELECTRO - HYDRAULIC CONTROLLED POPPET VALVES

A dissertation submitted by

Robert Stewart

In fulfillment of the requirements of

Courses ENG4111 and 4112 Research Project

Towards the degree of

Bachelor of Mechatronics

Submitted: November 2007

Abstract

The purpose of this report is to outline all the necessary aspects that must be considered in the design of a fully variable "poppet" valve within the current generation spark ignition or compression ignition engines. This encompasses all of the mechanical and electrical facets involved with the design, implementation, calibration and also investigates the predicted benefits of such a system. The report includes details of the pumping system, actuating system, control solenoids and control system.

Other considerations are given to the implications upon the immediate world technically, socially, environmentally and financially with the subsequent completion and integration of the design into the automotive industry.

Due to the time restrictions of this course, integration of the system onto a current generation engine was impossible, however the successful completion of the electronics involved with the control system can be seen to be working; causing the high voltage solenoids to pulse at the required time and duration on the test bench for the approximate desired output. Once this system is retrofitted the pulsed fluid from the high voltage solenoid would contain enough fluid force to lift the appropriate "poppet" valve from it seat.

In the completion of this report it is able to be seen that this system or similar will revolutionize both current spark ignition and compression ignition engines in terms of overall power, economy and emission reduction to a degree that has never been seen before in the automotive industry. The conservation of the environment is a high priority in Australia and internationally. Without delving into the intricacies of the debate of greenhouse gases, this report recognizes that vehicle emissions contribute a significant amount pollution. This is why it is imperative to design and implement more efficient engines globally via technologies such as this.

University of Southern Queensland

Faculty of Engineering and Surveying

ENG4111 Research Project Part 1 & ENG4112 Research Project Part 2

Limitations of Use

The Council of the University of Southern Queensland, its Faculty of Engineering and Surveying, and the staff of the University of Southern Queensland, do not accept any responsibility for the truth, accuracy or completeness of material contained within or associated with this dissertation.

Persons using all or any part of this material do so at their own risk, and not at the risk of the Council of the University of Southern Queensland, its Faculty of Engineering and Surveying or the staff of the University of Southern Queensland.

This dissertation reports an educational exercise and has no purpose or validity beyond this exercise. The sole purpose of the course "Project and Dissertation" is to contribute to the overall education within the student's chosen degree programme. This document, the associated hardware, software, drawings, and other material set out in the associated appendices should not be used for any other purpose: if they are so used, it is entirely at the risk of the user.

Prof Rod Smith
Dean
Faculty of Engineering and Surveying

Certification

I certify that the ideas, design and experimental work, results, analyses and conclusions set out in this dissertation are entirely my own effort, except where otherwise indicated and acknowledged.

I further certify that the work is original and has not been previously submitted for assessment in any other course or institution, except where specifically stated.

Name:

Student Number:

Signature

Date

Acknowledgments

A considerable thanks must be given to my current employer Lachlan Riddel of ChipTorque Performance for allowing time off work to complete this dissertation. Thanks must also be given to him for allowing the use of his dynometer cell, electronic hardware testing benches and acquisition of various parts in order for implementation and performance testing and also his practical hands on guidance.

Thanks must also be given to micro electronic wiz Rhys Foster for the help of circuit design and analysis.

Thanks to John Billingsley, my supervisor for his mathematical modeling assistance and patience during the compilation of this report.

Table Of Contents

1.0 Introduction ……………………………………………………………10

1.1 Introduction to EFI ……………………………………….……………12

1.1.1 EFI Sensors …………………………………………………………16

2.0 Preliminary Design ……….…………………………………………………….23

3.1 Actuating Mechanism …………………….……………………………….……..24

3.1.1 Design Consideration for Actuating Chamber…………………………….…….27

3.2 Pumping Mechanism…………………….……………………………………32

3.2.1 Pump demands due to injector Flow rates…...…………………………….35

3.2.2 Pump demands due to driver input……………………………………...38

3.3 Control Solenoids…………………………………………………………...40

3.3.1 Solenoid Injector Design …………………………………………….41

3.4 Control System…………………………………………………………...45

3.4.1 Control System Design ……………………………………………46

4.0 Results ……………..…………………………………………………….47

5.0 Avialable Information Resouces ………………….……………………………50

6.0 Project Methodology ………………………………………………..…..52

7.0 Consequential Effects ………………………………………………...53

References ……………………………………………………………………………...….54
Appendix A ……………………………………………………………………………..55

List of Figures

Figure 1 – Ignition Diagrams
Figure 2 – Regression of NOx level for Petrol Vehicle
Figure 3 – Regression of NOx and PM for Diesel Vehicles
Figure 4 – European Emission Standards for Passenger Cars
Figure 5 – Mass Airflow Meter
Figure 6 – Hot Wire MAF Sensor
Figure 7 – MAF Transfer Curve
Figure 8 – O2 Sensor
Figure 9 – Narrow Band Oxygen Senor Voltage Output
Figure 10 – Actuating Chamber Assembly
Figure 11 – Actuating Chamber Exploded View
Figure 12 – Simulink Model
Figure 13 – Scope Simulink Results
Figure 13A – Scope Simulink Results
Figure 13B – Scope Simulink Results
Figure 13C – Scope Simulink Results
Figure 14 – Rail Pressure Control Value
Figure 15 – Bosch Pump Exploded View
Figure 16 – Pump Section View
Figure 17 – Injector Flow vs Rail Pressure
Figure 18 – Required Valve Lift
Figure 19 – Toyota Diesel Injector Drive Scope
Figure 20 – Injector Flow vs Solenoid On Time
Figure 21 – High Pressure Solenoid Section View
Figure 22 – Bosch Injector Exploded View
Figure 23 – Value Lift vs Time at 6000rpm
Figure 24 – Fuel Consumption Gain
Figure 25 – Inlet Flow Speed

NOMENCLATURE

P	Pressure	MPa
F	Force	N
A	Acceleration	m/s^2
F_c	Current Flow	cm^3/ms
P_c	Current Pressure	MPa
F_o	Operating Flow	cm^3/ms
P_o	Operating Pressure	MPa
P_r	Rail Pressure	MPa

ABBREVIATIONS

SI	Spark Ignition
OEM	Original Equipment Manufacturer
VTEC	Variable Valve Timing and Lift Electronic Control
VVT	Variable valve Timing
VANOS	Variable Nockenwellensteuerung
EFI	Electronic Fuel Injection
DI	Direct Injection
EGR	Exhaust Gas re-circulation
MAF	Mass Air Flow
IAT	Intake air temp sensor
CKP	Crankshaft Position
ECT	Engine Coolant Temperature
ECU	Engine Control Unit
PCM	Power-train Control Module

1. 0 Introduction

The internal combustion engine powers almost the entire automotive industry worldwide. The two major forms of internal combustion engines are:

Spark ignition (SI) Compression Ignition

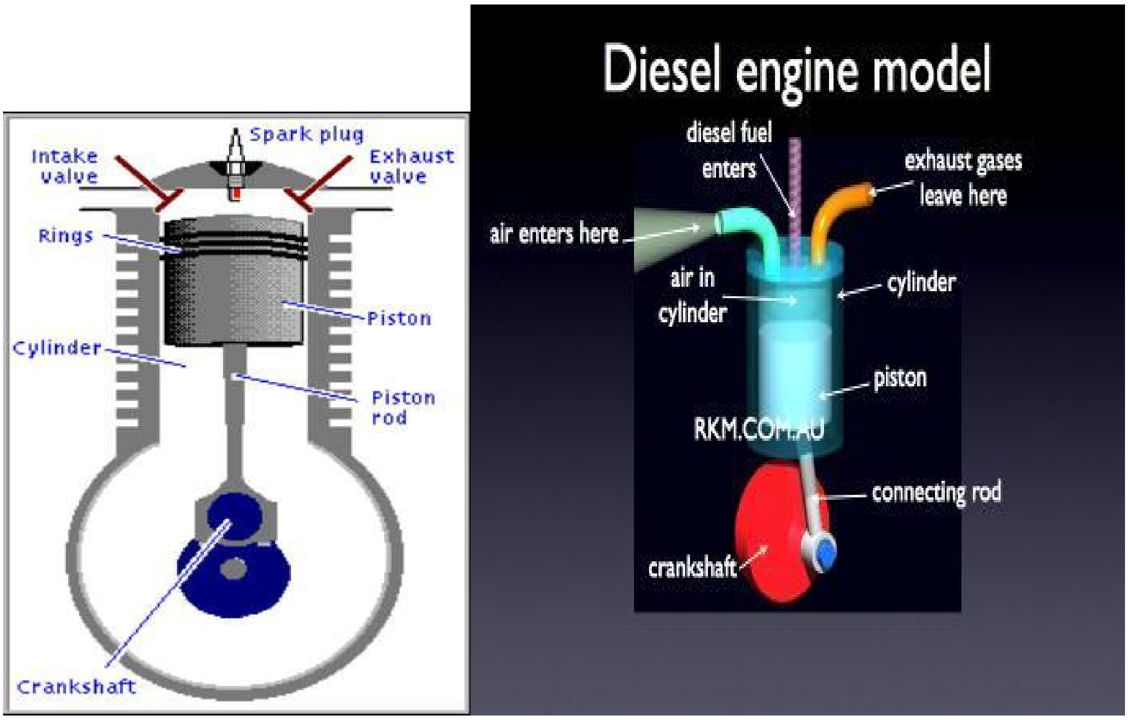

Figure 1 – Ignition Diagrams

Both engines introduce air into the engine and extract post combustion gases via one or more valves. The valves in the majority of cases are actuated via a mechanical camshaft. The camshaft has a certain profile that is usually determined via the required application of the particular engine.

The main difference between the two is how the combustion event is achieved. Diesel engines run very high compression ratios up into the 18-22:1, as the diesel fuel is injected into the chamber instantaneous combustion is achieved. Therefore the combustion event is governed by the fuel injection timing. However spark ignition engines introduce a desired fuel and air mixture into the chamber and the spark plug fires causing combustion. Both use valves that are actuated by camshafts and both are adversely affected by camshaft choice.

Camshafts come in many different forms however all have a particular profile that is ground to that particular camshaft. Valve timing, ramp rates, valve duration and lift are all variables that influence how an engine performs. The basic thinking is for maximum power the longer the valve is open at maximum lift the better the cylinder fill will be.

With the current technology both valve lift and timing can be altered via OEM manufacturers systems like VTEC, VVT-I, VANOS, VVT etc. While there are slight differences between the different systems they all still operate via a mechanical source so that complete valve control is not possible.

1.1 Introduction to EFI

Since the onset of EFI (Electronic Fuel injection) the main aim for car manufactures has been reducing emissions, increasing fuel economy, increasing drivability and achieving a higher overall power. The top priority in the past 5 years has been emissions reduction and better fuel economy. It is obvious that this is the driving force for new petrol engine technologies with the development and integration of DI (direct injection), post catalyst oxygen sensors and removal of EGR (Exhaust gas Re-circulation) due to camshaft variability just to name a few.

