# DEVELOPMENT OF VARIABLE INTAKE SYSTEM FOR SPARK-IGNITION ENGINE

MUHAMMAD HAFIZUDDIN BIN SALIM

BACHELOR OF MECHANICAL ENGINEERING
UNIVERSITI MALAYSIA PAHANG

2012

DEVELOPMENT OF VARIABLE INTAKE SYSTEM FOR SPARK-IGNITION ENGINE

MUHAMMAD HAFIZUDDIN BIN SALIM

Thesis submitted in fulfillment of the requirements
for the award of the degree of
Bachelor of Mechanical Engineering with Automotive Engineering

Faculty of Mechanical Engineering
UNIVERSITI MALAYSIA PAHANG

JUNE 2012

**UNIVERSITI MALAYSIA PAHANG**

**FACULTY OF MECHANICAL ENGINEERING**

I certify that the project entitled "Development of Variable Intake System for Spark-Ignition Engine" is written by Muhammad Hafizuddin bin Salim. I have examined the final copy of this project and in my opinion; it is fully adequate in terms of scope and quality for the award of the degree of Bachelor of Engineering. I herewith recommend that it be accepted in partial fulfillment of the requirements for the degree of Bachelor of Mechanical Engineering with Automotive Engineering.

Ahmad Basirul Subha bin Alias

Examiner Signature

## 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 with Automotive Engineering.

Signature :
Name : DR. RIZALMAN BIN MAMAT
Position : LECTURER
Date :

## STUDENT'S DECLARATION

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

Signature :
Name : MUHAMMAD HAFIZUDDIN BIN SALIM
Id Number : MH 08074
Date :

# DEDICATION

*My kind supervisor, Dr Rizalman bin Mamat*

*My beloved parent Salim bin Ismail & Ramlah binti Seman*

*My lovely, brothers and sisters*

*My entire friends*

*May Allah bless all of you*

## ACKNOWLEDGEMENT

I would like to express my gratefulness to Allah S.W.T for giving me strength and wisdom to finish in my project. I would like to express my sincere gratitude to my supervisor Dr. Rizalman Mamat for his germinal ideas, invaluable guidance, continuous encouragement and constant support in making this research possible. He always impressed me with his outstanding professional conduct, his strong conviction for science and motivation. 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

To my parent, Salim bin Ismail and Ramlah binti Seman thank you for all the support and sacrifice that both of you gave me. Your sacrifice is too great to be measured. It will be never forgotten.

Last but not least, to all people that involved directly nor indirectly in succeeded the thesis, I cannot find appropriate words that could properly describe my appreciation for their devotion, support and faith in my ability to attain my goals. Hopefully, they will continue to support me and thanks for making this possible to happen.

## ABSTRACT

This project is to study about the development of variable intake manifold for spark-ignition engine. Variable intake manifold is one of the methods in optimizing the performance of an engine. Some of manufacture have great interest on this system such as Volkswagen and Volvo companies. Further research was held by those companies. Each of them has different in design in order to race in technology of engine optimizing. This experiment was conducted by using flow bench that test on the flow rate of the new design intake manifold that has been fabricated. The test is on the intake manifold that is used by the 1600cc engine. Two intake manifolds were tested in this experiment, the Proton Waja intake manifold and the custom intake manifold that has been fabricated. The result found that the length of runner does affect the flow rate that produced by the intake manifold. The long runner will give better flow rate on the earlier phase of engine speed while the shorter runner will give better flow rate on the top end of engine speed. That is the reason why the variable intake manifold is better intake manifold compared to the standard intake manifold because it can be switch for the suitable length of runner depends on the engine speed condition.

**ABSTRAK**

Projek ini merupakan kajian mengenai pembuatan “*Variable Intake Manifold*” untuk enjin yang mengunakan sistem penyalaan pencucuh. “*Variable Intake Manifold*” merupakan salah satu kaedah dalam mengoptimumkan prestasi enjin. Beberapa syarikat pembuatan sangat berminat ke atas teknologi ini seperti syarikat Volkswagen dan Volvo. Mereka banyak melanjutkan kajian terhadap teknologi ini. Setiap pembuatan yang dilakukan sering berbeza dari segi reka bentuk didalam perlumbaan teknologi mengoptimumkan enjin ini. Ekseperimen in dilakukan dengan menggunakan bangku aliran bagi mengetahui kadar aliran yang terhasil daripada “*intake manifold”* itu tadi. Ujian yang dilakukan adalah berdasarkan enjin berkapasiti 1600cc. Dua “*intake manifold*” yang di uji di dalam eksperimen ini iaitu bagi penggunaan Proton Waja dan juga “*intake manifold”* yang telah di buat. Hasilnya didapati bahawa panjang pelari sememangnya mempengaruhi kadar aliran yang dihasilkan oleh “*intake manifold*”. Pelari yang panjang akan memberikan kadar aliran yang lebih baik pada fasa awal kelajuan enjin manakala pelari yang pendek akan memberikan kadar aliran yang lebih baik di hujung fasa kelajuan enjin. Disebabkan itu “*Variable Intake Manifold*” lebih baik berbanding “*intake manifold*” biasa kerana ia boleh mengubah laluan aliran mengikut kesesuaian kelajuan enjin.

# TABLE OF CONTENT

# LIST OF TABLES

## LIST OF FIGURES

# CHAPTER 1

# INTRODUCTION

## 1.1 PROJECT BACKGROUND

Intake system is one of the core systems in automotive engine system. It is the system that responsible for fuel and air to get into the block cylinder system to perform combustion phase. Improvement in air intake system is necessary in order to increase the performance of an engine. There are different process and component for an engine that using carburetor system compared to injection system. For an engine that using carburetor to mix air and fuel, it consist two components that is carburetor and intake manifold. Then, for an engine that using injection system, the system consists of fuel injector, intake manifold and throttle body. This project will only consider the injection type as the research topic. This project is utilizing the concept of the variable length intake manifold to develop a new design of intake manifold.

Variable intake system or so called variable length intake manifold (VLIM) is different with the old system of basic manifold. This system used to have two different path of air flow that will be used as combustion element in combustion chamber. The first path will have longer flow path than the second path. There will be controlled by a valve that will close one of the paths depending on it usage. The long path is used when the engine operates at low load or low rpm and the short path is used at the higher rpm and engine load.

