FINAL YEAR PROJECT REPORT

Modeling the performance characteristics of hydrogen engine with fuel injection systems using GT-Power

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2009

SUPERVISOR'S DECLARATION

I hereby declare that I have checked this project and in my opinion, this project is adequate in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering.

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Date:

STUDENT'S DECLARATION

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

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ACKNOWLEDGEMENTS

I am grateful and would like to express my sincere gratitude to my supervisor and co-supervisor Prof Dr Rosli bin Abu Bakar for his germinal ideas, invaluable guidance, continuous encouragement and constant support in making this research possible. He has always impressed me with his outstanding professional conduct, his strong conviction for science, and his belief that a degree program is only a start of a life-long learning experience. I appreciate his consistent support from the first day I applied to graduate program to these concluding moments. I am truly grateful for his progressive vision about my training in science, his tolerance of my naïve mistakes, and his commitment to my future career. I also sincerely thanks for the time spent proof reading and correcting my many mistakes.

I acknowledge my sincere indebtedness and gratitude to my parents for their love, dream and sacrifice throughout my life. I cannot find the appropriate words that could properly describe my appreciation for their devotion, support and faith in my ability to attain my goals. Special thanks should be given to my committee members. I would like to acknowledge their comments and suggestions, which was crucial for the successful completion of this study.

ABSTRACT

This study was addressed the effect of speed on engine performance for 1-cylinder for port and direct injection fuel systems. GT- power utilized to develop the model for port injection and direct injection hydrogen fuel systems. This port injection was installed before the cylinder head for port injection and injection timing are 74.01 g/sec at 5 º before top dead center for direct injection hydrogen fueled system and inducted with low pressure about 1bar – 2bar for port injection while 80bar for direct injection.
Air-fuel ratio was varied from rich limit (AFR=22.88) to a lean limit (AFR=68.66) when the engine speed constant at 3000 rpm. The rotational engine speed was varied from 1000 to 6000 rpm when the air-fuel ratio constant at 34.33 at stoichiometric condition. The obtained results seen that the engine speed and air-fuel ratio are greatly influence on the Brake Mean Effective Pressure (BMEP), Brake Specific Fuel Consumption (BSFC). It also seen that the decreases of the BMEP with increase of the engine speed, however, increase the brake specific fuel consumption (BSFC). The optimum minimum value of BSFC occurred within a range AFR from 38.14(φ=0.9 to 42.91 (φ= 0.8) for selected range of speed. The higher volumetric efficiency emphasizes that the direct injection hydrogen fuel system is a strong method solution to solve the problem of the low volumetric efficiencies of hydrogen engine. Maximum brake torque for hydrogen engine occurs at lower speed compared with gasoline. The present contribute suggests that the direct injection hydrogen fuel supply system as a strong method for solving the power, torque and abnormal combustion problems.

ABSTRAK

Kita mempelajari tentang prestasi engine dengan menggunakan pencucuk minyak terus dan pencucuk minyak tidak terus pada satu enjin silinder. Kite boleh membuat model satu engine silinder dengan menggunakan isian GT-POWER. Pencucuk minyak tidak terus dipasangkan sebelem kepala silinder dan tekanannya pada 1-2 bar manakala pencucuk minyak terus dipasangkan pada kepala silinder pada tekanan 80 bar dan masa minyak dimasukkan ke kepala silinder adalah 74.01 g/s pada 5 º sebelum TDC. Kadar minyak udara adalah dari kandungan kaya minyak (AFR=22.88) kepada kekurangan minyak (AFR=68.66) pada kelajuan enjin 3000 rpm. Kelajuan enjin adalah dari kelajuan 1000 to 6000 rpm apabila kadar minyak udara pada kadar yang tetap pada 34.33 iaitu pada kandungan udara cukup-cukup untuk membakar minyak dalam silinder enjin. Daripada keputusan perisian GT-Power, kandungan minyak udara dalam enjin mempengaruhi tekanan berkesan brek (BMEP) dan anggaran minyak specific brek (BSFC). Dapat diperhatikan dari perisisan GT-Power mendapati tekanan berkesan brek (BMEP) apabila kelajuan enjin meningkat, walaubagaimanapun, anggaran minyak specific brek (BSFC) menurun. Nilai maksimum dan minimum anggaran minyak specific brek (BSFC) berlaku pada jarak kandungan minyak udara AFR dari (φ=0.9) ke 42.91 (φ= 0.8) pada kelajuan enjin 3000 rpm. Keberkesanan isipadu didapati dari pencucuk minyak terus adalah cara yang paling bagus untuk selesaikan masalah kekurangan keberkesanan isipadu dalam enjin hydrogen. Maksimum torque brek untuk enjin hydrogen berlaku pada kelajuan enjin yang rendah berbanding enjin gasoline. Pada pendapat saya, pencucuk minyak terus adalah cara untuk menyelesaikan masalah kuasa, torque dan ketidaknormalan pembakaran dalam enjin.

TABLE OF CONTENT

LIST OF TABLES

LIST OF FIGURES

LIST OF ABREVIATIONS

BSFC	Brake Specific Fuel Consumption
BMEP	Brake Mean Effective Pressure
AFR	Air Fuel ratio
BP	Brake Power
DI	Direct Injection
PI	Port Injection

Chapter 1

Introduction

1.1 Project

Hydrogen is the chemical element with atomic number one. Hydrogen is in group one in chemical table reaction. Hydrogen is a colorless, odorless, tasteless, flammable and nontoxic gas at atmospheric temperatures and pressure. It is the most abundant element in the universe but is almost absent from the atmosphere as individual molecules in the upper atmosphere can gain high velocity during collision with heavier molecules, and becomes ejected from the atmosphere. Hydrogen does not exist naturally and so it must be extracted from other chemicals such as electrolysis of water, generation by partial oxidation of coal or hydrocarbon, recovery by product hydrogen from electrolytic cells used to produce chlorine and other products and dissociation of ammonia. Hydrogen has a big amount of energy for transportation and the substantially cleaner emission compare other internal combustion engine fueled. Hydrogen fueled has wide flammability limits and the high flame propagation speed, both allowing better efficiency. Besides, Flammability limits (volume % in air), auto ignition temperature in air (K) Flame velocity (m s−1)b, stoichiometric volume fraction %, Heat of combustion (MJ/kgair)b is higher compare other internal combustion engine fueled. On the other hand, we prefer fundamental of the hydrogen is vapor or gas in single cylinder engine with port injection hydrogen fueled compared liquid gaseous. In order to produce hydrogen liquid, we need a big power plant. The hydrogen gaseous needs to compress to produced liquid hydrogen but the energy will lost to 30-40% as a consequences. In addition, we need to modified the fueled tank storage and too expensive to using liquid hydrogen as a alternate fueled. This the disadvantages about the liquid hydrogen and that’s why we choose hydrogen gaseous as the alternate fuel. In addition, we choose port injection and direct injection. In this study the difference between two injection system at single cylinder engine using GT-Power.

1.2 Problem statement

Fossil fuels such as petroleum, natural gas and coal meet most of the world's energy demand, at present time. But combustion produce of these fossil fuels, such as carbon monoxide(CO), carbon dioxide (CO2), oxides of sulfur (SO_x), oxides of nitrogen (NO_x), hydrocarbon (HC), toxic metals which gave green house emission. In order to reduce the world pollution to zero the engineer found new alternative fuel called hydrogen to replace fossil fuel. The combustion of hydrogen in fuel engine is still under research and didn't practical in transportation because there still have some problem which has to solve to make dream come true. There some problem is still on research on how to solve the problem.

The problem is when hydrogen combust in engine the hydrogen was undergo abnormal combustion. One of the abnormal combustion is pre-ignition also called backfire. Pre-ignition is defined as combustion prior to spark discharge, and in general, results from surface ignition at engine hot spots, such as spark electrodes, valves or engine deposits. The minimum ignition energies of hydrogen–air mixtures will be much lower than propane–air and heptanes–air mixtures. The minimum ignition energy will lead the engine failure run because the heat was release in rapid pressure rise and higher heat rejection in cylinder.

On the other hand, knocking also known as abnormal combustion. When knocking occurs, there is a rapid release of the chemical energy in the remaining unburned mixture, causing high local pressures and generating propagating pressure waves in amplitude of several bars across the combustion chamber. The amplitude pressures in cylinder combustion chamber lead the engine failure.

Furthermore, the high combustion pressure is one of the problems. This occur when the hydrogen undergo pre-ignition and combustion knocking because the chemical release heat rate exceed the higher heat rejection that leads to higher in-cylinder surface temperatures and the engine will failure.

1.3 Objective

This hydrogen fuel injection port engine objectives are :-

i. Develop a model for H_2 ICE with direct and port injection fuel systems
ii. Compare between two delivery system direct and port injection
iii. Accessing the performance characteristics of each system

1.4 Scopes

The aim is analyzing the combustion in port injection hydrogen engine :

i. Developing a simulation model suitable for optimization and parameter studies.
ii. The model will be suitable for normal engine operation.
iii. The attention will be limited to the power cycle, i.e. the part of the engine cycle from intake valve closing time to exhaust valve opening time.

Chapter 2

Literature review

2.1 Introduction

This chapter discusses hydrogen abnormal combustion, hydrogen properties, hydrogen delivery system and some of related hydrogen fueled engine with port fuel injection. It also describes, summarize, evaluate and clarify the research that has been done for the benefit of this project. The aim of this research is to study the advantages and disadvantages of using hydrogen as alternate fueled replace gasoline using single cylinder gasoline engine besides verify the model experimentally.

2.2 Related work

Sixty years later, during his work with combustion engines in the 1860s and 1870s, N. A. Otto (the inventor of the Otto cycle) reportedly used a synthetic producer gas for fuel, which probably had a hydrogen content of over 50%. Otto also experimented with gasoline, but found it dangerous to work with, prompting him to return to using gaseous fuels. The development of the carburetor, however, initiated a new era in which gasoline could be used both practically and safely, and interest in other fuels subsided.

Hydrogen has since been used extensively in the space pro-gram since it has the best energy-to-weight ratio of any fuel. Liquid hydrogen is the fuel of choice for rocket engines, and has been utilized in the upper stages of launch vehicles on many space missions including the Apollo missions to the moon, Skylab, the Viking missions to Mars and the Voyager mission to Saturn. However the human awakens about the usage hydrogen as a alternate fueled is one of the invention internal combustion engine. Hydrogen was first suggested as an alternative fuel for internal combustion engines in the 1920s [1]. However, one problem with homogeneous charge hydrogen fuelled internal combustion engines is that they sometimes suffer from flash back phenomenon during the intake stroke, especially during high load operations Nowadays, many engineer has research and develop new technology in order to replace gasoline fueled to hydrogen fueled.

April 1994

Department of Mechanical Engineering, Seoul National University, Seoul describes the experimental results on a hydrogen fueled single cylinder engine to study the characteristics of a solenoid-driven intake port injection type hydrogen injection valve. The fuel-ail equivalence ratio was varied from the lean limit at which stable operation was guaranteed to the rich limit at which flash-back occurred and spark timing was also changed. As a consequence, a hydrogen intake port injection system can be easily installed on a spark ignition engine only with simple modification and the flow rate of hydrogen supplied can also be controlled conveniently. Furthermore, department of manufacturing engineering, City University of Hong Kong and department of energy engineering, zhejiang university, hangzhou has study about hydrogen supply system and combustion system to solve such problem abnormal combustion in hydrogen fueled engine. A fast response solenoid valve, which posses good switch characteristics and very fast response and its electronic control system are described. A high pressure hydrogen injector is designed to improved hydrogen jet penetration and mixture formation in the combustion chamber and to prevent backfire occurring in the hydrogen supply system pipe between the fast response valve and the combustion chamber. Besides they study about abnormal combustion such as backfire, pre-ignition high pressure and injection timing optimally.

1998

The characteristics of cryogenic hydrogen, such as high density and considerable cooling effect, favor the fuel injection, the mixing process and thus the combustion process. In addition to the preferred use of liquid hydrogen due to its range per tank filling and low amount of mass for storage in the vehicle, the cryogenic characteristics of hydrogen provide significant advantages. In addition to engine operation with external mixture formation, considerable success was achieved with internal mixture formation with injection of cryogenic high pressure hydrogen.

June, 2002

Center for Environmental Research and Technology, College of Engineering, University of California have been run experiment about NOx emission and performance data for a hydrogen fueled internal combustion engine at 1500 rpm using exhaust gas recirculation. This experiment including six experiments conducted on a 2-liter, 4-cylinder Ford ZETEC internal combustion engine developed to operate on hydrogen fuel. The experiments were conducted to ascertain the effect exhaust gas recirculation (EGR) and a standard 3-way catalytic converter had on NOx emissions and engine performance. All the experiments were conducted at a constant engine speed of 100 rpm and each experiment used a different fuel 8ow rate, ranging from 0.78 to 1:63 kg=h. These fuel 8ow rates correspond to a fuel equivalence ratio, _, ranging from0.35 to 1.02 when the engine is operated without using EGR (i.e. using excess air for dilution). The experiments initially started with the engine operating using excess air. As the experiments proceed, the excess air was replaced with exhaust gas until the engine was operating at a stoichiometric air/fuel ratio. The results of these experiments demonstrated that using EGR is an effective means to lowering NOx emissions to less than 1 ppm while also increasing engine output torque.