Current emissions standards such as the "EURO" levels show the increasing trend towards reducing emissions. Figure 2 shows the regression of NOx (Nitrous Oxide) levels since 1992 for petrol cars while Figure 3 shows the regression in PM (particulate Matter) and NOx since 1992 for diesel cars [1].

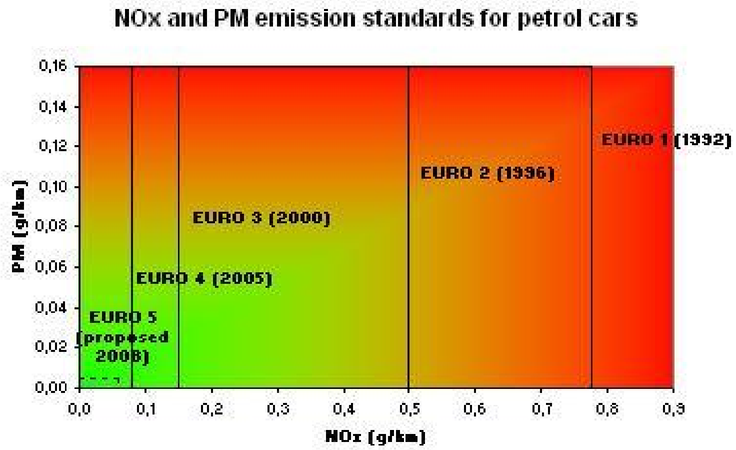

Figure 2 – Regression of NOx level for Petrol Vehicle

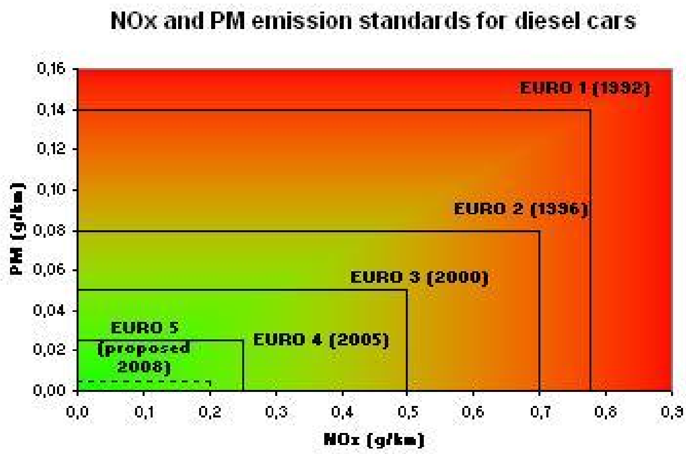

Figure 3 – Regression of NOx and PM for Diesel Vehicles

Figure 4 shows the actual emissions levels for each gas over the past 15 years. This illustrates how quickly the reduction in emissions is occurring with even more stringent levels to be introduced in the near future. This trend shows that the efficiency of current generation engines needs to increase even further, with the draw back being a more complex engine.

While a more complex engine is most undesirable for manufacturers as it employs more intricate parts, which means higher cost of production and engineering. The engine technology is only part of the problem in reducing certain emissions as some particular combustion byproducts are created from specific fuels and oils used. While this project delves into the efficiency of the combustion process the chemical composition of the exhaust from this design is beyond the scope of this project. It is however interesting to note in Figure 4 that while there is a regression in some emissions others plateau in the next few years. This is obviously the point at which higher efficiency may not be realistically achievable.

European emission standards for Passenger Cars (Category M_1*), g/km

Tier	Date	CO	HC	HC+NOx	NOx	PM
Diesel						
Euro I†	July 1992	2.72 (3.16)	-	0.97 (1.13)	-	0.14 (0.18)
Euro II, IDI	Jan. 1996	1.0	-	0.7	-	0.08
Euro II, DI	Jan. 1996[a]	1.0	-	0.9	-	0.10
Euro III	Jan. 2000	0.64	-	0.56	0.50	0.05
Euro IV	Jan. 2005	0.50	-	0.30	0.25	0.025
Euro V (proposed)	Sept. 2009	0.50	-	0.23	0.18	0.005
Euro VI (proposed)	Sept. 2014	0.50	-	0.17	0.08	0.005
Petrol (Gasoline)						
Euro I†	July 1992	2.72 (3.16)	-	0.97 (1.13)	-	-

Euro II	Jan. 1996	2.2	-	0.5	-	-
Euro III	Jan. 2000	2.30	0.20	-	0.15	-
Euro IV	Jan. 2005	1.0	0.10	-	0.08	-
Euro V (proposed)	Sept. 2009	1.0	0.10	-	0.06	0.005[b]
Euro VI (proposed)	Sept. 2014	1.0	0.10	-	0.06	0.005

* Before Euro V passenger vehicles > 2500 kg were type approved as Light commercial vehicle N1 - I

Figure 4 – European Emission Standards for Passenger Cars

1.1.1 EFI Sensors

Mass airflow meter

Figure 5 – Mass Airflow Meter

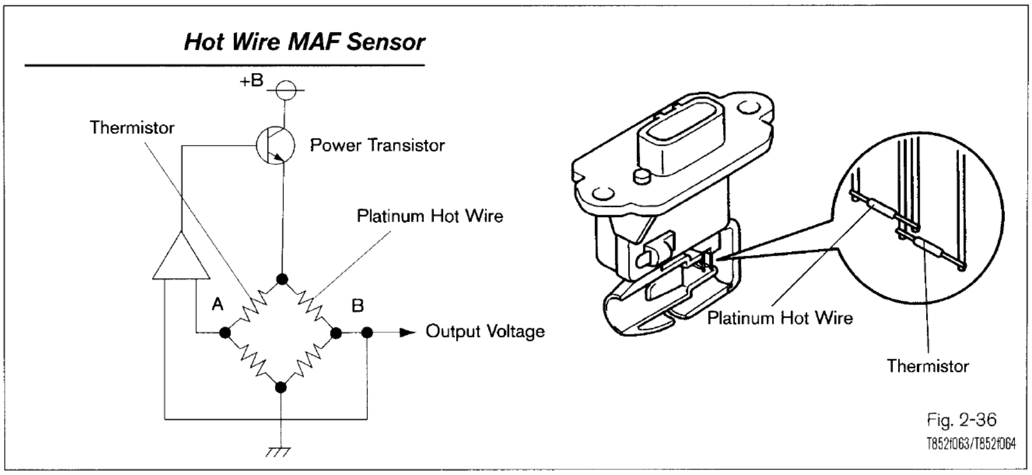

Figure 6 – Hot Wire MAF Sensor

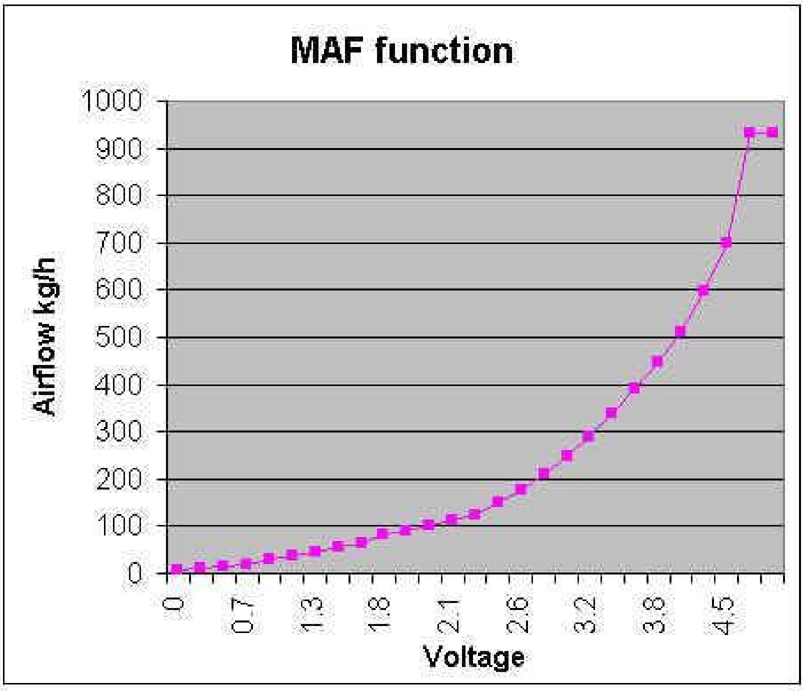

Figure 7 – MAF Transfer Curve

The hot wire airflow sensor determines the mass of air entering the engine by determining the heat loss over the platinum hot wire. As the volume of air entering the engine increases the flow rate also increases resulting in a greater cooling effect over the "hot wire". The voltage output of the device is a logarithmic scale of airflow usually in g/s of airflow vs voltage output. Depending upon the system and manufacturer this is usually a 0-5 volt signal. Current MAF sensors also incorporate and intake air temp (IAT) sensor into the device to minimise packaging.

The MAF sensor is responsible for the main air-mass calculation in steady state and dynamic situations however other sensors such as MAP (manifold absolute pressure) and TPS (throttle position) sensors can also be referenced usually in high transient conditions. Once the mass of air is known and the preprogrammed injector flow is also known the required air fuel ratio can be achieved. In modern engine ECU's the MAF is referenced that often that a model for each individual cylinder is known. This allows the ECU to change injector times and ignition angles based on the mass of air within each cylinder.

Oxygen sensors

Figure 8 – O2 Sensor

The oxygen sensors come in a variety of different types however there are two main types - narrow band and wide band. The narrow band zirconium sensor has a limited range of operation meaning that it is only reliable accurate at stoichiometric ratio of 14.7:1 air to fuel ratio or lambda 1. The voltage output at lambda 1is approximately 500mV and is asymptotic around this point, so that large voltage variations are seen with relatively low changes in AFR.

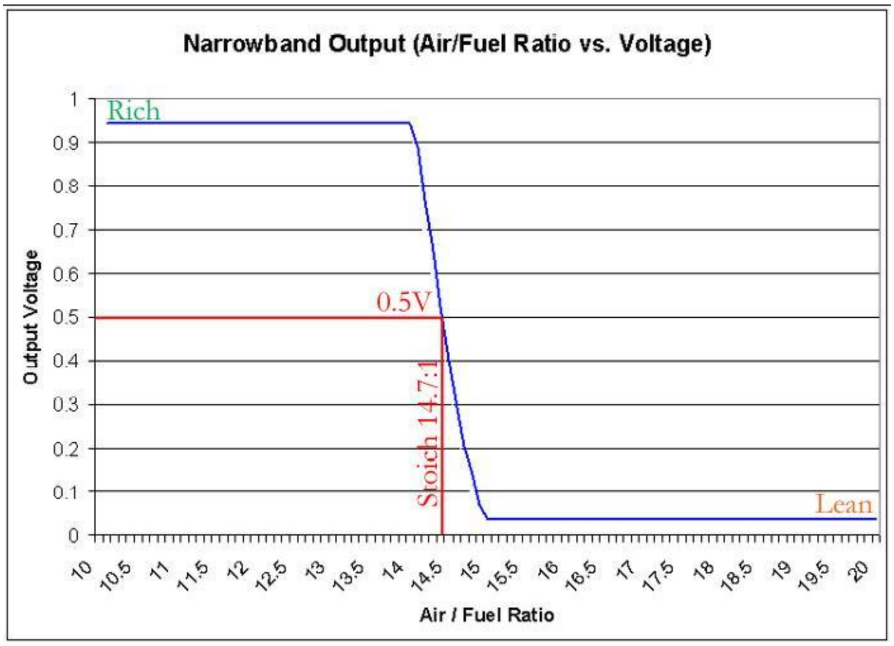

Figure 9 – Narrow Band Oxygen Senor Voltage Output

Wideband sensors display a linear voltage output with changes in AFR. The operable range depends upon the sensor in use however accuracies between 10:1 to 20:1 AFR can usually be measured.