Variable intake system invented is to optimize power and torque for the engine and performance and it always create better fuel consumption that make it more efficient than the basic intake system. This is because the variable intake is operating as a switch for an engine, it will switch only when the engine is need to be switch to used what path of manifold. The long path will increase of the flow speed for low rpm usage and the shorter path will create higher capacity of air that used at higher rpm as the fuel usage is increased.  This system will affect the swirl pattern and also it pressurization. Swirl pattern is very useful in the combustion process. A good swirl pattern will help the process to achieve complete combustion than it also will help to minimize the engine knocking. Pressurization is occurs when the manifold has the narrow part that will create dynamic pressure as it speed increased. So it will perform most likely the supercharge system at low pressure.

## 1.2 PROBLEM STATEMENT

The existing manifold will cause less fuel efficiency. With the variable length of manifold will create better fuel efficiency and create more and torque and power. Besides that, new design of variable intake may create better fuel consumption and also performance of the system better than late design. Better swirl pattern will be achieved with right shape of manifold that will depend on the narrow shape and angle of the manifold.

## 1.3 OBJECTIVES

The main objectives for this project are to achieve the following;

1. To propose a new design of variable intake system.
2. To test the propose design in term of it performance and effect.

## 1.4 SCOPE OF PROJECT

The scopes to cover for this project are listed as follows;

1. Design and fabricate the variable intake manifold for four cylinder four stroke engine (1600cc).
2. The intake manifold design is solely for the purpose of laboratory testing.
3. Data analysis will be studying on the effect of the intake manifold length.

# CHAPTER 2

# LITERATURE REVIEW

## 2.0 INTRODUCTION

In this chapter, discussion is more on development of a new design of intake that will effect on the flow efficiency majorly. The parameter that will involve on this study is flow capacity and swirl pattern that creates from the intake manifold. These parameters will determine the efficiency of the intake system that will contribute on the performance of an engine. For all the parameters that will be taking care of, the fuel consumption of the engine that using this manifold also will be determined by studying those effect.

Spark-ignition type of engine system will be used to be work on the manifold that will be designed. Thus, all basic principle of 4-stroke spark-ignition system that involved in internal combustion engine will be revealed.

Besides that, one of the journals will be studying is the performance and fuel consumption that considering of all those parameters by using manifold that have variables of length plenum instead of variables of at their runners length. This is because those two mechanisms have different in their principle of operation but does have similarity in principle of effect and its parameters.

## 2.1 INTERNAL COMBUSTION ENGINE PRINCIPLE OPERATION

### 2.1.1 SPARK IGNITION ENGINE

The spark-ignition engine is a system that operating the combustion process that initiated by a precisely time discharge of a spark across an electrode gap in the combustion chamber. The spark is igniting the air-fuel mixture may be either homogeneous (i.e., the fuel-air mixture ratio may be approximately uniform throughout the combustion chamber) or stratified (i.e., the fuel-air mixture ratio may be more fuel-air lean in some regions of the combustion chamber than in other portions). Most of all spark-ignition engine without considering the Direct Injection Stratified Charge (DISC) SI engine, the air flow rate is used to control the power output that depends on the volumetric efficiency of the flow. Besides that, in almost all operating condition the fuel-air ratio is approximately constant and stoichiometric. DISC SI engine's is differing because the performance is controlled by varying he fuel flow rate that makes the fuel-air ratio is in variable condition while the volumetric efficiency is approximately constant.

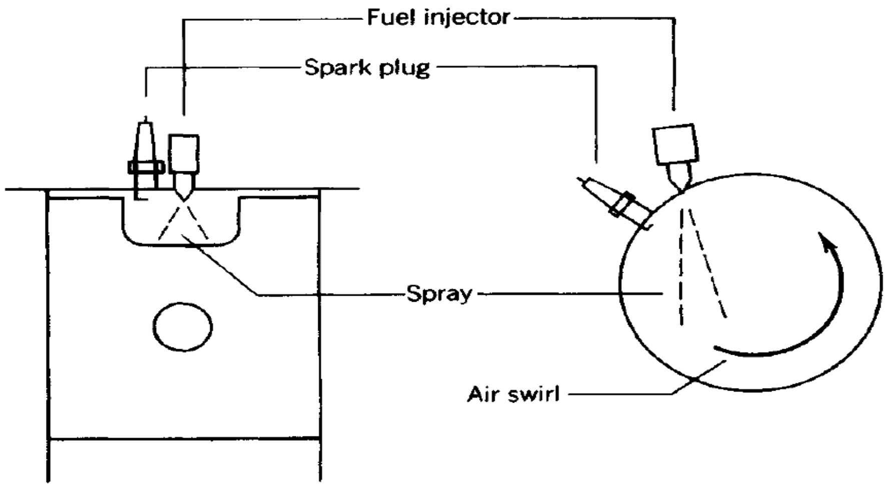


**Figure 2.1:** Schematic diagram of spark ignition engine

Source: (Matthews, 1990)

### 2.1.2 HOMOGENEOUS CHARGE SI ENGINE

Homogeneous charge SI engine had several phase from time to time. It is using method of inducting the fuel-air mixture during the intake process. Firstly, earlier invention that is by mixing fuel and air in the venturi section of a carburetor. Then, carburetor system is replaced by using throttle body fuel injection to give more precise control of the fuel-air mixture. As more precise control of the fuel-air mixture become more desirable, intake port fuel injection has almost entirely replaced throttle body injection system. This all process can be used in both of 4-stroke cycle or on a 2-stroke cycle.

