DESIGN AND DEVELOPMENT OF HIGH PRESSURE NATURAL GAS INJECTOR MODEL FOR SINGLE CYLINDER FOUR STROKE ENGINE USING MATLAB SIMULINK

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I hereby declare that the work in this thesis is my own except for quotations and summaries which have been duly acknowledged. The thesis has not been accepted for any degree and is not concurrently submitted for award of other degree.

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DEDICATION

To the one who made me stand til the last seconds.

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ABSTRACT

The aim of this project is to design and develop the hydraulic model for control oriented injector model for single cylinder four-stroke engine using Matlab Simulink. Injector plays significant role in determining engine performance by controlling the fuel supplied consequently the air to fuel ratio. In order to achieve acceptable accuracy for abroad range of engine speed the control of injection process must be carefully studied. The model consist of hydraulic, electromagnetic and mechanical model. This thesis is carried out with purpose to produces a reliable hydraulic model that can be used to study the control process. Reasonable mass flow rate data need to be determine by the model based on predetermined boundary condition. The fuel injector based on HDEV 5 were design using Matlab Simulink. The are two different Matlab Simulink design the fuel flow rate coresponding to the radius/diameter ratio and the fuel flow rate corresponding to the needle-seat relative displacement. Based on the graph and explanation, comparison between fuel flow rate and r/d ratio and fuel flow rate and needle-seat relative displacement are made. The result show that there is cavitation occur at r/d = 0.02 and effect the flow rate of the fuel injected. While in the injector opening phase, the fuel flow rate is slightly increase before the injector reached the transition value. The model of flow through the control volume feeding and discharge holes was further detailed as it was shown to play an important role in determining the flow regime in the orifice.

ABSTRAK

Tujuan projek ini adalah untuk merekabentuk dan membangunkan model hidraulik bagi model kawalan berorientasikan penyuntik untuk enjin satu silinder empat lejang yang menggunakan Matlab Simulink. Penyuntik memainkan peranan penting dalam menentukan prestasi enjin dengan mengawal bahan api yang dibekalkan menerusi nisbah mampatan udara. Bagi mencapai ketepatan yang diterima bagi rangkaian luar negara kelajuan enjin kawalan proses suntikan perlu dikaji dengan teliti. Model ini terdiri daripada model hidraulik, elektromagnetik dan mekanikal. Tesis ini dijalankan dengan tujuan untuk menghasilkan satu model hidraulik yang boleh dipercayai yang boleh digunakan untuk mengkaji proses kawalan. Kadar aliran minyak yang munasabah perlu ditentukan oleh model yang berdasarkan keadaan sempadan yang telah ditetapkan. Suntikan minyak dibina berdasarkan model injector HDEV-5 menggunakan Matlab Simulink. Dua bahagian Matlab Simulink yang berlainan di rekabentuk berdasarkan kadar jisim minyak dan nisbah r/d yang berlainan dan kadar jisim minyak berdasarkan perubahan jarum anjakan tempat duduk. Berdasarkan graf dan penjelasan lanjut, perbandingan antara kadar aliran bahanapi dan nisbah r/d dan kadar aliran bahan api dan jarum tempat duduk anjakan relatif dibuat. Menunjukkan hasil bahawa ada peronggaan berlaku pada r/d=0.02 dan mengakibatkan perubahan kadar aliran bahan api yang disuntik. Manakala dalam fasa pembukaan suntikan, kadar aliran bahan api sedikit peningkatan sebelum penyuntik mencapai nilai peralihan. Model aliran melalui isipadu kawalan dan lubang pelepasan seterusnya diperincikan kerana ia telah ditunjukkan untuk memainkan peranan yang penting dalam menentukan rejim aliran orifis.

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LIST OF SYMBOLS

Q	Flow Rate
g	Radial gap
p	Pressure
u	Discharge Coefficient
A	Area
ρ	Density
f	Wall friction
l	Length
d	Diameter
U_{vc}	Vena Contracta Discharge Coefficeint
P_{vc}	Pressure Vena Contracta
P_u	Pressure Inlet
P_d	Pressure Outlet
$u_h^d\,(\xi)$	Opening Phase Discharge Coefficient
ξ_M	Maximum needle lift

CHAPTER 1

INTRODUCTION AND GENERAL INFORMATION

1.1 Introduction

The combustion efficiency of an internal combustion engine is the completeness of vaporization of the fuel and the mixing of fuel and air in the mixture which is fed into the combustion chamber. Previously, the fuel was injected to the engine combustion chamber via the carburettor. However carburettor nowadays is completely replaced by the fuel injection system since it consumed more fuel and the air emissions are heavier than fuel injectors. Fuel injection systems are becoming popular for engines performance. There are two versions of fuel injection which are port fuel injection and direct injection. Direct fuel injection is the latest developed fuel injection. This system was designed specifically for four stroke or two stroke engines. Direct injection system helps to introduce fuel and air to be perfectly released and then injected into the cylinder according to the engine load conditions. It will result in a high power output, greater fuel efficiency and much lower emissions. In order to minimize the automotive pollution, natural gas has become the most effective fuel for vehicles. Natural gas is the most versatile alternative fuel available and can be used in speak ignited or compression engine from the smallest motorcycles to the largest rail locomotives. It is also becoming one of the most dominant bio fuels available worldwide. Natural gas can be used either as compressed natural gas (CNG) or liquefied natural gas (LNG). CNG is odourless, colourless and tasteless that mostly consists of methane. CNG vehicles store natural gas in high pressure fuel cylinders at 207 bar to 248 bar. LNG changed from a vapour to a liquid at room temperature by application of pressure. LNG has to be cooled to very low temperature in order to turn it to liquefy. LNG can be stored onboard a vehicle at 1 bar to 10 bar. The most important advantage of natural gas is that it is extremely clean burning when used in internal combustion engines. Exhaust emissions from NGV is much lower than those vehicles powered by gasoline. For instance, it is effectively reduced carbon monoxide (CO) and nitrogen oxides (NOx) production. Direct injection natural gas promise thermal efficiencies comparable to those attained by high compression ratio. There was an investigation towards the possibility of eliminating gasoline and diesel engines on natural gas engine by using fuel injection technology. Experiments on multi-point injection engines lead to several disadvantages because of the non-uniform distribution of air-fuel mixture, efficiency intake reduction and increase of fuel consumption(Jalaluddin, 2011).Potential benefits that associated with charge stratification can be obtained when direct injection of natural gas is used. It is occur when the gas is injected closely to the start of combustion. The benefits are enhance possible reduction of HC emission, and increased thermal efficiency and power. The diesel engine efficiency at all loads and speeds can be preserved by implement the late cycle injection of natural gas. This method needs gas supply at approximately 20 Mpa. Direct injection of high pressure natural gas can contribute high efficiency over a wide load range with diesel-to-gas ratio as minimum as 5% into the dual fuel diesel engines. There are few companies that manufactured low pressure CNG manifold injectors, for instances, IMPCO, Keihin AFS and Quantum companies. These injectors that are with 3.9bar typical maximum fuel pressure are also being used in a light-duty passenger car like dedicated CNG Honda Civic GX, and dedicated CNG Toyota Camry Sedan. Later, Cummins and Westport also producing direct injector for CNG applications. They are designed primarily for medium to large sized diesel engines. Daimler- Benz AG (Jorach*et al*, 1995), Westport Research Inc (Ouellette *et al*, 1999), Siemens Automotive Corporation (Pace *et al*, 2004), Ford Global Technologies (Reatherford and Johnson, 2006), Denso Corporation and Digicon (Kato and Date, 2006) have patented many gaseous direct injectors. Today, liquid injection becomes a well-developed technology. However, gaseous injection technology that is mainly for direct injection applications in low or high pressure is still its infancy. The density of natural gas varies appreciably with pressure and temperature even if it is stored in liquefied state (LNG). It is gaseous at the temperature and pressure required for combustion. The maximum flow rate for gas injection through a restriction is obtained during choked flow condition. This occurs at the smallest area in the system that is also known as the throat in the ideal gas. Due to the friction loses which has reduces stagnation pressure, real gas in small restrictions may not choke at the smallest geometrically area.

1.2 Problem Statement

Injector plays significant role in determining engine performance by controlling the fuel supplied consequently the air to fuel ratio. In order to achieve acceptable accuracy for high range of engine speed the control of injection process must be carefully studied. The model consists of hydraulic, electromagnetic and mechanical model. This thesis is carried out with purpose to produces a reliable hydraulic model that can be used to study the control process. Reasonable mass flow rate data need to be determined by the model based on predetermined boundary condition.

1.3 Objective

The aim of this project is to design and simulate mean value injector model for single cylinder four-stroke engine using MatLab Simulink.

1.4 Work Scopes

The project detail activities are carried out within the following scopes of work:

i) The injector model is designed for a single cylinder four-stroke 150cm^3 engine.

ii) The injection is only design for high pressure application when the injection pressure range is within 20 bar – 200 bars.

iii) Regardless of engine operating condition, the injector model is operated at different injection pressure, different r/d ratio and different needle-seat relative displacement.

iv) The focus of the model is on the hydraulic description of the injector.

v) The evaluated injector parameters are the volume flow rate and discharge coefficient.

1.5 Summary

Fuel injection system become popular replacing the used of carburettor in injecting fuel to the engine. There are two versions of injection systems and the latest injection system is direct injection system that specifically designed for four strokes and two strokes engines. Injectors are the most important mechanical devices that can enhance fuel efficiency by the assistance of electronic control unit (ECU). ECU is the part of injector that has the capability to receive and process to precise the amount of fuel delivered to the engine. The main goal of this project is to design and simulate the mean value injector model for single cylinder four-stroke engine using Matlab Simulink. The mean value is predicted by the mean value models. This method is developed using Matlab Simulink.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

An engine is defined as the machine that converts the chemical energy into mechanical energy through the combustion of a fuel. The combustion of engine may refer to internal combustion engine (ICE) and external combustion engine (ECE). The ICE is an engine in which the combustion of a fuel occurs with air in a combustion chamber. An external combustion engine (ECE) is a heat engine where an internal working fluid is heated by combustion in an external source through the engine wall or a heat exchanger.

Engine is a machine with two mechanisms which are two- stroke and four-stroke engines. Two- stroke engines are normally found in low power vehicle. The two-stroke engines refer its process of combustion of a fuel and the liberation of mechanical energy that takes place. Today, the development of four- strokes internal combustion engines are widely used in transportation. The cylinder of the four strokes engine differs from the two strokes engine by which the valves are located differently.

2.2 Engine

There are two types of internal combustion engine (ICE) which are spark ignition (SI) engine and compression ignition (CI) engine. The compression ignition (CI) engine is also known as diesel engine. Both of these types of engine are further classified as two- stroke and four-stokes engine.An internal combustion engine (ICE) is one in which the fuel is burnt within the engine. It involves the system where combustion of fuel and the conversation of heat energy from combustion to mechanical energy take place within the cylinder. Internal combustion engine is divided into two categories which are spark ignition engine and compression ignition engine. The spark ignition engine consumed gaseous or volatile fuels and operates on the two or four stroke cycle. While the compression ignition engine used distillate liquid fuel and work on either two or four stroke cycle. Compression ignition engine is normally designed to operate on the dual-combustion cycle in which Otto cycle and diesel cycle. The requirement design of the engine nowadays the engine must be an optimum performance, good fuel economy, low pollution, minimum noise level, easy cold starting, economic servicing, easy cold starting, economic servicing, acceptable durability, least weight and compact size (Nunney, 2000).

2.2.1 Internal Combustion Engine (ICE)

An internal combustion engine is much famous known as IC engines. The engine operates in which the combustion of fuel takes place inside the engine block itself. There are much great energy is generated after fuel combustion and then being converted into mechanical energy. IC engine can be divided into two kinds which are rotary in which there is a rotor that rotates inside the engine to produce power and reciprocating engines that converted piston into the rotary motion of the vehicles. They are the types of engine that are widely used. In reciprocating engine, they are classified into two other types which are spark ignition (SI) engine and compression ignition (CI) engine. In spark ignition (SI) engine, it is described that the burning of fuel occurred is generated by the spark plug. While the compression ignition (CI) engine concluded that the burning of fuel occurred because of the high pressure exerted on the fuel. The fuel is compressed in to high pressure and its start to burn. They are either two stokes or four strokes engines.

2.2.2 Four- stroke Engine Cycle

Stroke is refers to the movement of the piston in the engine. A four-stroke engine has one compression stroke and one exhaust stroke and each is followed by a return stroke. The compression stroke compresses the fuel-air mixture prior to the gas explosion. The exhaust stroke simple pushes the burnt gases out the exhaust. The four-stroke usually has a distributor that supplies a spark to the cylinder only when its piston is near top dead centre (TDC) on the fuel compression stroke. Some four-stroke engines do away with the distributor and make sparks every turn of the crank. This means a spark happens in a cylinder.