The main purpose of the O2 sensor is to monitor post combustion oxygen content. From this the PCM can determine if the mixture is richer (higher fuel content) or leaner (higher oxygen content) than the desired target lambda and adjust the injector on time to compensate. This is known as closed loop operation which up until recently only occurred at light to medium driving loads however with the tighter emission regulations full closed loop operation throughout the entire engine-operating region is now possible with the use of a wideband oxygen sensors. This can be seen on the 2006 BMW 335ci with great results in emissions reduction.

With the current EURO4 emissions in place post catalyst oxygen sensors have also been incorporated to monitor the efficiency of the chemical process within the catalytic converter and alert the driver if it is below a preset standard. This acts as a fail safe to guard against higher than normal emission levels.

Crank Shaft Position sensor

The CKP (crankshaft position) sensor is used to determine the position of the crankshaft. It does this via either a magnetic pickup or a hall effect sensor in conjunction with a toothed crank disk. The crank reference disk has multiple pickups to allow the crank position to be known. The separation of the teeth on the disk is known by the PCM with usually missing teeth at TDC (top dead center) of the number 1 cylinder. A typical disk may have a crank pattern of 36 –1 which means that there are 35 sensing teeth with one missing. The missing tooth usually denotes TDC. This means that the crank position is known every 10deg. Configurations that are in wide use are 60–2, 360-1 and 4 even.

The main purpose of the CKP is to determine the position of the crank in order for the PCM to correctly time both the ignition and fuelling events along with virtually all other electronic devices attached to a modern engine. If the PCM receives no CKP signal than the RPM of the engine is considered to be zero and the engine will not start. The CKP signal is also used to determine the velocity and acceleration of the crankshaft within current PCM's. This enables acceleration rates of the crank to be known after each combustion event. This data is then used to predict torque output of the motor per firing event and also in the detection of cylinder misfires.

Knock Sensor

Knock sensors are devices that monitor combustion noise via a piezo-electric element and are usually bolted to the side the engine block. The output voltage increases with engine noise. Depending upon the frequency and current engine conditions the PCM is able to determine if the noise is considered engine knock or only spurious noise.

Engine knock is usually from the combustion flame front being miss-timed with the position of the piston, an example is where the flame front is expanding towards the bottom of the chamber towards the piston and reaches the piston on its way towards the top of its stroke. There are multiple reasons for excessive engine knock or noise but usually from an uncontrolled burn event or a miss-timing ignition event.

Knock sensors are also now used on common rail diesel engines to monitor engine noise and to alter the fuelling event. It is also used to determine the pilot injection timing to progressively increase cylinder pressures so that rapid expansion does not occur.

Coolant Temp Sensor

The ECT (engine coolant temp) sensor outputs a linear voltage with respect to coolant temperature. The range depending upon the manufacturer is usually between –25 and 125 deg C.

The ECT sensors main purpose is to enable the PCM to determine the engine temperature in order to perform cold start functions such as extra injector enrichment (due to poor atomization of fuel mixture), increase idle speed, control thermo fans and perform catalyst light up functions. It is also used to determine the charge air temperature from the predicted air speed and manifold temp even though ECT is very close to manifold temperature.

2.0 Preliminary Design

The system that I am proposing will have complete control over all valve-operating parameters via an "Electro Hydraulic Actuation". In basic form the system will consist of a pumping mechanism a mechanical/hydraulic actuator per valve, multiple control solenoids and sensors will also be used in conjunction with a closed loop fuel and ignition control system.

There are four main parts of the system, the actuating mechanism, pumping system, control solenoids and the control system. The preliminary design will focus on controlling the inlet and exhaust valves of a four stroke, four-valve engine however with hands on experimentation the exhaust valves will be crank driven as per standard. The use of one solenoid per valve will be assumed for the theoretical design however the number of valves or solenoids per valve may change for testing purposes. The high-pressure pump will be driven via the crank with a gear ratio determined within "2.2 Pumping Mechanism". The solenoid operation and actuating specifications will be determined within their corresponding sections.

The sensory inputs to the control system include: mass airflow sensor, crankshaft position sensor, accelerator pedal position sensor, rail pressure sensor, engine temperature sensor, wideband oxygen sensor and knock sensor. The main output drivers from the PCM include thirteen pulse width modulated drivers – four injectors and eight valve control solenoids and a fuel rail control solenoid, four ignition drives and five auxiliary high/low drivers, one for fuel pump and four valve return solenoids. There may be more inputs and outputs required however these are the inputs and outputs of primary concern to this project.

The design prerequisites are based on being able to control the valve movement from a lift of 1mm to 12mm at full extension. The time at which the inlet valves are opened is assumed to be infinitely variable along with the closing time.

3.1 Actuating Mechanism

The actuating mechanism is a sealed fluid chamber. One end is fixed while the other end is a sliding plunger assembly that is attached to the valve. The valve is displaced from its seat when this chamber is flooded with pressurized fluid. The chamber contains two orifices one for allowing a calculated amount of fluid in via a high pressure control solenoid and the other is a safety bypass relief that will be positioned so that over extension of the valve doesn't occur. There is also an additional port or “T” piece between the input to the chamber and the high pressure control solenoid where the releasing of the fluid within the chamber can be controlled via an open/close solenoid to return the valve to its seat.

Design of this chamber will determine how much fluid pressure is required to lift the valve and the final flow rate of the pump. The fluid pressure required is inversely dependant upon flow rate so that the lesser the pressure the higher the required flow rate.

It may be required that a larger mass of fluid will be needed in order to take the combustion chamber temperatures away from the valve itself however this may most likely be monitored in operation. High temperature resistant seals should be used to reduce long-term fatigue such as composite material like **Viton®.** This material is resistant to gasoline and its blends.

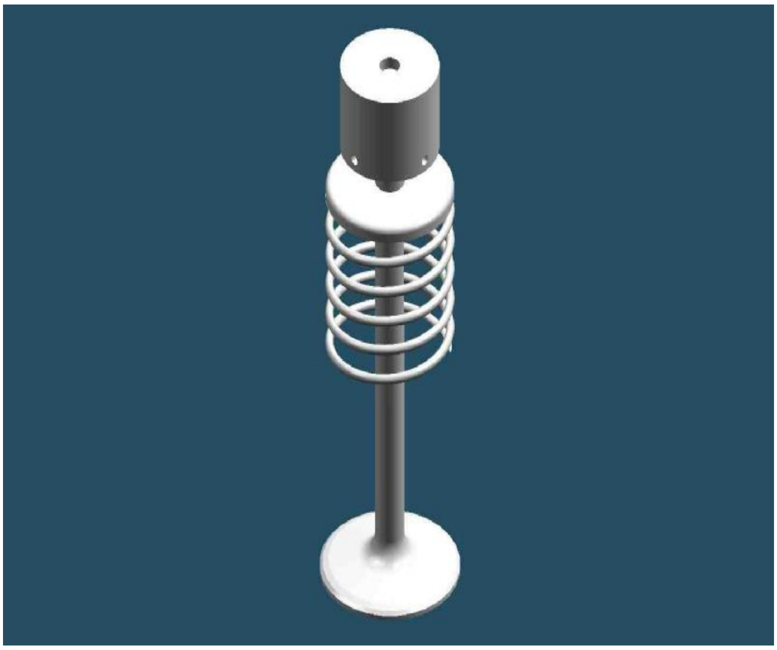

Figure 10 – Actuating Chamber Assembly

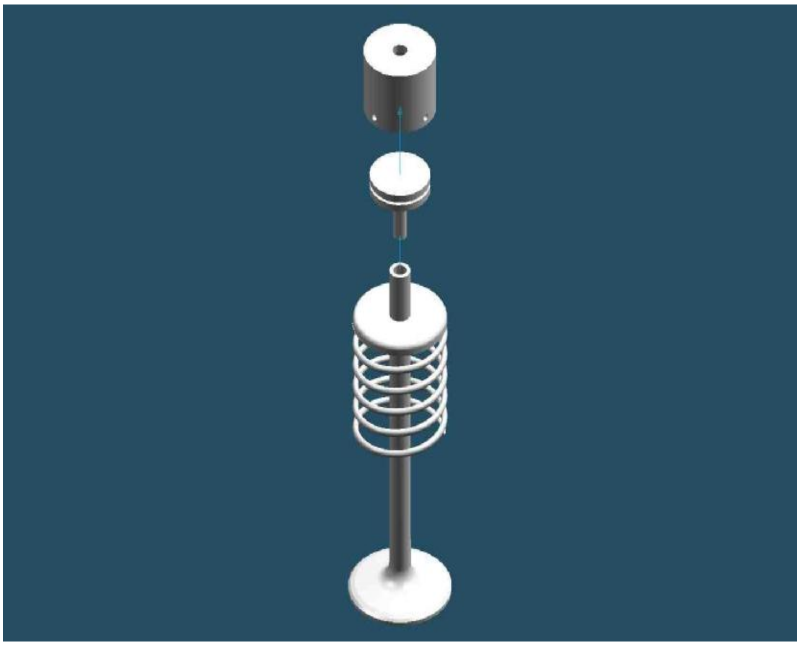

Figure 11 – Actuating Chamber Exploded View

From Figure 10 you can see the plunger is inserted onto to the end of the valve spring via an internal and external thread. There is also a 1mm grove in the plunger where the seal will be located. The retainer and valve spring are in the normal position and are held in via 2 collets. The valve spring is seated in the standard position within the cylinder head. The actuating chamber has 1 main feed where the injector is screwed in to and 4 smaller orifices to protect against over extension of the valve.

Depending upon the cylinder head in use the actuating chamber could be externally threaded so that it can be held in position within the head. Cylinder heads with full mechanical bucket actuating camshafts could be used in this situation as found in some Nissan and Toyota four valve engines.

3.1.1 Design Consideration of the Actuating Chamber

In the design of the actuating chamber consideration must be given to the final volume, as this will greatly affect the required flow rate of the pump per revolution. Consideration must also be given to the available pulse width requirements of the injector because if the volume is either too large or too small then the requirements of the injector may be outside of its operable range.