### 2.1.3 BASICS OF 4-STROKE HOMOGENEOUS CHARGE SI ENGINE

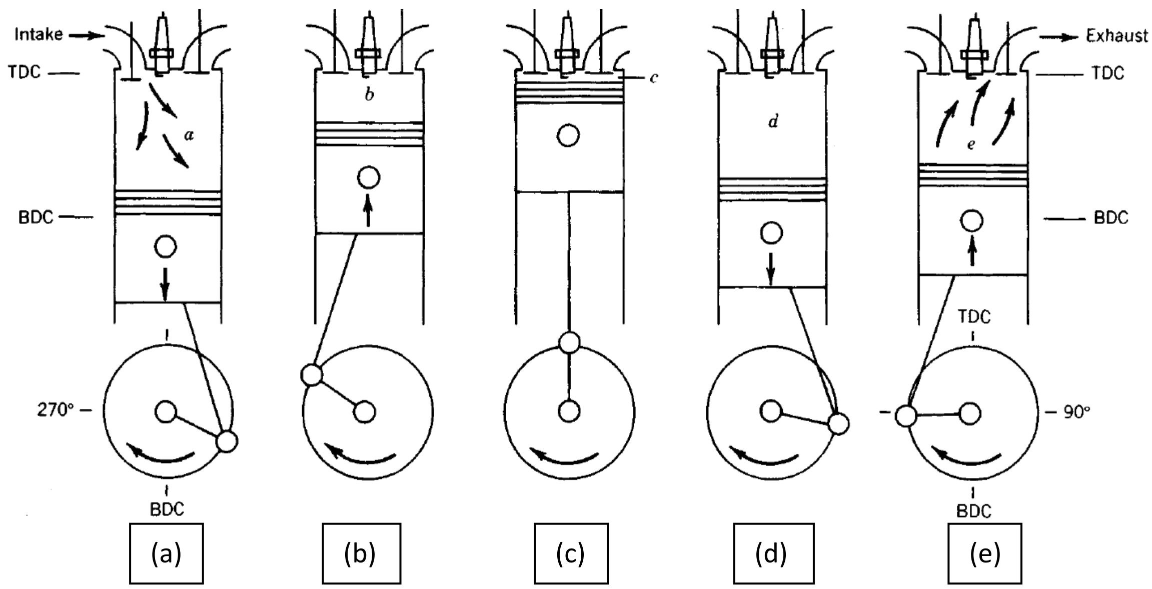


**Figure 2.2:** Four-Stroke Combustion Cycle

Source: (Matthews, 1990)

Today, 4-stroke system become more popular and also widely used in the automotive industry instead of 2-stroke system even though this 4-stroke system is more complicated but it produced better power efficiency and fuel consumption.

Base on Figure 2.2. During the first stroke(a), the piston move from top dead center (TDC – the closest gap between piston and cylinder head) where is at the minimum volume of combustion chamber to bottom dead center (BDC – the farthest gap between piston and the cylinder head) where is in the maximum volume of combustion chamber. As the piston travels down, the inlet valve is opened allowing the fuel-air mixture to be drawn into the combustion chamber. This stroke is called intake stroke.

The second stroke (b) is called compression stroke. During this process, both of the inlet and exhaust valve is closed. At this phase, the air-fuel mixture in the combustion chamber will be compress by movement of the piston back to the TDC. Just before the piston reaches the TDC a spark plug will ignite rapidly to combust the fuel air-mixture. The number of degrees before the top its stroke is the ignition advance. This process is to allow maximum pressure to occur slightly after TDC and producing torque.

The third stroke (d) is called power stroke where it consists of expansion process. It is during the movement to BDC after the second stroke gives power to producing torque.

Lastly, the fourth stroke (e) that called exhaust stroke. This is the process where the exhaust valve is opened to allow the products of combustion to be expelled by using movement of the piston from the BDC to the TDC. Then this process is being repeated afterward.

## 2.2 AIR INTAKE SYSTEM

Air intake system is widely used in internal combustion engine. This system is functional as the guider for the air that will be used in combustion chamber. Besides that, air intake system also functional as the filter for the air that will be used in the combustion process. The location of the air intake system always near to the engine especially the head cylinder part depends on the manufacturer. The components of the air intake system are including the air filter, intake manifold valve to cylinder and turbo with the intercooler for additional charge system. The air that needed for the combustion chamber will be increased as the engine is accelerates. That is why air intake system is optimized by manufacturer as the system will be used in the fuel consumption determination. The volumes of air that will be used were manipulated by a unit control system that was placed on the system.

As the second purpose of the system is to make sure the air that will enter into the combustion chamber is clean. A filter system is placed at the opening of the air intake system that will prevent incoming particles such as sand and dust. The standard requirement of a filter is to clean 99.8% volume of the incoming air (Nylander, 2008). Besides that, the filter also can function as the engine silencer for some product except for the performance type of filter. But the filter that silenced the noise of engine can reduce the performance of engine.

### 2.2.1 AIR INTAKE FILTER

Filter is one of the main components in the air intake system. This component is designed to remove moisture, dirt, dust, chaff and the like from the air before it reaches the engine. Effect of the flow restrictions in the air filter it increases the pressure ratio over the turbo which gives the same boost but to the cost of a higher working temperature. A low restriction intake system will be rewarded with more power and less heat (Bell, 1997). It must do this over a reasonable time period before servicing is required. The air cleaner also silences intake air noise. If dirt is allowed to enter the engine cylinders, the abrasive effects will result in rapid cylinder and piston ring wear. If the air cleaner is not serviced at appropriate intervals for the conditions in which it must operate, it will become restricted and prevent an adequate air supply for complete combustion from reaching the cylinders. Incomplete combustion results in carbon deposits on valves, rings, and pistons, which in turn cause engine wear and oil consumption problems. Air cleaner types include precleaners, dry types of various designs, and oil bath types. Air cleaner capacity (cubic feet per minute or litre per minute) may have to be up to twice as great on a turbocharged engine as compared with the same engine naturally aspirated.

### 2.2.2 TURBOCHARGING

Turbocharging system is one of the methods in enhancing the air intake system performance of an engine. It increasing the power output of an engine by introducing air into the engine cylinder with higher velocity and density compared to the natural aspirated engine that only depending on the ambient density. The system is using exhaust gas as the source of mechanical work that is used in order to generate the turbine of the system to perform suction of air from ambient to be drawn into the combustion chamber. That is why turbocharging plays a crucial role in utilizing exhaust gas energy efficiently (Kesgin, 2004). The operation of turbocharge is shown as Figure 2.3.

