Study on the Application of Electronic Fuel Injection to Carburetted Single Cylinder 4 Stroke Engine

by

Adibah Binti A. Jalil

Dissertation Draft submitted in partial fulfillment of
The requirements for the
Bachelor of Engineering (Hons)
(Mechanical Engineering)

JULY 2009

Universiti Teknologi PETRONAS
Bandar Seri Iskandar
31750 Tronoh
Perak Darul Ridzuan

CERTIFICATION OF APPROVAL

Study on the Application of Electronic Fuel Injection to Carburetted Single Cylinder Four Stroke Engine

by

Adibah Binti A. Jalil

A project dissertation submitted to the
Mechanical Engineering Programme
Universiti Teknologi PETRONAS
in partial fulfilment of the requirement for the
BACHELOR OF ENGINEERING (Hons)
(MECHANICAL ENGINEERING)

Approved by,

(Ir Idris Bin Ibrahim)
Idris bin Ibrahim, P.Eng. MIEM
Senior Lecturer
Mechanical Engineering Department
Universiti Teknologi PETRONAS

UNIVERSITI TEKNOLOGI PETRONAS
TRONOH, PERAK
July 2009

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

ADIBAH BINTI A. JALIL

ABSTRACT

Single cylinder 4 stroke engine usually use carburettor for the fuel system. Tuning carburettor to get the right amount of fuel for different engine operating condition is not easy. Comparing with Electronic Fuel Injection System (EFI), carburettor has high fuel consumption, produces less power, leads to fuel delivery instability and affects the drivability of the vehicle. The study is conducted to show that EFI system produce better performance than carburetted system for the single cylinder 4 stroke engine. The study is done by using engine simulation software, GT POWER. Carburetted engine model is built in GT POWER and validations are being made with previous testing and simulation data. The carburetted model is then converted to EFI model. The EFI model is further tuned to improve the engine torque at low speed range. The findings show that EFI system produce better performance than carburetted system. Finding also shows that engine torque at low speed range can be improved by intake length tuning and AFR tuning.

ACKNOWLEDGEMENTS

A special acknowledgement is reserved for my supervisor, Ir Idris Bin Ibrahim and also, my Engine Design lecturer, Dr Ir Masri Bin Baharum for their mentoring, supervision, knowledge and kindness in helping me to achieve my target and goals in completing this project.

A large measure of gratitude to the following person who contributed a lot through their help, suggestion and criticisms. I will always value their advices.

Mr. Kyairol Izwan b Ghazali, Senior Engineer, POWERTRAIN TECHNOLOGY, PETRONAS Research Sdn. Bhd.
Ahmad Nazri Bin Amiruddin, Senior Engineer, POWERTRAIN TECHNOLOGY, PETRONAS Research Sdn. Bhd.
Mr. Shaharuddin Hamid b Mustapha, Senior Engineer, POWERTRAIN TECHNOLOGY, PETRONAS Research Sdn. Bhd.

Also a special note of thanks to all the engineers and technicians of Powertrain Technologies who have freely taken the time to assist and advice me in completing this project. Since this list is too long to mention, it is hoped that those who have given help in this manner will accept this anonymous recognition.

Lastly, a very special recognition goes to both my parents, Mrs Siti Sabariah Binti A. Ralim and Mr A. Jalil Bin Kassim, and all my close friends for their strong moral support that have steeled my resolve. Without these two pillar of supports, I would have lost the patience and will to perservere.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF ABBREVIATIONS

WOT- Wide Open Throttle

BSFC- Brake Specific Fuel Consumption

RPM –Revolution per Minute

Dyno- Dynamometer

AFR- Air Fuel Ratio

EFI- Electric Fuel Injection System

ECU- Engine Control Unit

CHAPTER 1
INTRODUCTION

1.1 Background of Study

Single cylinder 4 stroke engines are usually used for motorcycle and karts. Most of them utilize carburetor instead of Electronic Fuel Injection as fuel delivery system as to reduce the overall production cost. The examples of single cylinder 4 stroke engine are Brigss and Straton and K200 engine.

K200 engine is one of the engines produced by PETRONAS. It is a single cylinder, four stroke engine with carburetted fuel system. This engine is used for Go Kart. Shown in Table 1.1 is the general technical specification of the engine.

Table 1.1: General Technical Specification of K200 Engine

Type	Single cylinder, 4-stroke engine	**Current Fuel System**	Carburetor
Displacement	199 cm^3	**Max. Power**	13.8 kW at 9000 rpm
Bore x Stroke (mm)	70 x 51.8	**Max. Torque**	16.5 Nm at 7000 rpm
Valve train system	2 valves SOHC	**Cooling System**	Air Cooled
Compression Ratio	10.0	**Lubrication System**	Dry Sump

This engine is still under development program and a lot of effort has been put to further improve the engine such as its durability, function, emissions and performance.

1.2 Problem Statement

Tuning of carburetor to get the right amount of fuel throughout its operating condition is exigent. Comparing to EFI fuel system, carburetor has high fuel consumption. The performance is affected by the ambient condition. It leads to fuel delivery instability and affects drivability of the vehicle.

When environmental factors change, the carburetors are less efficient. It produces less power, consume more fuel, or even damage the engine itself. The adaptive nature of electronic control systems allows an engine to always run at optimum performance.

Performance simulation is conducted to investigate the engine performance and behavior before the actual testing and development program is done. In order to do the simulation, model that is being used for the simulation must be reliable in order to obtain accurate simulation results. It is expected that the change will improve the performance of the engine.

1.3 Objectives and Scopes of Study

1.3.1 Objectives

The objective of the study is:

i. Develop, simulate and validate carburetted engine model with previous experimental and WAVE simulation data.
ii. Convert the validated carburetted engine model to EFI engine model, simulate and analyze the EFI engine model.
iii. Improve the performance of EFI engine at a speed range of 4000RPM to 7000RPM.

1.3.2 Scope of Study

This study only focuses on a single cylinder 4 stroke engine. The engine used for the study is a naturally aspirated spark ignition (SI) K200 engine. The study is only applicable for small engine. The simulation model is build based on the design parameters and also measurement data from previous dynamometer testing and WAVE simulation result. Other parameters that are not available are assumed and referred to database. The study is limited to the simulation using engine simulation software, GT POWER. No testing will be done for this project. The results obtained are prediction and may vary from real testing data. For the conversion, no injector selection will be done. The simulation is a steady state simulation. It is done with a constant load of 100% throttle.

This study is to be completed within approximately one year timeframe. The phase 1 of the project covers the modeling of carburetted engine in GT POWER and validation with previous dynamometer testing and simulation data. Phase 2 covers the conversion of the carburetted model to EFI model, simulation and analysis after the conversion, and increasing the torque at a speed range of 4000RPM to 7000RPM.

CHAPTER 2
LITERATURE REVIEW

2.1 Otto Cycle

Naturally Aspirated Spark Ignition (SI) 4 stroke engine cycle at WOT is shown in Figure 2.1. For analysis, the cycle is approximated with the ideal air standard cycle shown in Figure 2.2.

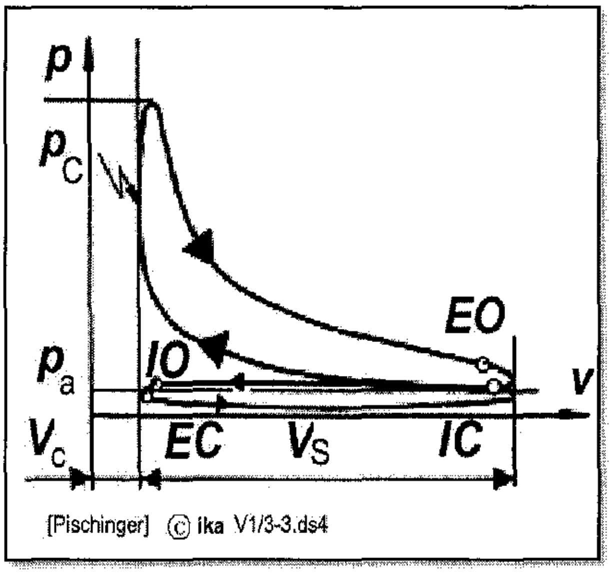


Figure 2.1: Actual Naturally Aspirated Spark Ignition (SI) 4 Stroke Engine Cycle (Benson)

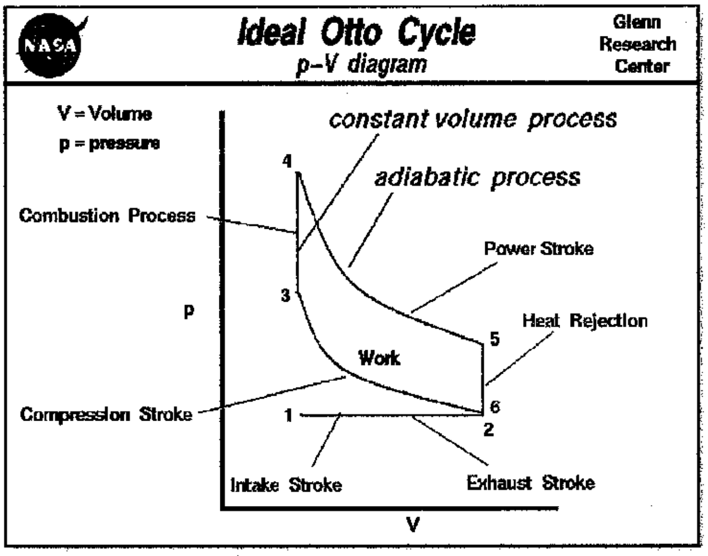


Figure 2.2: Ideal Air-Standard Otto Cycle (Benson)

A typical 4 stroke engine cycle, consist of intake stroke, compression stroke, power stroke and exhaust stroke. As mentioned, the cycle can be approximated using the Otto cycle. Referring to Figure 2.2, the intake stroke starts from TDC at point 1. The intake valve opens and allows for air fuel mixture. Piston moves to BDC at point 2 and compression stroke starts. During the compression stroke, the piston moves from BDC at point 2 to TDC which is at point 3, while compressing the air in the combustion chamber. Point 3 to point 4 is the combustion process where the spark plug ignites and burn the compressed air fuel mixture in the combustion chamber. After the combustion process, the piston moves to BDC at point 5 by the force produced from the combustion. This is the power stroke. Point 5 to 6 is the blowdown process. The exhaust valve start to open at point 5 to allow the exhaust gas out of the combustion chamber as piston moves from BDC to TDC.

2.2 Air Fuel Ratio and Engine Requirement

A mixture that will produce a *complete combustion* is called stoichiometric ratio. The stoichiometric ratio is 14.7:1 or simply 14.7. Term lean mixture is used when fuel is less than the air while term rich mixture is used to describe mixture that have more fuel than that air. When lean, the value of air fuel ratio is more than 14.7 and for rich mixture, the air fuel ratio value is less than 14.7.

Engine air fuel ratio must meet the requirement at its respective operating condition to get the desired engine performance, fuel consumption, drivability and emission. Different engine operating condition acquires different amount of mixture. Figure 2.3 illustrates typical air fuel ratio required by an engine at different operating condition.