2.2.3 Principle Operation of Four-Stroke Engine

i) Intake stroke

In which the air is admitted to the engine cylinder where the intake valve is opened and a mixture of gas and air is drawn into the engine.

ii) Compression stroke

The charge of fresh air is compressed by the piston, and the fuel is injected just before the point of maximum compression and it’s raise both the pressure and temperature as it is compressed into the lesser volume of the combustion chamber. During the compression stroke, the intake valve is closed and the piston travel s up compressing the fuel-air mixture. The spark occurs just before the piston reaches the top of its stroke.

iii) Power stroke

The air-fuel mixture is ignited by the sparking plug during which the gases expand. Then the pressure rises due to combustion and pushes piston downwards to drive the engine.

iv) Exhaust stroke

Exhaust valve is opened at the end of power stroke and the piston travel back up expelling the exhaust gases through the exhaust valve. At the top of this stroke the exhaust valve is closed.

2.2.4 The Difference between Gasoline, Diesel and CNG engine

Diesel engine and gasoline engine are quite similar. They are both internal combustion engines designed to convert the chemical energy available in fuel into mechanical energy. Both diesel engines and gasoline engines convert fuel into energy through a series of small combustion. The major difference between these two types of engines is the way the combustion happened. In a gasoline engine, fuel is mixed with air, compressed by pistons and ignited by sparks from spark plug. In diesel engine, the air is compressed first and then the fuel is injected. Gasoline engine is an internal combustion engine with spark ignition that is designed to run on gasoline and similar volatile fuels.

A diesel engine also known as a compression ignition engine which is an internal combustion engine that uses the heat of compression to initiate ignition to burn the fuel injecting into the combustion chamber. In a diesel engine, air and the fuel are infused into the engine at different stages, as opposed to gas engine where a mixture of air and gas are introduced. Fuel is injected into the diesel engine using an injector whereas in a gasoline engine, a carburettor is used for this purpose. In a gasoline engine, fuel and air are sent into the engine together and then being compressed. The air and fuel mixture limits fuel compression and the overall efficiency. A diesel engine compresses only air and the ratio can be much higher. A diesel engine compresses at the ratio of 14:1 up to 25:1, whereas in a gasoline engine the compression ratio is between 8:1 and 12:1.diesel engines can either be two cycles or four cycles and are chosen depending on mode of operation.

There are two types of natural gas engine which are ignited natural gas engine and compression ignition natural gas engine. Compared to gasoline, compressed natural gas (CNG) is cleaner and less expensive.

2.3 Engine Management System (EMS)

The Engine Management System (EMS) is a small-scale computer that controls the running of an engine by monitoring the engine speed, load, temperature and providing the injection at the right time for the prevailing condition (Indraguna V.E, 2006). The EMS is comprised of sensors for intake air and coolant temperature, intake manifold absolute pressure (MAP) and throttle position (TPS), as well as sensors for engine speed, and signals for the required injection and ignition sparks events and a sensor for information about the oxygen content in the exhaust.

Furthermore, there is an idle speed motor for adjusting and stabilizing the idle speed, or an electronic throttle body and finally a fuel pressure regulator and fuel injectors. The supplied high energy ignition coils are controlled by the integrated ignition module. The EMS is easy to calibrate, as its reduced time and cost to go from prototype to production. Automobile industries can even perform most of system integration in their own laboratory.

2.3.1 General Function

The Engine Management System (EMS) is responsible for controlling the amount of fuel being injected and for adjusting the ignition timing. It controls the running of an engine by monitoring engine speed, load, and temperature. Optimum functioning of the EMS assures maximum engine power, with the lowest amount of exhaust emissions and the lowest fuel consumption (Indraguna V.E, 2006).

2.3.2 Method of Air Control

An idle air control (IAC) motor is an electrically-operated valve which designed to adjust the engine idle RPM speed by opening and closing and air bypass passage inside the throttle body. The vehicle computer or electronic control unit (ECU) receives information from various sensors and will output signals to adjust the idle air control motor in or out to adjust engine idle speed by controlling engine idle air.

The intake manifold function is to receive the mixture from either the carburettor or the throttle body injector of a single-point fuel injection system and to distribute it evenly with least variation in air-fuel ratio, to the inlet valve ports.

2.3.3 Method of Fuel Control

The electronic fuel injection (EFI) accurately controlling the fuel mixture (A/F ratio) based on feedback from sensors connected to engine. The data such as engine speed, crankshaft position, air an engine load is required from the data sensor to provide optimum control of the system. The analogue signals from the sensor were converted to digital signal for computer to understand. The injector will injected the fuel with precise accuracy and a significant reduction in fuel droplet size due to the ability to meter fuel by the EFI system engine.

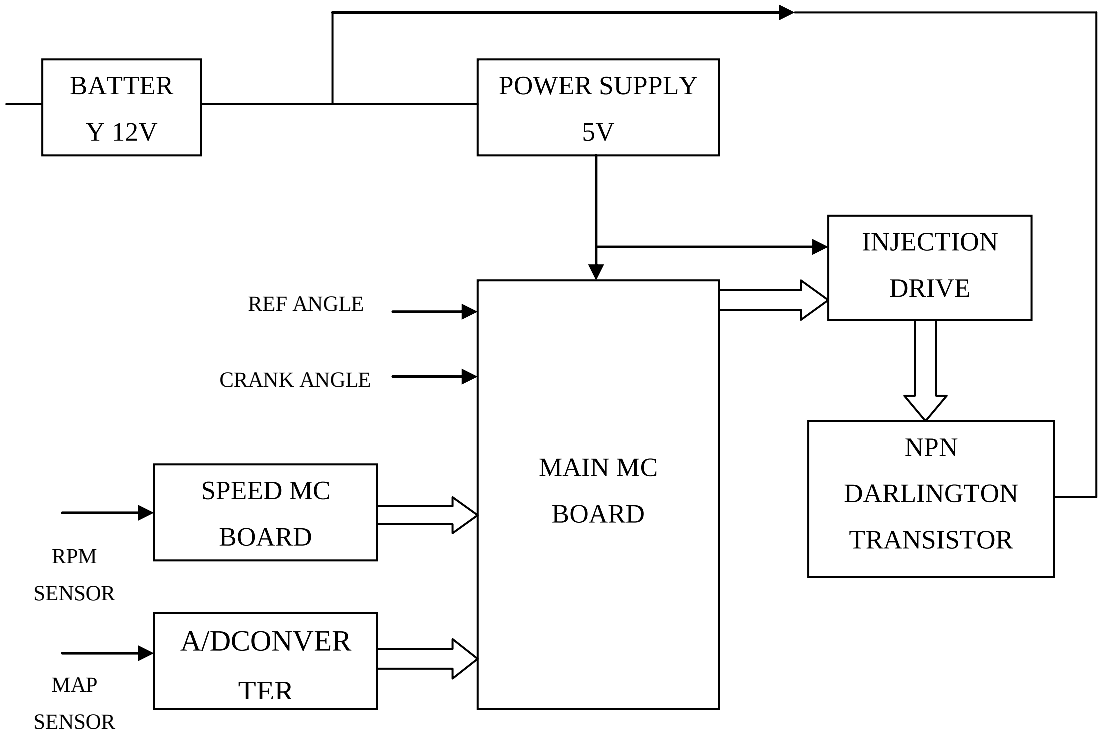


Figure 2.1: Overall EFI functional block

Source: Indraguna, 2006

2.3.4 Method of Ignition Control

The ignition system is necessary in engine motor vehicle to initiate the power stroke by ignite the combustible charge of air and fuel in the cylinder. It requires very high voltage from the ignition system in engine cylinder about 10,000 V. In actual practise the spark to ignite the combustible charge is not timed to occur exactly at the beginning of the power stroke but rather earlier than this and towards the end of the compression stroke. The ignition timing is necessary to allow sufficient time for the combustion process to take place, so the maximum cylinder gas pressure can be attained just after the piston passes top dead centre (TDC) (Nunney,2000).

2.4 Electronic Fuel Injection (EFI) System

Fuel injection (FI) technology is a technology that is widely used in vehicles these days. The technology is used to eliminate the need for carburettors. Carburettor is a mechanical system that mixes fuel with air (Indraguna V.E, 2006). The fuel injection system helps the engine to supply fuel directly to the cylinder in the intake manifold. The flow of fuel injected is regulated by the sensors located in the engine and then maintains it to appropriate levels.

The first fuel injection systems were throttle body fuel injection systems or single point systems, which had an electrically controlled fuel injector valve. Later, these fuel injection systems were replaced by more efficient multi-port fuel injection systems, which have a separate fuel injector for each cylinder. This latter fuel injection system is better at metering out fuel accurately to each cylinder. It also provides for a faster response.

Electronic fuel injection is much more complicated compared to carburettor since the fuel injection systems are developed to improve the fuel efficiency. It is slowly eliminate the use of carburettor that is clearly polluted the environment. In the fuel injection system, the fuel is efficiently mixes with air and supplied to the combustion chamber to produce efficient power to the vehicle.

The heart of the fuel injection system is the electronic control unit (ECU) which is entirely control the injection system. Various sensors transmit information to the control unit concerning air temperature, engine temperature, engine speed and engine load. The control unit used this information to determine the exact amount of fuel to be distributed to the cylinders (Clymer Publications, 1972).

2.4.1 Basic Component of EFI

The most important part of the whole electronic fuel injection (EFI) system is the electronic control unit (ECU) which controls the fuel map. Every sensor located in the engine and throughout the rest of the vehicle sends information to the ECU. Then ECU interprets the information and uses to keep the vehicles moving. ECU continually selects the engine control inputs such as air flow rate, fuel flow rate, and spark timing besides processing the information carried by several sensors (Pushkaraj, 2005). The ECU is though to be useless without it sensors. The five important sensors are:

i) **Mass Air Flow (MAF) Sensor**

MAF snsor measures the amount of air coming into the engine. When the engine is idling, less air is drawn. However, once the vehicle in motion, more air is drawn into the engine. So more fuel fuel in needed from the injectors.

ii) **Oxygen (O_2) Sensor**

The sensor is located in the exhaust system that detects the amount of unburned oxygen and fuel coming from the engine. ECU can adjust the amount of fuel injected into the engine to increase efficiency.

iii) **Throttle Position Sensor (TPS)**

TPS tells the ECU how hard and how quickly the gas pedal is pushed. The faster the pedal is pushed, the more the amount of fuel need to be added to the engine for furthetr speed.

iv) **Manifold Absolute Pressure (AMP) Sensor**

AMP snsor measures the changes in the engine manifold pressure which send signal to the ECU on how much load the engine need to bear. If it is in a high pressure, Ecu will lower the the engine vacuum and add mor fuel. However if there is a low pressure, the ECU will raise the vacuum and dial down the fuel injection.

v) Vehicle Speed Sensor (VSS)

This sensor works on by alerting the ECU on how fast the vehicle is moving and adjust the fuel to appropriate amount.

As being stated earlier, fuel injectors are the electro-mechanical devices that are used to meter fuel. The fuel injector is the small nozzle into which liquid fuel is injected at high temperature. The fuel injector is designed to introduce fuel at the correct time, at correct pressure and at correct quantity than a carburettor. Therefore, no fuel is wasted. The precise metering of fuel result in the increased power, lower emissions and lower the fuel consumption. A high pressure of fuel is essential to lift the nozzle valve for better penetration into the combustion chamber. The proper atomization also enhances the efficiency of the engine. The injector is a type of valve that is controlled electronically, which opens and closes and supplies atomized fuel to the engine. It sprays fuel into the intake valves directly in the form of fine mist. The injector opens and closes rapidly, and the pulse width determines how much fuel goes into the valve. Fuel is supplied to the injectors by a fuel rail.

There are two kinds of fuel pumps which are the mechanical fuel pump that was used in carburetted cars and the electric fuel pump that is used in vehicle with electronic fuel injection (EFI). In the EFI system, the fuel pump is electronic by means it is electronically controlled and powered. Others component is the fuel pressure regulator which a mechanical device that is inserted in automobile engines to maintain normal fuel pressure. In most combustion engines, the main goal of the fuel pressure regulator is to maintain the fuel system pressure at 2 bar. Fuel pressure regulator have an integral role in various mechanical devices because it serves as a buffer in keeping the fuel pump function constant despite changes in the fuel amount or the speed of the pump while it is running.