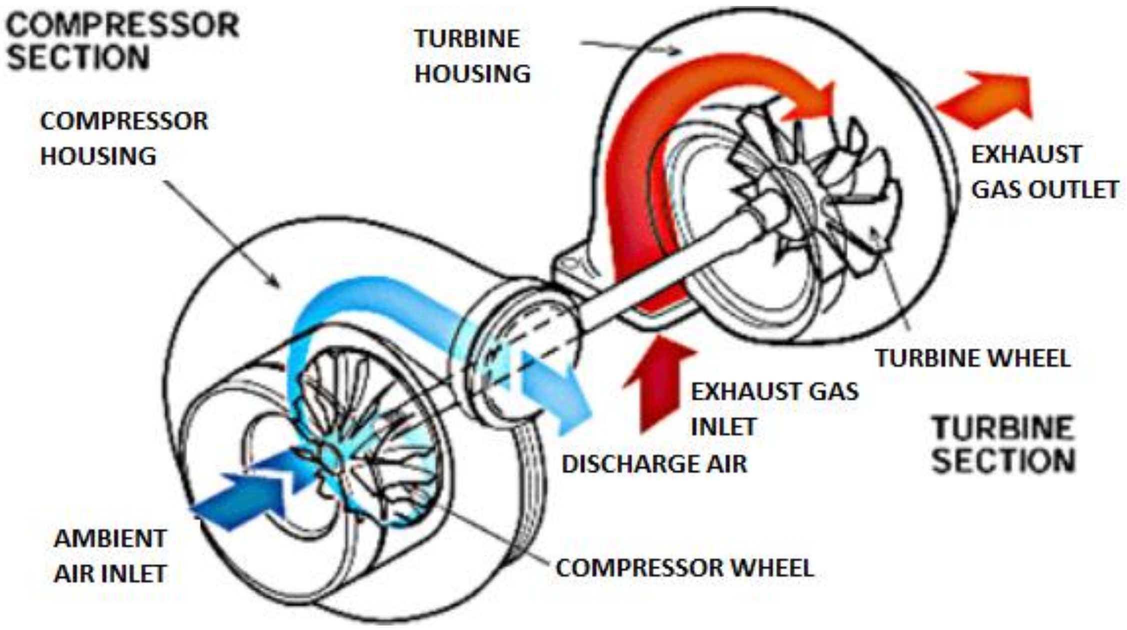


**Figure 2.3:** Turbocharging mechanism

Source: (Alsaeed, 2005)

In fact, increasing the power output of the engine (also called supercharging) does not always come with a better efficiency. There is a recognized practice to supercharge the automotive engine by using the mechanical power of the engine itself. In other words, a belt can transfer the engine torque to a compressor that boosts the pressure of the intake air going into the engine cylinder. Obviously that method would take from the engine power itself, while the supercharging is being performed. However, turbochargers can utilize the wasted hot exhaust gas to increase the engine power, without as much loss of efficiency (Alsaeed, 2005).

The commercially available turbochargers can be subdivided into two principal groups: those primarily designed for use on automotive and truck engines, and those for use on medium-speed and low speed diesel engines for railway traction, electricity-generating sets, industrial and marine applications. The difference between the two principal groups can be summed up as a contrast between small, simple and cheaply mass-produced units, and larger, more complex, expensive and reliable industrial or large marine units (Alsaeed, 2005).

### 2.2.3 INTAKE MANIFOLD

Intake manifold is the most important element in the air intake system. This is the main components that determine the value of flow that will be allowed to be used in the combustion chamber. The intake manifold is always works on with the throttle of the system that will use in air regulation. The manifold is always shape in piping as the function is to guide the air to be used into the combustion chamber. The air the passes through a section of piping, the length of this section of pipe is almost entirely dictated by geometric constraints around the engine (Porter, 2009). The theory of pressure waves is the common theory that will be used in the intake manifold construction. The focus of in designing the intake manifold is put on the plenum and runners of the intake manifold as this two components are the basic component of intake manifold. Figure 2.4 is showing the plenum and runners and position and also the flow of the air from the throttle body through the runners and plenum. Many researches were conducted on the plenum and runners in optimizing the intake manifold performance. The research is commonly studying about possible ways of increasing the volumetric efficiency of an engine. The approach studied has been to try to increase the kinetic energy of the air during the induction process thus achieving a “ram-charge” effect or supercharging effect in the cylinder (Shwallie, 1972).The intake manifold geometry has strong influence on the volumetric efficiency in IC engines (Winterbone and Pearson, 1999). One of the products of studying in the intake manifold field is the variable intake manifold or resonance intake manifold. The general idea is that for lower engine speed the length of the inlet pipe must be longer and as the speed increases the length should be shortened. That is exactly what systems of variable length intake manifold do in order to optimize power and torque across the range of engine speed operation, as well as to help provide better fuel efficiency(Varsos, 2010).

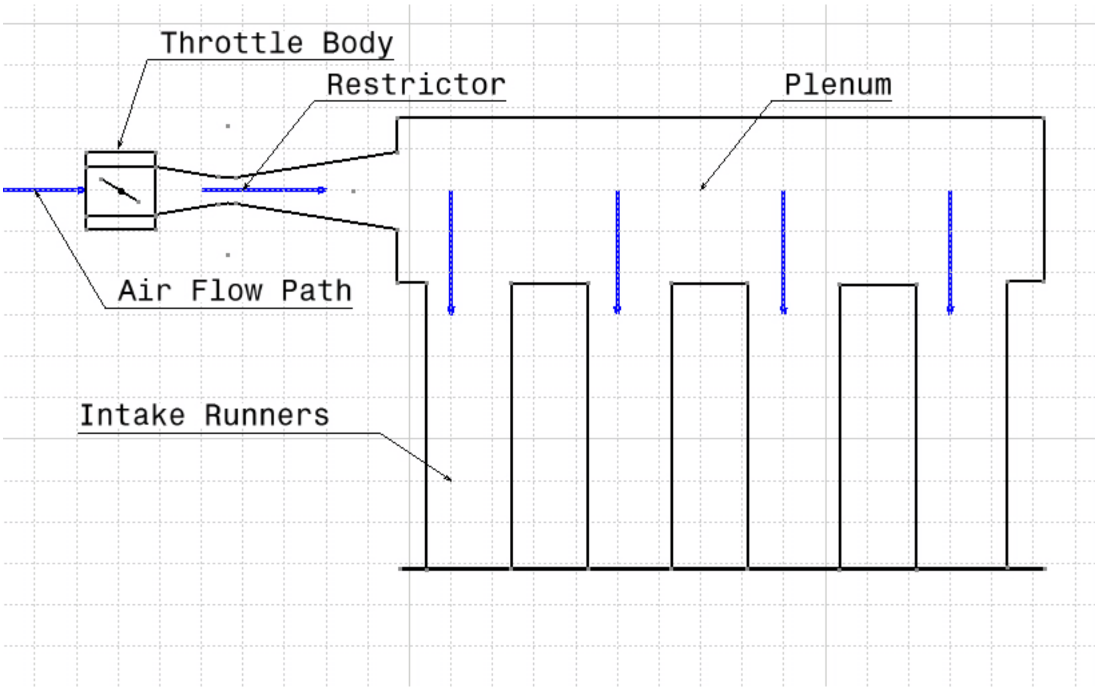


**Figure 2.4:** Basic intake manifold diagram.

Source: (Porter, 2009)