September, 2003

Department of Mechanical and Manufacturing Engineering, University of Calgary has discovery hydrogen abnormal combustion in engine application. They state that knock remains one of the prime limitations that need to be addressed so as to avoid its incidence and achieve superior performance. They research about hydrogen relating to the spark ignition hydrogen-fuelled engine, the e1ects of changes in key operating variables, such as compression ratio, intake temperature and spark timing on knock-limiting equivalence ratios are established both analytically as well as experimentally. The onset of knock, which is caused mainly by the auto-ignition of the unburned mixture in the "end gas region" of the charge which is at any moment yet to be burned, involves exceedingly rapid rates of combustion of the fuel–air mixture, increased heat transfer to the cylinder walls, excessively high cylinder pressure and temperature levels and increased emissions.

June, 2003

University of California, Riverside College of Engineering, Center for Environmental Research and Technology have been run experiment about NOx emission reduction in a hydrogen fueled internal combustion engine at 3000 rpm using exhaust gas recirculation. This experiment including five experiments conducted on a 2-l, 4-cylinder Ford ZETEC internal combustion engine (ICE) developed to operate on hydrogen fuel. The experiments were conducted to ascertain the effect exhaust gas recirculation (EGR) and a standard 3-way catalytic converter had on NOx emissions and engine performance. All the experiments were conducted at a constant engine speed of 3000 rpm and each experiment used a different fuel 7ow rate, ranging from 1.63 to 2:72 kg=h.

These fuel 7ow rates correspond to a fuel equivalence ratio, _, ranging from 0.35 to 0.75 when the engine is operated without using EGR (i.e. using excess air for dilution). The experiments initially started with the engine operating using excess air.

As the experiments proceed, the excess air was replaced with exhaust gas until the engine was operating at a stoichiometric air/fuel ratio. The results of these experiments demonstrated that using EGR is an effective means to lowering NOx emissions to less than 1 ppm while also increasing engine output torque

March, 2005

Department of Thermal Engineering, School of Mechanical Engineering, National Technical University of Athens has done the surveys the publications available in the literature concerning the application of the second-law of thermodynamics to internal combustion engines. A detailed reference is made to the findings of various researchers in the field over the last 40 years concerning all types of internal combustion engines, i.e. spark ignition, compression ignition (direct or indirect injection), turbocharged or naturally aspirated, during steady-state and transient operation. All of the subsystems (compressor, after cooler, inlet manifold, cylinder, exhaust manifold, turbine), are also covered.

October, 2005

A new method ingeniously adopting fuzzy-neural network (FNN) system to calculate the optimizing control laws for the optimizing control model has been designed, and a series connection control system has been set up, which is composed of FNN controllers combined with an adaptive controller for ignition timing to realize open-loop or closed-loop control pattern with stepping regulation of ignition timing.

November, 2005

Sandia National Laboratories, Combustion Research Facility have been study about low volumetric efficiencies and frequent pre-ignition combustion events, the power densities of premixed or port-fuel-injected hydrogen engines are diminished relative to gasoline-fueled engines, develop of advanced hydrogen engines with improved power densities and The ability for hydrogen to burn cleanly and operate efficiently is owed to the unique combustion characteristics of hydrogen that allow ultra-lean combustion with dramatically reduced NO*x* production and efficient low-engine load operation. Besides they undergo the examination of hydrogen properties relevant to engine operation and control to observe the hydrogen fueled at low and high engine load.

August, 2006

Program of Energy Engineering, Izmir Institute of Technology and Department of Mechanical Engineering, Izmir Institute of Technology, Turkey have been done an experimental study on performance and emission characteristics of a hydrogen fuelled spark ignition engine and investigate experimentally about the performance and emission characteristics of a conventional four cylinder spark ignition (SI) engine operated on hydrogen and gasoline. The test results have been demonstrated that power loss occurs at low speed hydrogen operation whereas high speed characteristics compete well with gasoline operation. Fast burning characteristics of hydrogen have permitted high speed engine operation. Less heat loss has occurred for hydrogen than gasoline.
July, 2007

Michigan Technological University Houghton and Czestochowa University of Technology have been done research about the Comparisons of hydrogen and gasoline combustion knock in a spark ignition engine. Their further engine studies examining combustion knock characteristics were conducted with hydrogen and gasoline fuels in a port-injected, spark-ignited, single cylinder cooperative fuel research (CFR) engine and characterization of the signals at varying levels of combustion knock from cylinder pressure and a block mounted piezoelectric accelerometer were conducted including frequency, signal intensity, and statistical attributes. Further, through the comparisons with gasoline combustion knock, it was found that knock detection techniques used for gasoline engines, can be applied to a H2-ICE with appropriate modifications. This work provides insight for further development in real time knock detection. This would help in improving reliability of hydrogen engines while allowing the engine to be operated closer to combustion knock limits to increase engine performance and reducing possibility of engine damage due to knock.

July, 2007

Engine studies examining combustion knock characteristics were conducted with hydrogen and gasoline fuels in a port-injected, spark ignited, single cylinder cooperative fuel research (CFR) engine. Characterization of the signals at varying levels of combustion knock from cylinder pressure and a block mounted piezoelectric accelerometer were conducted including frequency, signal intensity, and statistical attributes. Further, through the comparisons with gasoline combustion knock, it was found that knock detection techniques used for gasoline engines, can be applied to a H2-ICE with appropriate modifications. This work provides insight for further development in real time knock detection.This studies improving reliability of hydrogen engines while allowing the engine to be operated closer to combustion knock limits to increase engine performance and reducing possibility of engine damage due to knock.

2009

Universiti Malaysia Pahang was investigate the effect of fuel air ratio (AFR) and engine speed on performance of the single cylinder fueled port injection using GT-Power and develop computational model for port injection engine. Throughout the study, air fuel ratio was varied from stoichiometric mixture to lean. The engine speed was varied from 2500 to 4500 rpm. The result show that air fuel ratio(AFR) and engine speed were greatly influence on the performance of hydrogen fueled engine especially Brake Mean Effective Pressure(BMEP), Thermal Efficiency and brake specific fuel consumption (BSFC).

2009

University Malaysia Pahang was study the influence air fuel ratio and injection timing on the engine performance of 4-cylinder direct injection hydrogen fueled engine. The 4-cylinder direct injection hydrogen engine model was develop utilizing the GT-Power commercial software. This model represent one dimensional gas dynamics to represent the flow and heat transfer in the components of engine model.

2.3 Hydrogen properties

Wide flammability range

Compared to other fuels, hydrogen is not so fussy with its density mixture. It can ignite anywhere from a Fuel-to-Air mixture of 4 to 74 percent.

Easily ignitabily.
Hydrogen ignites easier than gasoline. This provides for efficient and prompt ignition but the drawback is hot spots within the combustion chamber can cause premature ignition.

High expansion mass.
The expanding gases formed by spent hydrogen have a much higher velocity and mass compared to gasoline.

Easily dispersed.
The ability of hydrogen to blend with air is greater than gasoline, thus forming a more uniform mixture.

Low density.
Hydrogen occupies a very large volumetric area in its gaseous state. To facilitate the storage of hydrogen gas, it has to be stored in its liquid form within high pressure tanks.

Low boiling temperature.

Liquid hydrogen cannot be mixed with other liquid fuels. Its low boiling point (-252 C) will freeze other fuels. This means a separate storage tank is needed to store liquid hydrogen.

Small quenching distance

Consequently, hydrogen flames travel closer to the cylinder wall than other fuel before they extinguish. Thus, it is more difficult to quench a hydrogen flame than a gasoline flame. The smaller quenching distance can also increase the tendency for backfire since the flame from a hydrogen-air mixture more readily passes a nearly closed intake valve, than a hydrocarbon-air flame.

Low Ignition Energy

Hydrogen has very low ignition energy. The amount of energy needed to ignite hydrogen is about one order of magnitude less than that required for gasoline. This enables hydrogen engines to ignite lean mixtures and ensures prompt ignition

High Auto ignition Temperature

The high auto ignition temperature of hydrogen allows larger compression ratios to be used in a hydrogen engine than in a hydrocarbon engine.

High Flame Speed

Hydrogen has high flame speed at stoichiometric ratios. Under these conditions, the hydrogen flame speed is nearly an order of magnitude higher (faster) than that of gasoline. This means that hydrogen engines can more closely approach the thermodynamically ideal engine cycle.

High Diffusivity

Hydrogen has very high diffusivity. This ability to disperse in air is considerably greater than gasoline and is advantageous for two main reasons.

2.4 Hydrogen delivering system

After reviewing the literature and balancing the advantages and disadvantages of the four basic types of known hydrogen fueling systems (carburetor, throttle body-injection, intake port fuel injection (PFI) and direct in-cylinder injection (DI) it was decided to observe the difference between two delivery system port injection (PI) and direct in-cylinder injection using commercial software, GT-Power.

Port injection is the most suitable choice because there no need to modified the gasoline engine. The port injection was installed before the fuel inlet to the cylinder and the pressure for inject the hydrogen fueled has a low pressure compare when the hydrogen inject directly to the cylinder engine which much higher about 80 bar compare to port injection about 2-4 bar. However, port injection gave less power output and less volumetric efficiency compare to direct in-cylinder injection. Beside, the gasoline engine need to modified and its too costly.

Furthermore we choose the port injection fuel because it has a simple modification to control the the fuel-ail equivalence ratio was varied from the lean limit at which stable operation was guaranteed to the rich limit at which flash-back occurred and spark timing was also changed called solenoid valve. As a consequence, a hydrogen intake port injection system can be easily installed on a spark ignition engine only with simple modification and the flow rate of hydrogen supplied can also be controlled conveniently.

A fast response solenoid valve which good switch characteristic and very fast response and a high pressure injector is designed to improve hydrogen jet penetration and mixture formation in the combustion chamber and to prevent backfire in hydrogen supply between the fast response valve and the combustion chamber. Besides that, the chip microprocessor is developing to control the ignition and injection timing optimally.

Otherwise, external mixture by mean of port injection also has been demonstrated to the result of the higher engine efficiency, extended lean operation, lower cyclic variation and lower NOx will produce. The most serious problem for port fuel injection is pre-ignition and back fire especially with rich mixture. However, condition with port fuel injection are much less severe and the probability for abnormal combustion is reduced because due to its imparts a better resistance to backfire.

On the other hand, direct injection is very expensive compare to port injection but gave much horsepower compare between two delivery fuel systems. Direct injection was installed at the fuel inlet to the cylinder on the spot and the pressure for inject the hydrogen fueled has a high pressure compare when using port injection to inject the hydrogen before the fuel inlet at the cylinder engine which much lower about 1-2 bar.

According to scientists, port injection and direct injection have very similar characteristics but volumetric efficiency for direct in-cylinder more higher compare to port injection and the fuel convert to mechanical energy high. In addition, direct injection was inject hydrogen into the cylinder head at high pressure around 80bar, so the engine didn’t need to compress hydrogen fuel in internal combustion engine and reduce the work done by engine gave higher power output consequently.