2.4.2 High Pressure Fuel Injector

Higher pressures mean greater atomization which results in better burning and more power from the injected fuel charge. Injection pressure has a significant influence on particulate emission levels. The higher the injection pressure, the better the fuel atomizes during injection and mixes with the oxygen in the cylinder. This results in a virtually complete combustion of the fuel with high energy conversion. In the last 30 years, a trend to high pressure fuel injection systems with an increase of maximum injection pressure from 800 up to 2000 bar is visible. In future, very flexible high pressure fuel injection systems are necessary with multiple injection and rate shaping capabilities and a maximum injection pressure beyond 2000 bar. In this case, the most important thing is the fuel injection system itself needs to reach low fuel consumption. The flexible injection system in each point of the engine map the optimum rate shaping, injection timing and multiple injections is possible to get the best compromise between emission trade off and fuel consumption.

The injection pressure for the engines that equipped with common rail technology is generated independently of the load and the engine speed. The injection pressure build up is separated from the injection process in order to enable this system. A high pressure pump generates the necessary fuel pressure to all injectors at all times. This fuel pressure is stored in a high pressure accumulator. These injectors have extremely fast solenoid valves that are actuated electronically that result in a great amount of flexibility in designing the injection process.

2.4.3 Injection Control Strategy

The control system is generally being categorized into two main groups which are open loop control system that has been commonly used since it first introduced and the other one is closed loop control system. Open loop system has eliminated the function of the driver Basic block diagram of open loop control system (Indraguna V.E, 2006).

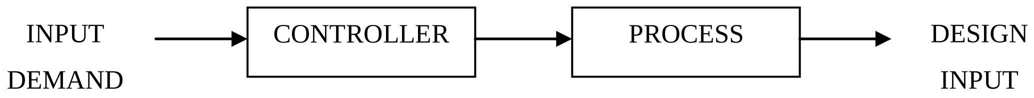


Figure 2.2: Open loop control system

Source: Indraguna, 2006

After few years, a greater precision of the timing needed that must relate the timing of the spark to the factors such as engine speed and engine load has been introduced by the electronic control unit (ECU). The timing requirements are stored in a memory unit. However in the modern loop open system the engine is in the same condition when the memory was programme n the memory unit no longer effective which caused the timing given by the look up table unsuitable for the engine. The air to fuel ratio is maintained with the need of efficient feedback control system. The differences of open loop system and closed loop system is the driver as sensor feedback by which the driver need to manipulate the main control, adjust the ignition timing and set the air to fuel ratio.

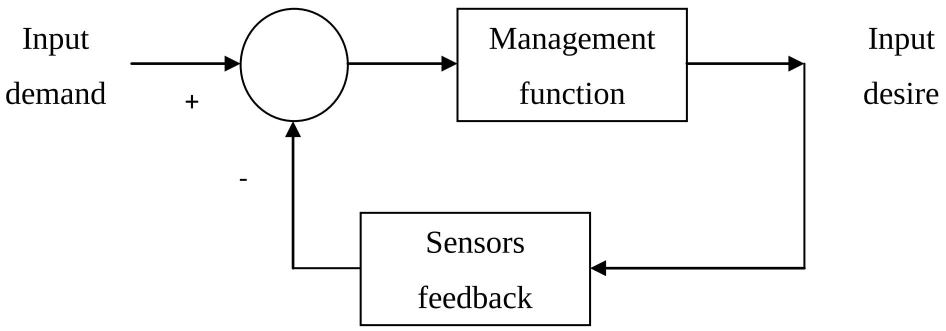


Figure 2.3: Closed loop feedback control system

Source: Indraguna, 2006

The used of early injection timing achieves a homogenous mixture because of the longer time available for mixture preparation. It was found that retarding the injection timing produces higher CO because the mixture become less homogenous and the combustion become less stable. In the PFI engine, the early injection generates a blue flame. However a yellow flame that is indicative from stratified combustion is observed when the injection timing is retarded to the end of the intake stroke. A later-injection timing improves the start of the combustion and advance the heat release point due to the present of a rich mixture. For the late injection, stratified charge mode, an excessively rich mixture near the spark gap must be avoided in order to maintain a stable ignition and minimize smoke.

2.5 Compress Natural Gas (CNG) Properties

Compressed Natural Gas (CNG) is used as a fuel for vehicles in which it is should be compressed. The common pressure for CNG is 207 bar to 248 bar. CNG fuel systems deliver fuel to the ICE as a result of the stored pressure in the fuel tank. There are no fuel pumps or compressors. As CNG moves from the tanks to the engine, its pressure reduce in steps to a pressure that is slightly above atmosphere (Bectold, 1997). Natural gas undergoes a rapid decrease in pressure and temperature as it moves through the fuel system.

A CNG leak can quickly produce a large gas/air mixture volume. A unit volume of 207 bar will expand by approximately 200 times when released into the atmosphere. The ignition energy required is in the range of 0.15 to 0.30 mJ. There is a danger of ignition if a CNG leak occurs because a portion of the gas/air mixture will be within the flammable range and an ignition source may be present.

The fact that the natural gas is stored at a high pressure for use as a vehicular fuel is the most unique physical characteristic of CNG. The presence of a gas that is stored and transferred at pressures that far exceed the normal experience of most fleet operations personnel raises the standard of precaution and training required. Inadvertent opening of valves or loosening of fittings containing high-pressure natural gas not only can create a fire hazard, but also can result in the high-velocity ejection of metal parts or fragments that could be lethal to nearby personnel (H. Montiel et al, 1996)

Table 2.1: Table of fuel properties

PROPERTIES	GASOLINE	DIESEL NO.2	LPG(HD-5)	CNG
Physical state	Liquid	Liquid	Gas	Gas
Boiling range (1atm)	80 to 420	320 to 720	-44 to 31	-259[a]
Density (lb/ft^3) (lb/gal)	43 to 49	49 to 55	31[b]	8[c]
	5.8 to 6.5	6.5 to 7.3	4.1[b]	1.07[c]
Net energy content Btu/lb	18700-19100	18900	19800	21300[a]
	112000-121000	123000-128000	82000	22800[a]
Autoignition temperature (^oF)	400-900	400-500	920-1,020	1,350
Flashpoint (^oF)	-45	125 (min)	-100 to-150	-300
Octane Number Range (R+M) 2	87 to 93	n/a	104[e]	120[e]
Flammability Limits (volume % in air)	L = 1.4	L = 0.7	L = 2.4	L = 5.3
	H = 7.6	H = 5.0	H = 9.6	H = 14
Human exposure limit fuel for fuel (ppm)	500	n/a	n/a	Nontoxic

Source: http://www.mckenziecorp.com/dehydration.htm

2.6 Fuel Injection (FI) Modelling

The fuel injection (FI) modelling includes a wide range of standard fuel and injection specific components. A range of simulation modes enables to model the complex system in order to predict fuel system to ensure the adequate fuel flow and pressure, accurate fuel metering, avoidance of interaction effects and control pressure transients.

2.6.1 Types for Modelling

2.6.1.1 Mean Value Model

Two main approaches can be found when considering the cylinder. The method that is normally used is mean value engine method which is the method that defines number of cylinders as one which occupies whole displacement volume. An average value over a cycle modelled the fluctuating flow through the inlet port. The aspects of this approach are the dynamics of speed, engine torque; pressure built up in the inlet and the exhaust manifolds.

2.6.1.2 Cylinder to Cylinder model

The other method is the cylinder by cylinder engine approach. The difference between the mean methods is, it describes each cylinder individually. The mean model generally adequate to be used in the process of control system design. However the cylinder by cylinder method is better in engine performance development because it derives from engine geometries which are very useful for the engine improvement in the future.

2.6.1.3Limit of Physical Properties

When deciding on the basis of a model, there are numeral approaches are available. Physical equation describing that the system is the most common method since it creates a general model working for many operating areas. The disadvantage of the system is, it might be difficult to be described correctly in theory. The black box is a table of two or more dimensions in which the measured data is stored into it. Since it is directly based on empirical formula this approaches often yield the result accurately. The major basis of the model rest on physical equations. While the equations of empirical are used to model certain complexities.

2.6.2 Overview of Engine / Injector modelling using MatLab Simulink

In this subchapter the gasoline engine will be described in detail bases on the pressure inside the cylinder prediction. The engine modelled was developed to describe effects of each parameter on the engine performance, the cylinder by cylinder approach is used to describe the modelled (Sitthiracha, 2006).