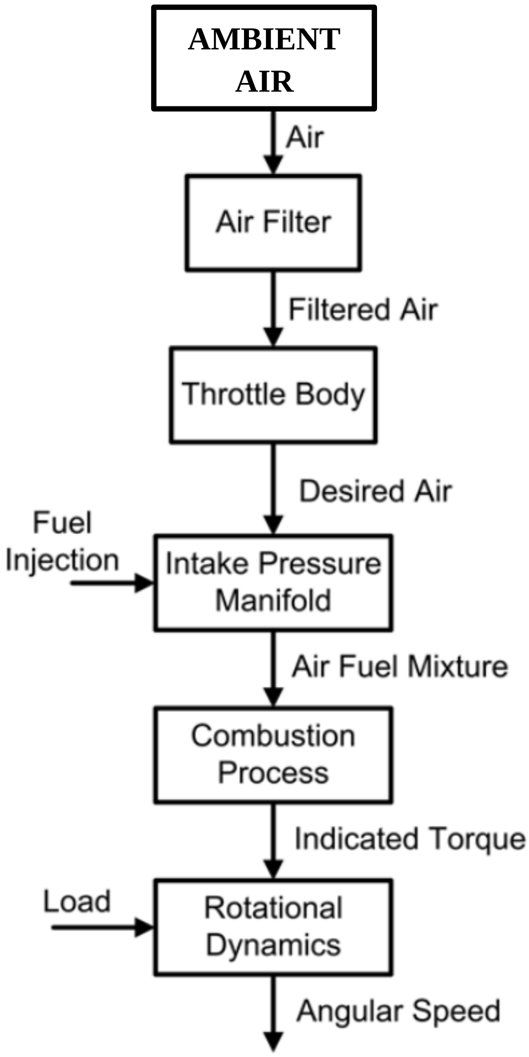


**Figure 2.5:** Component modeling flow

Source: (Ahmed, 2011)

### 2.2.4 VARIABLE INTAKE MANIFOLD

In order to get the good design of an intake manifold with variable length, the variable intake manifold from 3.2-liter V6 engine of Volkswagen is taken as the one of a reference. These intake manifold designs basically have two different path of air flow with using a valve that separating them. It consist of two port of air flow that one is called torque port and the other one is called performance port. This design used to optimize the usage of intake system. It will increase low rpm torque and high rpm power by taking advantage of the self-charging or “ram effect “that exist at some engine speed. This effect produced by tuning the intake manifold air duct length. At the higher rpm, short and larger diameter of air flow will be drawn in to the combustion chamber by using the performance port.

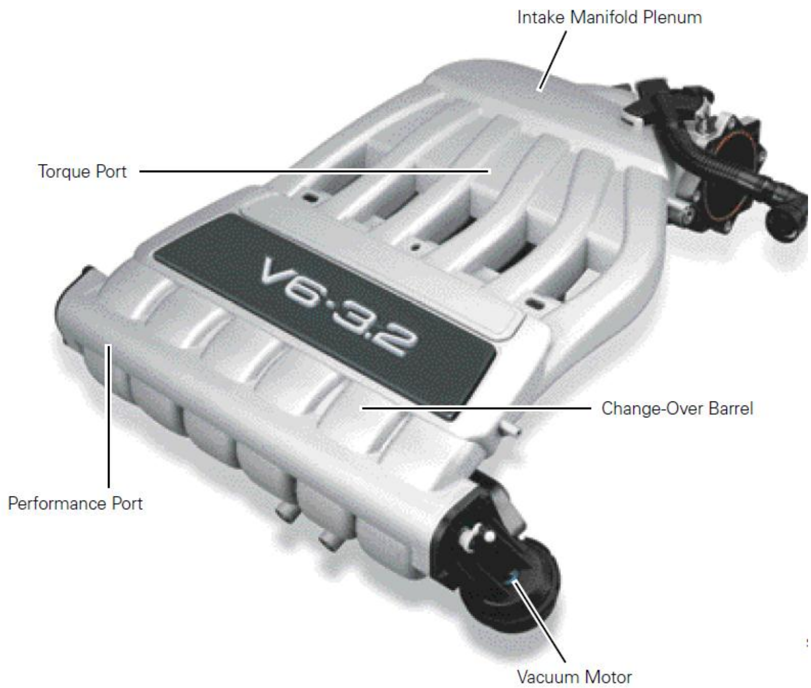


**Figure 2.6:** Volkswagen variable intake manifold

Source: (Volkswagen of America, 2006)

### 2.2.5 PRINCIPLES OF VARIABLE RESONANCE INTAKE MANIFOLD OPERATION

Combustion process that operates in the cylinder created pressure different between the cylinder combustion chamber and the intake manifold. The intake valves opening will create some sort of wave form in the intake manifold that moves from intake valve ports toward the torque at the speed of sound. Figure 2.7 as reference.

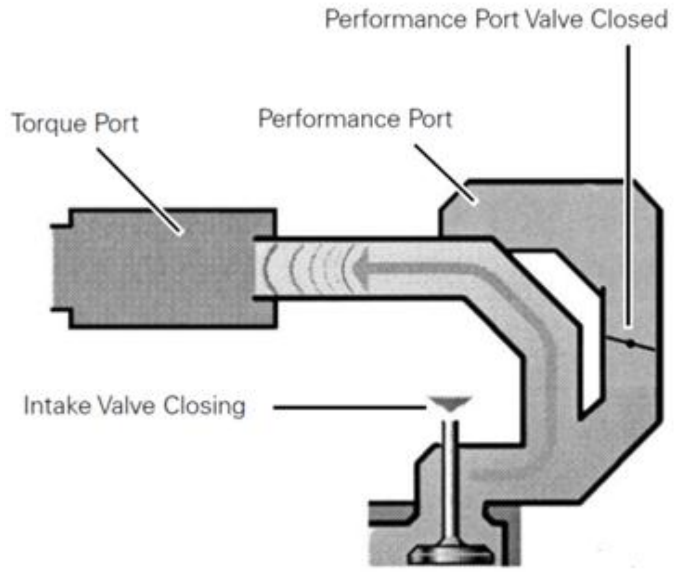


**Figure 2.7:** Intake manifold diagram for operation 1

Source: (Volkswagen of America,2006)

After that, opening the end of the intake duct at the torque port will rapidly form a intake wave same effect as a solid wall has on a ball. It will cause the wave to reflect back toward the intake valve ports in the form of a high pressure wave. Figure 2.8 as reference.