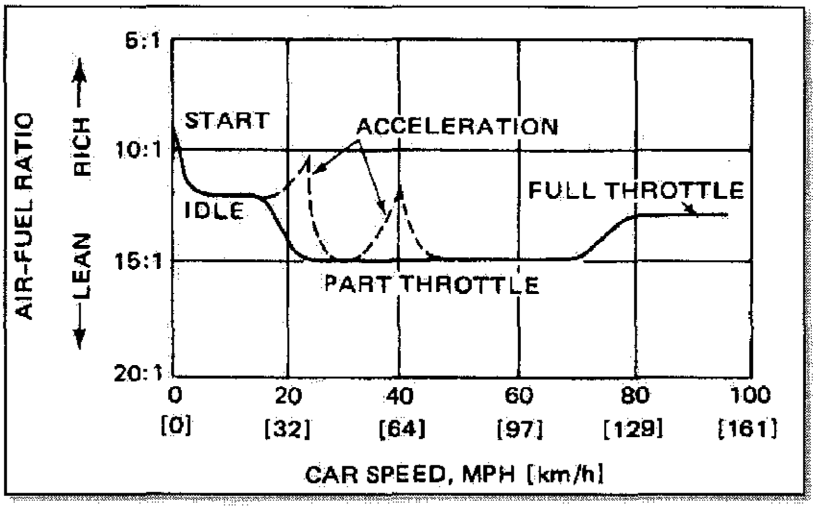


Figure 2.3: Typical air-fuel ratios required by an engine under different operating conditions (Crouse, 1993)

From Figure 2.3 when starting cold, the mixture is rich about 9:1. During idle, the mixture leans out to about 12:1. At medium speed, it leans out to around 15:1. If the driver steps on the gas to accelerate, the mixture is temporarily enriched as shown by the dashed lines. The mixture is also enriched at full throttle. The purpose of varying the air-fuel ratio is so that a combustible mixture always reaches the engine cylinders (Crouse, 1993).

2.3 Air and Fuel Induction

Three elements that are required for combustion in an engine are the air, fuel and ignition. The more the air, the more the fuel that can be burned; the more the energy that can be produced and be converted to power output.

Volumetric Efficiency is defined as

$$\eta_v = \frac{m_a}{\rho_a V_d} \quad \dots (1)$$

$$\eta_v = \frac{n\dot{m}_a}{\rho_a V_d N} \quad \dots (2)$$

Where

$m_a = mass\ of\ air\ into\ the\ engine\ for\ one\ cycle$

$\dot{m}_a = steady\ state\ flow\ of\ air\ into\ the\ engine$

$\rho_a = air\ density\ evaluated\ at\ atmospheric\ condition\ outside\ the\ engine$

$V_d = displacement\ volume$

$N = engine\ speed$

$n = number\ of\ revolution\ per\ cyle$

Typical value of volumetric efficiency at WOT is 75% to 90% and it goes down to much lower value as the throttle is closed. Restricting air flow causes the volumetric efficiency to be reduced. The example of restriction is closing the throttle and air cleaner.

The engine torque,

$$\tau = \frac{\eta_f \eta_v V_d Q_{HV} \rho_a (FA)}{2\pi n} \quad \ldots (3)$$

Where

$\eta_f = fuel\ conversion\ efficiency$

$\eta_v = volumetric\ efficiency$

$V_d = displacement\ volume$

$Q_{HV} = heating\ value\ of\ fuel$

$FA = fuel\ air\ ratio$

The engine power,

$$\dot{W}_b = \frac{\dot{m}_f}{bsfc} = \frac{(FA)\dot{m}_a}{(bsfc)} \quad \ldots (4)$$

$$\dot{W}_b = \frac{\eta_f \eta_v N V_d Q_{HV} \rho_a (FA)}{n} \quad \ldots (5)$$

Where

$bsfc = brake\ specific\ fuel\ consumption$

Both the engine torque and engine power equations above show that torque and power are directly proportional to volumetric efficiency. Volumetric efficiency increased, torque and power will increase.

For naturally aspirated engine, the volumetric efficiencies will always be less than 100%. This is due to the fuel vapor will displace some of the incoming air. Type of fuel, how and when it is supplemented will influence the value of volumetric efficiency. Carburetor adds fuel early in the intake flow and generally has low volumetric efficiency. Fuel will immediately start to evaporate and fuel vapor will displace incoming air. Fuel with high heat of evaporation will contribute to less loss of volumetric efficiency. The other factors that affect volumetric efficiency are intake temperature and fluid friction losses. When the intake temperature is high, it will improve the fuel mixing but it will sacrifice the volumetric efficiency. Intake bends, sharp corners, and flow restriction cause pressure losses and viscous drag. These will contribute to decrease in volumetric efficiency.

To have a good combustion, good fuel mixing and high volumetric efficiency is desirable. Good fuel mixing produces smaller fuel droplets. Poor fuel mixing leads to the unburned fuel in the combustion chamber. This unburned fuel will pass through piston ring and the unburned fuel can mix with the engine oil in the crankcase which is not desirable. The crucial part is that poor fuel mixing causes high fuel consumption. One of the methods to produce good fuel mixing is by roughing the surface of the intake manifold. It will increase the turbulence of the air fuel mixture and eventually produce better fuel mixing. However, it should be noted that better fuel mixing usually produces less volumetric efficiency since the fuel vapor has displaced the air in the mixture.

2.4 Carburetted Fuel System

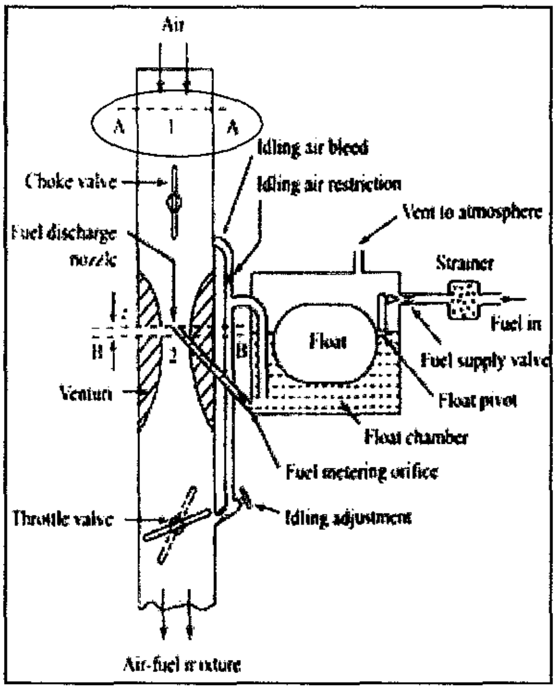


Figure 2.4: Simple Carburetor (Heywood, 1988)

Carburetor is a mixing device which supplies engine with a combustible air-fuel mixture. Figure 2.4 shows a schematic of a simple carburetor. Referring to Figure 2.4, the air flows from section A-A into the carburetor venturi. This is where the air velocity increases and pressure decreases. Fuel flows from fuel metering orifice as a result of pressure difference between the float chamber and the venturi throat.

It flows through fuel discharge nozzle to venturi throat where the air stream atomizes the liquid fuel. The fuel air mixture flows to the diverging section of the venturi where the flow decelerates and some pressure recovery occurs. The flows then pass through the throttle valve and enter the intake manifold.

The AFR delivered by the carburetor is given by

$$\frac{A}{F} = \frac{\dot{m}_a}{\dot{m}_f} = \frac{C_{DT}}{C_{DO}} . \frac{A_T}{A_O} . \left(\frac{\rho_{ao}}{\rho_f}\right)^{1/2} . \left(\frac{\Delta p_a}{p_a - p_f g z}\right)^{1/2} . \Phi \qquad \dots (6)$$

Where

$$\Phi = \left[\left(\frac{\gamma}{\gamma - 1}\right) . \frac{\left(p_T/p_o\right)^{2/\gamma} - \left(p_T/p_o\right)^{\gamma - 1/\gamma}}{1 - \left(p_T/p_o\right)}\right]^{1/2} \qquad \dots (7)$$

$\dot{m}_a = air\ mass\ flow\ rate$

$\dot{m}_f = fuel\ mass\ flow\ rate$

$C_{DT} = throat\ discharge\ coefficient$

$C_{Do} = throat\ discharge\ coefficient$

$A_T = throat\ area$

$A_o = Section\ A - A$

$\rho_{ao} = density\ of\ air\ at\ A - A$

$\rho_f = fuel\ density$

$p_T = pressure\ at\ throat$

$p_o = pressure\ at\ throat$

$\Delta p_a = p_o - p_T$

$g = gravitational\ acceleration$

$z = height\ difference\ between\ fuel\ discharge\ nozzle\ end\ and\ fuel\ level\ in\ float\ chamber\ see$ Figure 2.4

$\gamma =$ Speci ic heat ratio $^{C_p}/_{C_v}$

Φ is a flow compressible function where it accounts for the effect of compressibility. Appendix A shows the effect of Φ as a function of pressure drop. For normal carburetor operating range, where $\Delta^{p_a}/_{p_o} \leq 0.1$ the effect of compressibility which reduce the Φ below 1.0 is small.

A_o, A_T, ρ_f and ρ_{ao} are all constant for given carburetor, fuel and ambient condition. However, the discharge coefficient varies with flow rates. Hence, the AFR delivered by the carburetor is not constant. Figure 2.5 shows the performance of carburetor. Note that, when $\Delta p_a \leq \rho_f g h$ there will be no fuel flow. When $\Delta p_a > \rho_f g h$, fuel starts to flow more rapidly than air flow. Carburetor delivers a mixture of increasing fuel to air ratio as the flow rate increase.

In summary, some of the carburetted deficiencies are the mixture cannot be enriched during the startup and warm up, cannot adjust to changes in ambient air density which is due to primarily change in altitude, and as the air flow approaches the maximum wide open throttle, the equivalent ratio remains essentially constant. This can be seen in Figure 2.5. The mixture should increase to 1.1 or greater to provide maximum engine power.

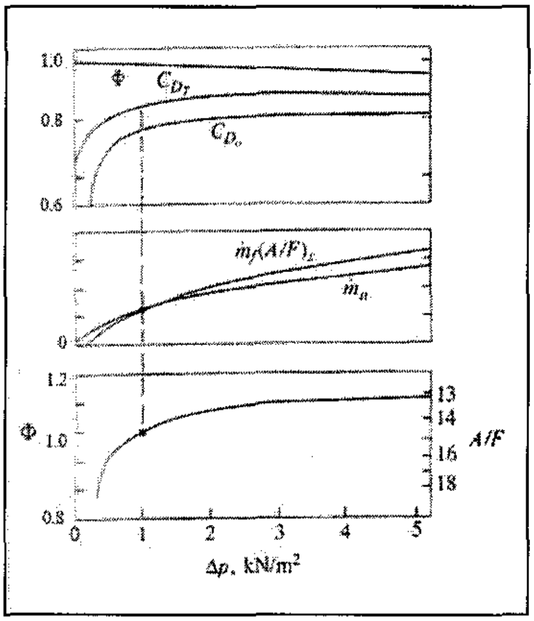


Figure 2.5 : Performance of Carburetor (Heywood, 1988)

2.5 Electronic Fuel Injection

The fuel injection system is controlled by the Engine Control Unit (ECU). The fuel injector spays fuel to be mixed with air in the intake system. The amount of fuel injected by the injector and the time it sprays the fuel is controlled by the ECU. Shown in Figure 2.6 is the simplified EFI system. Sensors for the measurement of engine speed, throttle position, manifold vacuum, coolant temperature, air intake temperature, and oxygen works as an input to the Electronic Control Module. The ECM then determines the amount of fuel needed and open the injectors to produce the desired air fuel ratio. The ECM also determine how long and when to open the injectors.