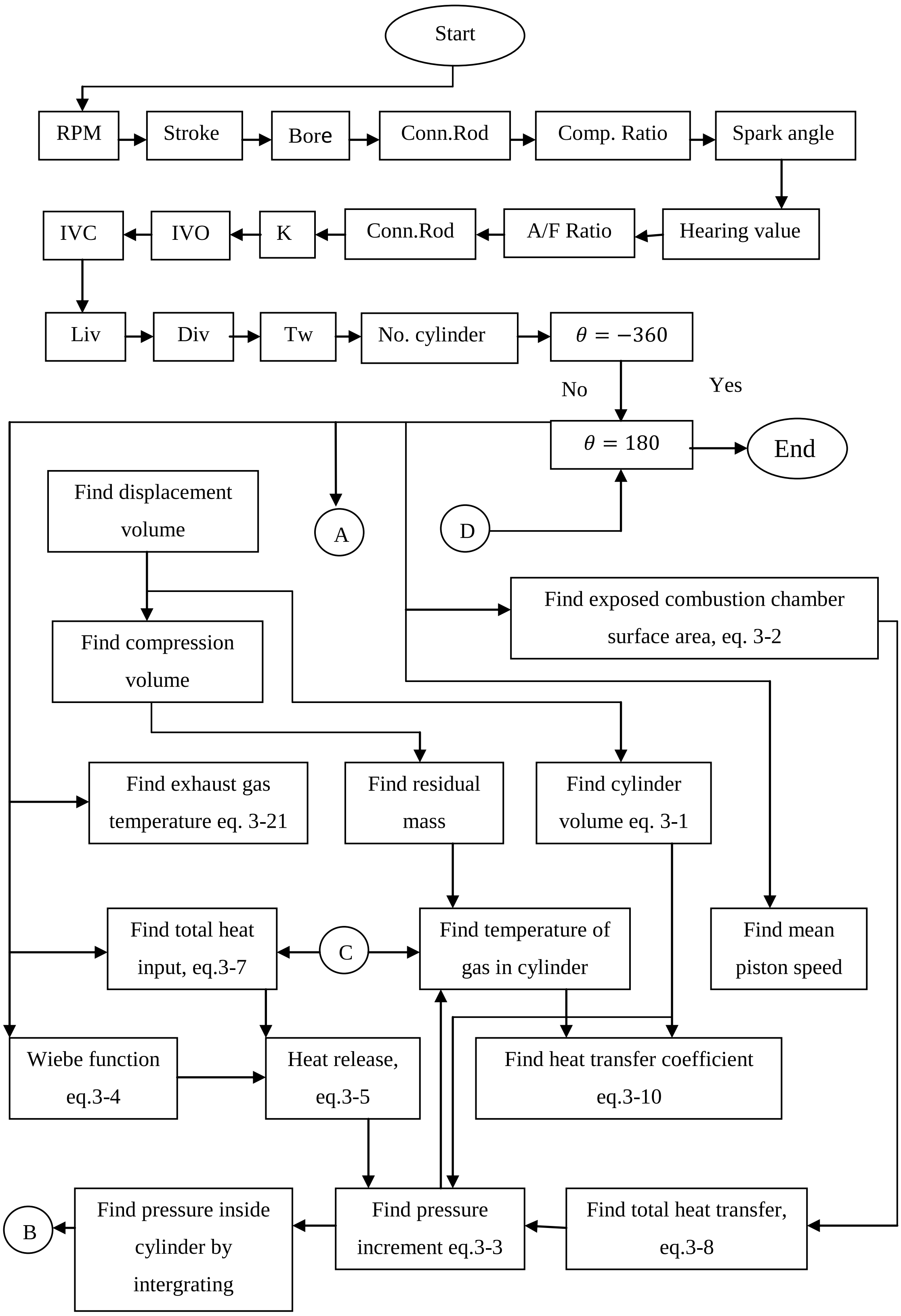


Figure 2.4: Model overview

Source: Indraguna, 2006

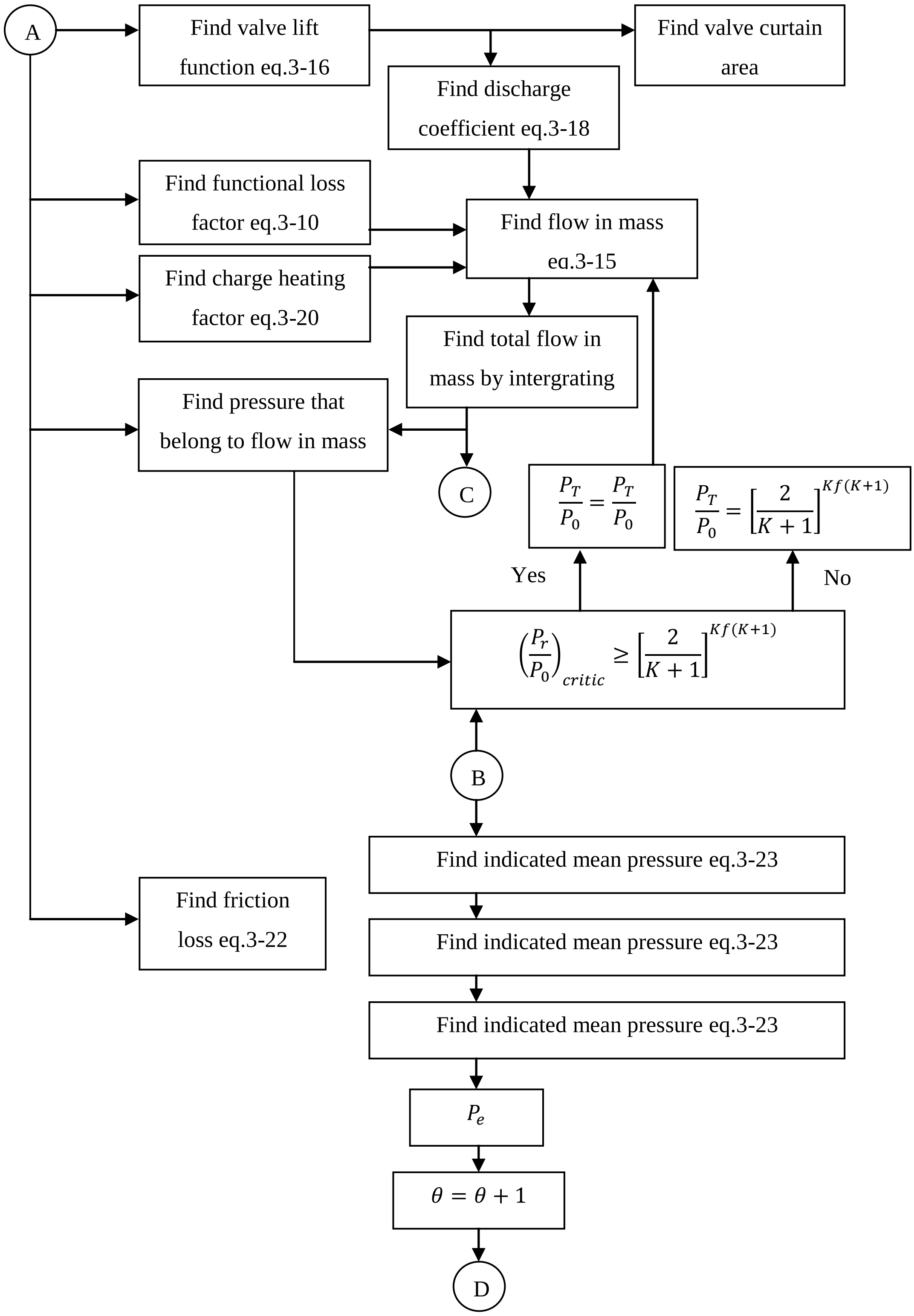


Figure 2.5: Model overview (continued)

Source: Indraguna, 2006

2.7 Summary

Engine is an important machine that runs the duty of fuel combustion. Fuel combustion of engine is categorized by two which are internal combustion engine (ICE) and external combustion engine (ECE). Internal combustion engine (ICE) is an engine whose fuel is burnt inside the engine itself. The internal combustion engine (ICE) is separated into two types which are spark ignition (SI) engine that is also known as petrol and gasoline engine, while the other one is compressed ignition (CI) engine that is also known as diesel engine. Both of these types of engine involved the two strokes and four strokes engine cycles. Four strokes engine cycle refers to the movement of the piston in the engine that involves four stages of cycle which are intake stroke, compression stroke, power stroke and exhaust stroke. It is many parts of engine and its system that completed the delivery of fuel in vehicles. For instance is the engine management system (EMS) that monitored the engine speed load, temperature and providing the fuel injection at the correct time. With the advance of mechanical technology, the fuel injection system is widely used in vehicles and thus eliminating the used of carburettors. The first fuel injection system is throttle body fuel injection system or single point system. However the latest multi-points injection system came out that is more efficient. It efficiency is proven by the accuracy of fuel delivers to each cylinder. The basic component of electronic fuel injection (EFI) is electronic control unit (ECU). This electronic control unit (ECU) will be functioned well with the assistance of five important sensors that sent information to ECU to be interpreted to keep the transport moving. The five important sensors are mass air flow (MAF), oxygen (O_2) sensor, throttle position sensor (TPS), manifold absolute pressure (Amp) sensor and vehicle speed sensor (VSS). The injection control strategy is categorized into two main groups which are open loop system and closed loop system. The fuel injection modelling is based on the standard fuel and injection specific components. This will enable to model complex fuel system with enough fuel metering, flow and pressure. It involved two types of modelling which are mean value models and cylinder to cylinder model.

CHAPTER 3

METHODOLOGY

3.1 Introduction

Bosch HDEV 5 is one of a manufactured high pressure injector. It was developed to be used as a port or a direct injection. HDEV 5 functioning to meter fuel and obtain well defined mixture of a fuel and air. It is an inward opening solenoid injector that ensures the linearity at short injection times in a very stable condition. This injector is considered a very beneficial injector as it has a high spray variability that concerning spray angle and shape. Besides that, the flow rate can be defined in a huge range.

3.2 Structural Design of an Injector

Figure 3.1: HP injection Valve HDEV 5

Source: http://www.bosch-motorsport.com

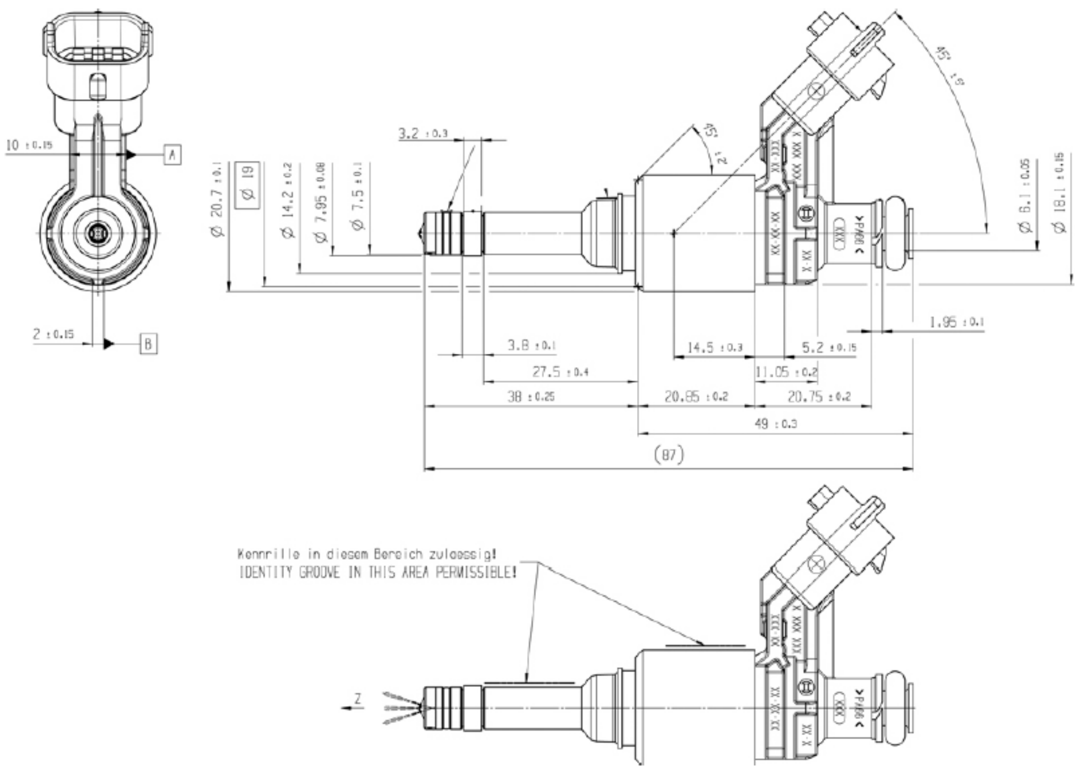


Figure 3.2: HDEV 5 dimension diagram

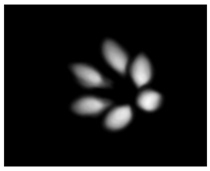
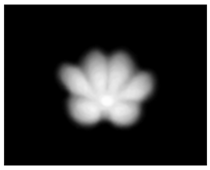
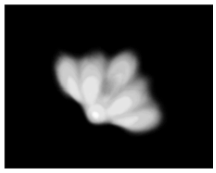
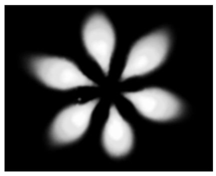
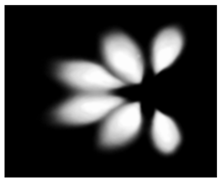
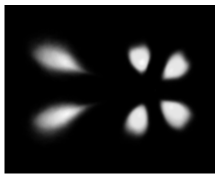
Source: http://www.bosch-motorsport.com

Table 3.1: HDEV 5 specification table

Application	
Application	308 – 1,026 g/min @100 bar
Fuel input	Top-feed-injector
Fuel	Gasoline
Operating pressure	200 bar
Operating temperature range	$-31 - 130$℃
Storage temperature range	$-40 - 70$ ℃
Max. Vibration	$600\ ms^{-2}$
Electrical Data	
Booster supply	65-90 V
Booster current	13.2 A
Booster time	$500\ \mu s$
Power supply	12 V
Pick-up current	9.6A
Pick-up time	$800\mu s$
Hold power supply	12V
Hold current	3 A hysteresis 0.8 A
Coil resistance	$1{,}500\ m\Omega$
Mechanical Data	
Weight w/o wire	68g
Diameter	20.7mm
Length	87mm
Characteristics	
Spray type	Multihole
Number of holes	4-7 holes
Spray angle overall	110°
Spray angle single beam	$8 - 20°$
Static flow tolerance	$\pm$ 5%
Dynamic flow tolerance	$\pm$ 6% @ $t_1 -$ 1.5 ms
Leakage	$\leq$ 2.5 mm^3/min @ 23℃
Connectors and Wires	
Connector	Kompakt
Mating connector	A 928 000 453

Source: http://www.bosch-motorsport.com

Table 3.2: HDEV 5 variation spray targeting

Examples of variations, further variations on request	
	Variations spray targeting
	
	
	
	
	
	

Source: http://www.bosch-motorsport.com

3.3 Physical Model of an Injector

3.3.1 Hydraulic Model

For injector, the mathematical model leads to the definition of hydraulic model, mechanical model and electromagnetic model. According to zero-dimensional approach, the hydraulic parts of the injector that have limited spatial extension are modelled with ideal components for instance uniform pressure chambers and laminar hydraulic resistances. While according the one-dimensional approach, the nozzle delivery chamber and injector inlet are connected by the internal hole. This is due to the wave propagation phenomena in these parts that play an essential role in determining injector performance. In zero-dimensional approach, the continuity and compressibility equation for every chamber in the model is:

$$\sum Q = \frac{V}{E_l}\frac{dp}{dt} + \frac{dV}{dt} \tag{1}$$

The unknown $\boldsymbol{\sum Q}$is the net flow rate that comes into the chamber. While $\boldsymbol{\frac{V}{E_l}\frac{dp}{dt}}$ is the increasing rate of the fluid volume in the chamber due to the fluid compressibility and $\frac{dV}{dt}$ is the deformation rate of the chamber volume.

The laminar flow hydraulic resistances are characterized by a flow rate proportional to the pressure drop $\boldsymbol{\Delta p}$ across the element, is used to model the fluid leakages occurred between the coupled mechanical elements in relative motion for example the needle and its liner or control piston and control valve body.

$$Q = K_L \Delta p \tag{2}$$

The theoretical value K_L of an annulus shaped cross-section flow area can be obtained by:

$$K_L = \frac{\pi d_m g^3}{12 l \rho v} \tag{3}$$

In this case, the leakage flow rate can be underestimated by the equation 3. The rate may be as low as one third of the real one. Equation 2 and equation 3 depend on the third power of the radial gap $\boldsymbol{g}$. The value $\boldsymbol{K_L}$ has to be experimentally evaluated in the real injector working condition in order to take these effects of the leakage flow rate into account.