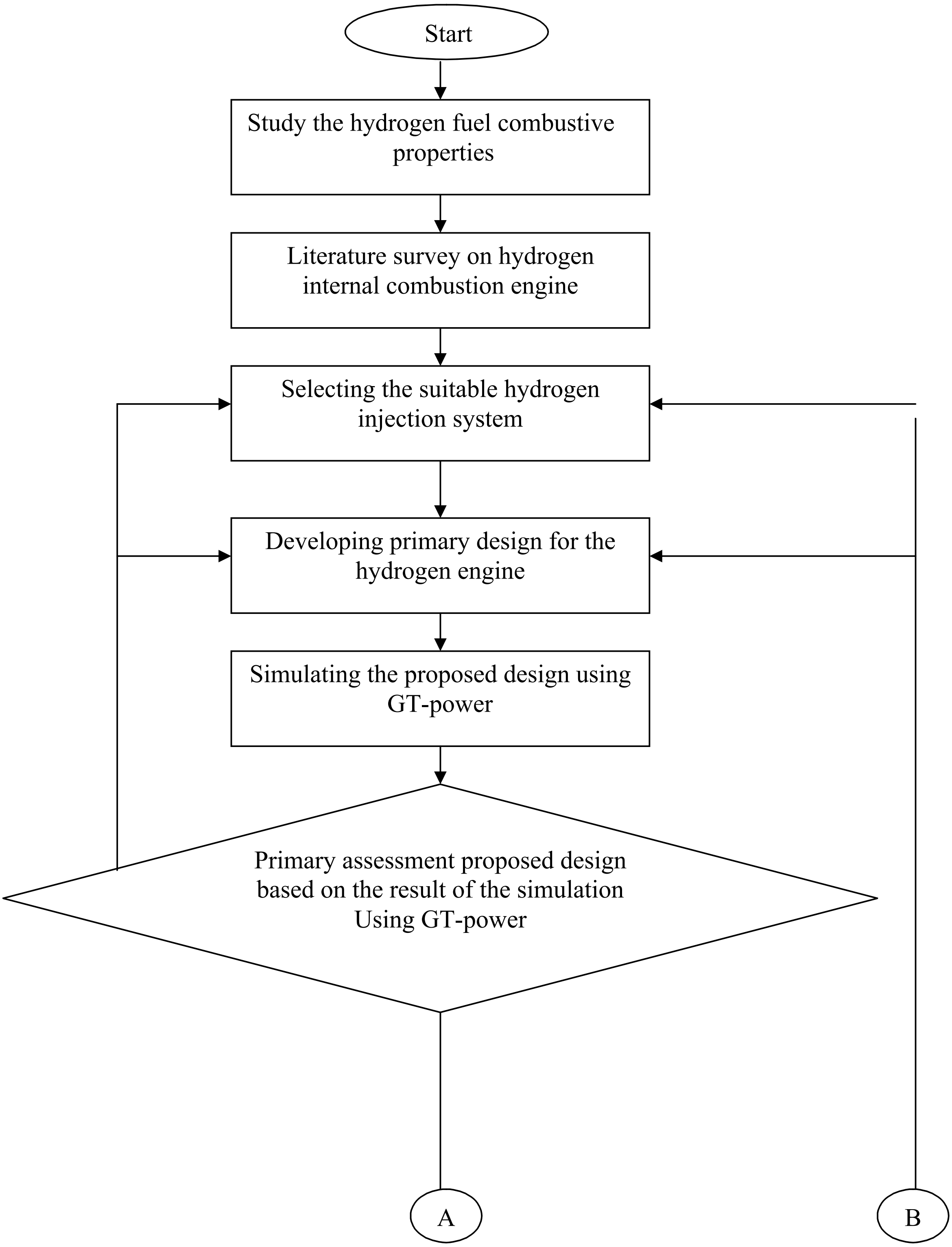
Chapter 3

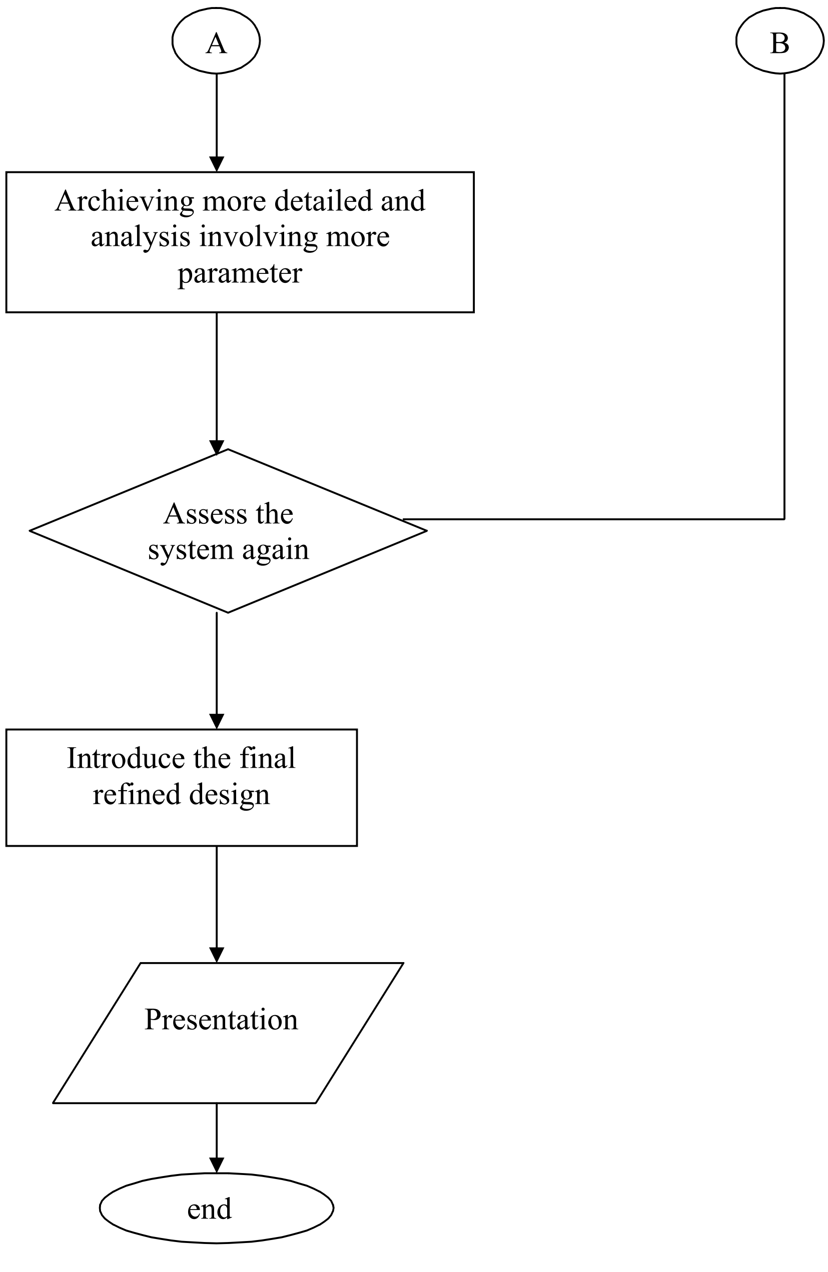
Methodology

3.1 Introduction

Methodology refers to the study of methods applied to a specific field of knowledge. A methodology includes explanation of theories or approaches, comparative study of different approaches and review of the individual method, in its study. This chapter represents the method and process occurred when carried out this project. All of the process like measuring the parameter, mapping the engine and simulate the model will be explained clearly in this chapter. Also, the tools and software used for this project also will be described.

3.1.2 Flow chart




A
B
Archieving more detailed and analysis involving more parameter
Assess the system again
Introduce the final refined design
Presentation
end

3.2 Modeling in GT power

The injection of hydrogen was located in the midway of the intake port while direct injection install at the head cylinder. The model of the hydrogen fueled single cylinder engine as shown in figure 3.1. Engine specification for the base engine is shown in table 1. The specific value of input parameter including the air fuel ratio (AFR) ,engine speed (RPM) and injection timing were defined in the model. The boundary condition of the intake air was defined first in the entrance of the engine. The air enters through a bell mouth orifice to the pipe. The discharge coefficient of the bell mouth orifice were set to 1 to ensure the smooth transition as in the real engine.

Delivery system	**port injection**	**Direct injection**
Bore/Stroke(mm)	100/100mm	100/100mm
Volume(L)	0.785L	0.785L
Compression ratio	9.5 :1	9.5 :1
piston pin offsets(mm)	1.0	1.0
Parameter	AFR 34.33 Variable AFR	Variable RPM RPM fix

3.3 One-dimensional computer simulation

We using the GT power software to develop the model engine for port injection and direct injection hydrogen fueled in single cylinder engine. The each component in the engine 4stroke all have in GT POWER library with each dimension. Each dimension need to be jot down in the template to ensure the simulation running as the actual engine and the hydrogen engine performance can be determined experimentally.

Using the GT POWER software, we can develop a model because in this software the general delivering system is built into designated components which hydrogen was injected into intake port while direct injection install at head cylinder. After that, hydrogen through the intake valve, cylinder, crank train for internal combustion energy to produce the mechanical energy for transport moving. Hence, when the water vapor can be out using the exhaust valve, exhaust port, exhaust runner and to the environment. The GT power library of all the engine component data will be input the engine dimension data. Select Window and then Tile with Template Library from the menu will create the GT-POWER model. All the available templates in GT-POWER contains in the Template Library. We need to copied the template from GT POWER library and click on the icons listed and drag them from the template library into the project library for the purpose of this model. Some of these objects and templates have already been defined and included in the GT-POWER template library. For first case of the simulation, all of the parameter in the model listed automatically in the case setup and each one must be defined.

After that, open case setup for using different RPM in cranktrain template. The stoichiometric air fuel ratio for hydrogen is 34.33 must be changed in SI inject template to ensure the hydrogen combust exactly. Consequence, the hydrogen engine performance can be determine experimentally.

After clicked case setup with different RPM, we can run the simulation. Hence, the GT POWER produces several output files after we run the simulation run. We can take out the result using the GT POST and is the post processing application for the most output available and order analysis output. In addition, the same method was using in direct injection and the injection timing for the direct injection is 74.01 g/sec at 5° at crank angle to gave the optimum power in 1-D experiment using commercial software, GT-Power. The difference between to delivery fuel system can be observe using GT-Power software.

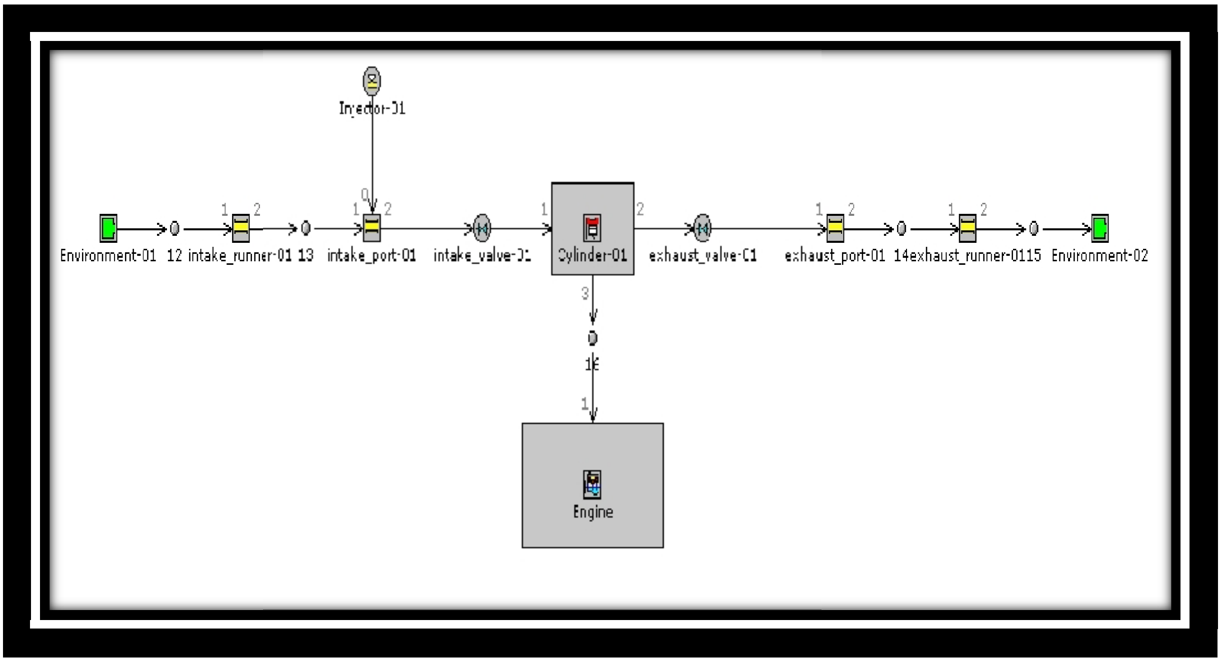


Figure 3.1: Single cylinder with port injection hydrogen fueled

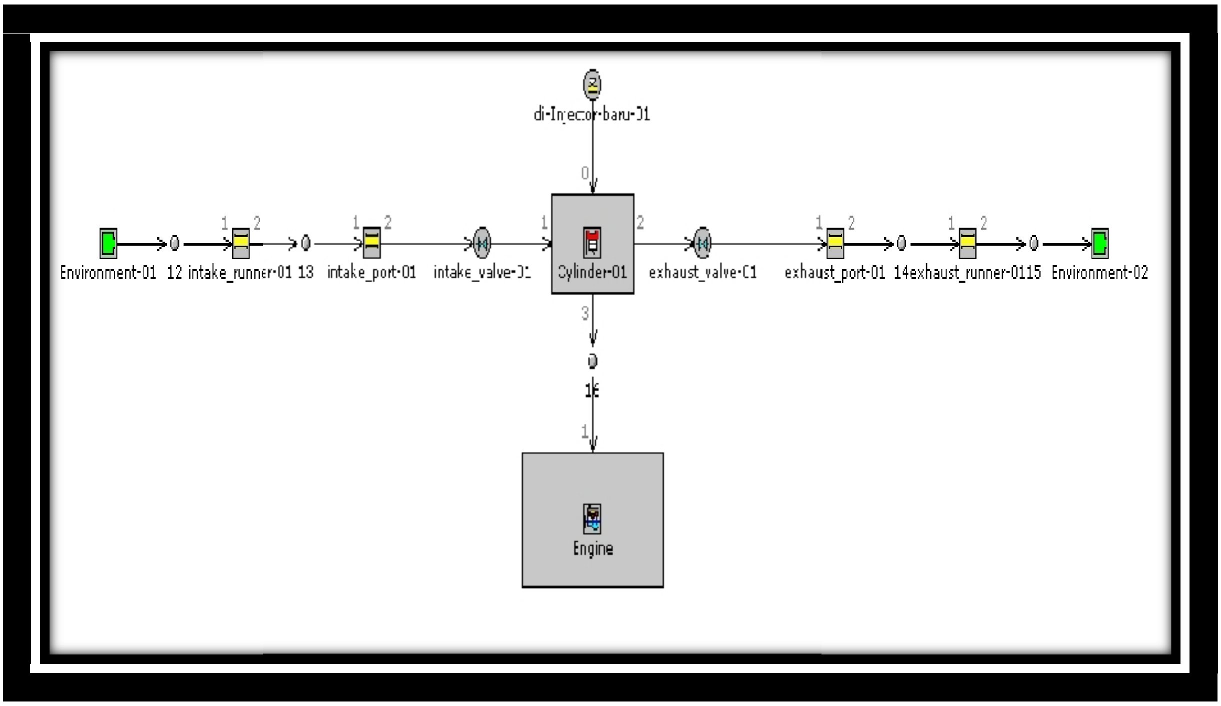


Figure 3.2: Single cylinder with direct injection hydrogen fueled

- The ambient condition designate ‘Env’ will start the intake system. For the intake and exhaust system, the ‘Env’ is used to describe the boundary condition of pressure, temperature and the mixture composition. Then it is connect to the default orifice to an inlet pipe. This object describe end environment boundary conditions of pressure, temperature and composition. The pressure flag, the standard (total) had been selected due to the pressure and temperature will be imposed as total conditions at the inlet of the attached flow component. Then, the composition of the end environment is defined as air.