Since the chamber height is determined by the overall required lift the only other factor is the cross sectional area of the chamber. Consideration must be given to the cross sectional area as the larger the area the less the pressure required however the flow through the injector is not linear as shown in figure 17. After multiple values of cross section areas were trialed a cylinder diameter of 20mm was chosen. This gives a cross sectional area of $3.14cm^3$ and a volume at full valve extension of $3.77cm^3$.

Since the valve spring in use has a rate of 57N/mm of lift and has a seat pressure of 30N then the minimum required pressure to raise the valve 1mm and 12mm is equal to:

$P = F/A = (57+30)/3.14x10^{-4}$
$P = 277.1$ kPa for 1mm lift

$P = F/A = ((57x12)+30)/3.14x10^{-4}$
$P = 2.274$ MPa for 12mm lift

Cross-referencing this with the injector flow rate at that particular pressures from figure 17 it can be seen that at 277.1kPa a flow of 2.12 cm^3/ms is possible. At 1mm of lift the volume within the chamber is $0.314cm^3$, therefore the required injector on times is 0.148ms or 148µs. At 2.274MPa a flow rate of 6.18 cm^3/ms is possible. At 12mm of lift the volume within the chamber is $3.77cm^3$, therefore the required injector on time is 0.61ms or 610µs. The input force to the actuator however is modeled as an impulse force as shown below which is probably a more accurate description of the system.

The valve spring is more appropriately modeled as an over damped second order system, as the port flow through the input tube will throttle the fluids entry and escape. The weight of the valve and spring is considered to be around 50grams each and the input force is considered to be a unit input or injector on time. The system can be modeled using:

$$m\ddot{x} + c\dot{x} + kx = f(t)$$

Rearranging the equation to determine the acceleration.

$$\ddot{x} = \frac{1}{m}\left(f(t) - c\dot{x} - kx\right)$$

This system can then be modeled using Simulink to find the valves response to the input force over a specified time period.

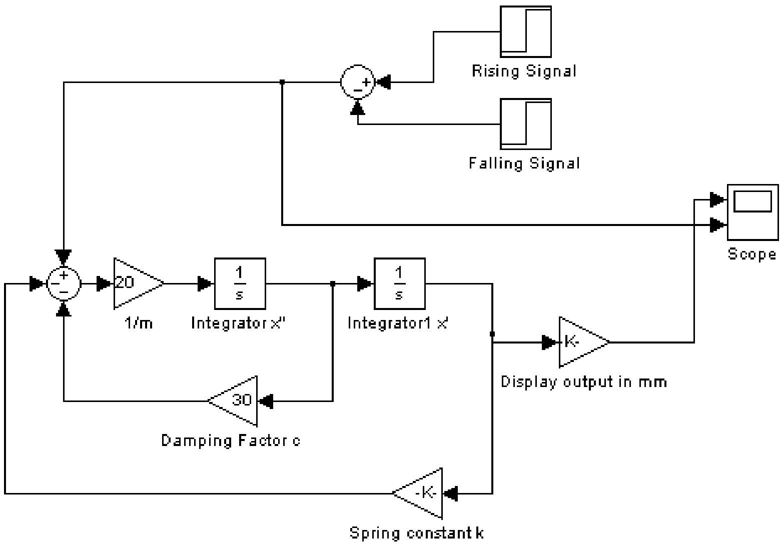

Figure 12 – Simulink Model

Figure 13 shows the response to a step input of 350µs at 2MPa of rail pressure with a dampening constant of 30. Figure 13A shows the response to a step input of 350µs at 5MPa of rail pressure and a dampening constant of 30.

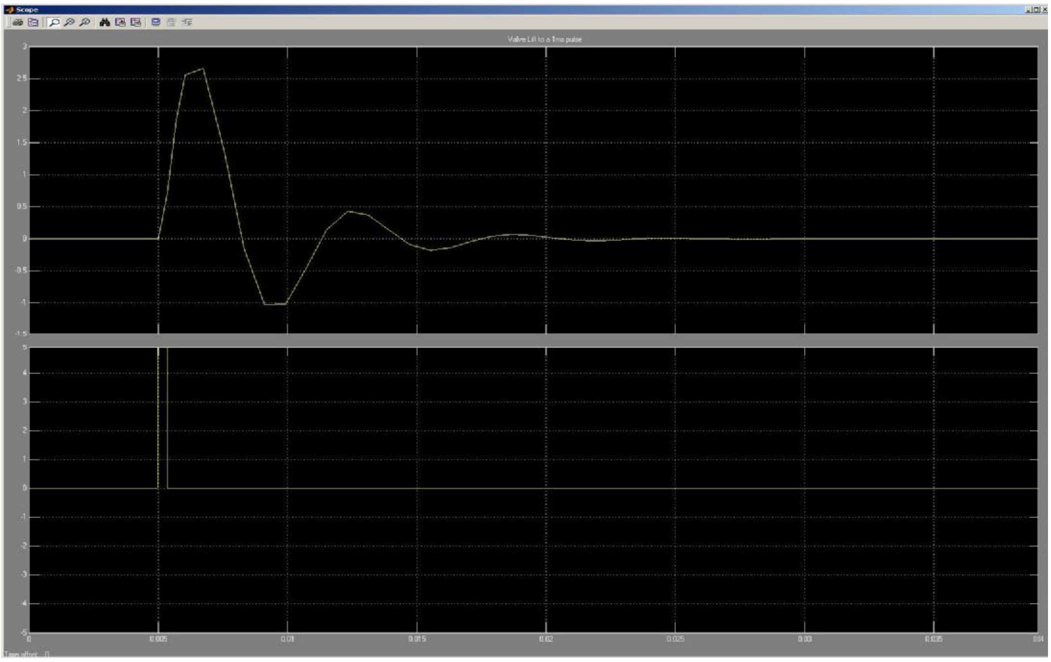

Figure 13 – Scope Simulink Results

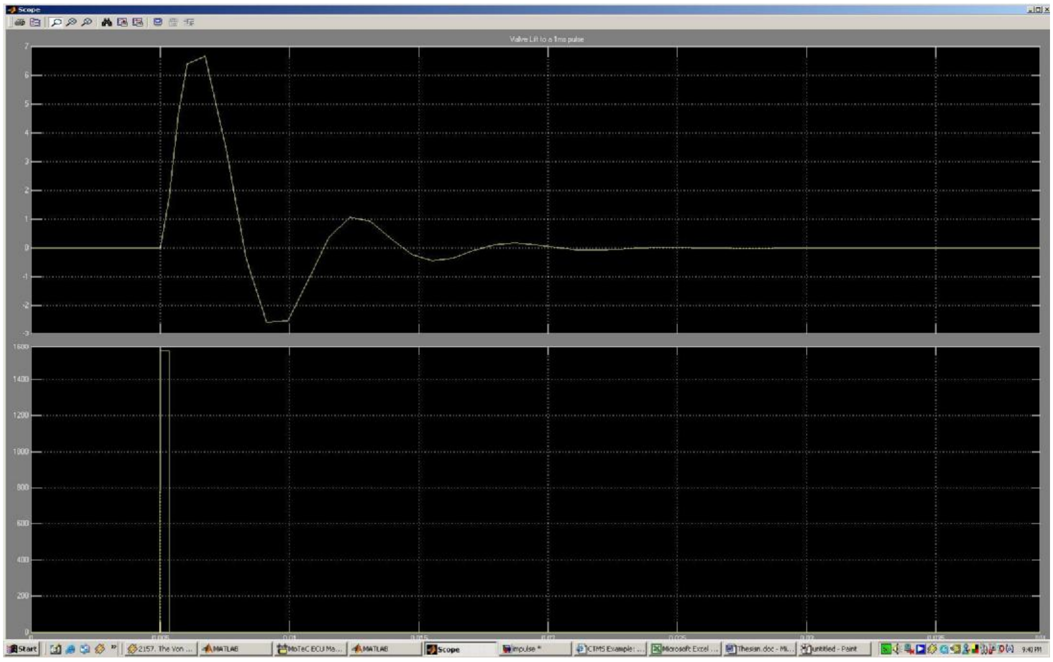

Figure 13A – Scope Simulink Results

The top graph in each figure shows the amount of valve lift in mm on the y-axis against time in seconds on the x-axis and the bottom graph shows the unit step input with the y-axis showing the force in Newton's and the x-axis once again being time related in seconds.

The effects of injector pulse width can be seen in figure 13B with a Rail pressure of 2MPa and a dampening constant of 30 and an injector on time of 150µs. Figure 13C shows the effects of increasing the injector pulse width 850µs to 1ms with an increase in lift of 6mm.