In the control feeding and discharge holes, there will be a turbulent flow that is assumed to be accoutred in the nozzle and in the needle- seat opening passage. According to Bernoulli's law, the flow rate through these orifices result in is proportional to the square root the pressure drop, $\Delta\boldsymbol{p}$ across the orifice

$$Q = \mu A \sqrt{\frac{2\Delta p}{\rho}} \tag{4}$$

3.3.2 The Discharge Coefficient

The discharge coefficient of control volume orifices A and Z are evaluated according to the model by considering four flow regimes inside the holes that are laminar, turbulent, reattaching and fully cavitating. A premiliminary estimation of the holes discharge coefficient can be obtained as the following equation by neglecting cavitation occurrence.

$$\frac{1}{\mu} = \sqrt{K_I + f\frac{l}{d} + 1} \tag{5}$$

The variables of $\boldsymbol{K_l}, \boldsymbol{l}, \boldsymbol{d}\ and\ \boldsymbol{f}$ represent the inlet loss coefficient, hole axial length, hole's diameter and wall friction coefficient respectively.

The wall friction coefficient $\boldsymbol{f}$ can be evaluated as follows:

$$f = MAX\left(\frac{64}{Re}, 0.316Re^{0.25}\right) \tag{6}$$

The Re variable is the Reynolds number

The ratio between the cross section area of the vena contracta and the geometrical holes area is $\boldsymbol{\mu_{vc}}$. It is written as

$$\frac{1}{\mu_{vc}^2} = \frac{1}{\mu_{vc_0}^2} - 11.4\frac{r}{d} \tag{7}$$

The value of $\boldsymbol{\mu_{vc_o}}$ is equal to 0.61 and the variable $\boldsymbol{r}$ is the fillet radius of the hole inlet.So that vena contracta pressure can be determined as

$$p_{vc} = p_u - \frac{\rho_l}{2}\left(\frac{Q}{A\mu_{vc}}\right)^2 \tag{8}$$

$\boldsymbol{p_{vc}}$ is the pressure in vena contracta. While $\boldsymbol{p_u}$ is the oil vapour pressure.

Cavitation does not occur if the pressure in vena contracta p_{vc} is higher than oil vapour pressure p_u**.** In this case, hole discharge coefficient can be found in the equation 5 given. However, if cavitation occurs the discharge coefficient is evaluated as in the equation given below.

$$\mu = \mu_{vc}\sqrt{\frac{p_u - p_v}{p_u - p_d}} \tag{9}$$

3.3.3 Discharge Coefficient of Nozzle Holes

The discharge coefficient model of the nozzle holes are designed on the base of the unsteady coefficients. The pattern of this coefficient versus needle lift has three difference phase which is in the first phase, the moving needle tip influences the efflux through nozzle holes. This phase occurs during the opening of injector. The second phase occurs when the needle is at maximum stoke and the discharge coefficient increased in time. While the last phase shows the discharge coefficient remains constant. This last phase happened during the needle closing stroke.

The nozzle hole discharge coefficient is modelled by considering two limits curves which are lower limit trend $(\boldsymbol{\mu_h^d})$and upper limit trend$(\boldsymbol{\mu_h^s})$. Lower limit trend models the discharged coefficient transient efflux conditions. While upper limit trend represents the steady-state value of the discharged coefficient for a given needle lift. The first order system dynamics shows how the evolutionary of transient to stationary values are modelled. The transient trend presents a first region and this was experimentally observed (Catania et al,1994; 1997). The brief equation is

$$\mu_h^d(\xi) = \begin{cases} \mu_h^d(\xi_0)\sin\left(\dfrac{\pi}{2\xi_0}\right) & 0 \leq \xi < \xi_0 \\ \dfrac{\mu_h^d(\xi_M) - \mu_h^d(\xi_0)}{\xi_M - \xi_0}(\xi - \xi_0) + \mu_h^d(\xi_0) & \xi \geq \xi_0 \end{cases} \tag{10}$$

The variable$\boldsymbol{\xi}$ is stands for the needle-seat relative displacement. The unknown$\boldsymbol{\xi_o}$ is the transition value of ξ between the sinusoidal and the linear trend. While the variable $\boldsymbol{\xi_M}$ is the maximum needle lift.

The different level of deformation cause the maximum needle lift,$\boldsymbol{\xi_M}$ varies with rail pressure. The relation between ξ_M and reference rail pressure $\boldsymbol{p_{r0}}$ is shown in the equation 11. It is noted as linear.

$$\xi_M = K_1 p_{r0} + K_2 \tag{11}$$

$\boldsymbol{K_1}$and $\boldsymbol{K_2}$ are constant. While the value of ξ_0 can be determined by the equation below.

$$\xi_0 = K_3 P_{r0} + K_4 \tag{12}$$

In equation 10, when the needle seat relative displacement approaches its relative maximum value$\boldsymbol{\xi_M^r}$, the discharge coefficient increase in time. This is result in the formed of transition phase beteen the unsteady and the stationary values of hole discharge coefficient.

$$\mu_h = \mu_h^d(\xi_M^r) + \left[\mu_h^s \xi_M^r - \mu_h^d \xi_M^r\right]\left[1 - exp\left(-\frac{t - t_0}{\tau}\right)\right] \tag{13}$$

The variable $\boldsymbol{t_0}$ is the instant in time when needle seat relative displacement approaches its value. While the variable $\boldsymbol{\mu_h^s}(\boldsymbol{\xi_M^r})$ is the unsteady and stationary value discharge coefficient.

3.3.4 One-dimensional Model

To model the fluid flow in the pipe connecting injector and rail in the injector internal duct that carries the fluid from the inlet t the delivery chamber, it must follow the one- dimensional modelling approach. In the common rail injection system cavitation, the pipe flow conversation equations are written for a single- phase fluid as there is no connecting pipe appeared. In this approach an isothermal flow is assumed and need to solve the momentum and mass conversation equation.

$$\frac{\partial \mathrm{w}}{\partial \mathrm{t}} + \mathrm{A}\frac{\partial \mathrm{w}}{\partial \mathrm{x}} = \mathrm{b} \tag{14}$$

The unknown $w = \begin{Bmatrix} u \\ p \end{Bmatrix}, A = \begin{Bmatrix} u & 1/\rho \\ \rho c^2 & u \end{Bmatrix}, b = \begin{Bmatrix} -4\tau/\rho d \\ 0 \end{Bmatrix}$

The variable $\boldsymbol{\tau}$is the wall shear stress. This is has been evaluated from the assumption of the steady-state friction.

Eigenvalues of hyperbolic system of partial differential equation 14 are$\boldsymbol{\lambda} = \boldsymbol{u} \pm \boldsymbol{c}$

The wave propagation celerity, c can be evaluated as

$$c = \sqrt{\frac{c_l}{\left(1 + K_p \frac{E_l}{E_p} \frac{d_p}{t_p}\right)}} \tag{15}$$

The unknown $\boldsymbol{K_p}$ is the pipe constrain factor that is depending on pipe support layout and $\boldsymbol{E_p}$ is the Young's modulus of elasticity of pipe material. While the variables of $\boldsymbol{d_p}$ and $\boldsymbol{t_p}$ are equal to pipe diameter and pipe wall thickness.

The variable $\boldsymbol{K_p}$can be evaluated as the following equation.

$$k_p = 1 - v_p^2 \tag{16}$$

The variable $\boldsymbol{v_p}$is the Poison's modulus of the pipe material. As stated above, if there is civatation occurs, this simple pipe model is not suitable to use.

The oil thermodynamic properties are affected by temperature and pressure. These are reported different in the operation field of the common rail injection system. In the analytic functions of the exponential type, the oil properties were approximately in the range of 0.1 to 200 MPa pressure and 10℃ to 120℃ temperature. The analytic functions were derived by using the least-square method for non linear approximation. The formulae are as below:

$$\rho_l(p,T) = K_{\rho 1} + \left[1 - exp\left(-\frac{p}{K_{\rho 2}}\right)\right] K_{\rho 3} p^{K_{\rho 4}} \tag{17}$$

$$E_l(p,T) = K_{E1} + \left[1 - exp\left(-\frac{p}{K_{E2}}\right)\right] K_{E3} p^{K_{E4}} \tag{18}$$

$$v_l(p,T) = K_{v1} + K_{v2} p^{K_{v3}} \tag{19}$$

The variables $\boldsymbol{K_{Ei}}, \boldsymbol{K_{\rho i}}$, and $\boldsymbol{K_{vi}}$ are polynomial functions of temperature **T**.

$$K_i = \sum_{j=0}^{l_i} K_{i,j} T^j \qquad i = 1,2,3,4 \tag{20}$$

Table 3.3: Polynomial Coefficient

$\boldsymbol{K_\rho}$	$\boldsymbol{j=0}$	$\boldsymbol{j=1}$	$\boldsymbol{j=2}$	
$\boldsymbol{K_{\rho 1,j}}$	8.3636*e*2	-6.7753*e*-1	-	
$\boldsymbol{K_{\rho 2,j}}$	1.5063*e*2	-2.4202*e*-1	-	
$\boldsymbol{K_{\rho 3,j}}$	1.7784*e*-1	1.4640*e*-3	1.5402*e*-5	
$\boldsymbol{K_{\rho 4,j}}$	7.8109*e*-1	-8.1893*e*-4	-	
$\boldsymbol{K_E}$	$\boldsymbol{j=0}$	$\boldsymbol{j=1}$	$\boldsymbol{j=2}$	
$\boldsymbol{K_{E1,j}}$	1.7356*e*3	-1.0908*e*1	2.2976*e*-2	
$\boldsymbol{K_{E2,j}}$	7.5540*e*1	-	-	
$\boldsymbol{K_{E3,j}}$	1.5050	-3.7603*e*-3	-	
$\boldsymbol{K_{E4,j}}$	9.4448*e*-1	3.9441*e*-4	-	
$\boldsymbol{K_v}$	$\boldsymbol{j=0}$	$\boldsymbol{j=1}$	$\boldsymbol{j=2}$	$\boldsymbol{j=3}$
$\boldsymbol{K_{v1.j}}$	6.4812	-1.5847*e*-1	1.6342*e*-3	-6.0334*e*-6
$\boldsymbol{K_{v2.j}}$	4.0435*e*-4	-2.3118*e*-6	-	-
$\boldsymbol{K_{v3.j}}$	1.4346	-6.2288*e*-3	3.3500*e*-5	-

Source: Indraguna, 2006

Table 2 shows the numerical coefficients that are appeared with its SI units where pressure $[p] = bar$, temperature $[T] = ℃$, density $[\rho_l] = \frac{kg}{m^3}$, bulk modulus $[E_l] = MPa$ and kinetic viscosity $[v_l] = mm^2/\mathbf{s}$. The celerity of the air free oil is evaluate by $c_l = \sqrt{E_l/\rho_l}$. It has been estimated that the examined range of temperature and pressure for the maximum deviation between experimental and analytical values for density, bulk modulus, celerity and kinematic viscosity are $\pm\mathbf{0.2}\%, \pm\mathbf{1.2}\%, \pm\mathbf{0.6}\%$ and $\pm\mathbf{18}\%$ respectively.

3.3.5 Electromechanical Model

The electromechanical actuator is a model that drives the control valve must be realized to work out the net mechanical force applied by the solenoid in it s armature. The principle of energy conservation to the armature-coil is used to find the magnetic force applied by the solenoid on the armature $\boldsymbol{F_{Ea}}$.

$$V\ I\ dt = F_{Ea} dx_a + dW_m \tag{21}$$

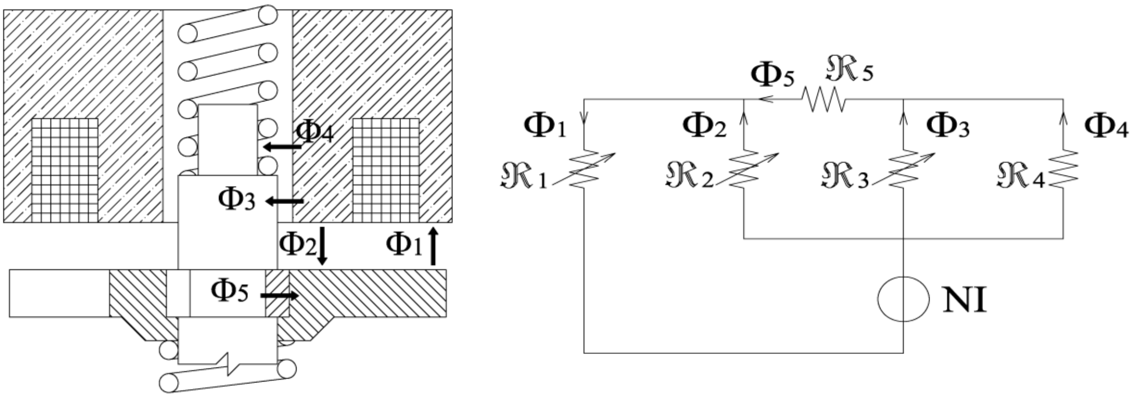
$\boldsymbol{V\ I\ dt}$is the input of electrical energy.While $\boldsymbol{F_{Ea} dx_a}$ and $\boldsymbol{dW_m}$ are the mechanical work done on the armature and the change in the magnetic energy.