- The intake port which the injector was installed at intake runner while direct injection was installed at cylinder head in order to injected the hydrogen into the cylinder for internal combustion when mixture with air. This template using to determine the wall temperature, surface roughness, heat transfer multiplier and reverse pressure loss coefficient

- The template ValveCamConn is the template when the hydrogen enter the cylinder for combustion process. The time for combustion need to exactly enough to avoid the exceed or less oxygen for combustion. The valve is defined such that high pressure at the inlet vlave causes the valve to open and high pressure at the outlet valve causes the valve to close. Valve reference diameter is used to calculate the effective flow area from the discharge coefficient arrays. The upstream pressure airea is the pressure area on the valve that is acted upon to open the valve that is acted upon to close the valve.

- 'InjAF-RatioConn' Connection is modeled as hydrogen injector to cylinder. It describes as an Injector-01. This object describes an injector that's injects fluid at a specific air-fuel ratio mixture into a pipe. It is used the local airflow sensor type that means the airflow sensor location will be the same as the point of injection. The injector is connected straight to the throttle body, so the location of the injector is at the inlet of the throttle body. The air fuel ratio needs to be added to gave maximum performance engine.

- 'InjAF-SeqConn' Connection is modeled as hydrogen injector into cylinder head. It described as Di-injector. This object describes an injector that's injects fluid at a specific air-fuel ratio mixture into a pipe. It is used the local airflow sensor type that means the airflow sensor location will be the same as the point of injection. The injector is connected straight to the cylinder head body, so the location of the injector is at the inlet of the throttle body. The air fuel ratio needs to be added to gave maximum performance engine.

- 'EngineCrankTrain' is the object attributer. The cranktrain models the crankslider mechanism and crankshaft which translate the torques generated directly from the pressure acting on each piston in the cylinders into the crankshaft output torque. The important parameters, need to be defined are engine type whether 2-stroke or 4-stroke engine, no of cylinder, configuration of cylinder whether in- line or v, the engine speed, and the start of cycle.

- EngCylinder' is modeled as the enginecylinder. This is the engine speed need to manipulated using different RPM to experiment the suitable engine speed using hydrogen. The important parameter that need to be defined are the cylinder geometry, wall temperature, heat transfer and combustion objects. Most of the parameters in the cylinder need to be defined by another reference such as "geom.", "air –fuel", "intake", "cyltwall", "htr","comb", and "scav".

Chapter4

Results and discussions

4.1 Introduction

In this chapter, all the raw results will be rearranged and the selected findings will be discussed briefly to give the proper explanation about the process and the important point in the results.

4.2 RPM=3000

In first case, we fix the Engine speed at (RPM=3000) and run the simulation with different Air fuel ratio (AFR) for every injection fuel system port injection (PI) and direct injection (DI). The result as observed as the graph below

RPM = 3000 AFR variable for both fuel injection system

Table 4.1: Different Air-fuel ratio when engine speed fixed

AFR	**BMEP (PI) (Bar)**	**BMEP (DI) (Bar)**	**BSFC (PI) g/kW-h**	**BSFC (DI) g/kW-h**	**Brake Torque (PI) N-m**	**Brake Torque (DI) N-m**	**Brake Power (PI) HP**	**Brake Power (PI) HP**
22.88	10.16	12.68	127.52	122.36	63.53	79.22	19.96	24.88
24.52	10.18	12.54	120.10	115.42	63.63	78.39	19.99	24.63
26.41	10.18	12.40	112.10	108.40	63.68	77.51	20.00	24.35

28.61	10.19	12.25	105.10	101.30	63.67	76.56	20.00	24.05
31.21	10.17	12.08	97.60	94.18	63.55	75.50	19.97	23.72
34.33	9.98	11.71	91.40	88.24	62.39	73.25	19.60	23.01
38.14	9.15	10.62	90.79	87.70	57.20	66.35	17.97	20.84
42.91	8.24	9.45	90.68	87.55	51.51	59.08	16.18	18.56
49.64	7.19	8.15	91.03	87.81	44.93	50.92	14.12	15.99
57.22	6.24	7.00	91.93	88.59	39.03	43.78	12.26	13.76
68.66	5.14	5.71	94.13	90.55	32.14	35.68	10.10	11.21

AFR	**Indicated Torque(PI) N-m**	**Indicated Torque(DI) N-m**	**Indicated Power(PI) kW**	**Indicated Power(DI) kW**	**Vol eff(PI) fraction**	**Vol Eff (DI) fraction**
22.88	74.16	90.12	23.29	28.31	0.71	0.8516
24.52	74.25	89.27	23.33	28.07	0.72	0.8518
26.41	74.28	88.36	23.34	27.76	0.73	0.851857
28.61	74.27	87.38	23.33	27.45	0.73	0.851957
31.21	74.12	86.28	23.29	27.10	0.74	0.852019
34.33	72.92	83.95	22.91	26.37	0.75	0.852103
38.14	67.66	76.94	21.26	24.17	0.76	0.852147
42.91	61.87	69.55	19.44	21.85	0.77	0.852204
49.64	55.19	61.25	17.34	19.24	0.78	0.852138
57.22	49.20	54.00	15.45	16.97	0.79	0.851883
68.66	42.19	45.76	13.26	14.38	0.80	0.851359

4.2.1 Brake Mean Effective Pressure (BMEP)

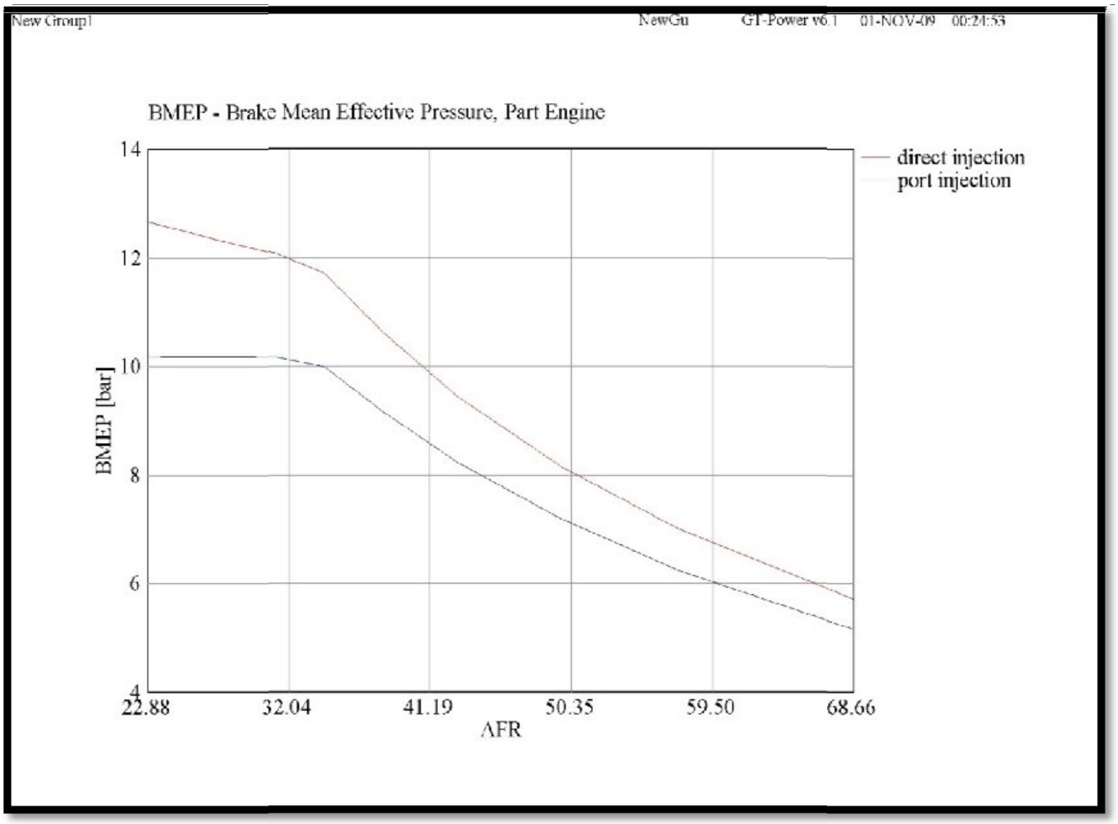


Figure 4.1: Brake Mean Effective Pressure (BMEP) vs AFR

Based on table 4.1 and figure 4.1 the Air-fuel ratio (AFR) was varied from rich limit (AFR= 22.88) from equivalent ratio (φ= 1.5) to a very lean limit (AFR = 171.65) from equivalent ratio (φ= 0.2) and the engine speed is constant with 3000 rpm just like the above figure. The variation of the analysis of AFR at constant speed, we can observe the indicated power output from the engine. The differences of BMEP gradually decrease with increase of the AFR and RPM from rich condition to lean condition. The highest BMEP occurred when the Air Fuel ratio (AFR) at rich mixture at 22.89 and gradually decrease after that. The maximum Brake Mean Effective Pressure(BMEP) is 10.19

while the Brake Mean Effective Pressure (BMEP) for direct injection is 12.68. Due to dissociation at high temperature combustion, molecular oxygen is present in the burned gases under stoichiometric conditions. Thus some additional fuel is added and partially burned. This increases the temperature and the number of moles of the burned gases in the cylinder. These effects increase the pressure; those were given increase power and mean effective pressure (Heywood, 1988).

4.2.2 Brake Power (BP)

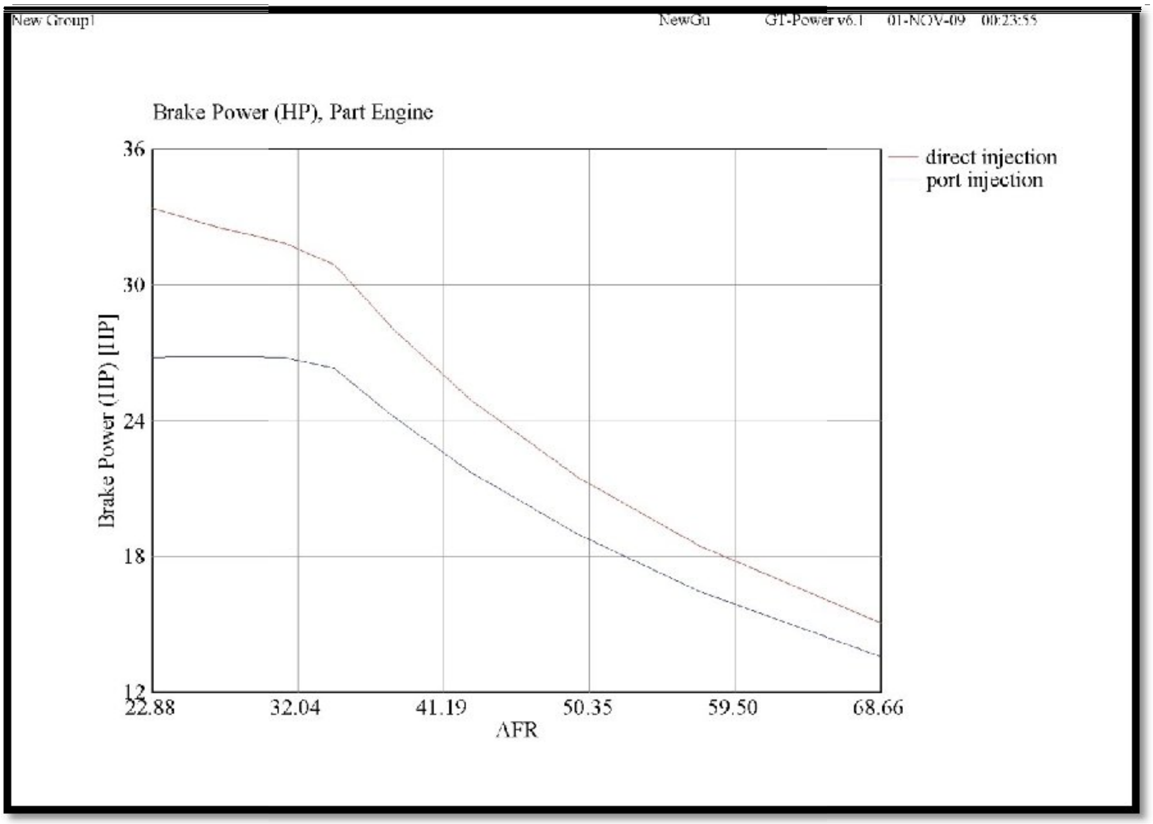


Figure 4.2: Brake Power vs AFR

Based on table 4.1 and figure 4.2 shows the Brake power (BP) with Air-fuel ratio. Brake power (BP) increasing with air fuel ratio decrease at certain level. The brake power decrease at certain level due to friction losses and become dominant factor at lean limit condition. The maximum Brake Power(HP) for port injection is 20.00HP while maximum Brake Power(HP) is 24.88HP.