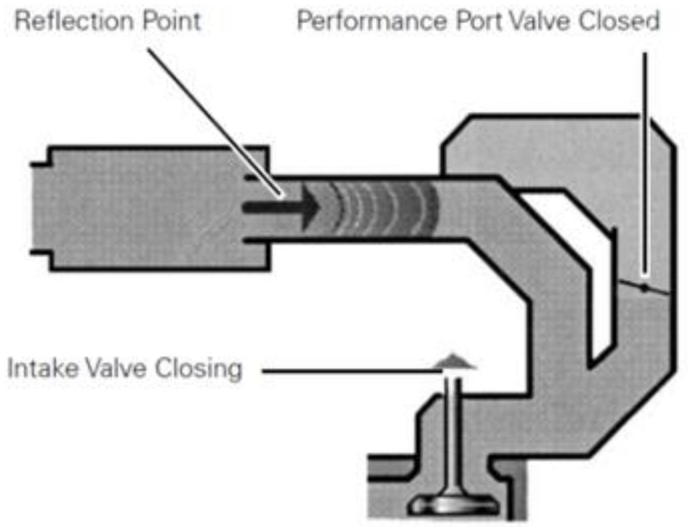


**Figure 2.8:** Intake manifold diagram for operation 2

Source: (Volkswagen of America,2006)

As the engine in still in low rpm, the source of the air flow is still from the torque ports. This is the phase where the wave forms the maximum pressure behind the valve waiting to enter the combustion chamber. At this time, the piston is in top dead center position. After the late fuel-air mixture is entering the combustion chamber, the new wave form that consists of maximum charging will be form rapidly as it called self-charging or “ram effect.” Figure 2.9 as reference.

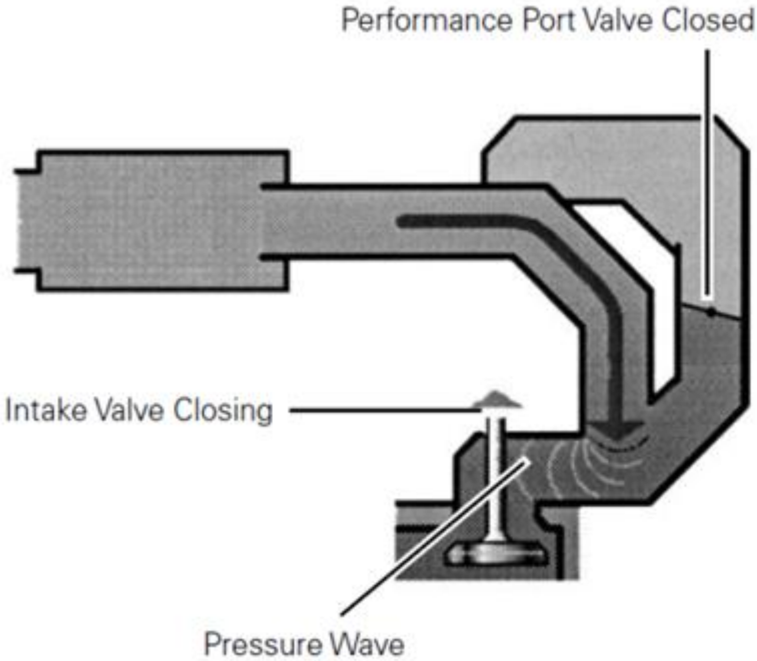


**Figure 2.9:** Intake manifold diagram for operation 3

Source: (Volkswagen of America,2006)

When the speed of the engine increase, there are certain phase that make the high pressure will have less time to reach the inlet port. It is because the wave is limits to only at speed of sound. There is where the shorter intake must be taking place to solve the problem. Figure 2.10 as reference.

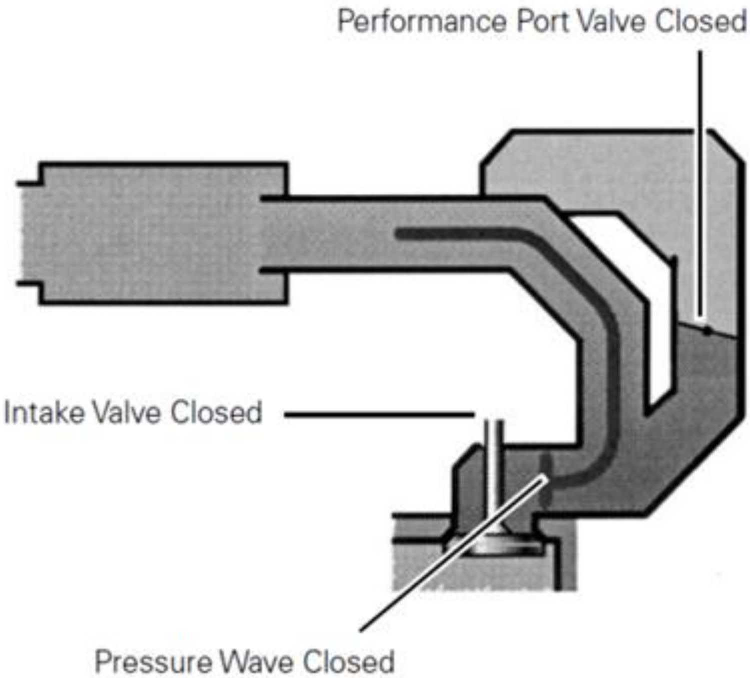


**Figure 2.10:** Intake manifold diagram for operation 4

Source: (Volkswagen of America, 2006)

For 3.2 and 3.6-liter v6 engine type, the function of performance will be used at the below 900 rpm and above 4100 rpm it is where the performance port valve is opened. The performance port is designed to overcome the problem on pressure wave; it will use the shorter path so that the intake and pressure waves will have sufficient power to be used in combustion chamber. Figure 2.11 as reference.

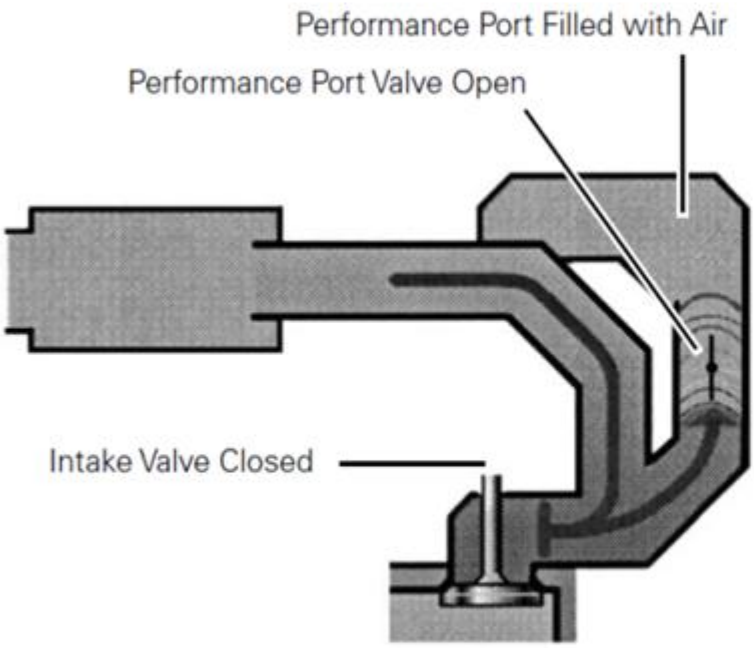


**Figure 2.11:** Intake manifold diagram for operation 5

Source: (Volkswagen of America, 2006)

When the intake valves are closed the performance port is filled with air. Then, after the intake valve is opened again, intake wave form to move to both of intake ducts toward the torque port and the performance port at the same speed. Wave form from performance will reflect first as it has shorter path than the torque port path. Figure 2.12 as reference.