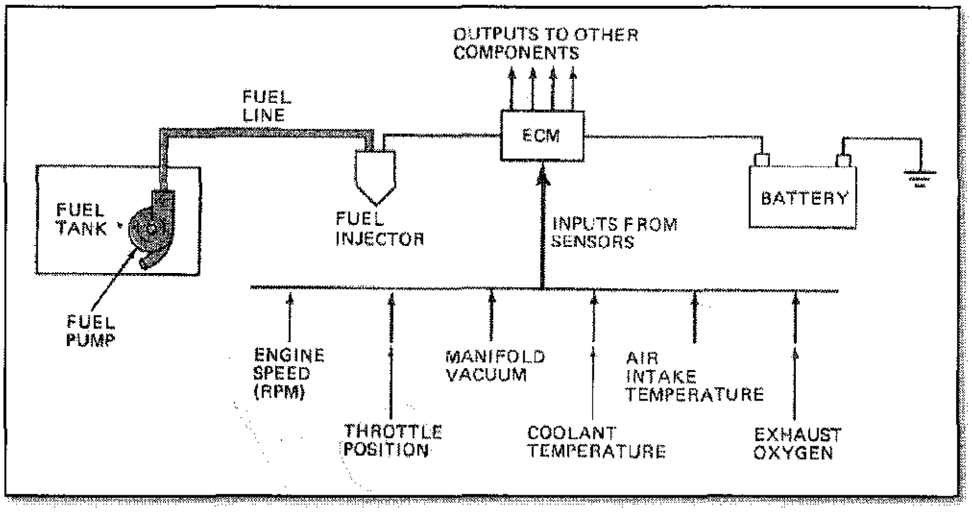


Figure 2.6: Simplified Electronic Fuel Injection System (Crouse, 1993)

The Electronic fuel injection system supplies the engine with a combustible air fuel mixture. It varies the richness of mixture to suit different operating conditions. When a cold engine starts, the fuel system delivers a very rich mixture. This has a high proportion of fuel. After the engine warms up the fuel system leans out the mixture. It then has a lower proportion of fuel. For acceleration and high speed operation, the mixture is again, enriched (Crouse, 1993).

With EFI, there's no venturi throat to create pressure drop as with a carburetor system. Because of little and no air fuel mixing occurs in the intake manifold, high velocity is not as important, and larger diameter runners with less pressure loss can be used. There is also no displacement of incoming air with fuel vapor in the manifold (Pulkrabek, 1997).

Fuel injection is able to atomize fuel better than carburetor and thus improving fuel economy. Just as performance can be changed with a few simple environmental changes, so can the emissions output. Carburetors are tuned with screws, where EFI is calibrated digitally. It is difficult for manufacturers to guarantee emissions compliance from mechanical devices that could easily be altered in the field. EFI can compensate for engine wear and degradation over time, where mechanical parts cannot (Electrojet, Driving Technology, 2008).

2.6 GT POWER

GT POWER is one of the engine simulation tools used by leading engine and vehicle makers all around the world and is recognized as industry standard. It's part of GT Suite, a computer-aided engineering (CAE) tools developed by Gamma Technologies, Inc. to address engine and powertrain design. By using GT POWER, user can model advance concept. It is design in both steady state and transient simulation. For this project, the simulation will run with the steady state simulation.

Among several of its applications are torque curve and fuel consumption, transient performance and response, valve profile and timing optimization and real time engine modeling. Fast engine model can be created from the Detailed GT POWER Model to have a lower simulation time.

Outputs of GT POWER include time variation quantities such as flow rates, flow velocities, pressure and temperature in system, engine volumetric efficiencies, power and torque. Some of the applications of GT POWER are manifold design and tuning, combustion analysis, EGR system performance, and valve profile and timing optimization. Other example of engine simulation software is WAVE which is produced by RICARDO.

2.7 Computer Modeling and Simulation of Four Stroke Engine

Typically, the aim of the engine design is to achieve its target performance characteristic for the application required. One needs to understand thoroughly on the filling of combustion chamber with air, emptying the exhaust gas from it, and the combustion that happen inside the combustion chamber of an engine. Real geometry of an engine and measured test data from actual engine will be produced to illustrate a design point being made. Pragmatic approach is desirable to simulate the unsteady gas dynamic and thermodynamic within the engine and the physical geometry of that engine should be defined to the finest detail, starting from where air enters the engine initially to where the exhaust gas finally exits from the engine.

Computer modeling and simulation brings all theoretical model together and solve it on a digital computer. Sub model of engine i.e. intake manifold, cylinder, is brought together to illustrate the effectiveness of a complete simulation of the four stroke engine (Blair, 1999).

The most useful aspect of engine simulation is that it allows imagining the unimaginable. Outputs are in forms of numbers and graphs and they will then be used to conduct analysis. By analyzing the output, one could comprehend the ramification of altering the design variables on an engine such as compression ratio, valve timing manifold tuning etc.

CHAPTER 3
METHODOLOGY

3.1 Research Methodology

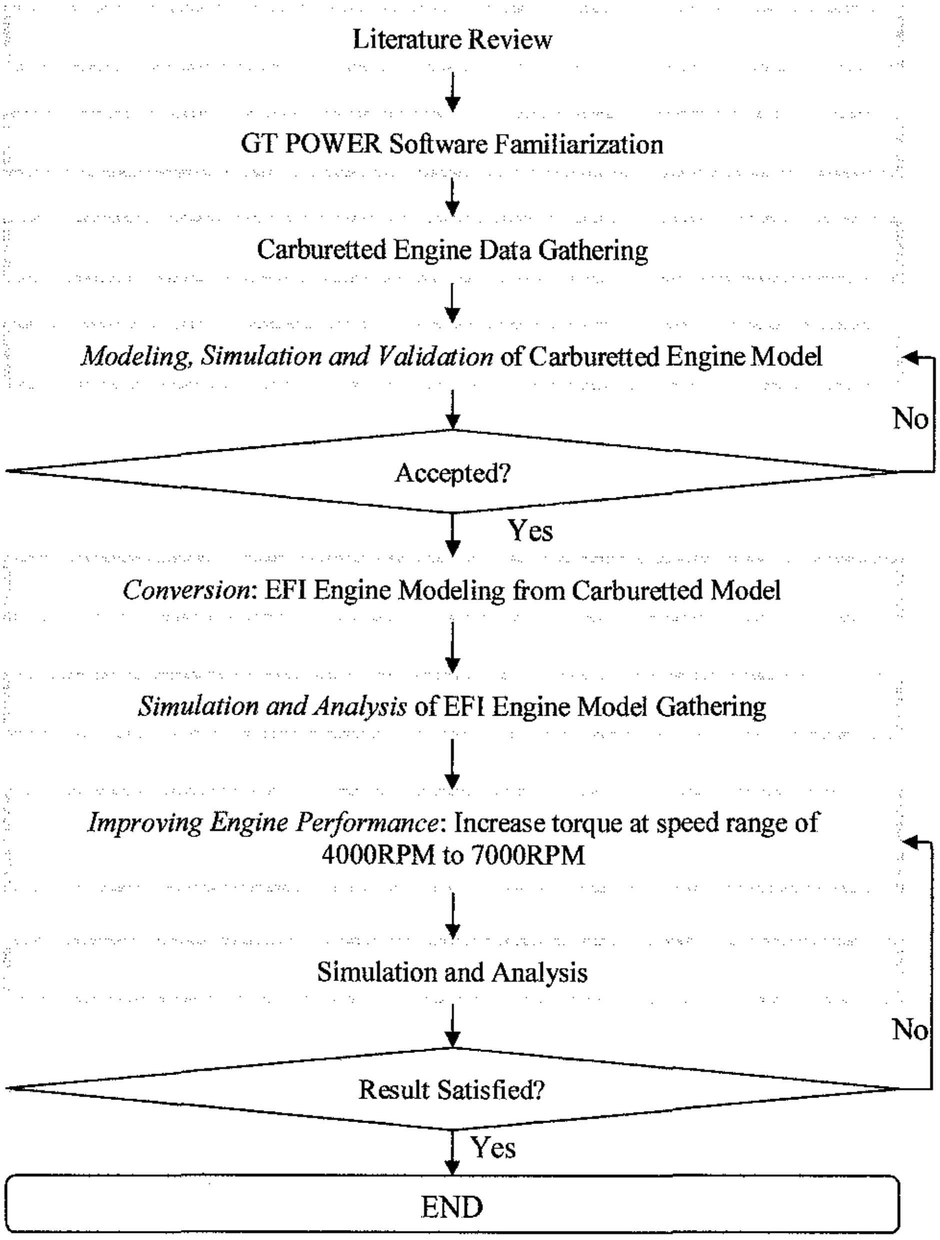


Figure 3.1: Research Methodology

Shown in Figure 3.1 is the flow chart on how to achieve the objective of FYP. Below are the explanations.

Literature Review: study and get knowledge on Internal Combustion Engine and GT POWER

GT POWER Software Familiarization: software familiarization was done by learning how to model and simulates a basic single cylinder four stroke engine. The simulation run in this project is a steady state simulation.

Carburetted Engine Data Gathering: The data will be obtained from the project collaborator, POWERTRAIN Technology, PETRONAS Research Sdn Bhd (PRSB). Data gathered to model and validate the engine model are:

- *Engine parameters,* such as dimension of engine parts, lift arrays and flow arrays of the valves.
- *Previous simulation data and testing data* such as torque curve and power curve. This is important in validating the model simulation.

Modeling, Simulation and Validation of Carburetted Model: the gathered data collected in Collaborator Company is used to model and validate the carburetted model. This is a cyclic process and model is refined until the carburetted model is satisfied.

Shown in Figure 3.2 is the model diagram of the engine GT POWER.

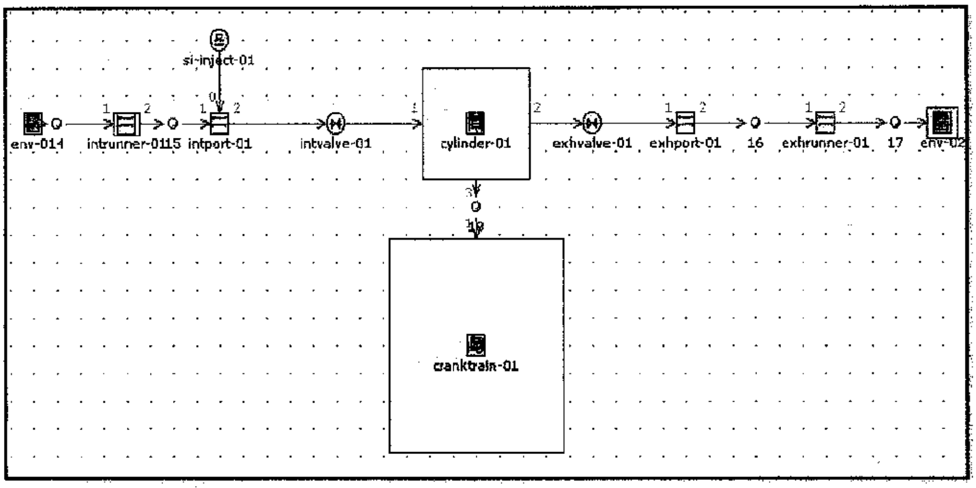


Figure 3.2: Engine Model Diagram in GT POWER

Assumptions made for the model are:

- Piston modeled as mass less (inertia value is ignored)
- The model did not account for the effect of swirl and tumble
- For simplicity, the intake runner, intake port, exhaust runner and exhaust port are being modeled as a straight pipe.
- Restriction in the intake and exhaust system is neglected.