Voltage **V** may be expressed in terms of flux linkage$\left(\boldsymbol{N\ \frac{d\Phi}{dt}}\right)$, so from Faraday's law voltage the equation can be expressed as

$$N\ I\ d\Phi = F_{Ea} dx_a + dW_m \tag{22}$$

$\boldsymbol{\Phi}$and$\boldsymbol{x_a}$ can be consider as independent variables (Chai, 1998; Nasar, 1995). The equation 22 can be simplified as

$$F_{Ea} = -\left.\frac{\partial W_m}{\partial x_a}\right|_{\Phi} \tag{23}$$



a) Magnetic path

b) Magnetic equivalent cicuit

Figure 3.3 : Magnetic Model Skecth

Source: Dongiovanni and Coppo, 2007

Figure 3.6a shows the path of significant magnetic flux, having neglected secondary leakage fluxes and flux fringing. To evaluate the magnetic energy stored in the gap, the magnetic circuit geometry of the control valve need to be analysed. Figure 3.6b shows the magnetic equivalent circuit obtain by exploiting the analogy between Ohm's and Hopkinson's law where $\boldsymbol{NI}$ is the ampere-turns in the exciting coil and$\boldsymbol{\Re}_j$ $(j = 1,..,5)$ are the magnetic reluctances.

The value of the ***j*-th** reluctance can be obtained by:

$$\Re_j = \frac{l_j}{\mu_o A_{aj}} \quad (j = 1,2) \tag{24}$$

The unknown $\boldsymbol{A_a}$ is equal to magnetic fluxs that acrossed the cross section area. While ***l*** is the path length.

When the flux flows across a radial path, the reluctance can be evaluated as

$$\Re_j = \frac{1}{2\pi\mu_o t_j}\left(\frac{d_e}{d_i}\right)_j \; (j = 3,4,5) \tag{25}$$

The variables of $\boldsymbol{t}$, $\boldsymbol{d_e}$ and $\boldsymbol{d_i}$ represent the radial thickness, external diameter of the volume and the internal diameter of the volume respectively.

Reluctance of the ferromagnetic components was neglected because it is several of magnitude lower than the corresponding gap reluctance. The equivalent circuit reluctance connected to the magnetomotive force generator is determined as

$$\Re = \Re_1 + \frac{\Re_2\Re_3\Re_4 + \Re_2\Re_5(\Re_3 + \Re_4)}{\Re_3\Re_4 + (\Re_2 + \Re_5)(\Re_3 + \Re_4)} \tag{26}$$

The portion of energy stored in the gap between control-valve body and magnetic core depends on the armature lift $\boldsymbol{x_a}$ while the magnetic energy $\boldsymbol{W_m}$ is stored in the volume of the electromechanical actuator. The magnetization curve of non-ferromagnetic materials (oils in the gaps) linear, equation 23 above can be simply written as

$$F_{Ea} = -\frac{1}{2}\Phi^2\frac{d\Re}{dx_a} = -\frac{1}{2}\left(\frac{N\,I}{\Re}\right)^2\frac{d\Re}{dx_a} \tag{27}$$

It is important to notice the saturation phenomenon occurred in every ferromagnetic material. The materials present a high magnetic flux density and caused the magnetic flux not to be increased. This is after the curve $\boldsymbol{B - H}$ almost at flat. The magnetization curve is given by the following equation.

$$B = \begin{cases} \mu H & H < H^* \\ \mu H^* + \mu_0(H - H^*) & H \geq H^* \end{cases} \tag{28}$$

The maximum force of attraction is limited as the maximum magnetic flux is approximately

$$\Phi_{Mj} \approx \mu H_j^* A_j \tag{29}$$

being$\boldsymbol{\mu_0}$ negligible with respect to $\boldsymbol{\mu}$**.** The maximum magnetic flux can be determined in the ***j*-th** branch of the circuit.

The model is to evaluate the inductance of the solenoid that result in the evaluation of an inductance of $134\mu H$ where the valve actuator in a closed position. $5kHz$sinusoidal wave generator frequency is high to make the mechanical system movements is negligible. Hence, the measured inductance value is $137\mu H.$ It is described that the measurement of the injector coil inductance $\boldsymbol{L = N/\Re^2}$ could be used to evaluate the control valve lift indirectly. According to Ohm's Law the inductance ***L*** could be examined as in equation 30. In calculating the value of ***L***, there are only the values of solenoid current, ***I*** and voltage, ***V*** are needed.

$$L = \frac{\int (V - RI)dt}{I} \tag{30}$$

3.3.6 Mechanical Model

The mechanical devices that can move during injector functioning like needle, control piston and control valve are modelled using the conventional mass-spring-damper scheme that is governed by mechanical equilibrium equation.

$$m_j \frac{d^2 x_j}{dt^2} + \overline{\beta_J} \frac{dx_j}{dt} + \overline{k_J} x_j + \overline{F_{oJ}} = F_j \tag{31}$$

The variables $\boldsymbol{m_j}$, $\boldsymbol{\overline{\beta_J}}$, $\boldsymbol{\overline{k_J}}$ and $\boldsymbol{\overline{F_{oJ}}}$ represent the displacing mass, the damping coefficient, the spring stiffness and the spring preload respectively.

In the common rail injection system, the high working pressures stress its components and cause an appreciable deformation. The nozzle is modelled by means of a conventional mass spring damper scheme. While the injector body is modelled by means of a simple spring having appropriate stiffness. Three degree of freedom mechanical system are formed by injector needle, control piston and needle. Three equilibrium equations are needed to describe the system motion.

$$F_p = -F_c + p_T S_p + F_{R(Pn)} - F_{R(Pb)}$$

$$F_n = -p_T S_n - F_{R(Pn)} + F_S + F_{R(nN)}$$

$$F_N = -F_S + p_e S_n - F_{R(nN)} \tag{32}$$

$$F_c = p_{uA} S'_P + p_{dz}(S_P - S'_P) \tag{33}$$

And

$$F_s = p_S S_S + p_D S_D + [\gamma\, p_s + (1-\gamma) p_D](S_n - S_D - S_S) \tag{34}$$

When $\gamma = 0$, its indicates that the nozzles is closed and when $\gamma = 1$, its indicates that the nozzles is opened.

For control piston

$$\begin{array}{llll} x_p < X_{MP} - l_p & \overline{\beta_P} = \beta_P & \overline{k_p} = 0 & \overline{F_{OP}} = 0 \\ X_p - l_p \le x_p & \overline{\beta_p} = \beta_b + \beta_P & \overline{k_p} = k_b & \overline{F_{OP}} = -k_b(X_{MP} - l_p) \end{array} \tag{35}$$

For needle

$$\begin{array}{llll} x_n - X_N < 0 & \overline{\beta_n} = \beta_b + \beta_n & \overline{k_n} = k_b + k_n & \overline{F_{On}} = F_{On} \\ 0 \le x_n - x_N < X_{Mn} - l_n & \overline{\beta_n} = \beta_n & \overline{k_n} = k_n & \overline{F_{On}} = F_{On} \\ X_{Mn} - l_n \le x_n - x_N & \overline{\beta_n} = \beta_b + \beta_n & \overline{k_n} = k_b + k_n & \overline{F_{On}} = F_{On} - k_b X_{Mn} \end{array} \tag{36}$$

For nozzle

$$
\begin{array}{llll}
x_n - x_N < 0 & \overline{\beta_N} = \beta_b + \beta_N & \overline{k_N} = k_b + k_N & \overline{F_{ON}} = 0 \\
0 \leq x_n - x_N & \overline{\beta_N} = \beta_N & \overline{k_N} = k_N & \overline{F_{ON}} = 0
\end{array} \quad (37)
$$

We can divide the control valve in two mobile parts; the pin element having mass m_c and the armature element of mass, $\boldsymbol{m_a}$. Two degree of freedom mechanical system are form where $j = a$ indicates the armature and $j = c$ the control pin. The external force $\boldsymbol{F_j}$ can be evaluated as

$$F_a = F_{Ea} - F_{R(ca)}$$

$$F_c = (p_{dA} - p_T)S_c + F_{R(ca)} + F_{R(cb)} \quad (38)$$

$\boldsymbol{F_{Ea}}$variable is the electromagnetic action that the current generates when it flows in the solenoid coil.

Damping coefficient $\overline{\boldsymbol{\beta_J}}$**,** stiffness $\overline{\boldsymbol{k_J}}$, and preload $\overline{\boldsymbol{F_{OJ}}}$ are evaluated as follow:

Pin element

$$
\begin{array}{llll}
x_c < 0 & \overline{\beta_c} = \beta_b + \beta_c & \overline{k_c} = k_b + k_c & \overline{F_{Oc}} = F_{Oc} \\
0 < x_c < X_{Mc} - l_c & \overline{\beta_c} = \beta_c & \overline{k_c} = k_c & \overline{F_{Oc}} = F_{Oc} \\
X_{MC} - l_c \leq x_c & \overline{\beta_c} = \beta_b + \beta_c & \overline{k_c} = k_c & \overline{F_{Oc}} = F_{Oc} - k_b(X_{Mc} - l_c)
\end{array} \quad (39)
$$

Armature

$$
\begin{array}{llll}
l_{MC} - X_{Mc} + x_c \geq x_a & \overline{\beta_a} = \beta_b + \beta_c & \overline{k_a} = k_a & \overline{F_{Oa}} = F_{Oa} \\
x_a > l_{Mc} - X_{Mc} + x_c & \overline{\beta_a} = \beta_b + \beta_a & \overline{k_a} = k_b + k_a & \overline{F_{Oa}} = F_{Oa} - k_b(l_{Mc} - X_{Mc} + x_c)
\end{array} \quad (40)
$$

3.4 Model Development

The model of the bosch HDEV-5 injector was implemented in a Matlab Simulink environment based on the equation presented in sub-chapter 3.3. The implementation follows the basic structure of the mathematical model that consists of three blocks namely as hydraulic, electromagnetic and mechanical.

Using this block as a structure for MatLab Simulink model, it allows observation of inputs and output of individuals blocks, which helps in the debugging stage of the program development. It also allows independent evaluation and modification of individual blocks without changing the structure of whole model (X.T Tran, B. Milton, T. White, M. Tordon, 2003). With a strategic placed scope it allows tracing of individual signal and thus gives a valuable graphical insight into the working of the bosch HDEV-5.