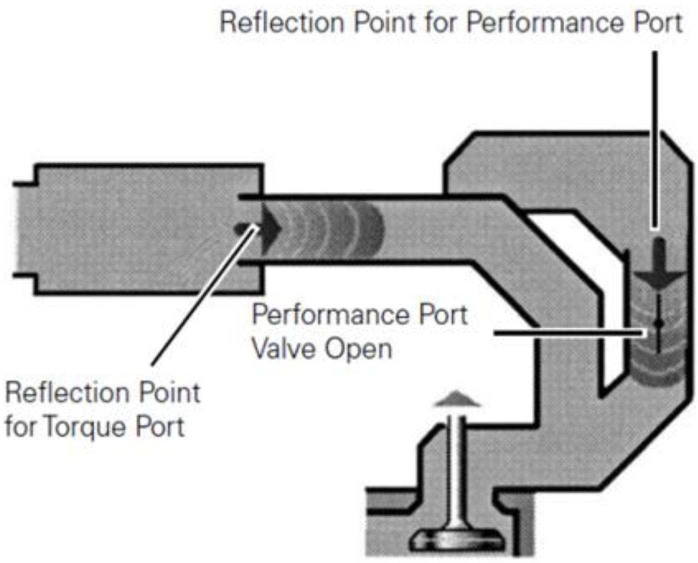


**Figure 2.12:** Intake manifold diagram for operation 6

Source: (Volkswagen of America, 2006)

After the pressure wave is reflected toward the intake valves port, this where the pressure wave is needed to be forced into the combustion chamber rapidly just before the intake valves closed. Figure 2.13 as reference.

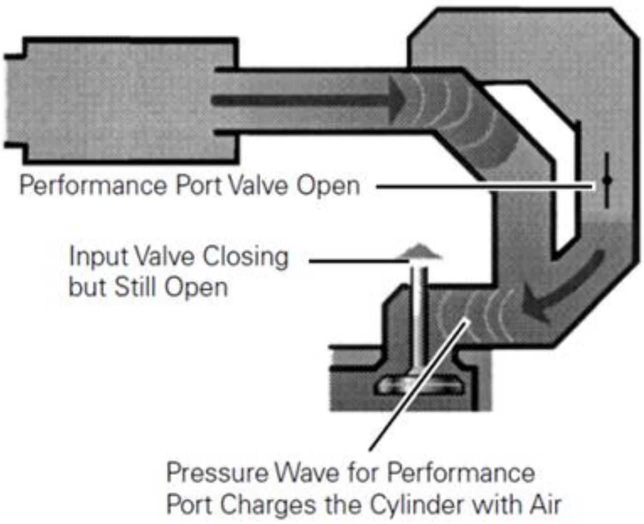


**Figure 2.13:** Intake manifold diagram for operation 7

Source: (Volkswagen of America, 2006)

For the next operation, as the wave pressure reflected by intake valves is late from the torque port and pushes its air charge up the intake duct. Then, the operation is ready for the next cycle upcoming. Figure 2.14 as reference.

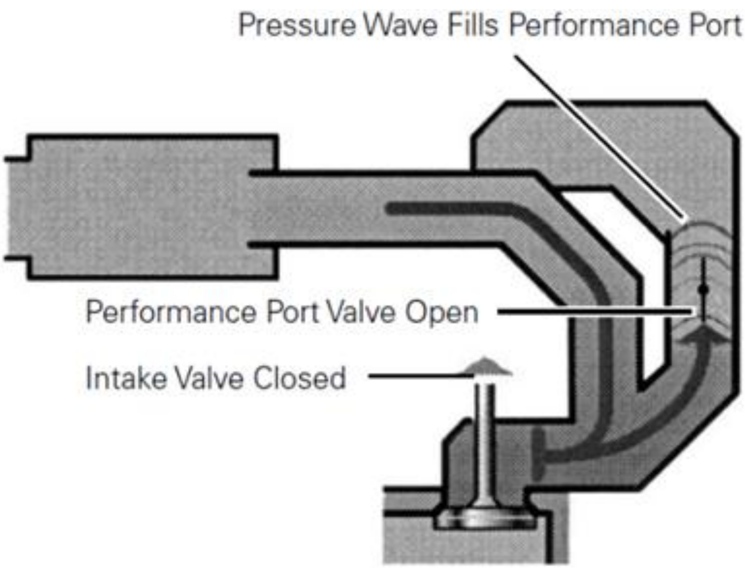


**Figure 2.14:** Intake manifold diagram for operation 8

Source: (Volkswagen of America, 2006)

### 2.2.6 INTAKE MANIFOLD DESIGN

In designing a new intake manifold, the performance characteristic that produced from the intake manifold must be put as a high consideration because it will determined whether this design is good or not. This intake will be test on an engine and the performance characteristic that have been discussed is through its brake torque, brake power, thermal efficiency and specific fuel consumption. The data is being compared by using three different length of plenum to be compared with each other to differentiate its performance characteristic.

In order to get the result, some operating parameters must be taking care is the relative air-fuel ratio, spark timing, fuel injection timing, valve timing, exhaust gas recirculation ratio and compression ratio in spark ignition.

Besides that, important design criteria such as low at flow resistance; good distribution of air and fuel between cylinder; runner and branch lengths that take advantage of ram and tuning effect; sufficient heating to ensure adequate fuel vaporization with throttle body injected engines.

By having a good design of intake manifold, the good air flow so called resonance flow can be achieve. Resonance is very useful and it occur at some speed of engine, this is because resonance can enhance the performance of an engine characteristic but its highly depends on the intake manifold dimensions and shape. Previously, conventional intake manifold has fixed air flow geometry and static intake manifold. Along the way of engine running, the intake can be optimizing the usage of its performance at certain engine speed. That’s why this test is very important to have variable length of plenum to optimize and to enhance the performance of an engine (Ceviz and Akin, 2010).

## 2.3 AIR INTAKE SYSTEM OPTIMIZATION EFFECT

It has long been known that an optimized design of the intake manifold is essential for the optimal performance of an IC engine (Heywood, 1987). As we know, developments of intake manifold creating better efficiency on some variables. Many researchers were conduct on further study of intake system such as Ceviz in the ‘Intake plenum volume and its influence on the engine performance, cyclic variability and emissions,’ and also Hamilton and Lee in the ‘The effect of intake plenum volume on the performance of a small normally aspirated restricted engine’.