The model is then validated with the previous testing and simulation data which are obtained by the collaborator company.

Conversion of Carburetted Model to EFI Model: the conversion in this project implies to the change of the fuel supply system of the engine. Carburetor part in the model diagram is replaced with injector. However, in reality, one needs to replace the carburetor with the fuel injector and the intake pipe because there is no injector port in the intake pipe for carburetted fuel system. Location of the injector on the intake pipe will be specified in the injector part in the model.

Simulation and Analysis of the EFI Model: the converted engine model, EFI will then be simulated and the result of simulation will be analyzed. Result difference of carburetted model and EFI model will be measured and analyzed.

Improving Engine Performance: Increase torque at speed range of 4000RPM to 7000RPM: Tuning of AFR and intake length will be done to improve the torque at the specified speed range.

3.2 Tools

The tool used to model the engine is GT POWER Software.

3.3 Project Timeline

Shown in Table 3.1 and Table 3.2 is the project timeline of phase 1 and phase 2.

Table 3.1: Project Timeline of Phase 1

No	Detail/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Selection of Project Topic														
2	GT POWER Software Familiarization														
3	Submission of preliminary report														
4	Carburetted K200 engine data gathering														
5	Literature review and methodology writing														
6	Modeling of K200 engine in GT POWER Software														
7	Simulation of the model														
8	Comparison and Validation of simulation result with previous testing data														
9	Seminar 1														
10	Submission of Progress Report														
12	Submission of Interim Final Report Draft														
13	Oral Presentation														

Table 3.2: Project Timeline of Phase 2

No	Detail/Week	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
1	Conversion of the carburetted model to EFI model																
2	Simulation and analysis of EFI model																
3	Submission of progress report 1																
4	Engine parameter change to improve EFI performance																
5	Simulation and analysis																
6	Submission of progress report 2																
7	Seminar (compulsory)																
8	Poster submission																
9	Submission of dissertation draft																
10	Oral presentation																
11	Submission of project dissertation																

Mid semester Break

CHAPTER 4
RESULTS AND DISCUSSIONS

The results are obtained from the simulated model of the single cylinder engine. The simulation was executed throughout the engine speed range of 3000RPM to 10000RPM. Comparison is being made for the validation of the model and comparison between the carburetted model and EFI model. The nomenclature of the graphs shown in this chapter can be obtained in appendix B.

4.1 Modeling, Simulation and Validation of Carburetted Model

4.1.1 Results

Modeling simulation and analysis are done on carburetted engine model using GT POWER. Comparison are being made on the torque, power, volumetric efficiency and BSFC as shown in Figure 4.1, Figure 4.2, Figure 4.3, and Figure 4.4 respectively for the purpose of validation with the previous WAVE simulation and testing data.

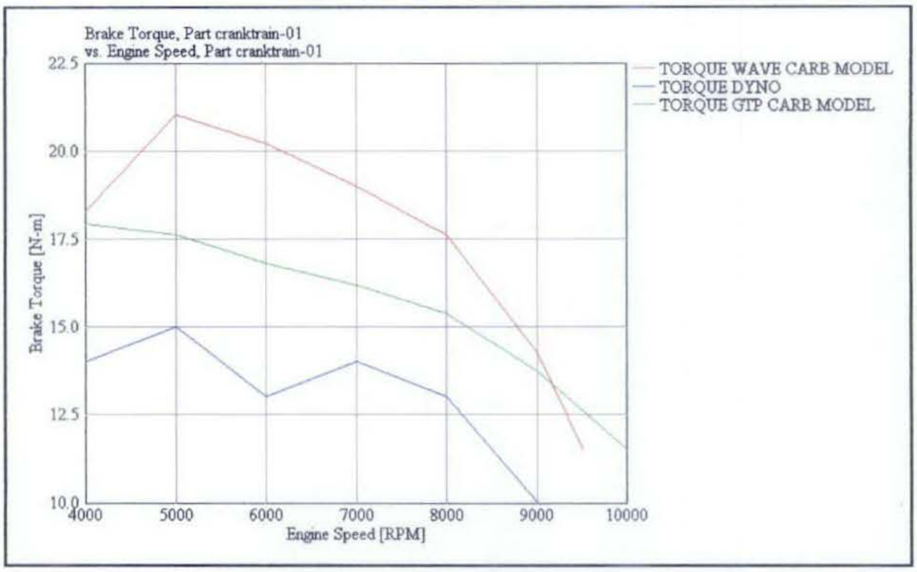


Figure 4.1 : Torque Validation

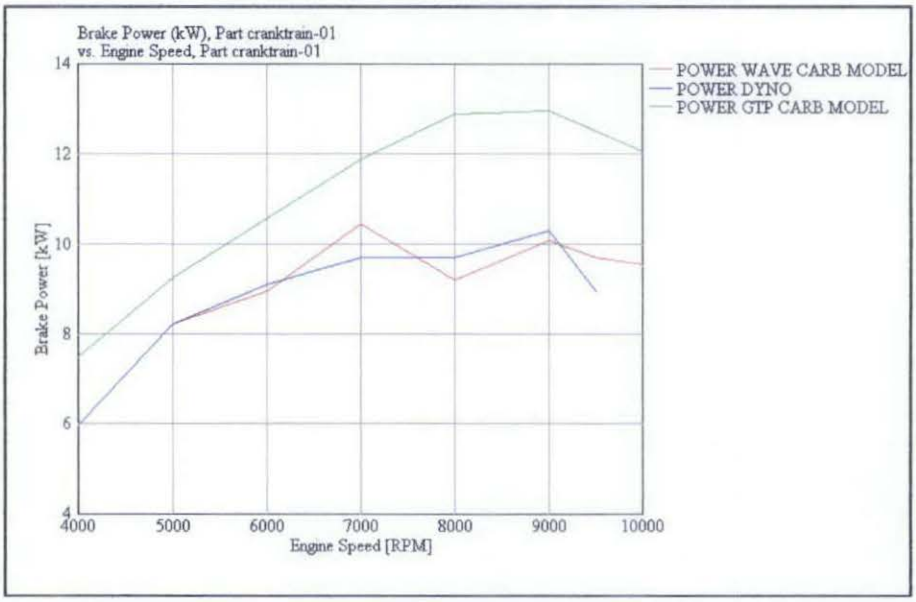


Figure 4.2 : Power Validation

Figure 4.3 : Volumetric Efficiency Validation

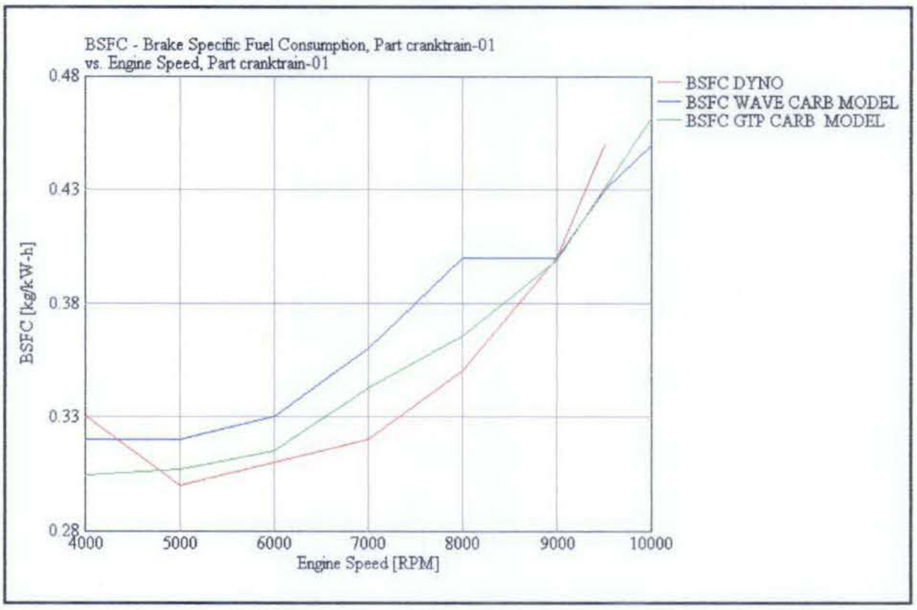


Figure 4.4 : Brake Specific Fuel Consumption Validation

From the results, it can be seen that the simulation results are in between the previous dynamometer and WAVE result except for the power curve. For the torque curve, GT POWER simulations show decreasing trend all the way through from 4000RPM to 10000RPM. The trend follows the WAVE simulation above engine speed of 5000RPM.

So does the previous dynamometer result, except for speed range 6000RPM to 7000RPM where the torque of the dynamometer increases a bit while the GT POWER simulation result decreases.

For the power curve result, the GT POWER results shoot up an average of 25% from both previous dynamometer and WAVE result. However, the result follows the trend of the previous WAVE simulation and testing result.

Looking at the volumetric efficiency graph, the results shows typical volumetric efficiency graph and the range is about 0.8 to 1 which is also a typical value for a naturally aspirated engine. Again the results is in between both the WAVE and dynamometer result. It does follow the trend of both previous results where it starts to increase from 4000RPM to 5000RPM and decrease from 5000RPM onwards.

From the BSFC, it can be seen that the GT POWER simulation results are nearly the average of both testing and WAVE result.

4.1.2 Discussion

As can be seen in the validation result in Figure 4.1, Figure 4.2, Figure 4.3, and Figure 4.4, the simulation results varies with the previous simulation and testing result in terms of the values and the trending in certain speed range. As for example, the trend in dyno torque (Figure 4.1) at speed range of 5000RPM is different from the GT POWER result. However, the trending in previous simulation result (WAVE simulation) is similar to the current simulation result (GT POWER simulation). All these errors are probably due to the assumptions, limitations and unavailability of certain data.

As mentioned, assumption has been made to develop the simulation model. The intake system used carburetor for fuel supply and this cannot be modeled accurately. There was also no input from carburetor to correlate the volume of fuel being supplied with respect to the pressure inside the venturi.

Therefore, the values of AFR through the speed range at wide open throttle are being assumed so that the result which can be seen in Figure 4.1, Figure 4.2, Figure 4.3 and Figure 4.4, will meet the previous testing and simulation data.

Shown in Table 4.1 are the values of AFR being assumed in the GT POWER model. The exhaust pipe, intake port, and exhaust port are modeled as straight pipes. Restrictions in the airflow on the intake side including air filter, throttle body, plenum and exhaust system including exhaust header, the catalyst converter, and the mufflers are also neglected. As mentioned in section 3.1 , the other assumptions made are piston is modeled as mass less, and the effect of swirl and tumble are neglected.