4.2.3 Brake torque (BT)

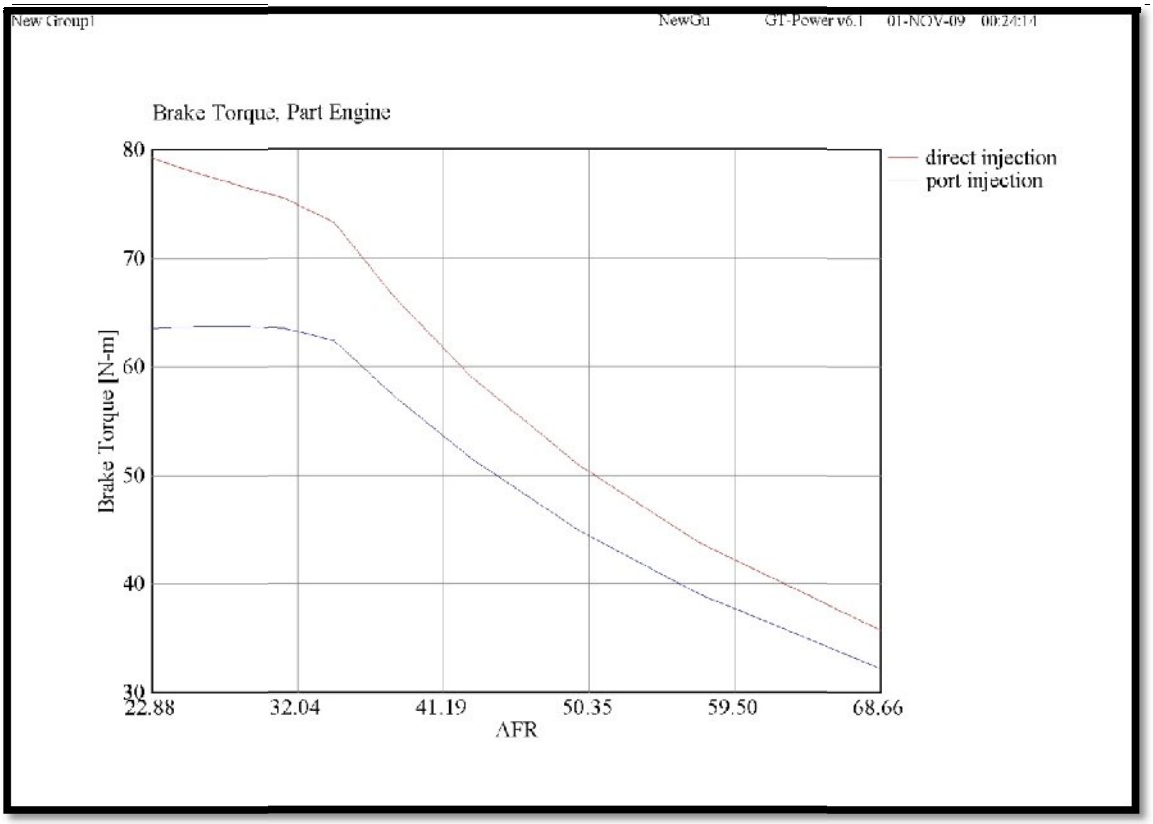


Figure 4.3: Brake Torque vs AFR

Based on table 4.1 and figure 4.3, Brake Torque (BT) for variable AFR is following the conventional engine torque. The brake torque will increase and decrease at certain point due the inability of the engine work at lean mixture that leads torque losses and the maximum brake torque port injection is 63.68Nm while direct injection has a maximum brake torque 78.39Nm.

4.2.4 Brake Specific Fuel Consumption (BSFC)

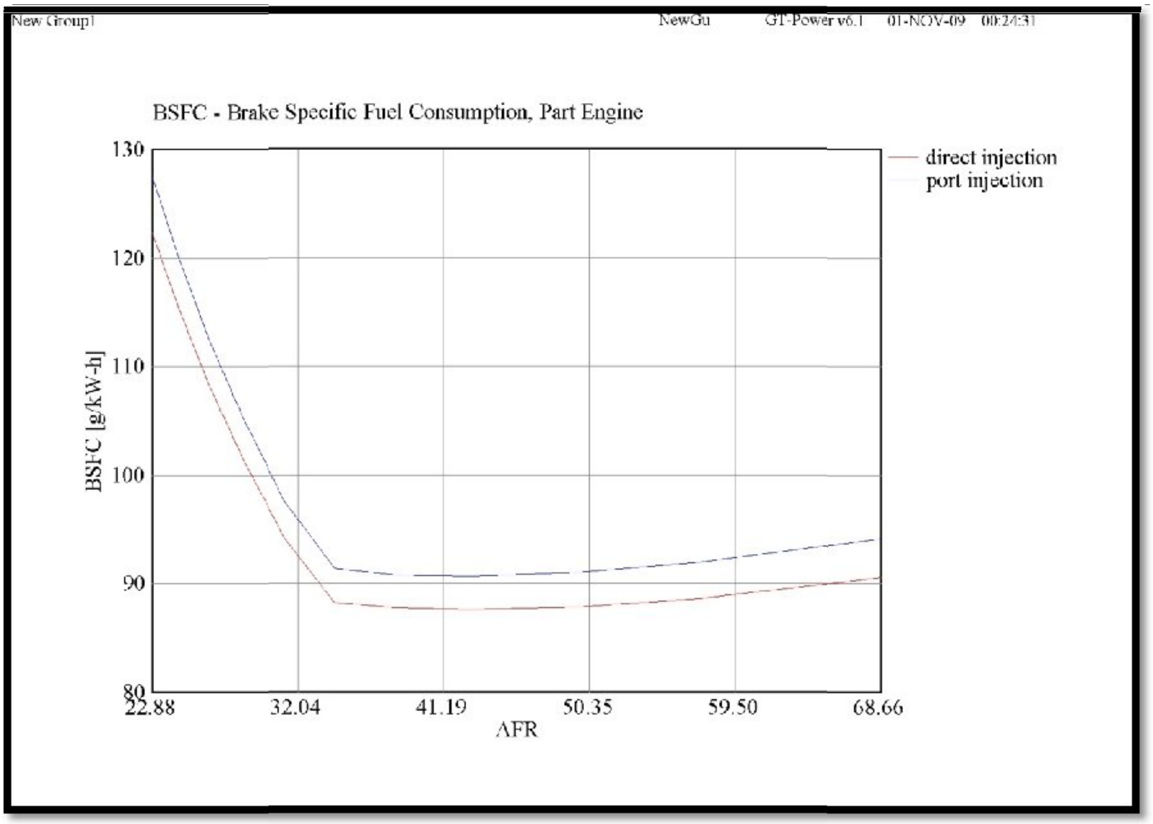


Figure 4.4: Brake Specific Fuel Consumption vs AFR

Based on table 4.1 and figure 4.4 we can conclude that increase of BSFC with increases in the rotational speed and increases the value of AFR. This figure shows that the maximum BSFC at rich condition at air fuel ratio (AFR) at 28.61. This is because of very lean operation conditions, which leads the unstable combustion and more lost power due to a reduction in the volumetric heating value of the mixture. This behavior can be more clarified by figure 4.4.

4.2.5Indicated Power (IP)

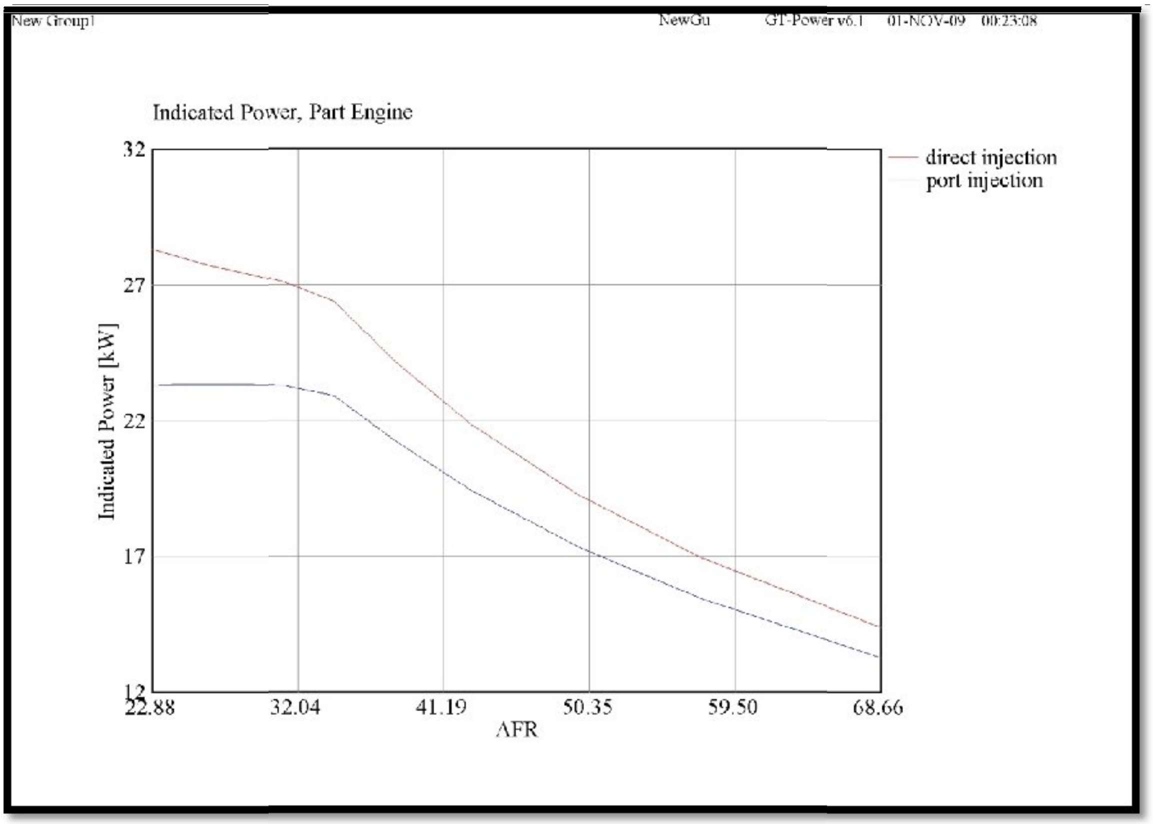


Figure 4.5: Indicated Power vs AFR

Based on table 4.1 and figure 4.5 the maximum indicated power was observed at air fuel ratio at lean conditions (AFR=22.89) and after the graph gradually decrease and at certain limit air fuel ratio at stoichiometric condition AFR 34.33 and linearly decrease after the that because at lean mixture the combustion is not complete in order to give the maximum engine power consequently the indicated power gradually decrease. The maximum Indicated Power (IP) for port fuel injection 23.34 kW while direct fuel injection is 28.31 kW.