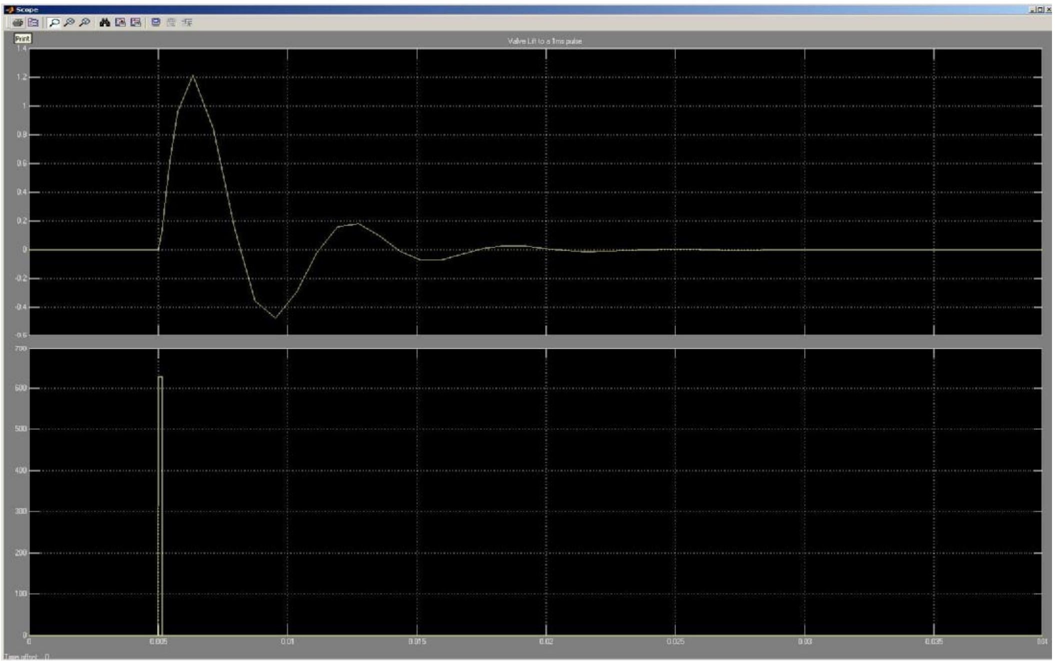

Figure 13B – Scope Simulink Results

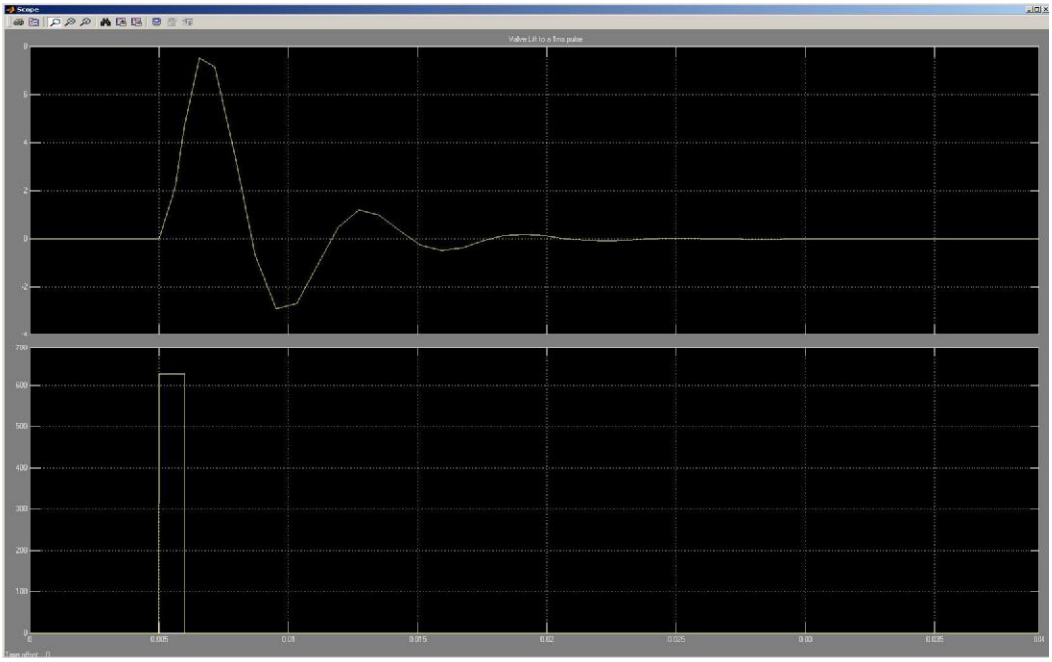

Figure 13C – Scope Simulink Results

Various values for damping coefficients and pressure inputs where trialed to determine the best compromise between overshoot and injector pulse width or step time. It's obvious that a higher the pressure will cause the valve to open faster with the drawback being overshooting of the desired valve lift, this can be compensated for by increasing the damping factor.

3.2 Pumping Mechanism

The pumping mechanism is used to supply a pressurised fluid to a common rail. From here a control solenoid will allow a metered amount of fluid into a valve displacement chamber in order to open and close the valves.

The characteristics of the pumps is critical to the success of the project as an incorrectly chosen pump may take too much power away from the engine in order to drive it. From preliminary calculations a relatively large amount of pressure will be required to overcome the valve spring force. In the order of 10Mpa will be required to extend a valve to a lift of about 12mm. While the pump will have to generate large pressures only small flow rates are required, as the valve displacement chambers will only contain about $3.77cm^3$ of fluid at full lift. So depending upon amount of valves used, total desired lift and final engine rpm the total flow rate of the pump is to be determined. While these pressures sound high current common rail diesels run in the order of 22000Psi of fuel pressure so currently available high-pressure pumps are widely available off the shelf.

The pump itself will most likely be crank driven, as the energy required to create these kinds of pressures may be impossible with currently available electrical pumps. The fluid medium to be used is also an unknown at this point as engine oil may be the incorrect viscosity at low temperatures. A highly viscous fluid with good temperature stability and anti corrosive properties and lubricating abilities would be required.

As the rail pressure is dependent however not entirely upon amount of valve lift and only small amounts of valve lift is required at low rpm and load it will be necessary for a control system to monitor rail pressure and adjust pressure to minimize pumping losses.

The pressure is adjusted by a pulse width modulated control valve that allows bypasses fluid from the rail and returning it to the low-pressure side of the pump. Figure 14 shows the internals of a current Bosch control valve.

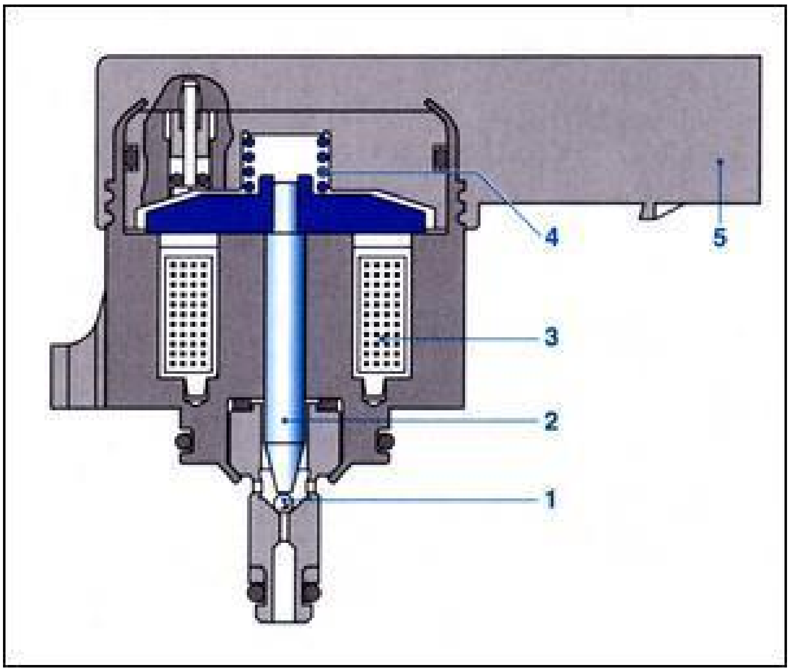

Figure 14 – Rail Pressure Control Value

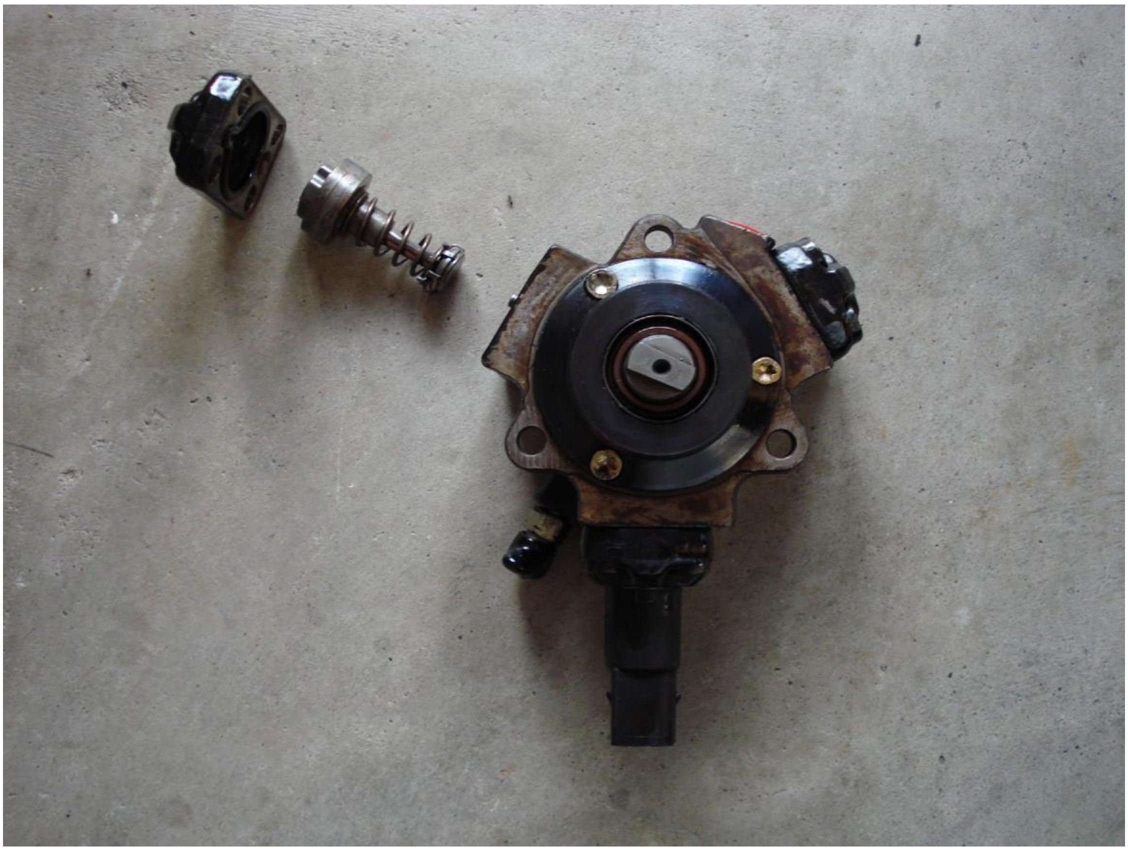

Figure 15 – Bosch Pump Exploded View

3.2.1 Pump Demands due to injector flow rates

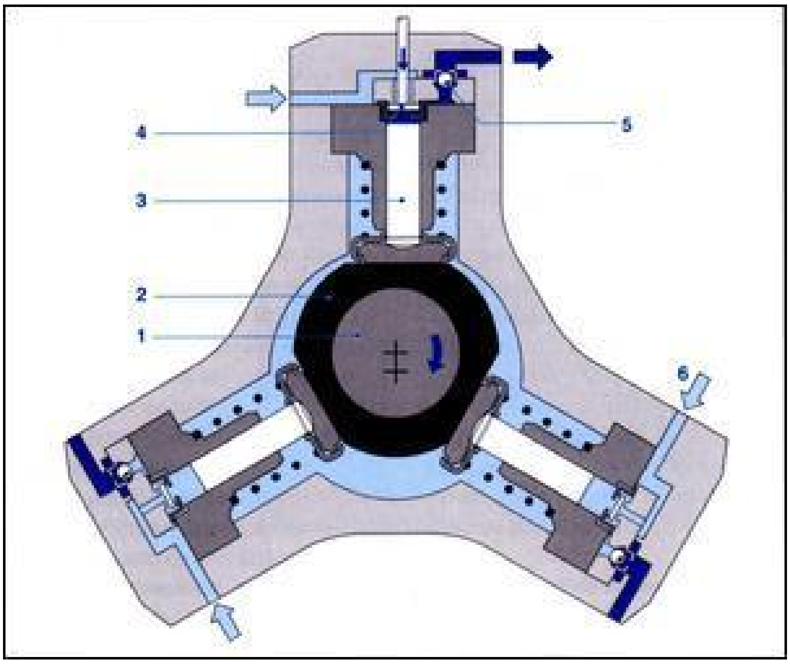

Figure 16 – Pump Section View

The injector flow rate is dependant upon the pressure in the rail and can be approximated by the equation:

$$F_C = F_O \times (P_C/P_O)^{(1/2)}$$

Where F_C and P_C is the current flow rate through the injector while F_O and P_O is the measured flow at a known operating pressure. Figure 17 shows the injector flow verses rail pressure, which has been determined from the data acquired in figure 20.