During the previous engine dynamometer testing, there was no measurement of air flow being done. This could be one of the contributing factors to the errors between the simulation and testing result.

Other required parameters that are not available are obtained from database. Constant and some values used are through prediction based on typical value. For example, the FMEP (friction mean effective pressure), the values used in the model are typical value which is obtained in the model example of GT POWER since the testing data is not available.

4.2 Conversion to EFI Model, Simulation and Analysis

4.2.1 Results

Shown in Figure 4.5, Figure 4.6, Figure 4.7 and Figure 4.8 are the results upon the conversion from the carburetted model to the EFI model. For the conversion, the AFR values from the carburetted model are being incorporated in the EFI model. Shown in Table 4.1 are the AFR values over the speed ranges that has been taken from carburetor model and which are also being incorporated in injector part in EFI model. These values are for wide open throttle condition.

Table 4.1 : AFR Value throughout Engine Speed

Engine Speed	AFR
4000	11.8
5000	11.8
6000	11.8
7000	11.2
8000	11.0
9000	10.8
10000	10.3

Table 4.2 shows the initial condition of the EFI System, specifically the initial condition of the injector. As mentioned before, a few assumptions have been made. The injector delivery rate is assumed to be 5.5. This assumption has been made according to the database from GT POWER example. The value is taken from a 4 cylinder port injected gasoline engine. The vaporized fuel fraction is 0.3. According to the reference, this value is a typical value for port injected gasoline engine (GT-ISE version 6.1 Help Navigator).

Table 4.2: Initial Condition of EFI System in GT POWER

Parameter	Value
Injector delivery rate	5.5 g/s
Fuel ratio	As specified in Table 4.1
Injection timing angle (crank angle relative to the TDC firing)	360^0 (at the start of intake stroke)
Injector location	Middle of intake pipe
Vaporized fuel fraction	0.3

Figure 4.5, Figure 4.6, Figure 4.7 and Figure 4.8 shows the results after the conversion in terms of torque, power, volumetric efficiency and brake specific fuel consumption respectively. With the same air fuel ratio assumed in carburetor model, it is proven that EFI improves the performance of the engine. This can clearly be seen in the graphs in Figure 4.5, Figure 4.6, Figure 4.7, and Figure 4.8.

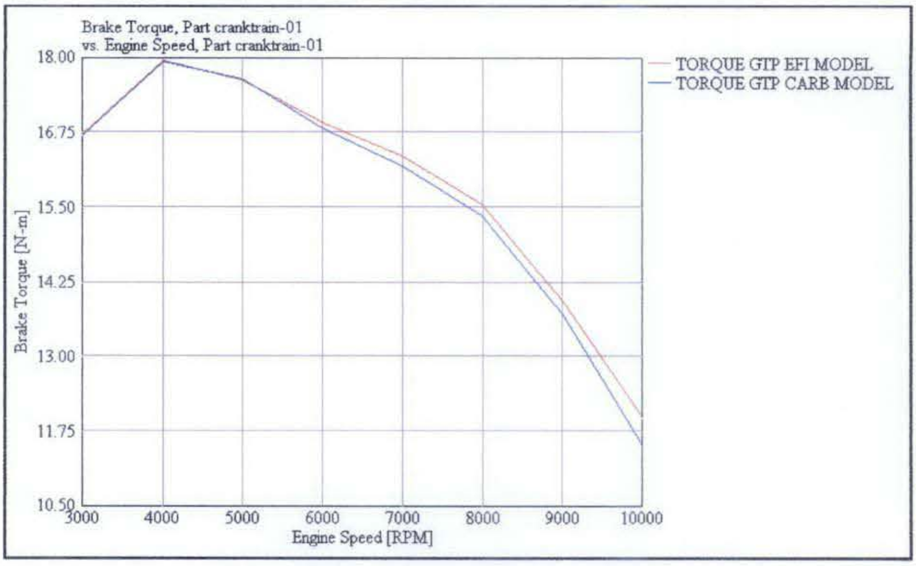


Figure 4.5 : Conversion-Torque Comparison

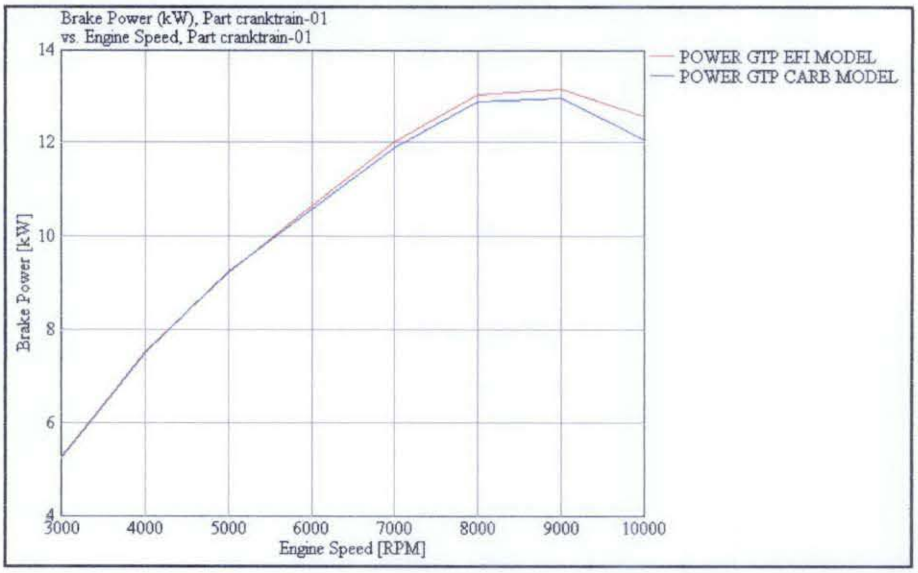


Figure 4.6 : Conversion-Power Comparison

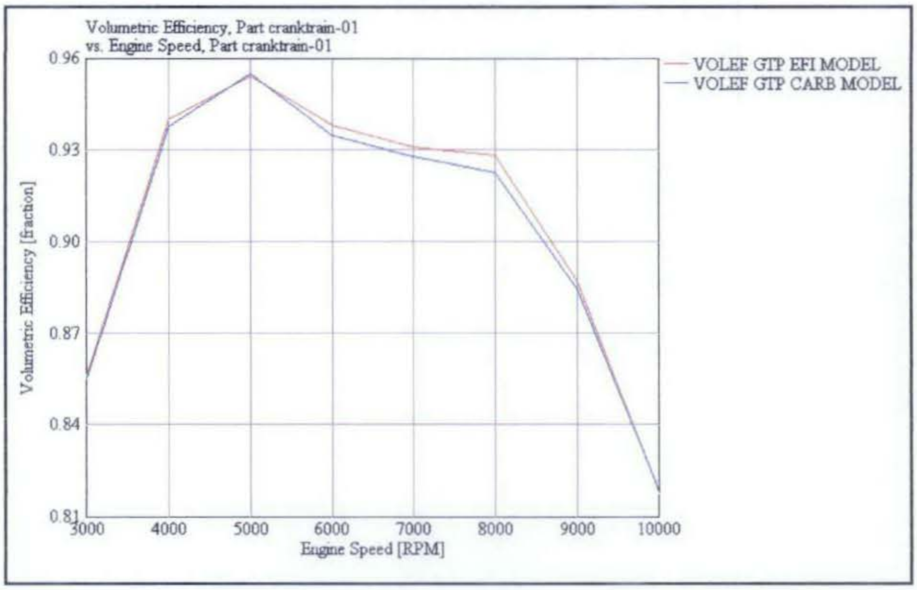


Figure 4.7 : Conversion-Volumetric Efficiency Comparison

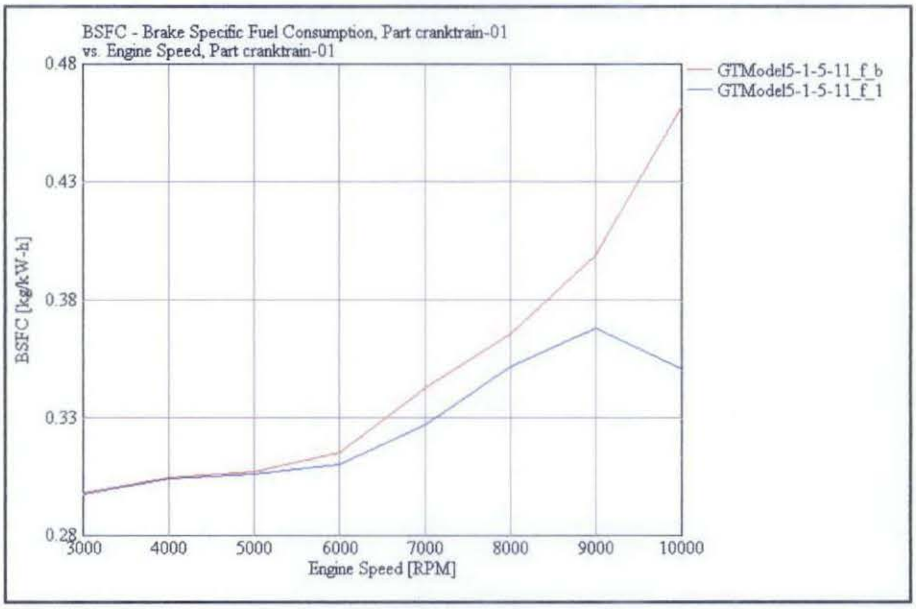


Figure 4.8 : Conversion-Brake Specific Fuel Consumption Comparison

4.2.2 Discussions

The difference between carburetor and EFI is that fuel supply (AFR) values of the carburetor are dependent on the ambient condition. This effect can be seen in the equation 6. The density of air changes with the change of pressure, temperature and altitude of the ambient. From equation 6, when the air density changes, the AFR value changes.

In EFI system, the AFR values are being controlled by the ECU (engine control unit). Calibrating the ECU allows one to set the amount of the fuel to be injected with respect to the engine speed and the engine load. Regardless of the ambient condition, the amount of the mixture delivered to the combustion chamber is governed by the value that has been set in the ECU according to the defined operating condition.

Basically, when the ECU recognizes the engine speed and the load respective sensors, it then calculates the amount of the fuel to be injected and convert the values into control unit current pulses which are then transmitted to the injector valve. The longer the time the valve opens the more amount of fuel to be injected.

It should be noted that the conversion result in this project are using the AFR value from the carburetor model. As mentioned before, with ECU one could calibrate this value to optimize the performance of the engine.

4.3 Improving Torque at Speed Range of 4000RPM to 7000RPM

4.3.1 Results

The reason behind this change is to improve the throttle responds during cornering, thus, improve the drivability of the vehicle. Appendix C shows the example of the relation between the engine speed versus time of a go kart track test. This appendix explained the issue of throttle responds during cornering, which also affects the drivability of the vehicle. One of the ways to counter this issue is by improving the torque at low speed range.

There are thousands of methods that can be done to improve the performance of an engine. For this project, the objective is to improve the torque at speed range of 4000RPM to 7000RPM as mentioned in section 1.3 . This is to improve the throttle response during cornering, thus, improving drivability of a vehicle. According to equation 3 in section 2.3 , the torque is directly proportional to the volumetric efficiency.