4.2.6 Indicated Torque (IT)

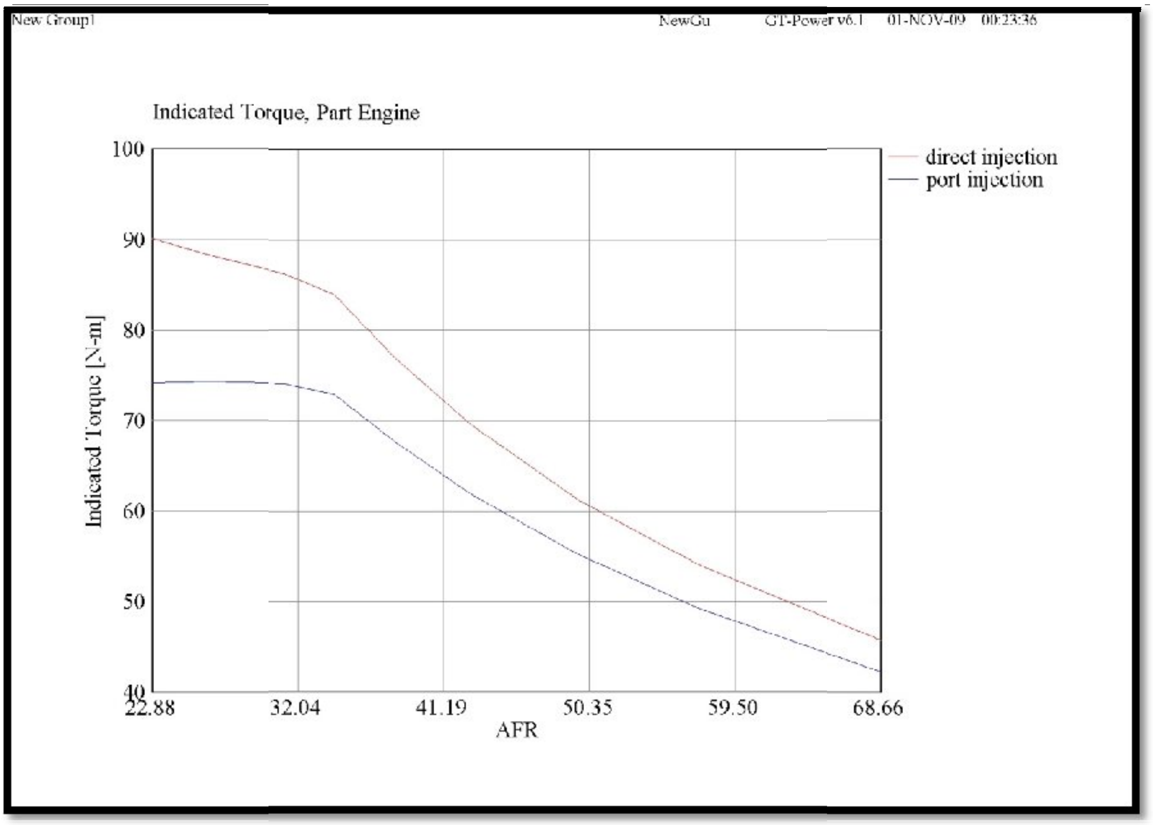


Figure 4.6: Indicated torque vs AFR

Based on the table 4.1 and figure 4.6 show that rotational engine speed (RPM) and air fuel ratio increase torque increase and decrease certain limit. The maximum torque the graph decreases because torque doesn’t support the load when engine speed and air fuel ratio still increase. At higher engine speed, engine not withstand dynamometer load thus increase the friction.

4.2.7 Volumetric Efficiency(%)

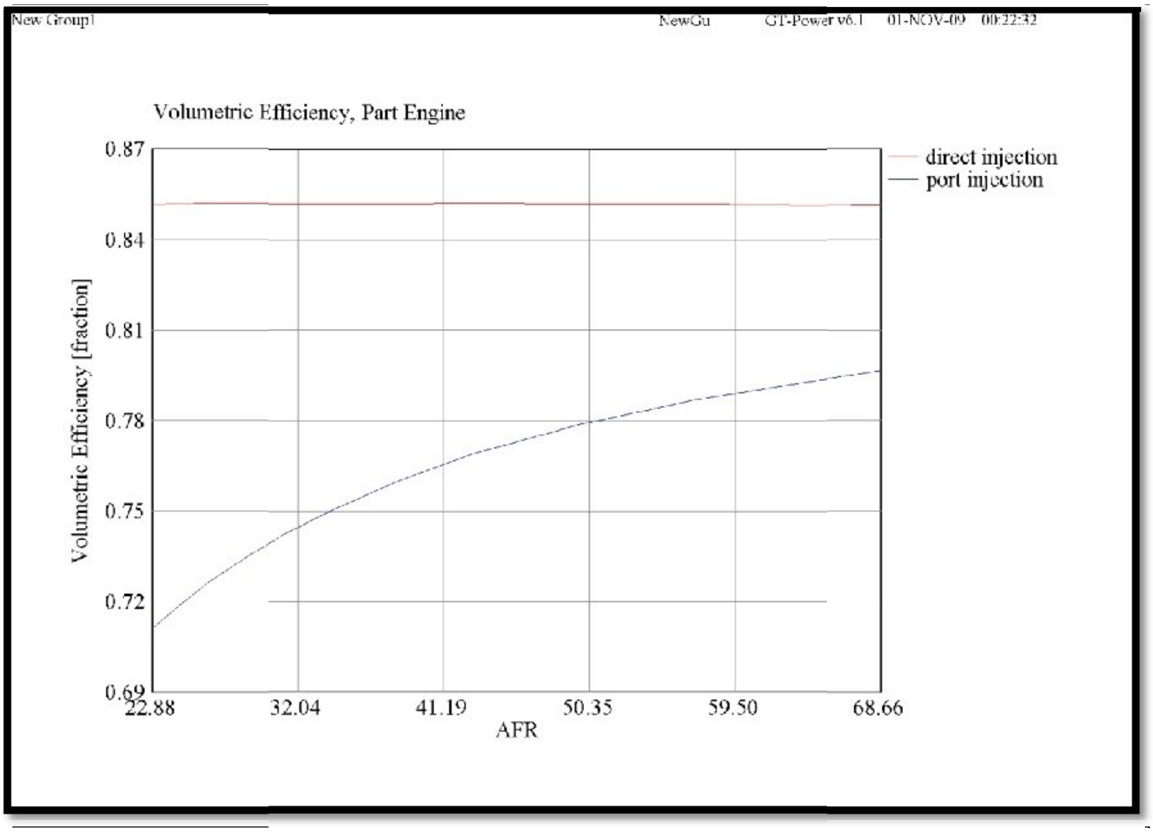


Figure 4.7: Volumetric Efficiency vs AFR

Based on table 4.1 and figure 4.7 show that the maximum volumetric efficiency was observed at 85.2103% at stoichiometric condition at air fuel ratio 34.33 at engine speed 3500 rpm where the volumetric efficiency decrease at high engine speed rotational due to friction losses for direct injection while the port injection at 80% .

4.3 AFR=34.33

In second case, we fix air fuel ratio at (AFR=34.33) and run the simulation with different engine speed (RPM) for every injection fuel system port injection (PI) and direct injection (DI). The result as observed as the graph below

AFR=34.33 Engine speed variable (RPM) variable for both fuel injection system

Table 4.2: Different RPM when AFR fixed

RPM	**BMEP (PI) Bar**	**BMEP (DI) Bar**	**BSFC (PI) g/kW-h**	**BSFC (DI) g/kW-h**	**Brake Torque (PI) N-m**	**Brake torque (DI) N-m**	**Brake Power (PI) HP**	**Brake Power (DI) HP**
1500	9.38	11.02	89.00	86.84	58.65	68.87	12.37	14.52
2000	9.62	11.28	88.92	86.38	60.09	70.52	16.89	19.83
2500	9.79	11.53	89.87	86.98	61.17	71.06	21.49	25.32
3000	9.98	11.72	91.39	88.24	62.39	73.25	26.31	30.89
3500	10.00	11.69	93.72	90.33	62.55	73.07	30.77	35.95
4000	9.72	11.29	97.16	93.49	60.74	70.56	34.15	39.68
4500	9.16	10.55	102.00	97.95	57.27	65.97	36.22	41.73
5000	8.29	9.41	108.73	104.11	51.81	58.80	26.41	41.33
5500	7.17	8.09	117.84	112.15	44.80	50.55	34.63	39.08
6000	5.93	6.71	130.67	123.19	37.14	41.91	31.33	35.35

RPM	Indicated Torque(PI) N-m	Indicated Torque(DI) N-m	Indicated Power(PI) kW	Indicated Power(DI) kW	Vol eff (PI) fraction	Vol eff (DI) fraction
1500	65.79	76.17	10.33	11.97	0.69	0.79
2000	68.32	78.91	14.31	16.53	0.70	0.80
2500	70.53	81.58	18.46	21.36	0.73	0.83
3000	72.92	83.95	22.91	26.37	0.75	0.85
3500	74.27	84.94	27.22	31.13	0.77	0.87
4000	73.66	83.62	30.85	35.03	0.78	0.87
4500	71.37	80.19	33.63	37.79	0.77	0.85
5000	67.09	74.17	35.13	38.83	0.74	0.81
5500	61.25	67.05	35.28	38.62	0.68	0.75
6000	54.76	59.57	34.41	37.43	0.64	0.68

4.3.1 Brake Mean effective pressure (BMEP)

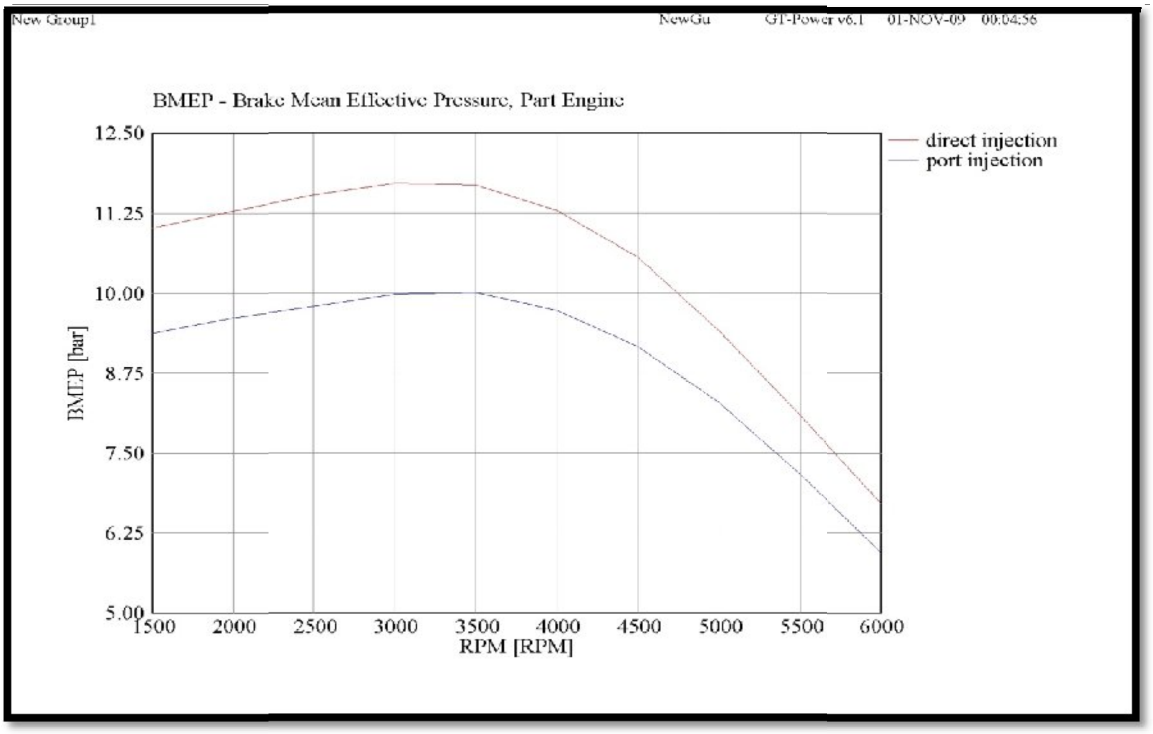


Figure 4.8: Brake Mean Effective Pressure vs RPM

It is obvious that the brake mean effective (BMEP) decreases when RPM and air fuel ratio increases. Due to its association at high temperatures following combustion, molecular oxygen is present in the burned gases under stoichiometric conditions. Thus some additional fuel can be added and partially burned. This increases the temperature and the number of moles of the burned gases in the cylinder. These effects increase the pressure; these were given increase power and mean effective pressure (Heywood, 1988).

4.3.2 Brake Power (HP)

Figure 4.9**:** Brake Power (HP) vs RPM

Based on table 4.2 and figure 4.9 shows the Brake power(BP) with Engine Speed (RPM). Brake power (BP) increasing with Engine speed decrease at certain level. The brake power decrease at certain level due to friction losses and become dominant factor at lean limit condition. The maximum Brake Power(HP) for port injection is 36.22HP while maximum Brake Power(HP) is 41.73HP.