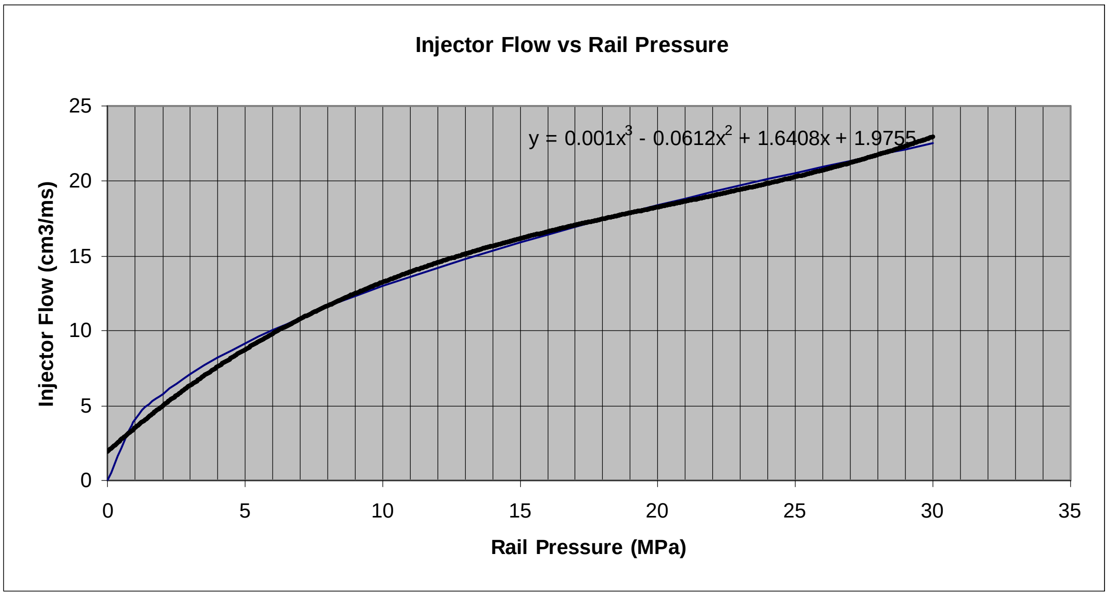

Figure 17 – Injector Flow vs Rail Pressure

A third degree polynomial trend line can be added to the graph in order to determine the Injector flow. From Figure 17 the injector flow in cm^3 can be determined:

Injector flow = Pulse width (ms) x ($0.001P_r^3 - 0.0612P_r^2 + 1.6408P_r + 1.9755$)

Therefore

Lift = Injector flow/Chamber Area

This formula along with the modeled response of a mass spring damped system to an impulse force that was found in section 2.1 can be used to determine the required pulse width requirement along with the rail pressure.

Since the valve displacement chambers are $3.77cm^3$ in volume at full extension, the variables required in determining final fluid flow is the maximum engine speed, number of valves and the number of cylinder within the engine. If a maximum engine speed is assumed to be 6000revs/min and the engine used is a 4valve per cylinder four stroke four cylinder. Then the require flow rate is equal to:

Flow rate = lift (cm) x 3.14 (cm^2) x 4/2 (cm^3/rev)

Therefore the required flow rate is 7.54 cm^3/rev. So at 6000 rpm or 60 revolutions per second the required flow rate at 12mm of lift is $4.524x10^{-4}$ m^3/s. If a maximum pressure of 15Mpa is assumed then the energy consumed by the pump is equal to:

Energy consumed = Pressure (Pa) x Flow rate (m^3/s)
Energy consumed = $15x10^6$ x $4.524x10^{-4}$ m^3/s
Energy consumed = 6.786 kW

The generic pump I have acquired from a common rail diesel has a bore and stroke of 6.5mm and three cylinders that create a total flow rate of 0.65cm^3/rev. Therefore to achieve the required flow rate a drive ratio of at least 11.6:1 is required. Depending upon the design this may cause cavitation of the pump due to the return spring not being able to hold the piston to the pump cam profile. A better approach would be to enlarge the bore and stroke diameters in order to compensate. Increasing the bore and stroke diameters to 15mm would allow for 7.95 cm^3/rev at a drive ratio of 1:1, which should be adequate for 12mm of lift to any desired rpm, as long as other componentry can cope. Obviously this is the approach that would be taken by manufactures for higher horsepower requirements so there may be other pumps commercially available.

Note that all thermodynamic implications on pump design has been omitted due to factors such as rail temperature, pump temperature and engine operating conditions all being variables that will greatly affect the result and may only be accurately determined by hands on testing.

3.2.2 Pump demands due to Driver request

The required valve lift is somewhat dependant upon the desired torque request from the driver at that particular rpm. It can be assumed however that less lift is generally required at lower engine speeds as can be seen in Figure 18 from research performed by BMW.

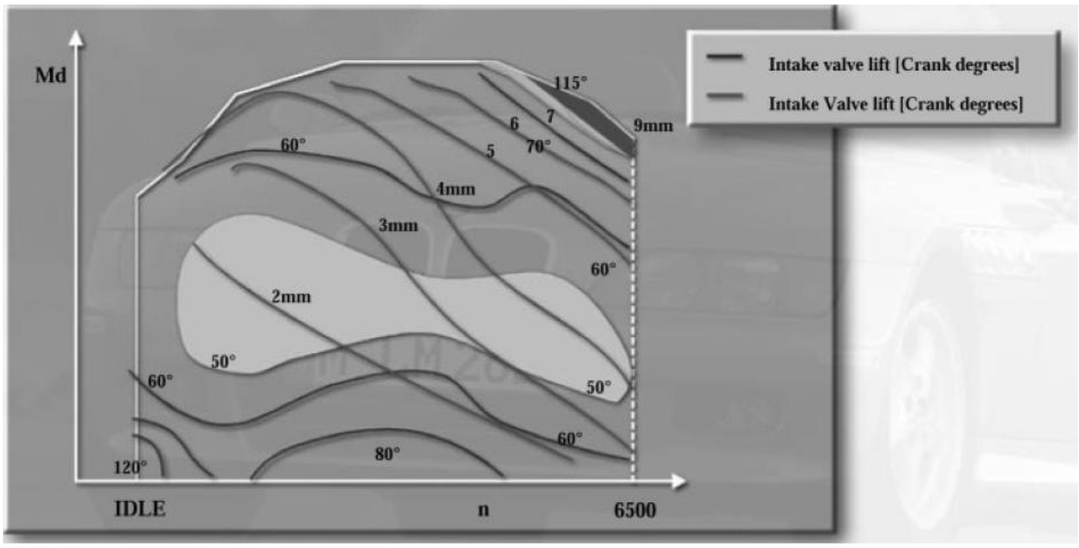

Figure 18 – Required Valve Lift

Assuming that there is no temperature change of the fluid and the pump is 100 percent efficient two 3d models can be made that approximately describe the pressure required verse Driver torque request verses required lift. The second 3d map would outline the pump requirements verses Driver torque request verses required lift.

From the graphs derived by BMW the relationship between desired torque and lift can be plotted. The trend line reviled is a an exponential function approxiamated by:

Lift = $0.7786\exp^{0.0043TR}$

This formula will be the main driving force in determining the required pressure and injector on times to achieve the drivers torque request.

3.3 Control Solenoids

The overall benefits that the project hopes to achieve are dependent upon the accuracy of the valve control. The electro-hydraulic solenoids that will be used need to have high levels of accuracy in their operation. Current commercially available control solenoids may not have the necessary precision in order to cope with the large pressures involved and the frequencies at which they will be used. However current diesel injectors have a reasonable precision at similar pressures so adopting a similar solenoid may be possible. In fact Denso have an injector that can deliver a 0.4ms pulse at up to 1800bar pressure.

There are many various ways on how to control the fluid entering and exiting the actuating chamber. The first line of thinking was to use a three-port valve, with one port supplying pressurized fluid the other being the input to lift the valve and the other the output to return the valve to its seat. In this configuration the solenoids on time or pulse width would represent the valve lift while the valve duration would be the time until the solenoid is de-energized. The problem with this design is that there is no dampening of the valve spring when returning the valve to the seat. The same situation is seen when a particular valve spring cannot keep the valve to the camshaft and the valve "floats". Long term "Valve Float" causes degradation in stem strength and eventually the valve fails. The other approach is to use one three way control valve so that fluid can be meter in and out to so that control of the valve back to the seat is possible. This will allow the valve to be seated at the correct rate. The downside to this approach is that the valve will be harder to control and the solenoid will be energized the entire time the valve is to be opened which may dramatically reduce its life.

The simplest approach at this stage is to use a single high-pressure solenoid to control the valve opening time and lift via the solenoid pulse width and return the valve via a highly damped on/off solenoid that bleeds fluid back to the pump. This approach was selected, as it is the most plausible design to implement on a real vehicle in order to determine if the benefits are warranted.

3.3.1 Solenoid Injector design

The solenoid that was available to me was sourced from a common rail diesel engine. It is a Bosch injector number: 0445120. The solenoid has been designed to operate up to pressures of 160Mpa. Information available about the flow rates and operating configurations is almost non-existent however I have scoped the injector drive output from the PCM of the 2006 Toyota D4D.