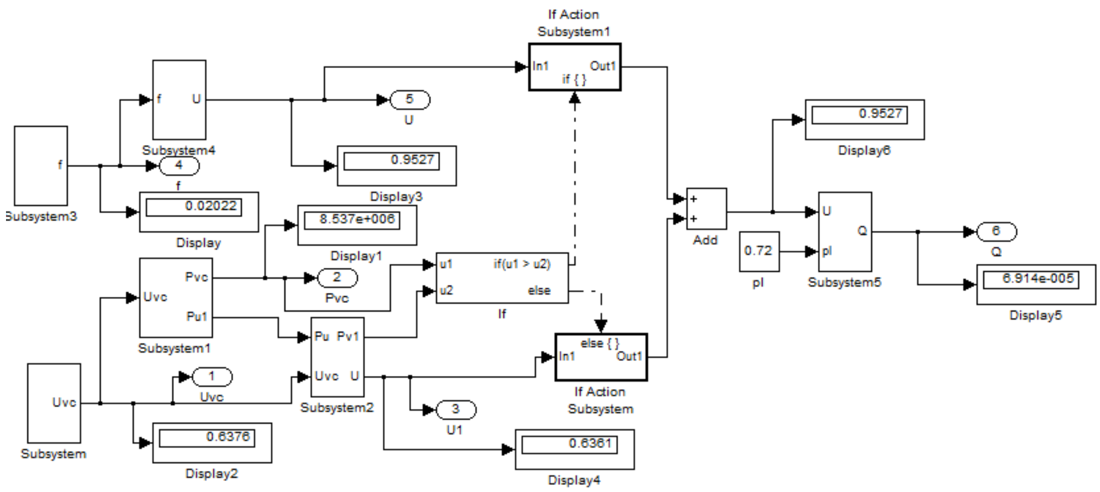


Figure 3.4: Model for flow rate prediction at different r/d ratio

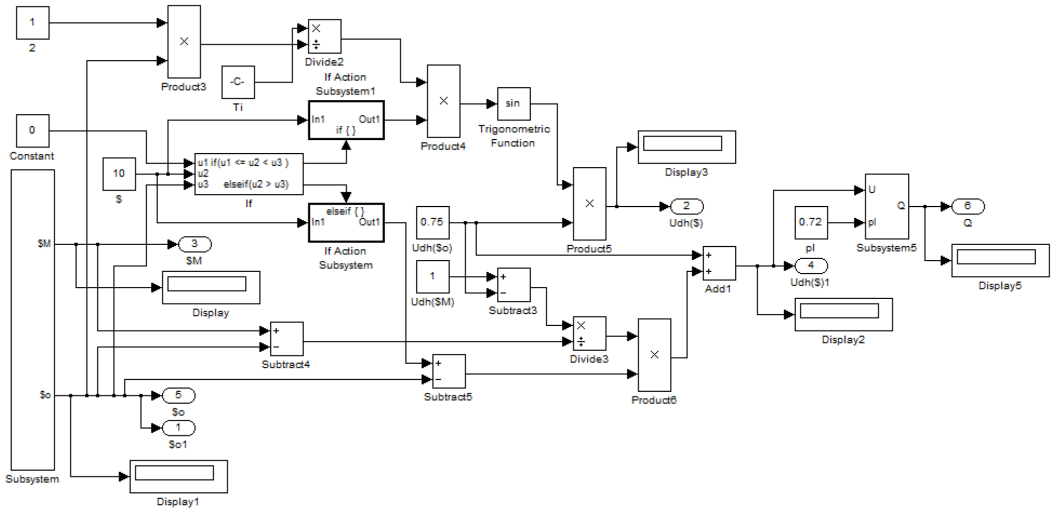


Figure 3.5: Model for flow rate prediction at different needle-seat relative displacement

3.5 Problem setup/Case study

The control system must be program with very accurate information relating to the setting that is required for each condition under which the injector expected to operate in order to achieve precise control of an injector. The engine map can be used to achieve possible engine performance to accommodate suitable air and fuel flow.

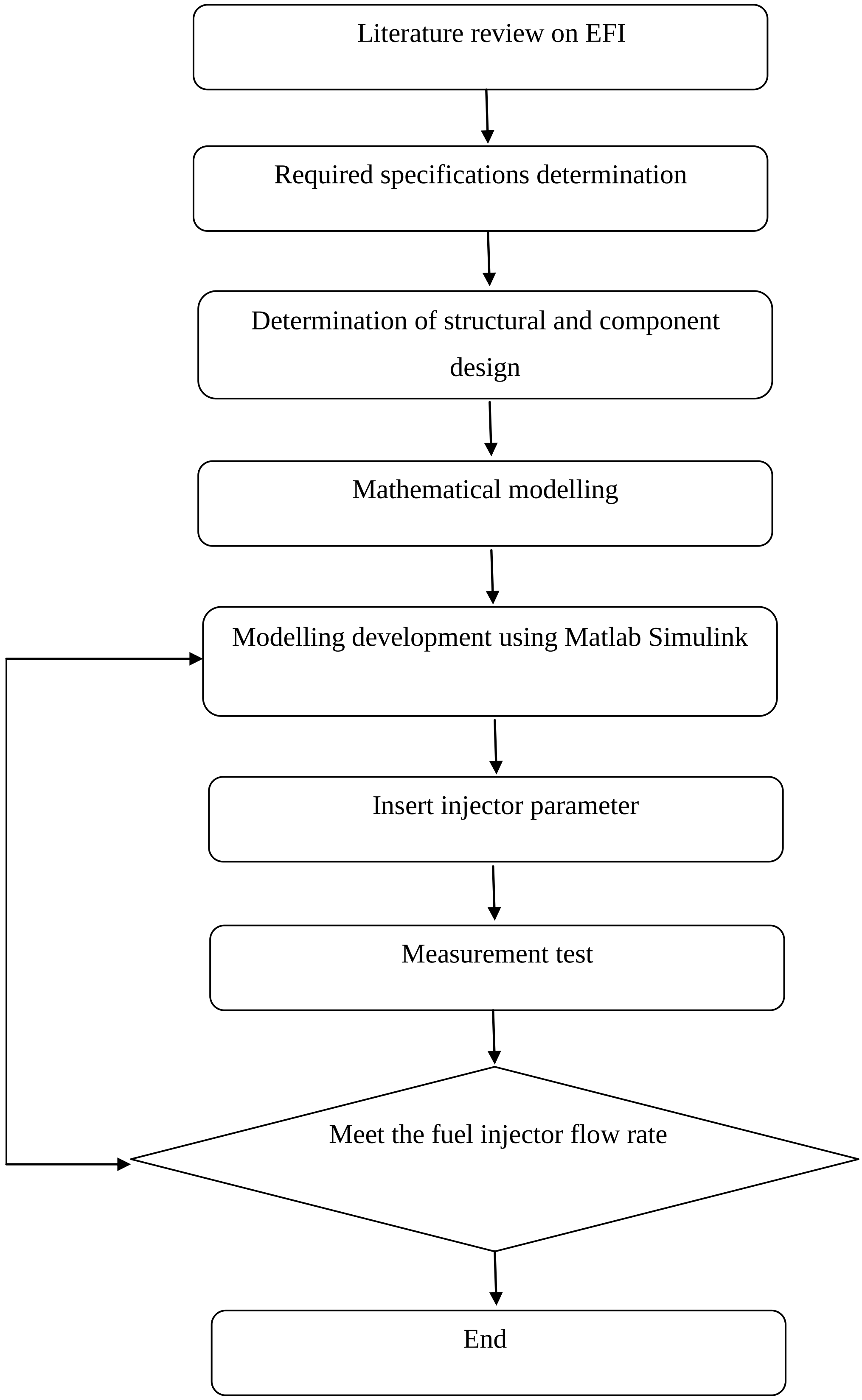
3.5.1 Testing procedures one

1. Design a fuel injector based on HDEV 5 using Matlab Simulink for a single cylinder 4-stroke engine.
2. After the MatLab Simulink design of injector is completely build up, set up the r/d ratio at 0.2 (absence of cavitation)
3. Then, the injection pressure of the fuel injector is set to be at 0-200 bar.
4. When all the parameters have been set up, run the simulation to get the result.
5. Finally, plot the graph result on the injection fuel flow rate corresponding to the injection pressure of the fuel injector.
6. The experiment testing procedures (2-5) is repeated by changing the r/d ratio at 0.02 (cavitation occurs).

3.5.2 Testing procedures two

1. Design a fuel injector based on HDEV 5 using MatLab Simulink for a single cylinder 4-stroke engine.
2. After the MatLab Simulink design of injector is completely build up, set up the r/d ratio at 0.2 and rail pressure at 100 bar.
3. Then, the needle-seat relative displacement of the injector is set to be at 0-300 um.
4. When all the parameters have been set up, run the simulation to get the result.
5. Finally, plot the graph result on the injection fuel flow rate corresponding to the injection pressure of the fuel injector.
6. The experiment testing procedures (2-5) is repeated by changing the rail pressure at 200 bar.

3.6 Injector Flowchart Design



CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 Introduction

This chapter present the result and discussion on Matlab Simulink analysis. The result discuss are the injection fuel flow rate and the discharge coefficient based on the engine operating condition. The injector model operated at a different injection pressure, different radius and diameter ratio and different needle-seat relative displacement. The results were compared between gaseous fuel predictions with liquid fuel properties.

4.2 Flow rate of different r/d ratio

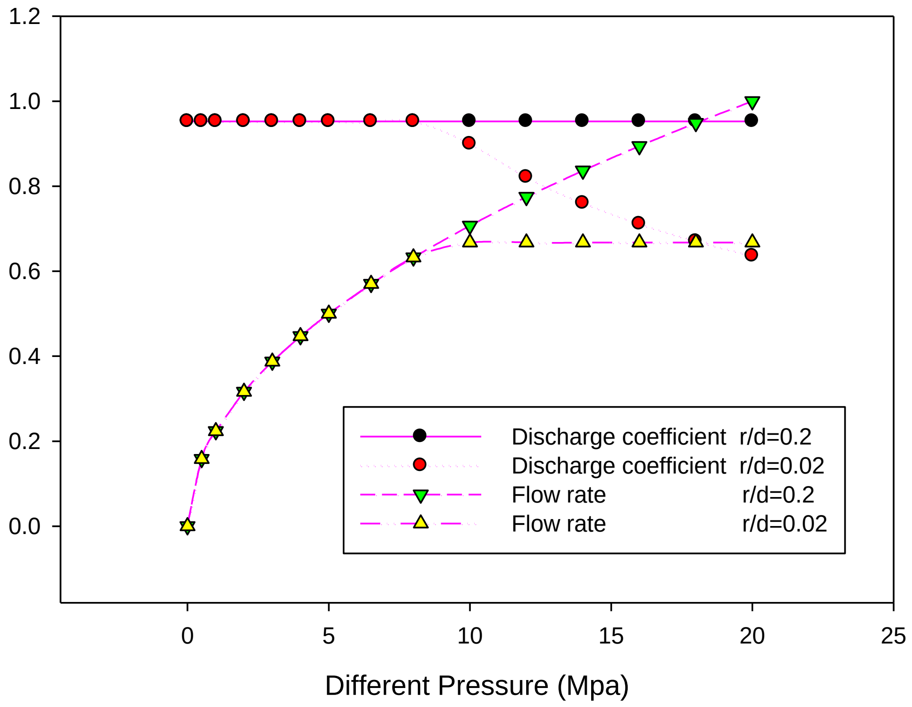


Figure 4.1: Graph of flow rate prediction at different r/d ratio

Fig. 4.1 shows two trends of predicted flow rate $\left({}^{Q}/_{Q_o} \right)$ in functions of pressure drop (Δp) through the holes with the same diameter and length, but characterized by two different values of the r/d ratio.

In absence of cavitation (r/d = 0.2), the relation between flow rate and pressure drop are monotonic, while if cavitation occur (r/d = 0.02), the hole experiences a constant after the cavitation occur starting at 10 Mpa.

The cavitation strongly effect on the injector performance. Although the pressure drop further increased, the flow rate is decreased. It is affected by the discharge coefficient that decreased after the pressure reached 10 Mpa.

In order to design a good injector model, it was necessary to study the geometrical profile of the control volume holes such as r/d ratio and l/d ratio of the injector to avoid cavitation.

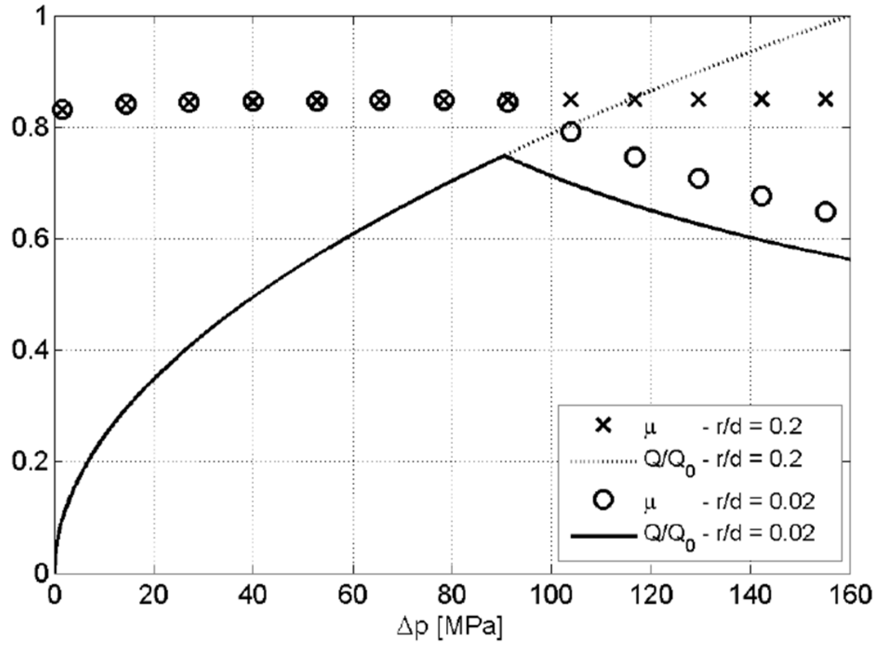


Figure 4.2: Graph of flow rate predictions at different r/d ratio (Diesel Fuel)

Fig 4.2 shows two trends of predicted flow rate $\left({}^{Q}/_{Q_o}\right)$ in a function of pressure drop (Δp)on different r/d ratio taken in the thesis of accurate modelling of injector for common rail system. In this thesis used Diesel as a fuel to be analysed.