4.3.3 Brake Torque

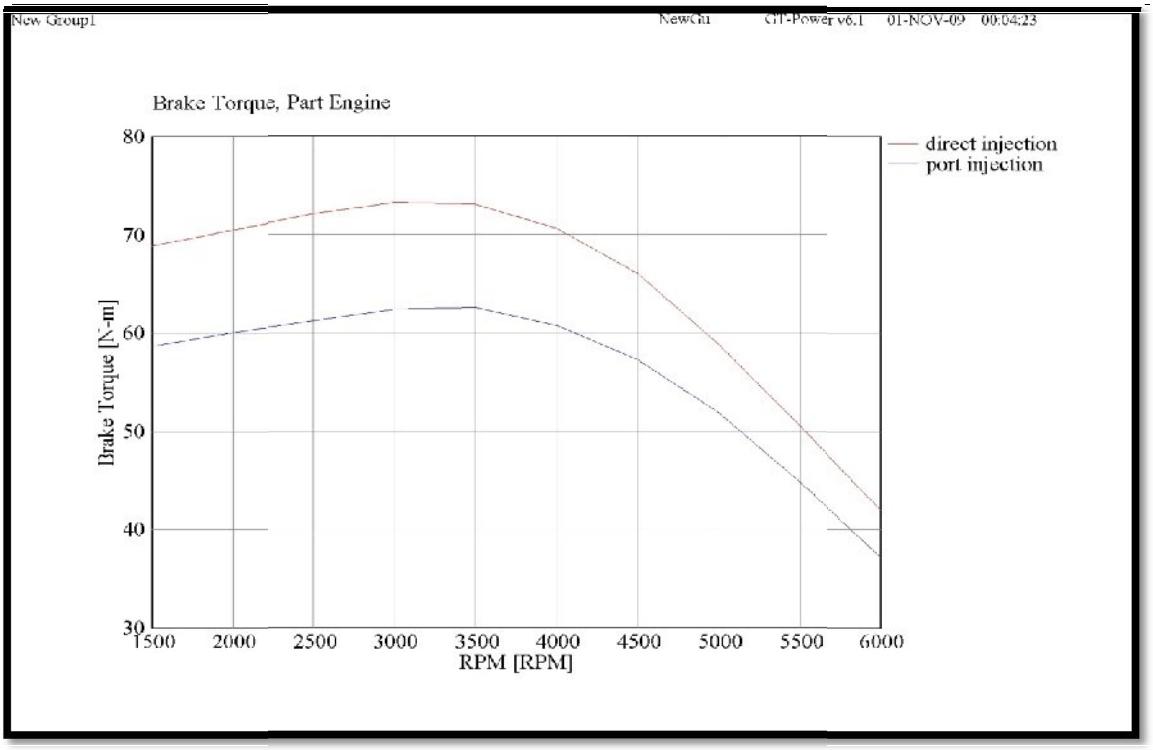


Figure 4.10: Brake Torque vs Rpm

Based on table 4.2 and figure 4.10, Brake Torque (BT) for variable Engine Speed(RPM) is following the conventional engine torque. The brake torque will increase and decrease at certain point due the inability of the engine work at high speed engine that leads torque losses because the engine exceed the overload of the engine and the maximum brake torque port injection is 62.55Nm while direct injection has a maximum brake torque 73.25Nm.

4.3.4 Brake Specific Fuel Consumption (BSFC)

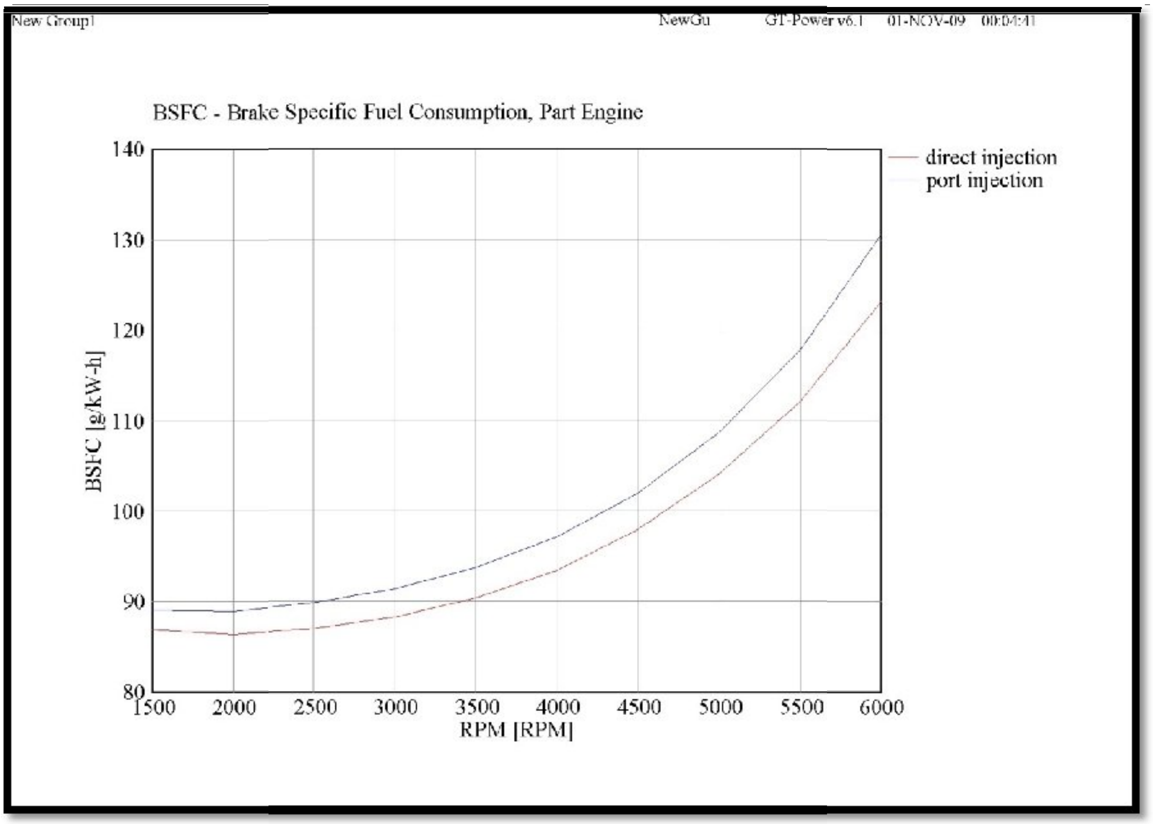


Figure 4.11: Brake Specific Fuel Consumption vs RPM

Based on table 4.2 and figure 4.11 above we can conclude that increase of BSFC with decreases in the rotational speed and increases the value of AFR. This is because of very lean operation conditions, which leads the unstable combustion and more lost power due to a reduction in the volumetric heating value of the mixture. This behavior can be more clarified by figure above. The Maximum Brake Specific Fuel Consumption for port injection is 130.67 gkW-h while direct injection is 123.19 gkW-h.

4.3.5 Indicated Power

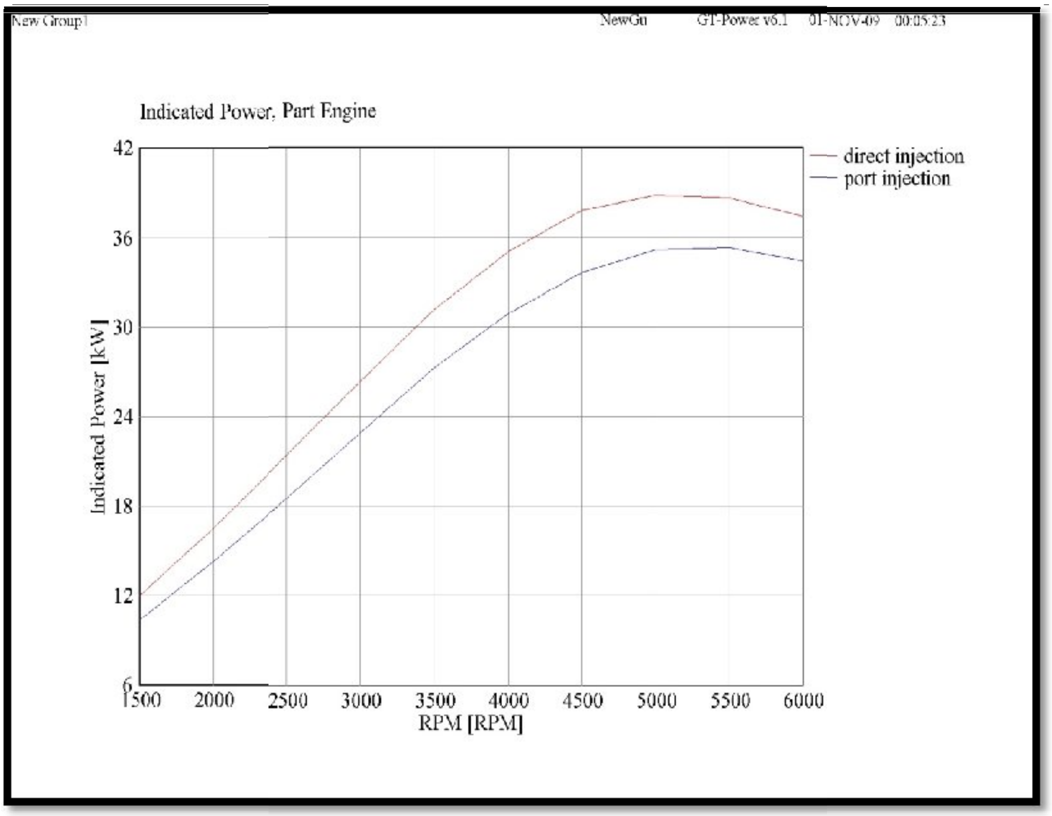


Figure 4.12: Indicated Power vs RPM

The maximum indicated power was observed at engine speed at 5500 rpm for port injection while 5000 rpm for direct injection and after that at certain level the indicated power decrease at higher engine speed after certain level due to the friction losses and becomes the dominant factor at very high engine speed.

4.3.6 Indicated Torque

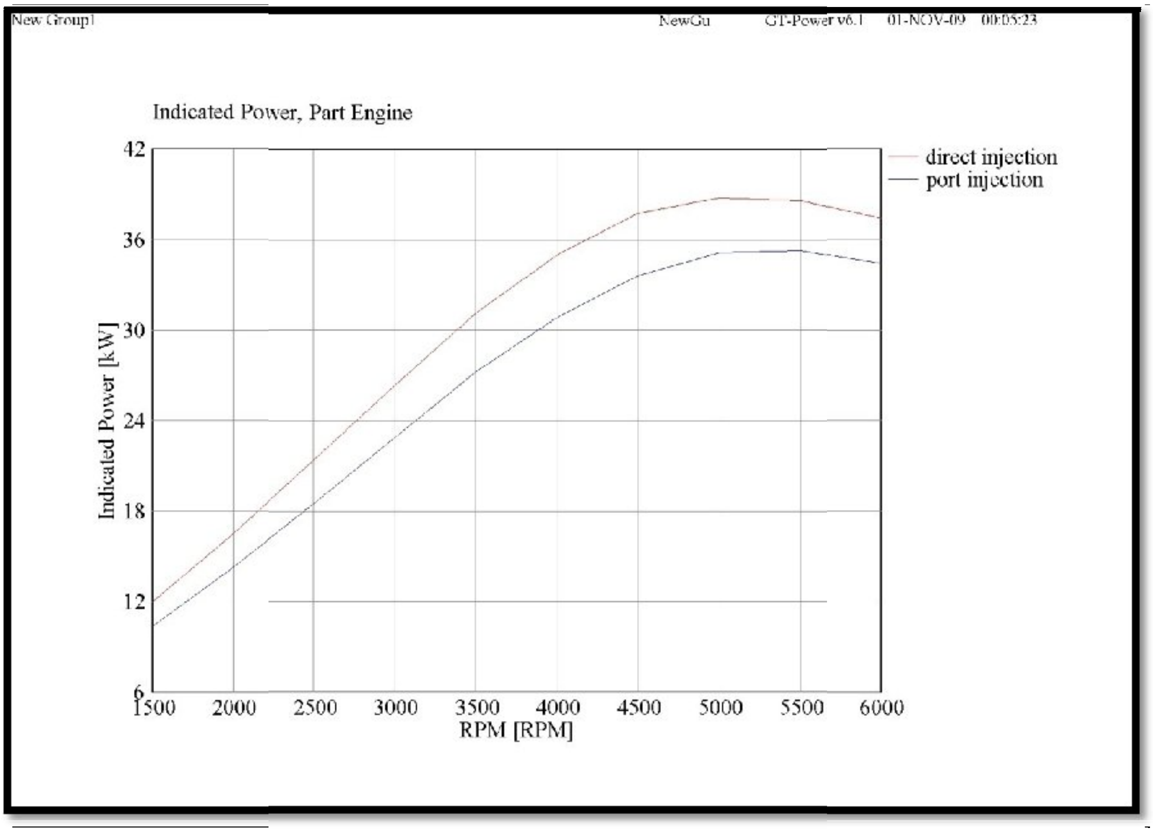


Figure 4.13: Indicated Torque vs RPM

The maximum indicated power was observed at engine speed at 5000 rpm for direct injection is 38.83 kW while port injection was observe at engine speed at 5500 is 34.41 kW and after that at certain level the indicated power decrease at higher engine speed after certain level due to the friction losses and becomes the dominant factor at very high engine speed.