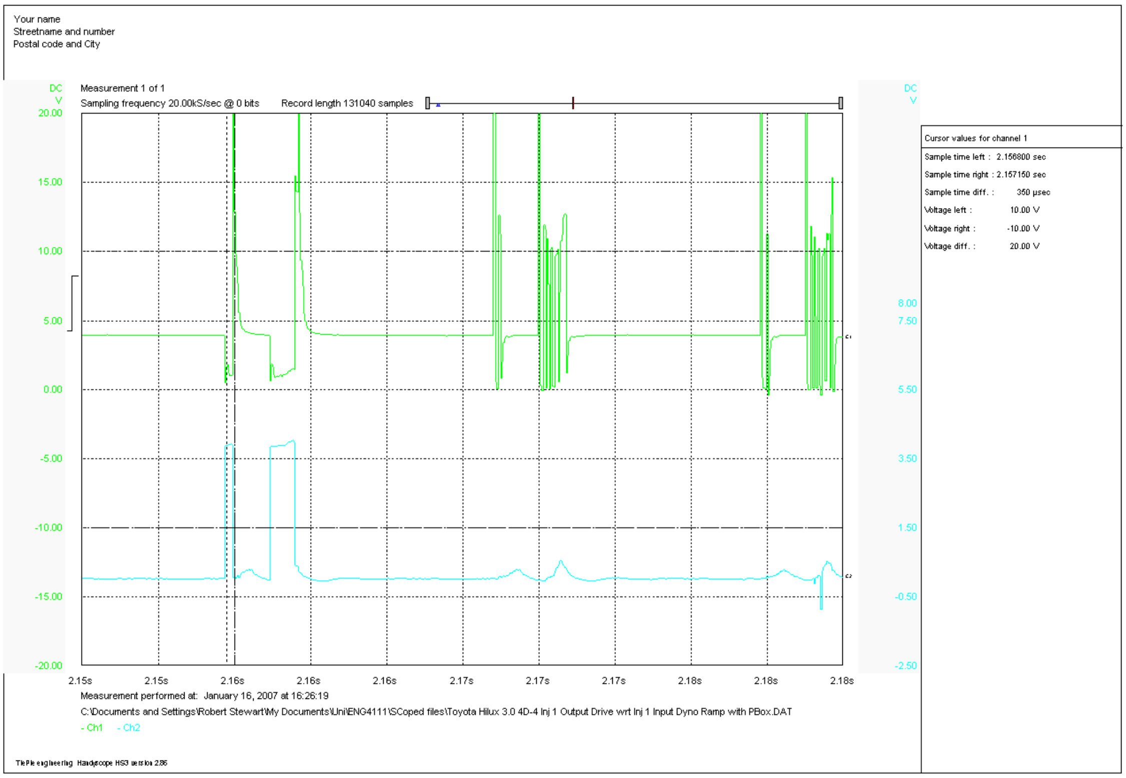

Figure 19 – Toyota Diesel Injector Drive Scope

As can be seen from Figure 19 that the blue signal is the injector drive voltage signal from the PCM while the green signal is actual voltage signal at the injector. The output from the PCM is a square wave 0-4 volt signal, which initiates the injector to pulse on the rising edge. The first blue pulse of 350µs initiates the pilot pulse while the secondary pulse of 1.1ms is used for the main injection. This scan was from a full throttle dynometer ramp. From the scoped data it is noted that a maximum injector on time was recorded at around 1.25ms. The injector on time is obviously restricted by the time of the flame front to reach the injector nozzle so while an injector may be able to be held on for longer durations it is assumed that 1.25mS is the maximum on time. The minimum pilot pulse was approximately 160μs.

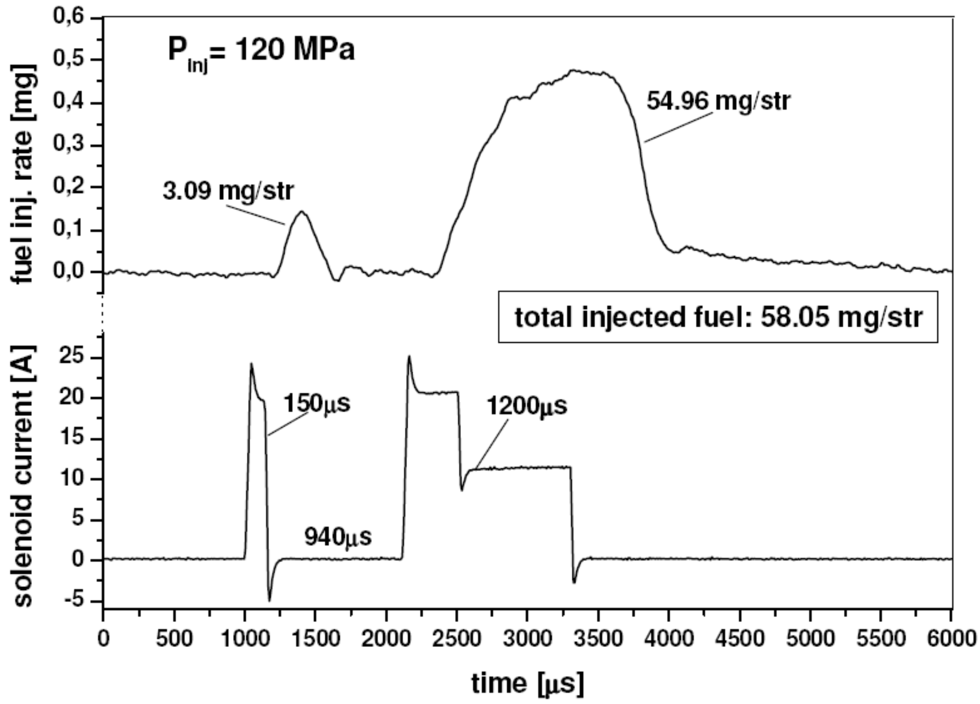

Figure 20 – Injector Flow vs Solenoid On Time

A plot of injector volume flow verses solenoid current for the injector is shown in Figure 20 [2]. This shows the various volume flow vs injector duration for both a pilot pulse of 150μs and a main pulse of 1.2ms. A straight-line approximation is assumed between these two points, which yields:

Injector flow = 0.0471 x injector on time –1.8568 (mg/μs).

Since the maximum on time and the injection quantity at this duration is known the volume of the actuating chamber can now be determined. From Figure 20 it can also be seen that the cracking "peak" current required is around 25amps while a "holding" current of around 10amps is needed. From the graph it can also be seen that there is an inherent delay between the solenoid being pulsed electrically and the start of actual fuel delivery, this is normally referred to as solenoid latency and is in the order of 250μs for this solenoid at normal operating temperature. This delay however minor must be taken into account within the control system.

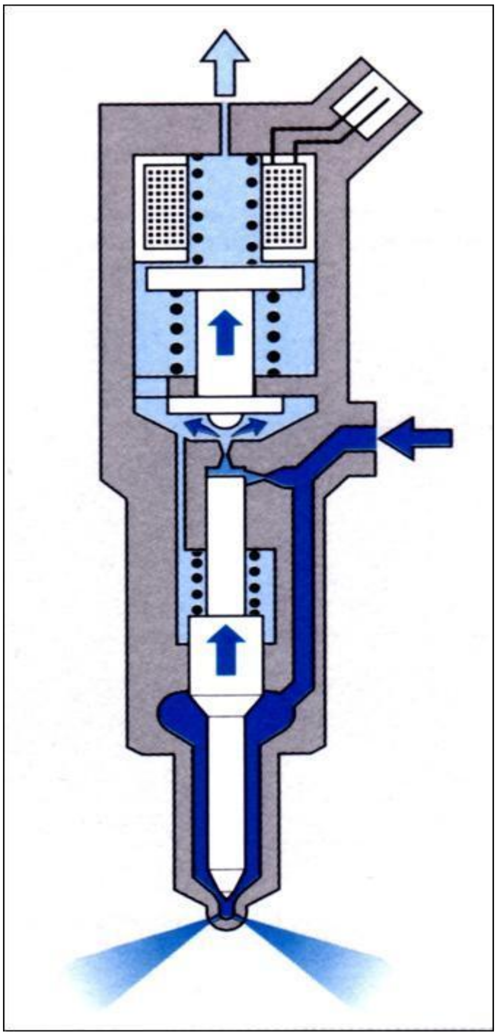

Figure 21 – High Pressure Solenoid Section View

The injector cross section can be seen in Figure 21. It shows that the high-pressure fluid (dark blue) is fed all the way down to the nozzle. The injector is currently on however if the solenoid is de-energized the actuator closes with spring force and blocks the small high pressure feed into the main body. The pressure differential either side of the needle is then equal however the spring-force in the bottom of the injector body forces the pintle to close. The unused fluid within the main body is fed back into the system on the low-pressure side of the pump.

Figure 22 – Bosch Injector Exploded View

3.4 Control system

The control system is needed in order to control both the valve timing in relation to crankshaft position and also the desired lift. Since there is complete operation of the valve a throttle will not be required. Since a physical throttle blade is no longer required a torque based mapping will be required to deliver the necessary acceleration in any driving condition.

The valve control unit (VCU) will take various sensory inputs to calculate the optimum valve lift, timing and duration; mainly crankshaft position, common rail pressure, accelerator pedal position and mass airflow. The VCU will then calculate the desired pulse widths to control amount of valve displacement and duration. This system would be considered open loop as there is no sensory feedback of the amount of lift however until testing is concluded a closed loop feedback system using a valve position sensor might be needed. This would allow a fully closed loop system to be used.

Obviously with this system a fair amount of tuning may be require to obtain optimum efficiency in which case a closed loop self tuning function may be needed. This could be achieved by using a mass airflow sensor to measure current airflow in steady state.

Another tuning option that is currently available is “displacement on demand”. This function allows two of more cylinders to be disabled sequentially by cutting injector pulse. The problem being that the engine will need to compresses that cylinder and in doing that it will loose efficiency. With this system a similar approach may be used however leaving the valve open a small amount will remove the force required to compress that cylinder. It will also aid in catalytic converter efficiency by having extra oxygen in the exhaust.

3.4.1 Control system design

If the engine in use is a 16valve four cylinder four stroke and there is one high pressure solenoid per valve and one return solenoid per inlet and exhaust valves then there will be 16 pulse width modulated outputs and 8 high/low drivers. This control system excludes the injector and ignition control, as it is not directly related to this design. The injector drivers need to cope with at very least 25amps peak over a duration of 150µs and around 15amps hold for around 1.2ms. Due to the high current/voltage characteristics of the solenoid back EMF protection circuitry would be considered a must.

There are two ways in which the solenoids could be driven. The first way would be to use a high power transistor to drive the signal high while the other pin is grounded. The second being that one pin of the solenoid is powered while the other pin is earthed via a suitable transistor within the PCM. Both ways have their own advantages and disadvantages however preference would be given to driving the solenoid low. This kind of approach is warrantable as the new 42Volt "Powernet" will most likely be implemented in the next few years. This should be able to supply sufficient current and voltage in order to drive the solenoids accurately.

4.0 Results

The results of the actual implementation of this design onto a current generation engine is beyond the time frame outlined in this course and would consume far too much of my time. The theoretical results however speak for themselves in terms of performance as shown in Figure 23. In this diagram the massive difference between a standard camshaft and this design can be seen.

A rail pressure of around 5Mpa is required to extend the valve to a lift of 13mm in 2ms, which is much faster than say a Nissan Skyline RB26DETT engine that takes around 5ms to create a valve lift of 8mm at 6000rpm as shown in figure 23. It should be noted however that at lower engine speeds the valve opens even slower within a typical cam driven engine whereas the electro-hydraulic driven cams still open at the same rate no matter the engine speed.