Increasing volumetric efficiency will increase torque. To increase the volumetric efficiency, there are two methods that have been studied to achieve the objective of this project. The methods are:

- AFR tuning with respect to the engine speed at WOT
- Varying intake length

- **AFR Tuning With Respect To The Engine Speed At WOT**

Shown in Table 4.3 is the tuned AFR value with respect to the engine speed at WOT.

Table 4.3: Tuned AFR Value with respect to Engine Speed at WOT

Engine Speed	Initial AFR	Tuned AFR
4000	*11.8*	*14*
5000	*11.8*	*14*
6000	*11.8*	*13*
7000	*11.2*	*13.5*
8000	11.0	11.0
9000	10.8	10.8
10000	10.3	10.3

Note that the tuned AFR values are at speed range of 4000RPM to 7000RPM. Figure 4.9, Figure 4.10, Figure 4.11, and Figure 4.12 shows the results after the conversion in terms of torque, power, volumetric efficiency and brake specific fuel consumption respectively.

Figure 4.9: AFR Tuning of EFI System-Power Comparison

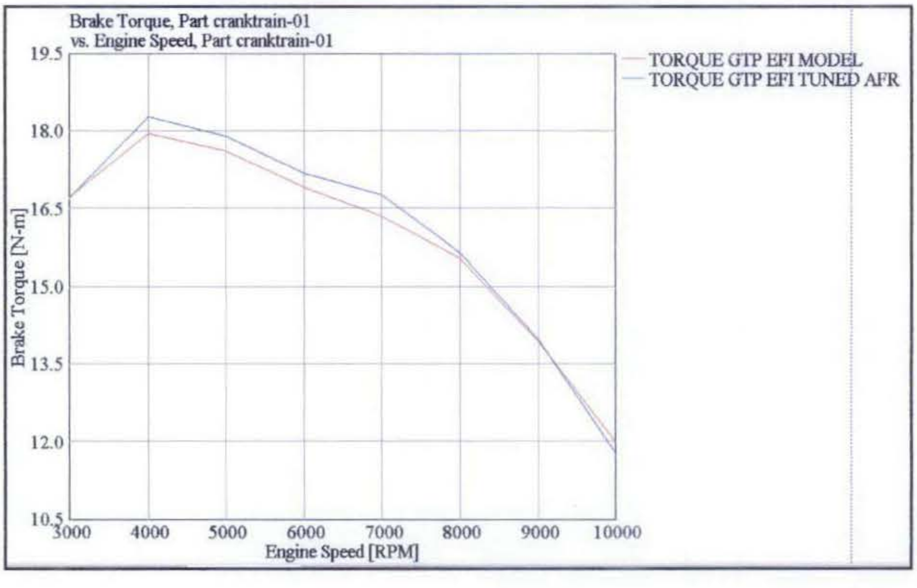


Figure 4.10: AFR Tuning of EFI System-Torque Comparison

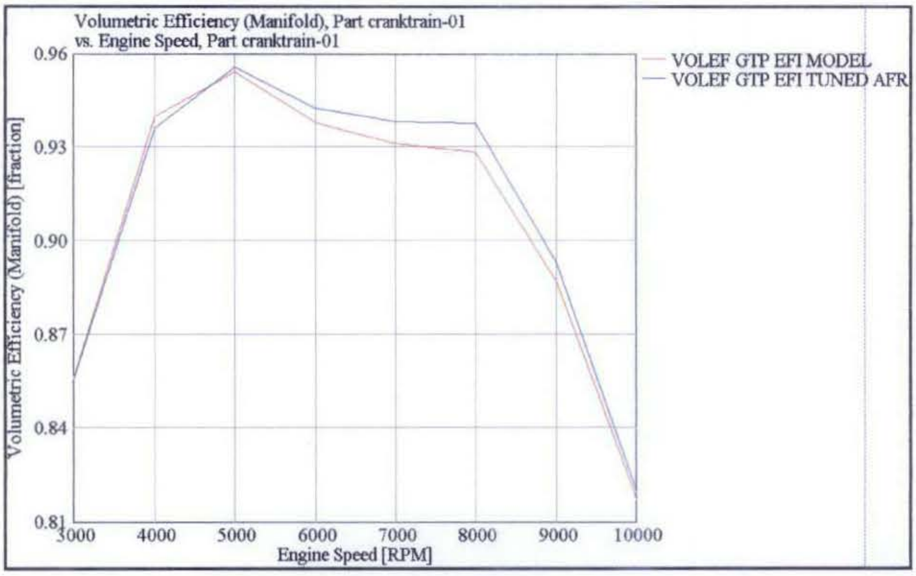


Figure 4.11: AFR Tuning of EFI System-Volumetric Efficiency Comparison

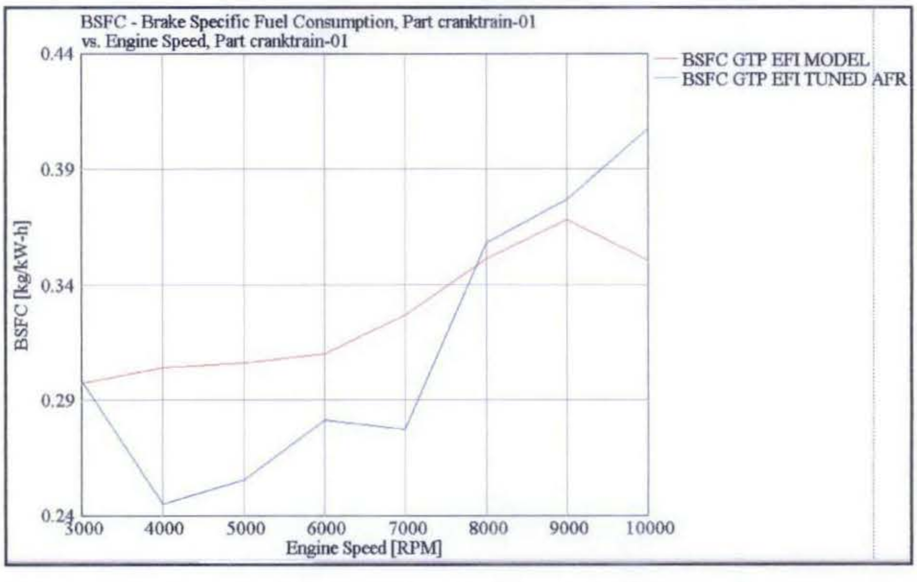


Figure 4.12: AFR Tuning of EFI System-Brake Specific Fuel Consumption Comparison

- **Varying intake length**

From the Tuned AFR Model, the intake length of the model is increased from 100% to 500% of its initial length as an intention to increase the volumetric efficiency at lower speed range. Table 4.4 shows the amount of percentage increase in intake lengths which are 100%, 200%, 300%, 400% and 500% from the original engine intake length. Analysis is being done on power, torque, volumetric efficiency and BSFC of all the models and the results can be seen in Figure 4.13, Figure 4.14, Figure 4.15 and Figure 4.16 respectively. Each of the models is being compared with the original intake length in terms of percentage difference. The percentage difference for each of the model (for torque, power and BSFC) can be seen in Table 4.4.