4.3.7 Volumetric Efficiency

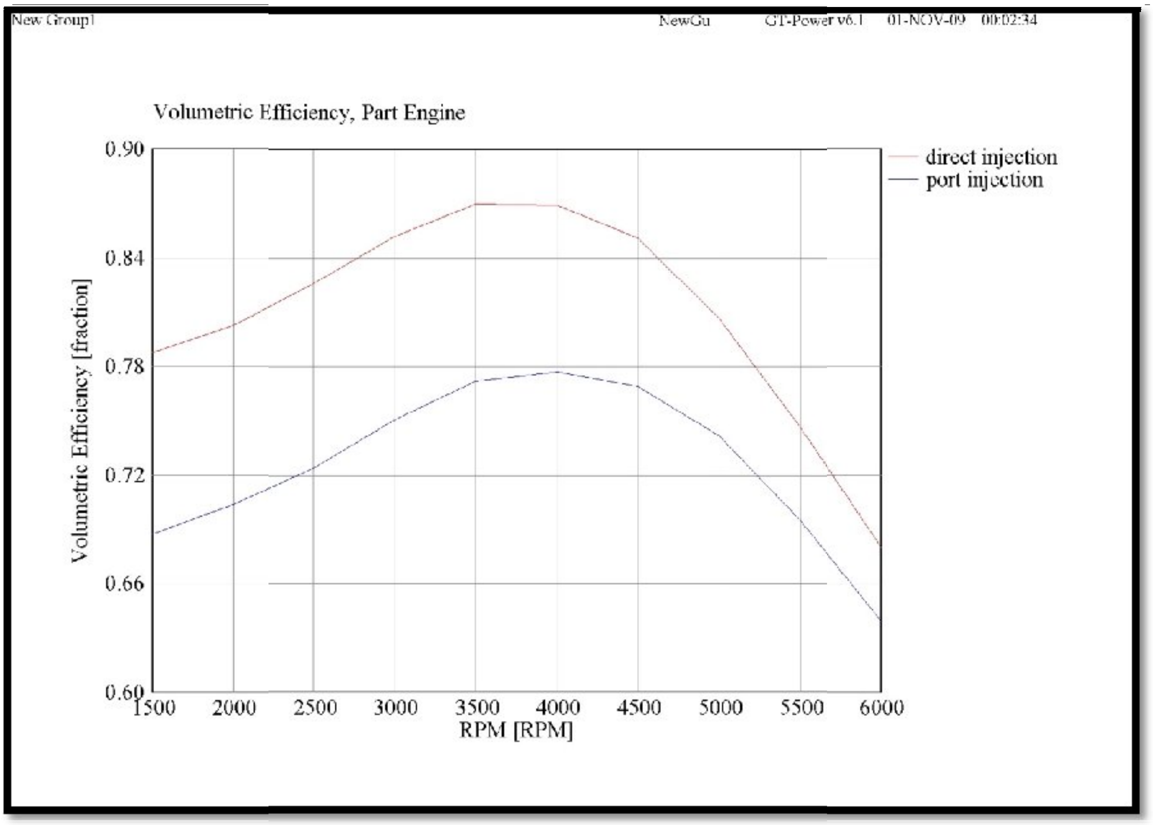


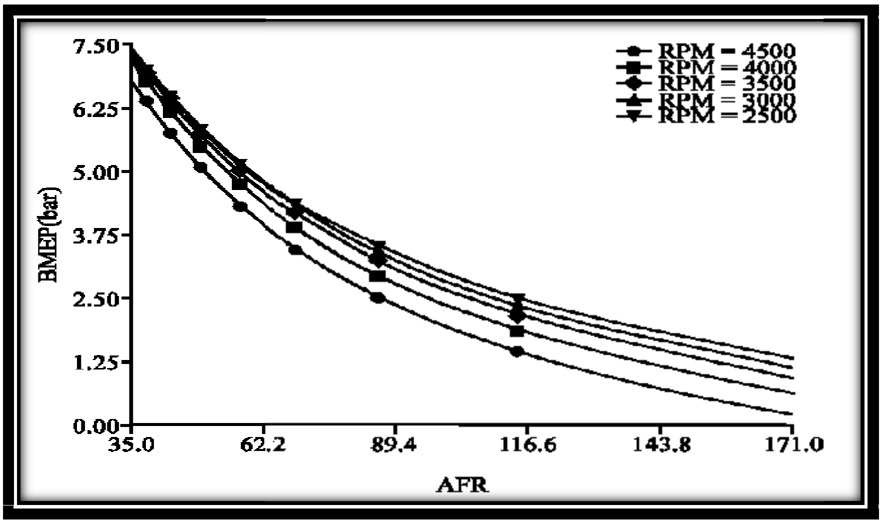
Figure 4.14: Volumetric efficiency vs RPM

The maximum volumetric efficiency was observed at 87% at stoichiometric condition at air fuel ratio 34.33 at engine speed 3500 rpm while port injection is 0.78 at engine speed 4000 rpm where the volumetric efficiency decrease at high engine speed rotational due to friction losses. In the first part of the curve, higher speed lead to higher volumetric efficiency because of the high speed gives high vacuum at the intake port and consequent large air flow rate that goes inside the cylinder.

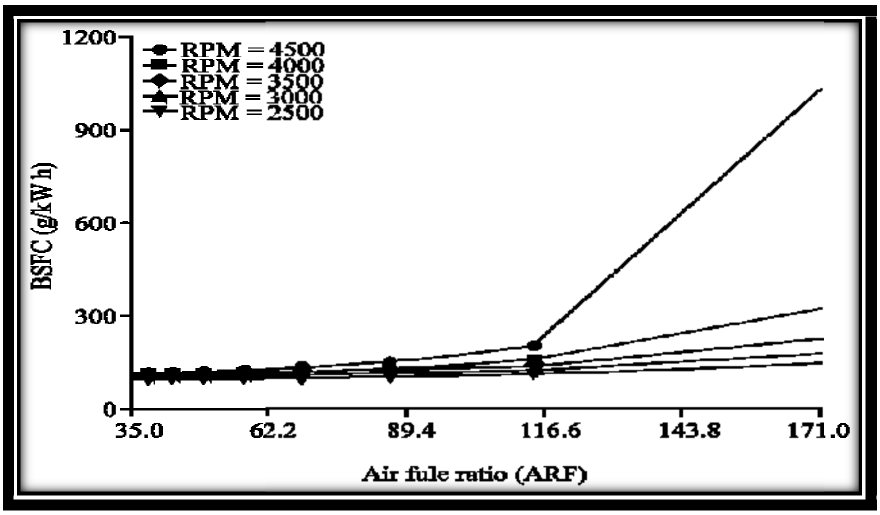
4.4 Validation Result

4.4.1 Port injection

Brake Mean Effective Pressure (BMEP)

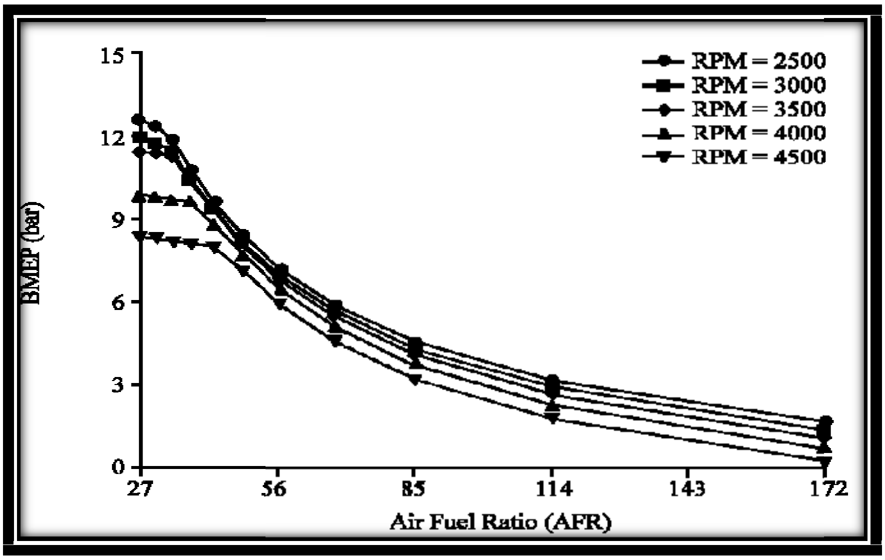


Brake Specific Fuel Consumption (BSFC)

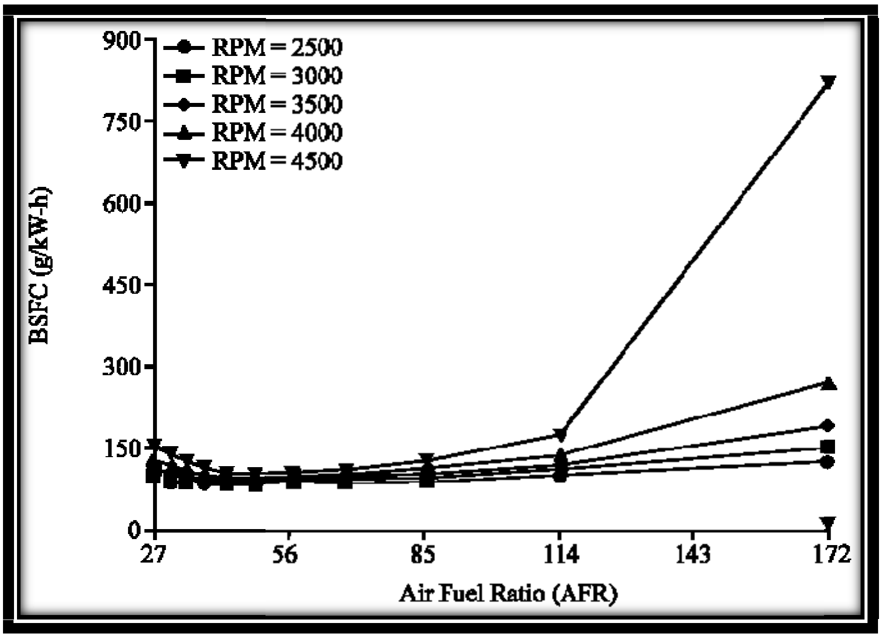


4.4.2 Direct Injection

Brake Mean Effective Pressure (BMEP)



Brake Specific Fuel Consumption (BSFC)



Chapter 5

Conclusion and recommendation

Recommendation

A computational model was developed for 1-cylinder direct and port injection hydrogen fueled engine. The influence of air fuel ratio and engine speed with injection timing for direct injection was fixed at 74.01 g/sec. The main results are summarized as follows;

1. Air Fuel Ratio (AFR) has strong effect on the injection timing. Adding more hydrogen to the mixture (ascent towards richer mixture) retards optimum injection timing from TDC timing.
2. As a compromise, injection timing of 60 BTDC can be considered as optimum for the present engine. However, this is for constant injection timing. The recommended operation is with different injection timing based on AFR.
3. Interaction between injection duration and spark timing is strongly undesired and can result in unstable operation. This is was apparent by the unaccepted performance parameters during interaction period. Avoiding of this interaction should take priority in specifying injection timing
4. Spark timing is another parameter that should be optimized for hydrogen engines, especially direct injection hydrogen fueled engines. The optimum value spark timing are related strongly. The best way of optimizing injection timing is to fix spark timing on maximum brake torque timing.

Conclusion

The performance characteristics of single cylinder port injection hydrogen fueled engine were investigated. The following conclusions are drawn:

1. At very lean conditions with low engine speeds, acceptable BMEP can be reached, while it is unacceptable for higher speeds. Lean operations leads to small values of BMEP compared with rich conditions.
2. Maximum brake thermal efficiency can be reached at mixture composition in the range of (φ = 0.7 to 0.9) and it decreases dramatically at leaner conditions
3. The desired minimum BSFC occurs within a mixture composition range of (φ = 0.7 to 0.9). The operations with very lean condition (φ =0.2) and high engine speeds (>4500) consumes unacceptable amounts of fuel
4. Lean operation conditions results in lower maximum cylinder temperature. A reduction of around 1400K can be gained if the engine works properly at (φ =0.2) instead of stoichiometric operation
5. The low values of volumetric efficiency seems a serious challenge for the hydrogen engine and further studied are required

It has been adequately emphasized that the hydrogen fuel posses some properties which are uniquely different from the corresponding properties of conventional hydrocarbon fuels. This is was primarily the reason why initially the research occurs under heavy load conditions. These often cause engine to stop. The symptoms of unsteady combustion the most pronounced effect in an internal combustion engine.

In general, it is desirable to have maximum volumetric efficiency for engine. The importance of volumetric efficiency is more critically for hydrogen engines because of the hydrogen fuel displaces large amount of incoming air due to its low density (0.0824 kg m at 25 °c and 1 atm). This reason reduces volumetric efficiency to high content. A sotichiometric mixture of fully vaporized gasoline and air consists of approximately 2% gasoline by volume.

Therefore, the low volumetric efficiency for hydrogen engine is expected compared to gasoline engine works with same operating conditions and physical dimension. However, the higher volumetric efficiency can be gained with direct injection of hydrogen which all show figures above. The maximum value of volumetric efficiency for selected range of speed was around 85%. At further higher engine speed beyond these values, the flow into the engine during at least part of the intake process becomes choked.

Once this conditions occurs, further increase in engine speed decrease the flow rate significantly. Thus, the volumetric efficiency decrease sharply because of the higher speed is accompanied by some phenomenons that have negative influence on volumetric efficiency. These phenomenons include the charge heating in the manifold and higher friction losses which increase as the square of the engine speed.

From this study I conclude, the model engine for H_2 ICE with direct injection and port injection able to develop using GT-power commercial software. Besides, we can verify the performance characteristic of direct injection and port injection

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