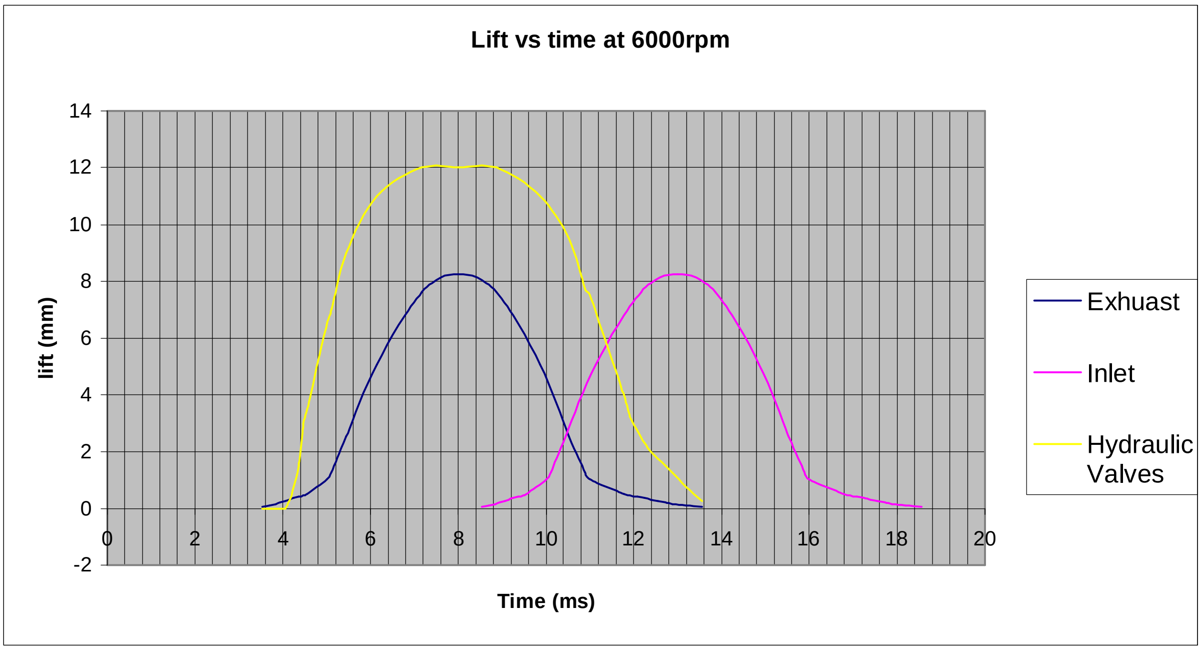

Figure 23 – Value Lift vs Time at 6000rpm

The energy requirements in order to drive the pump is virtually the only source of parasitic loss. Pump energy consumed is the multiple of flow rate and rail pressure so at low lift values the pump loss is even less than the full throttle requirements of 6.786 kW.

Without measuring the exact torque required to rotate the cam over it is hard to compare the losses in tuning the camshaft within a normal crank driven engine.

Figure 24 shows the fuel efficiency gain over a map of load and RPM of a cam less engine developed by BMW. The regions that seem to have shown the most improvement is the sections under high vacuum conditions as this is where the pumping losses are greatest. Gains of 10 percent and above are shown in the regions of the map where most daily driven cars will operate the majority of their time.

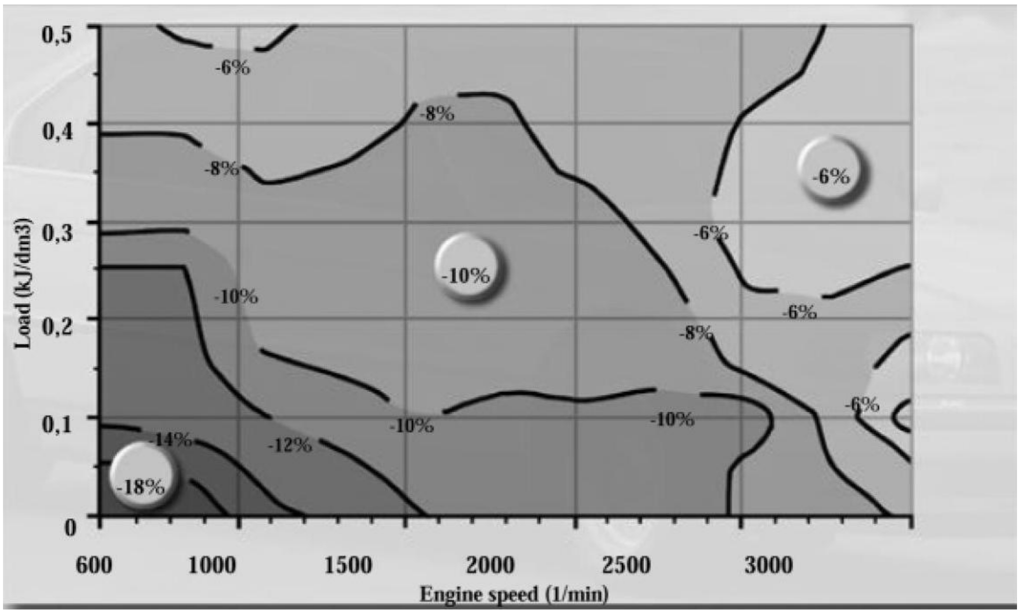

Figure 24 – Fuel Consumption Gain

The fuel consumption gain might also be from massive increase port velocity shown in Figure 25. This will have significant effect on mixture preparation and atomization that is require for a complete burn.

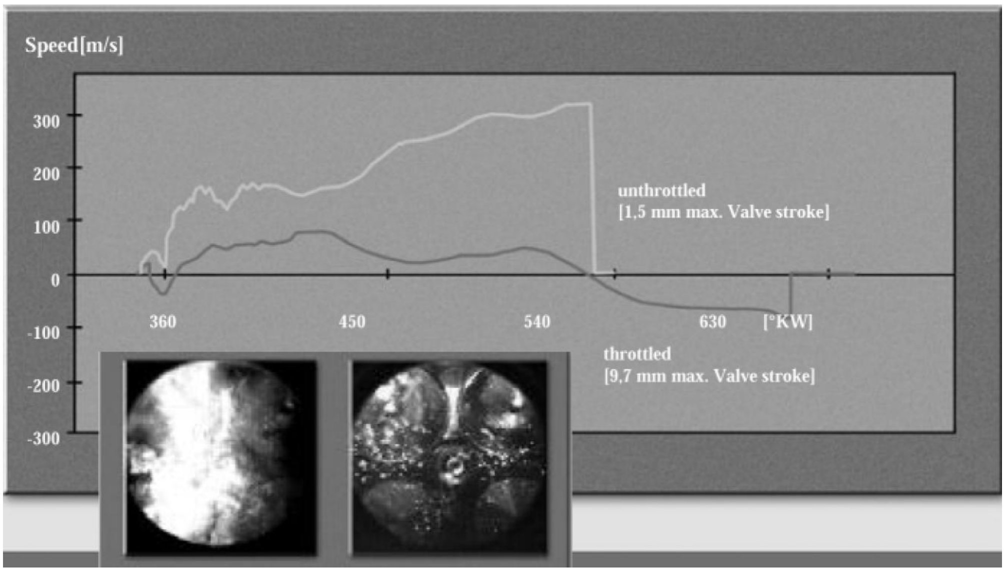

Figure 25 – Inlet Flow Speed

While the primary focus of the project was on the solenoid selection, the current generation of direct injection diesel injector's are somewhat limited in the use as an actuating device. This is not discrediting the high degrees of accuracy in metering fluid but a redesign is most likely required to be able to use the solenoid as a forward and reverse actuator. This however is beyond the scope of this project.

5.0 Available Information Resources

Current information available publicly on a “cam less” engine is virtually impossible to acquire. Similar technology is said to be used in current Formula 1 racecars however the industry is tightly guarded about divulging any useful information. Mercedes holds the patent that basically covers the electronic actuation of a valve via a direct solenoid. In using hydraulics it may be plausible to gain the current private owner ship of the patent that extends to “Electro-Hydraulic poppet valve Actuation” as a lucrative OEM manufacturer does not hold it. It may also be a loop hold for other manufacturers to be able to employ this type of technology.

The information regarding the types of solenoids to be used may also be very difficult to come by as most of the electromechanical device used on current motor vehicles is funded by major manufactures and the information is paid by the guaranteed large production runs. This is usually due to the fact that these devices are in a highly stressful environment and are prone to fatigue.

The measuring of how theses devices work is however a main part of this project. This approach is somewhat flawed by the time take but has endeavor to help understand certain aspects.

At this stage it seems that a retrofit of current devices seems more plausible as the devices used by manufactures have already been tested and are known to work in this environment.

The background information that is essential to the projects success maybe due to the large Research and Development program that is utilized at my current workplace and which has been made available to be published for this project.

Other source of good information was found from the Society of Automotive Engineers (SAE) documents. These are technical papers that have been submitted by practicing engineers for the work that have completed. Before the work is deemed worth it is put before a panel of Engineers to check it validity. Papers where found that have described the results of a moderately variable “cam less” engine performed by BMW.

6.0 Project Methodology

I think in the initial stages the retrofitted engine will have a camshaft installed with a mock up of the final design that will add lift and duration to the original camshaft. In this way the engine should still be able to be run and hopefully the benefits of the design will be seen at this point.

For preliminary testing the control of one camshaft may be easier with the expected benefit still able to be seen. The engine would need to have an overhead camshaft configuration with fewer valves the better. So a 4 cylinder 2 valve overhead cammed engine would be preferable. In this configuration the exhaust valves would be driven from the crank and the inlet valves could be driven via the control system.

The MOTEC M800 aftermarket ECU's has an injector drive circuit that has a fully adjustable phasing time that can be mapped against RPM and throttle position. This would make it possible to drive the hydraulic solenoids in order to control the pulse width (valve lift) and the valve opening angle vs crank position. The rail pressure control valve would also be able to be controlled via one of the many pulse width modulated outputs. This feature can also be mapped via RPM and throttle request.

7.0 Consequential Effects

The outcome of the project is to increase the efficiency of the internal combustion engine for both spark ignition and compression ignition engines. This will mean less emissions lower fuel consumption and gain overall power. The positives may however be outweighed by the possibly higher engine production costs due to the extra control devices. It is impossible to make a calculated judgment, as final testing of the engine is not completed.

There are no ethical implications that I can foresee, as there are numerous amounts of research and developments into gaining engine efficiency. However depending upon my final design it may encroach on current patents surround this idea and careful investigation may need to be sought.

7.0 Resource Management and Risk assessment

The overall cost of the thesis may run into the hundreds if not thousands of dollars due to the high amount of highly engineer parts required. Also the use of laboratory testing facilities is very expensive. My employer and I are willing to cover all costs of development and testing as my current workplace is very well setup. With the use of our current four wheel and two wheel drive laminar flow dynometer cells it is possible to measure practically all the benefits of the design including a five gas analyzers. There is also a high level of bench top scoping devices.

The main risks involved are with damaging expensive engine parts and devices. I will build into my current and final design safety features that will hopefully save any potentially damaging failures.

References

[1] Transport and Environment as Viewed 20 Sep 2007
http://ec.europa.eu/environment/air/transport.htm

[2] Laboratory test as Viewed 25 Oct 2007
http://www.simech.unimo.it/Download/Terza%20Relazione%20Semestrale/R3.4.2.pdf

BMW Group, 2000 The Third generation of Valve trains – New Fully Variable for Throttle-Free Load Control SAE Technical Papers 2000-01-1227

Claus Brustle and Dietmar Schwarzenthal, 2001 VarioCam Plus – A highlight of the Porsche 911 Turbo Engine SAE technical Paper 2001-01-0245

Compliance Engineering - The 42-V Powernet and its Influence on Automotive EMC as viewed on the 11 Oct 2007 http://www.ce-mag.com/archive/04/Ohara.html

American High Performance Seals – Materials as viewed 13 Oct 2007
http://www.ahpseals.com/new/tech/materials.php