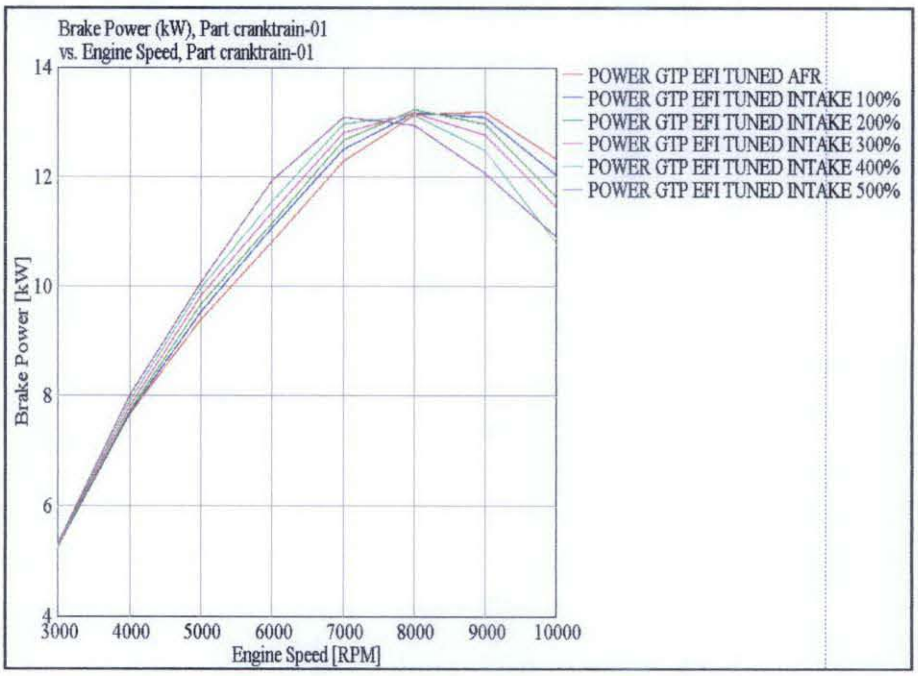


Figure 4.13: Intake Length Tuning of EFI System-Power Comparison

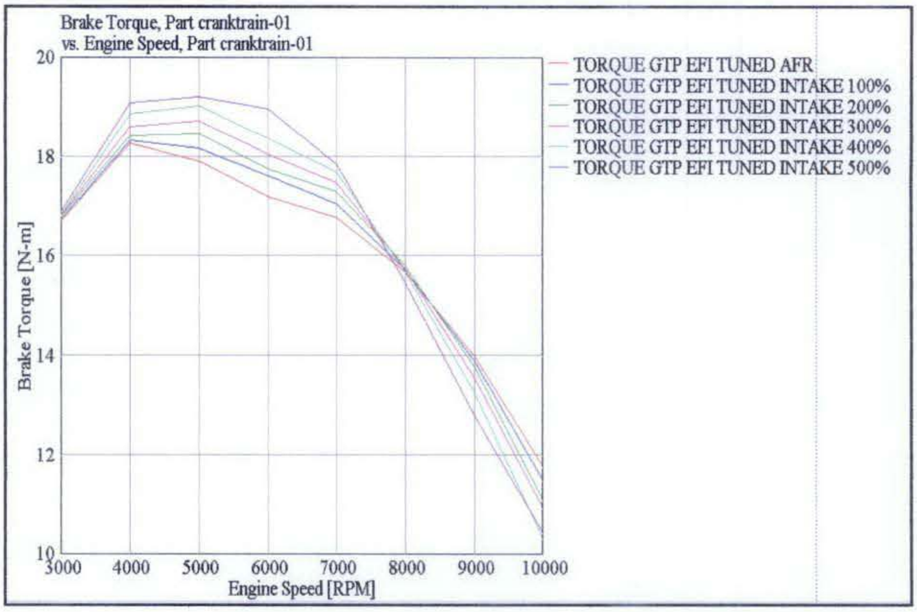


Figure 4.14: Intake Length Tuning of EFI System Torque Comparison

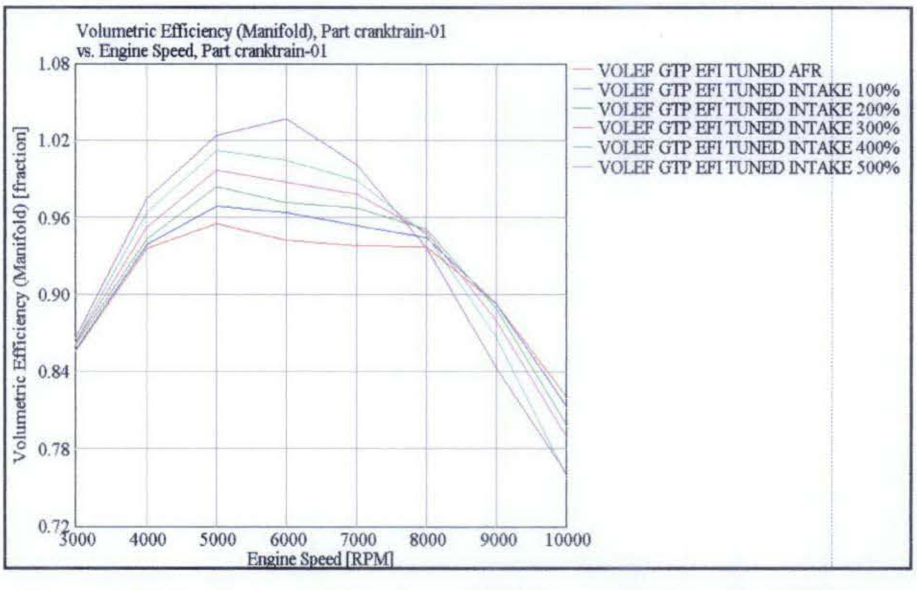


Figure 4.15: Intake Length Tuning of EFI System-Volumetric Efficiency Comparison

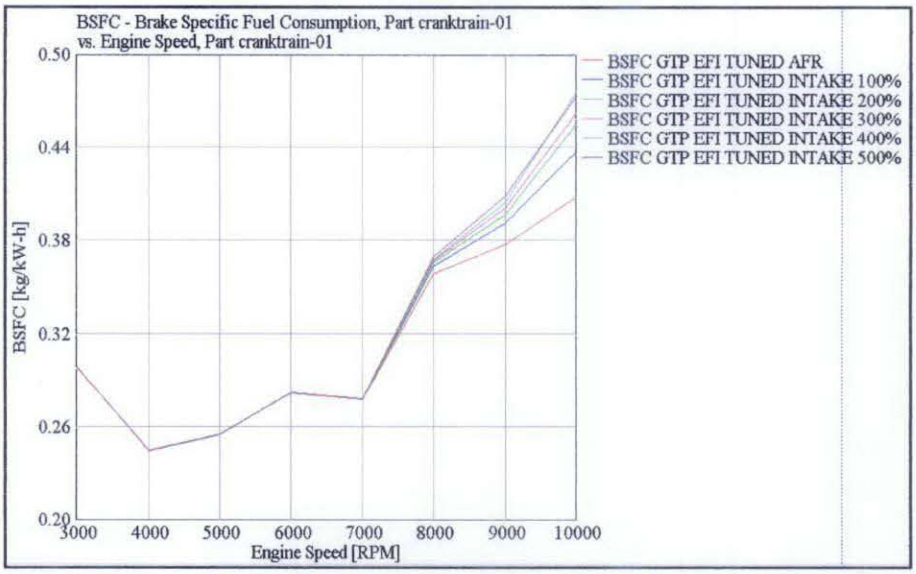


Figure 4.16: Intake Length Tuning of EFI System-Brake Specific Fuel Consumption Comparison

Table 4.4: Analysis on Intake Length Tuning

%Increase in Length	100%	200%	300%	400%	500%
Torque	3.198	4.423	6.381	4.952	8.557
Power	3.198	4.425	6.383	4.954	8.556
BSFC	12.503	19.973	22.252	26.459	27.425

4.3.2 Discussion

- **AFR Tuning With Respect To the Engine Speed at WOT**

The interesting part of EFI is that the amount of fuel can be tuned to optimize the engine performance. As mentioned, the target is to increase the volumetric efficiency and one way to increase volumetric efficiency is by tuning the AFR value.

Table 4.3 shows the tuned AFR value with respect to the engine speed at WOT condition. Notice that the tuned AFR is leaner than the initial AFR. In a lean mixture, provided that the fuel is injected at the same position and the surface roughness of the intake pipe is the same, the volumetric efficiency is higher than that of with richer mixture. Figure 4.11 shows the volumetric efficiency result after the AFR has been changed in EFI system.

It can be seen that the volumetric efficiency increases after the AFR is tuned to the leaner value. According to the equation 1 in section 2.3 , the amount of air is directly proportional to the volumetric efficiency. Obviously, the amount of air in leaner mixture is more than the amount of air in richer mixture. Figure 4.10 justify that when the volumetric efficiency increases, the torque will also increase.

In a lean mixture, flame speed is slow and combustion lasts well past TDC. This keeps the pressure high well into the power stroke, which produces a greater power output. Equation 5 in section 2.3 justifies the result in Figure 4.9. Increasing the volumetric efficiency will also increase the power of the engine.

The BSFC of the engine is also improve (reduced) up until 7900 RPM. With the result, the objective in improving the torque at the speed range of 4000RPM to 7000RPM is achieved by tuning the AFR. On top of that, the results also conclude that it will improve the power and the BSFC up until 7900RPM. However, careful measure should be taken into account since the exhaust gas temperature increase when engine is running towards lean mixture.

- **Varying Intake Length**

Flow in the inlet and exhaust is unsteady due to periodic motion of piston and valve. This motion creates finite amplitude compression and refraction pressure wave that propagate at sonic velocity through intake and exhaust airflow. As engine speed increases, the frequency and amplitude of the pressure wave increase.

Exhaust and inlet manifold are sized and tuned to use the pressure wave to optimize the volumetric efficiency at a chosen engine speed. A tuned intake manifold will have a locally higher pressure when the intake valve is open, increasing the charge density. While a tuned exhaust manifold will have locally lower pressure when the exhaust valve is open, increasing the exhaust flow (Ferguson & Kirkpatrick, 2001).

Figure 4.15 shows the effect of intake runner length on the volumetric efficiency plotted as a function of engine speed. It can be seen that increasing the intake length will improve greatly the volumetric efficiency at lower speed range. However, it drops off sharply at the higher speed range. Note that in Figure 4.14 and Figure 4.15,the volumetric efficiency change also greatly affects the torque and power.

To choose the best length, analysis has been done by taking the percentage different of the model curve with a certain intake length increase relative to the curve with the original intake length at a certain speed range. Equation 8 shows the calculation of the percentage different for the torque curve.

At a certain speed,

$$percentage\ difference for\ x\%increase =$$

$$\Sigma\left[\frac{\text{Torque value with } x\% \text{ increase in intake length} - \text{torque value with original intake length}}{\text{torque value with original intake length}}\right] X\,100 \quad \ldots(8)$$

Where x = percent increase in intake length from original intake length (100%~500%)

The torque value for both original intake length and tuned intake length are obtained from the torque curve from the respective $x\%$ percent increase in length. Calculations are also made for the power and BSFC of all models with tuned intake length. The results can be seen in Table 4.4. The least percentage relative to the original intake length is desirable since the gain and loss can be said to be balanced.

From Table 4.4, 200% increase in intake length has the best result when comparing the percentage difference of torque, power and BSFC. The percentage differences for power and torque have only slight difference. The power, torque and BSFC for model with 300% and 500% increase in intake length has a very large percentage difference which is not desirable since loss in torque and power are very large at the high speed range. The 100% increase in intake length has the least percentage relative to the original intake length.

However, looking at Figure 4.14, the torque gain is not much although the gain at lower speed range is balanced with the loss in higher speed range. Take note that the objective is to improve the torque at speed range of 4000RPM to 7000RPM. The 200% increase in intake length is the best where the gain at the required speed range is enough and the loss at higher speed range is acceptable.

CHAPTER 5
CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

From the results, it is concluded that the results of the simulation model correlate with the previous testing and simulation result in terms of the trending. The errors are due to unavailability of other parameter which are being assumed and referred to database. Besides that, the testing results are limited and correlation with the testing result can hardly made.

In comparing the EFI and carburetted system, the EFI system produce better performance compared to the carburetted fuel system. The amount of fuel to be injected by the carburetor is affected by the ambient condition while the EFI system does not.

From the results stated in section 4.3 , torques at the lower speed range can be improved by improving the volumetric efficiency. Two methods that can be done are tuning the AFR and intake length.

5.2 Recommendation

For more accurate validation, it is recommended that the model should account for bends for the intake pipe, intake port, and exhaust pipe and exhaust port. Results then should be validated with other previous simulation and testing result instead of only comparing it with torque, power, volumetric efficiency and BSFC only. Testing should be done with more instrumentation such as mass air flow sensor so that measurement of the air flow can be done to obtain more accurate testing data.

In modeling the EFI system, further research should focus more on proper selection of fuel injector for the single cylinder 4 stroke engine. To validate the EFI model, further research should also concentrate on testing the engine with EFI system on the engine test bed.

It is recommended to further tune the AFR value so that the performance can be optimized. It is also recommended to run the simulation in transient state. The AFR value defined is recommended to be specified with respect to the engine speed and load.

REFERENCES

Technical Data

1. Technical Data Sheet of K200 engine

Books

2. Blair, G. P. (1999). *Design and Simulation of Four Stroke Engine.* U.S.A: SAE Publication Group.

3. Crouse, A. (1993). *Automotive Mechanics, Tenth Edition* (Tenth Edition ed.). Singapore: Mc Graw Hill.

4. Ferguson, C. R., & Kirkpatrick, A. T. (2001). *Internal Combustion Engine, Applied Thermosciences.* New York: John Wiley and Sons.