This graph is a bit different from the graph measured fuel flow rate that used CNG as a fuel. It's because of the density of the fuel used in the experiment is based on the constant value of 0.72 ${}^{kg}/_{m^3}$ while in the thesis, the density of diesel fuel is varies depends on the pressure inside the injector.

4.3 Flow rate of different needle-seat relative displacement

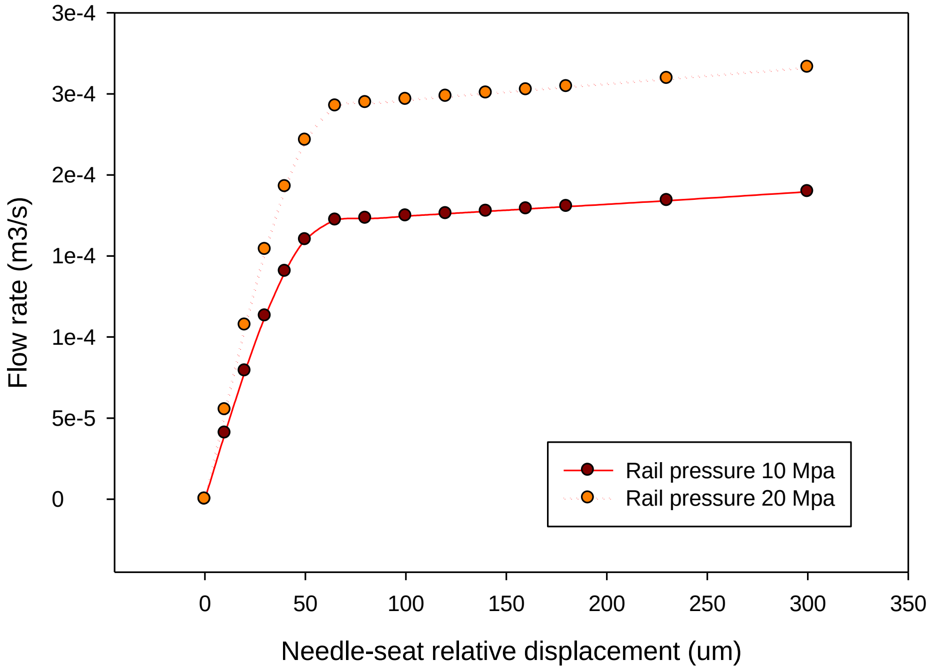


Figure 4.3: Graph of flow rate predictions at different needle-seat relative displacement

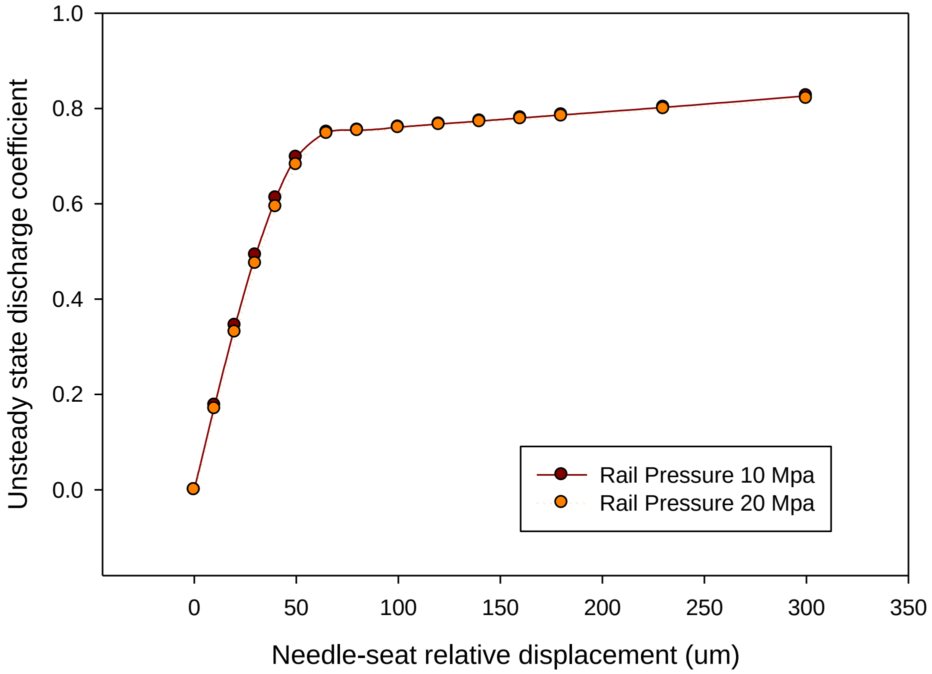


Figure 4.4: Graph of measured trends of unsteady nozzle hole discharge coefficient

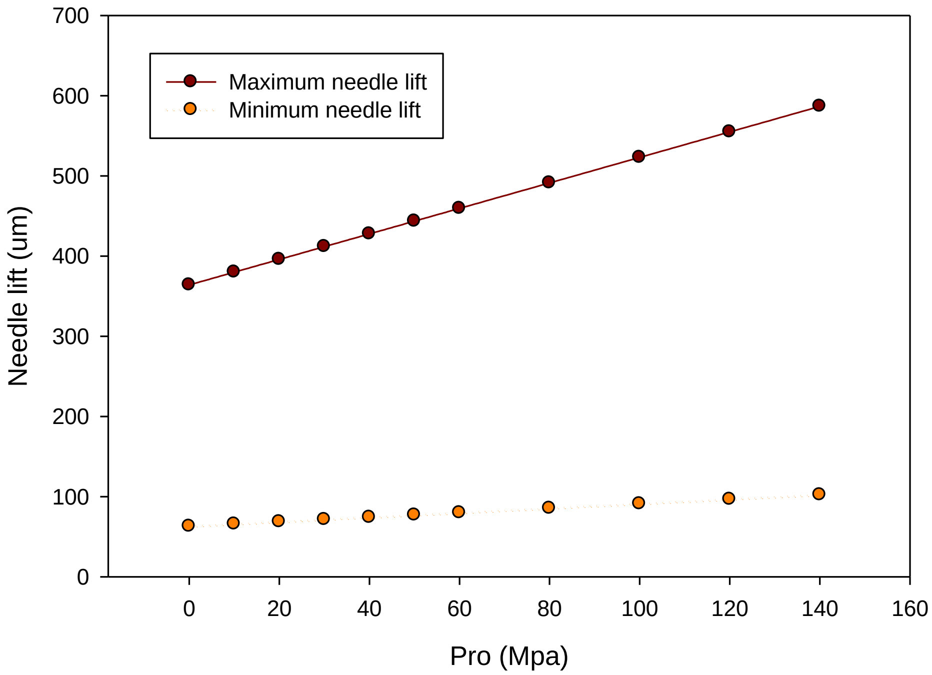


Figure 4.5: Graph of effect of pressure on the max moving element lift and the transition value of ξ

Fig 4.3 shows the trends of unsteady flow rate versus needle lift at rail pressure of 100 bar and 200 bar. During injector opening phase, the unsteady effects are predominant and the sinusoidal-linear trend of the hole discharge coefficient is considered.

From the graph we can see the flow rate is slightly increased until it reaches the transition value of ξ. From Fig 4.5 we can see the pattern of the maximum needle lift and the transition value of ξ between the sinusoidal and the linear trend and the graph is linear depends on the rail pressure. From the graph, the transition value of ξ is at 65.8 µm for $p_{ro} = 10\ Mpa$ and 68.6 µm for $p_{ro} = 20\ Mpa$. After the transition value of ξ is reached, the flow rate is linearly increased until needle lift reaches the maximum.

From Fig 4.4 shows the measured trends of unsteady nozzle hole discharge coefficient. There is no much different of discharge coefficient in different rail pressure of $p_{ro} = 10\ Mpa$ and $p_{ro} = 20\ Mpa$. So, the flow rate during injector opening phase is not dominant depends on the discharge coefficient of the needle-seat relative displacement.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

This thesis is to study the hydraulic model part of the high pressure natural gas injector. Its focused on the flow rate of the fuel inject to the combustion chamber depends on the radius/diameter ratio and the needle relative displacement of the injector.

A previously developed numerical model of a high-pressure natural gas injector model was refined here, with the aim of obtaining accurate predictions of injector operation in its entire application field.

The model of flow through the control volume feeding and discharge holes was further detailed as it was shown to play an important role in determining the flow regime in the orifice.

5.2 Recommendation

Injector mathematical model can be divided by 3 main parts that is hydraulic, electromagnetic and mechanical model. This thesis only focus on the hydraulic model of the injector. For accurate modelling of an injector it should also consider electromagnetic and mechanical model.

As far as the hydraulic model was concerned, the electromagnetic and mechanical model should be studied in order to get an accurate modelling of high pressure natural gas injector. So, the whole operation of the injector can be determined.

In mechanical model, the way an electronic fuel injector functions is relatively simple, however the complexity of the mathematical description of the physical actions and of the equations involved is considerable.

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APPENDIX A1

Table of Flow rate of different r/d ratio

ΔP (Mpa)	r/d = 0.2			r/d = 0.02		
	μ	Q (m^3/s)	Q/Q_o	μ	Q (m^3/s)	Q/Q_o
0.0000	0.9527	0.0000	0.0000	0.9527	0.0000	0.0000
0.5000	0.9527	6.9140e-5	0.1581	0.9527	6.9140e-5	0.1581
1.0000	0.9527	9.7780e-5	0.2236	0.9527	9.7780e-5	0.2236
2.0000	0.9527	1.3828e-4	0.3162	0.9527	1.3828e-4	0.3162
3.0000	0.9527	1.6936e-4	0.3873	0.9527	1.6936e-4	0.3873
4.0000	0.9527	1.9556e-4	0.4472	0.9527	1.9556e-4	0.4472
5.0000	0.9527	2.1864e-4	0.5000	0.9527	2.1864e-4	0.5000
6.5000	0.9527	2.4929e-4	0.5701	0.9527	2.4929e-4	0.5701
8.0000	0.9527	2.7656e-4	0.6325	0.9527	2.7656e-4	0.6325
10.0000	0.9527	3.0920e-4	0.7071	0.8994	2.9190e-4	0.6675
12.0000	0.9527	3.3872e-4	0.7746	0.8211	2.9193e-4	0.6676
14.0000	0.9527	3.6585e-4	0.8366	0.7602	2.9193e-4	0.6676
16.0000	0.9527	3.9112e-4	0.8944	0.7111	2.9193e-4	0.6676
18.0000	0.9527	4.1484e-4	0.9487	0.6704	2.9192e-4	0.6676
20.0000	0.9527	4.3728e-4	1.0000	0.6360	2.9192e-4	0.6676

APPENDIX A2

Table of Flow rate of different needle-seat relative displacement

ξ (μm)	$p_{ro} = 10\ Mpa$		$p_{ro} = 20\ Mpa$	
	μ	$Q\ \left(m^3/_s\right)$	μ	$Q\ \left(m^3/_s\right)$
0	0.0000	0.0000	0.0000	0.0000
10	0.1773	4.0689e-5	0.1698	5.5109e-5
20	0.3446	7.9084e-5	0.3307	1.0733e-4
30	0.4924	1.1300e-4	0.4745	1.5400e-4
40	0.6122	1.4050e-4	0.5936	1.9266e-4
50	0.6973	1.6002e-4	0.6820	2.2135e-4
65	0.7500	1.7212e-4	0.7472	2.4251e-4
80	0.7545	1.7315e-4	0.7535	2.4455e-4
100	0.7609	1.7462e-4	0.7596	2.4653e-4
120	0.7673	1.7609e-4	0.7657	2.4851e-4
140	0.7736	1.7754e-4	0.7718	2.5049e-4
160	0.7800	1.7901e-4	0.7779	2.5247e-4
180	0.7864	1.8047e-4	0.7840	2.5445e-4
230	0.8023	1.8412e-4	0.7993	2.5942e-4
300	0.8264	1.8965e-4	0.8207	2.6636e-4