5. Ganesan, V. (2007). *Internal Combustion Engines.* New Delhi: Tata McGraw-Hill Publishing Company Limited.

6. Heywood, J. B. (1988). *Internal Combustion Engine Fundamental.* Singapore: Mc Graw Hill.

7. Pulkrabek, W. (1997). *Engineering Fundamental of the Internal Combustion Engine.* Republic of Singapore : Prentice Hall

8. TECHNOLOGIES, G. (AUGUST 2004). GT POWER USER'S MANUAL GT SUITE VERSION 6.1. USA: GTI.

Websites

9. Benson, T. (n.d.). *Ideal Otto Cycle: PV Diagram*. Retrieved October 20th, 2008, from National Aeronautic and Space Administration: http://www.google.com.my/imgres?imgurl=http://www.grc.nasa.gov/WWW/K-12/airplane/Images/otto.gif&imgrefurl=http://www.grc.nasa.gov/WWW/K-12/airplane/otto.html&h=531&w=710&sz=11&tbnid=Fy7fjbppdZQJ::&tbnh=105&tbnw=140&prev=/images%3Fq%3Dotto%2Bcycle&hl=en&u

10. *Electrojet, Driving Technology*. (2008, November 1). Retrieved March 16 , 2009, from FAQ: http://www.electrojet.org/faq.html

11. *GTISOFT*. (2007). Retrieved August 12, 2008, from Gamma Technologies: http://www.gtisoft.com/broch_gtpower.html

12. Integrity Automotive Misoula. (2006). *Integrity Automotive*. Retrieved August 12, 2008, from integrityautomissoula:
13. http://www.integrityautomissoula.com/fuel_injection.asp

14. Jenks, D. (2006, July 6th). *Mikuni Carburetor Operation and Tuning*. Retrieved September 11th, 2008, from Ian Williams Tuning Kartshop homepage: http://www.iwt.com.au/mikunicarb.htm

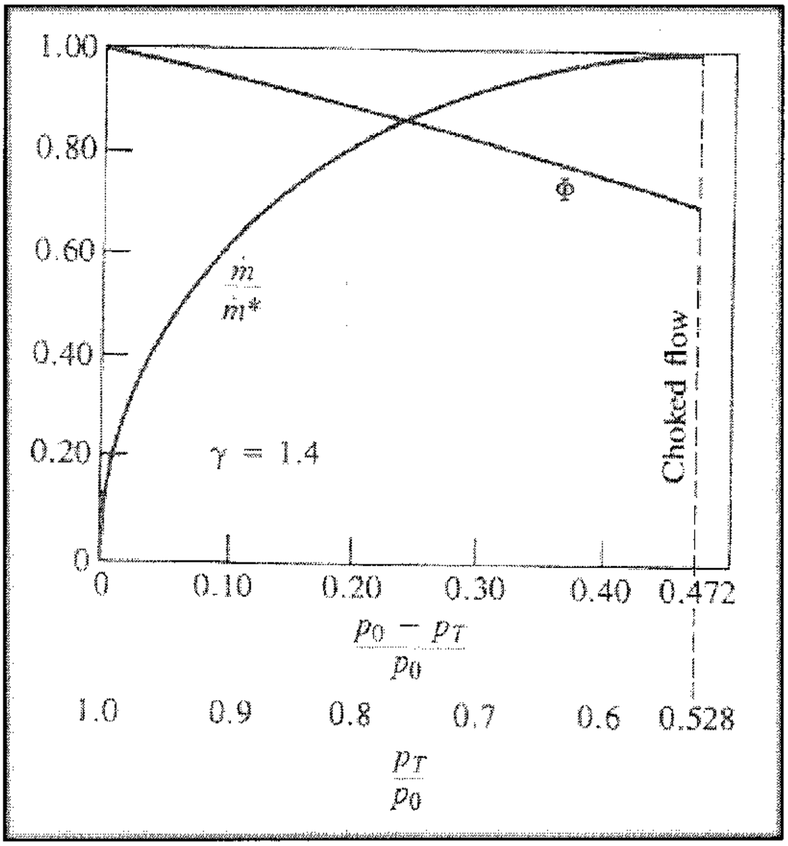
Help Navigator

15. GT Suite (GT-ISE) version 6.1 Help Navigator

16. TECHNOLOGIES, G. (AUGUST 2004). *GT POWER USER'S MANUAL GT SUITE VERSION 6.1*. USA: GTI.

APPENDIX A

Relative Mass Flow Rate and Compressible Flow Function Φ as a Function of Nozzle or Restriction Pressure Ratio For Ideal Gas With γ =1.4 (Heywood, 1988)



APPENDIX B

Nomenclature of Graphs

Result Name	Description
POWER WAVE CARB MODEL	Previous Simulation Power Curve (Carburetted System)
TORQUE WAVE CARB MODEL	Previous Simulation Torque Curve (Carburetted System)
VOLEF WAVE CARB MODEL	Previous Simulation Volumetric Efficiency Curve (Carburetted System)
BSFC WAVE CARB MODEL	Previous Simulation Break Specific Fuel Consumption (Carburetted System)
POWER DYNO	Previous Dynamometer Testing Power Curve
TORQUE DYNO	Previous Dynamometer Testing Torque Curve
VOLEF DYNO	Previous Dynamometer Testing Volumetric Efficiency Curve
BSFC DYNO	Previous Dynamometer Testing Break Specific Fuel Consumption
POWER GTP CARB MODEL	GT POWER Simulation Power Curve (Carburetted System)
TORQUE GTP CARB MODEL	GT POWER Torque Curve (Carburetted System)
VOLEF GTP CARB MODEL	GT POWER Volumetric Efficiency Curve (Carburetted System)
BSFC GTP CARB MODEL	GT POWER Break Specific Fuel Consumption (Carburetted System)
POWER GTP EFI MODEL	GT POWER Simulation Power Curve (EFI System)
TORQUE GTP EFI MODEL	GT POWER Torque Curve (EFI System)
VOLEF GTP EFI MODEL	GT POWER Volumetric Efficiency Curve (EFI System)

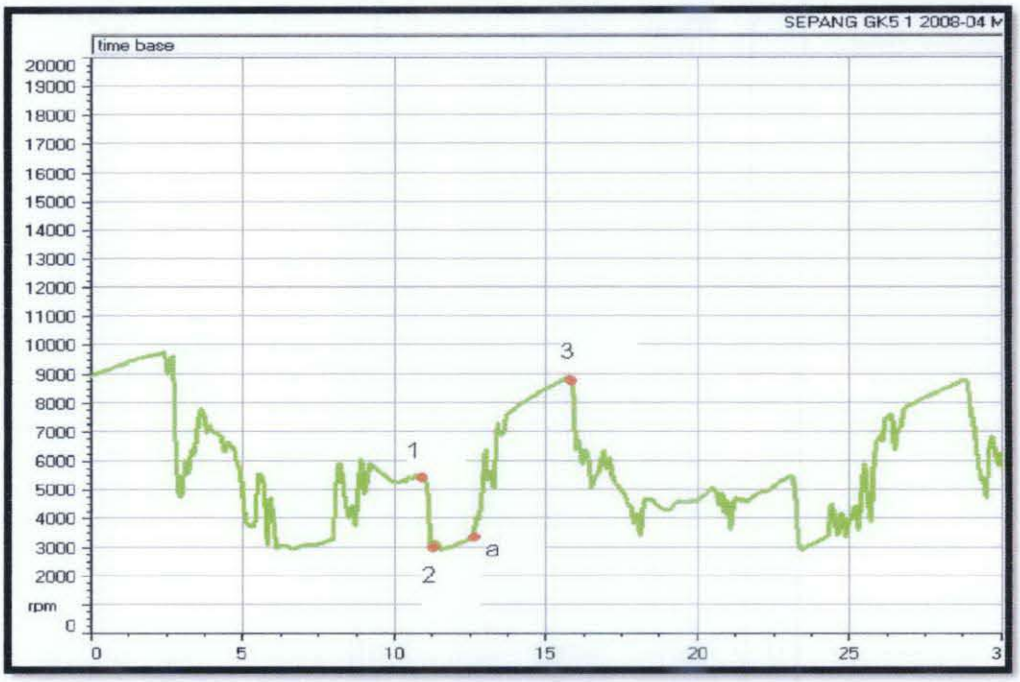
BSFC GTP EFI MODEL	GT POWER Break Specific Fuel Consumption (EFI System)
POWER GTP EFI TUNED AFR	GT POWER Simulation Power Curve (EFI System with tuned AFR)
TORQUE GTP EFI TUNED AFR	GT POWER Torque Curve (EFI System with tuned AFR)
VOLEF GTP EFI TUNED AFR	GT POWER Volumetric Efficiency Curve (EFI System with tuned AFR)
BSFC GTP EFI TUNED AFR	GT POWER Break Specific Fuel Consumption (EFI System with tuned AFR)
POWER GTP EFI TUNED INTAKE 100%	GT POWER Simulation Power Curve (EFI System with Tuned Intake-100% from Original Intake Length)
POWER GTP EFI TUNED INTAKE 200%	GT POWER Simulation Power Curve (EFI System with Tuned Intake-200% from Original Intake Length)
POWER GTP EFI TUNED INTAKE 300%	GT POWER Simulation Power Curve (EFI System with Tuned Intake-300% from Original Intake Length)
POWER GTP EFI TUNED INTAKE 400%	GT POWER Simulation Power Curve (EFI System with Tuned Intake-400% from Original Intake Length)
POWER GTP EFI TUNED INTAKE 500%	GT POWER Simulation Power Curve (EFI System with Tuned Intake-500% from Original Intake Length)
TORQUE GTP EFI TUNED INTAKE 100%	GT POWER Simulation Torque Curve (EFI System with Tuned Intake-100% from Original Intake Length)
TORQUE GTP EFI TUNED INTAKE 200%	GT POWER Simulation Torque Curve (EFI System with Tuned Intake-200% from Original Intake Length)

TORQUE GTP EFI TUNED INTAKE 300%	GT POWER Simulation Torque Curve (EFI System with Tuned Intake-300% from Original Intake Length)
TORQUE GTP EFI TUNED INTAKE 400%	GT POWER Simulation Torque Curve (EFI System with Tuned Intake-400% from Original Intake Length)
TORQUE GTP EFI TUNED INTAKE 500%	GT POWER Simulation Torque Curve (EFI System with Tuned Intake-500% from Original Intake Length)
VOLEF GTP EFI TUNED INTAKE 100%	GT POWER Simulation Volumetric Efficiency Curve (EFI System with Tuned Intake-100% from Original Intake Length)
VOLEF GTP EFI TUNED INTAKE 200%	GT POWER Simulation Volumetric Efficiency Curve (EFI System with Tuned Intake-200% from Original Intake Length)
VOLEF GTP EFI TUNED INTAKE 300%	GT POWER Simulation Volumetric Efficiency Curve (EFI System with Tuned Intake-300% from Original Intake Length)
VOLEF GTP EFI TUNED INTAKE 400%	GT POWER Simulation Volumetric Efficiency Curve (EFI System with Tuned Intake-400% from Original Intake Length)
VOLEF GTP EFI TUNED INTAKE 500%	GT POWER Simulation Volumetric Efficiency Curve (EFI System with Tuned Intake-500% from Original Intake Length)
BSFC GTP EFI TUNED INTAKE 100%	GT POWER Simulation Brake Specific Fuel Consumption Curve (EFI System with Tuned Intake-100% from Original Intake Length)
BSFC GTP EFI TUNED INTAKE 200%	GT POWER Simulation Brake Specific Fuel Consumption Curve (EFI System with Tuned Intake-200% from Original Intake Length)
BSFC GTP EFI TUNED INTAKE 300%	GT POWER Simulation Brake Specific Fuel Consumption Curve (EFI System with Tuned Intake-300% from Original Intake Length)

BSFC GTP EFI TUNED INTAKE 400%	GT POWER Simulation Brake Specific Fuel Consumption Curve (EFI System with Tuned Intake-400% from Original Intake Length)
BSFC GTP EFI TUNED INTAKE 500%	GT POWER Simulation Brake Specific Fuel Consumption Curve (EFI System with Tuned Intake-500% from Original Intake Length)

APPENDIX C

Carburetted Engine Test Track Result



The figure above is a track test result of engine speed versus time of a 4 stroke single cylinder *carburetted engine*. From point 1 to point 2, the driver enters the corner. Driver presses the brake and the engine speed drops from 5500RPM to 3000RPM. Point 2 to point 3 is a straight line track. The driver again fully presses the throttle. It can be seen that the engine speed did not smoothly went up from point 2 to point 3 (3000RPM to 9000RPM). Instead, it 'stuck' at point a (3500 RPM) instead of rev up smoothly to 9000RPM. This shows that the drivability is affected during cornering.