### 2.3.1 FUEL CONSUMPTION

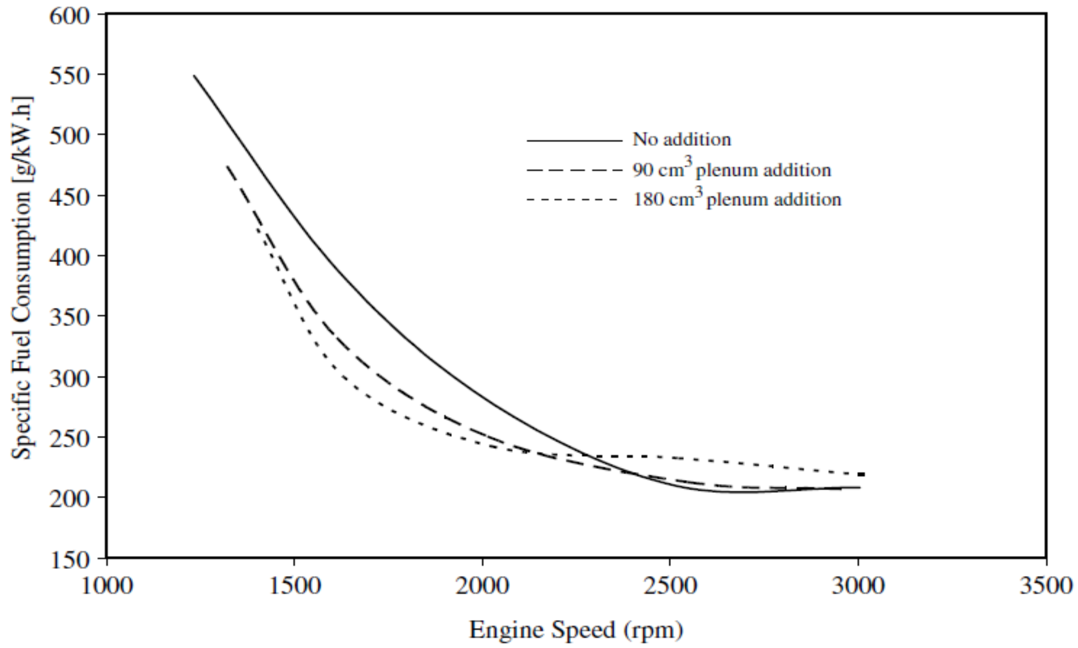


**Figure 2.15:** Basic fuel consumption graph

Source: (Ceviz, 2006)

Figure 2.15 is the graph of further research on the fuel consumption base on the intake plenums as the variables. This research is conduct by Ceviz in the ‘Intake plenum volume and its influence on the engine performance, cyclic variability and emissions’. This graph show that the specific fuel consumption that used by conduct in three different length of intake plenum. By this graph, we can see that different variable of plenum or volume of the intake manifold is very affecting the fuel consumption. For the result, we can conclude that if we have fix volume plenum the fuel consumption cannot be optimized. This is because for a bigger plenum size volume or length the fuel consumption is very efficient at 2300 rpm and below but after that, the fuel consumption is increased. Differ with smaller volume of intake manifold plenum, the fuel consumption is very high at 2300 rpm and below and after that it become more efficient. Thus, with this data we can design variable manifold that can be used to minimize the fuel consumption of this engine.

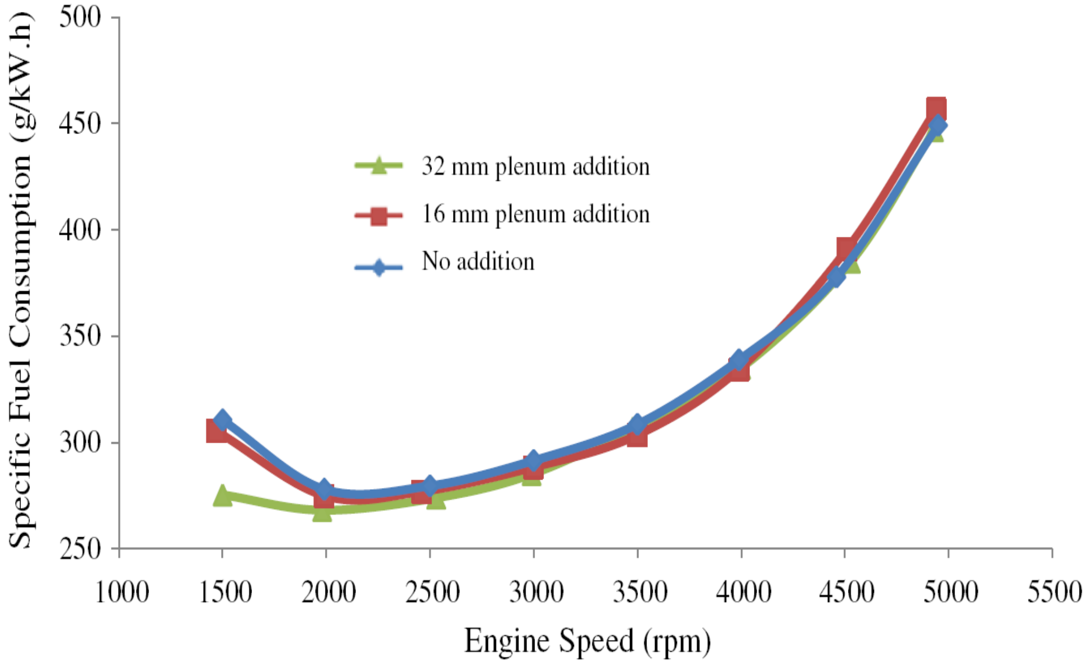


**Figure 2.16:** Variation of fuel consumption with engine speed for three different intake plenum volumes

Source: (Ceviz and Akin, 2010)

Figure 2.16 is showing the further research on the effect of intake plenum on fuel consumption by Ceviz and Akin. This graph shows three variations of different intake plenum volume used. It can be seen from this figure that the fuel consumption per engine power output was the lowest for the engine speed range of 1500–3000 rpm by using 32 mm plenum addition and for 3000–4000 rpm range by using 16 mm plenum addition. The higher engine speeds, it is necessary to use the original engine manifold as it has lower fuel consumption compared to others variables. This graph show that by increasing the volume of plenum or intake manifold it will create good efficiency on fuel consumption at the top end speed of an engine.

### 2.3.2 ENGINE PERFORMANCE

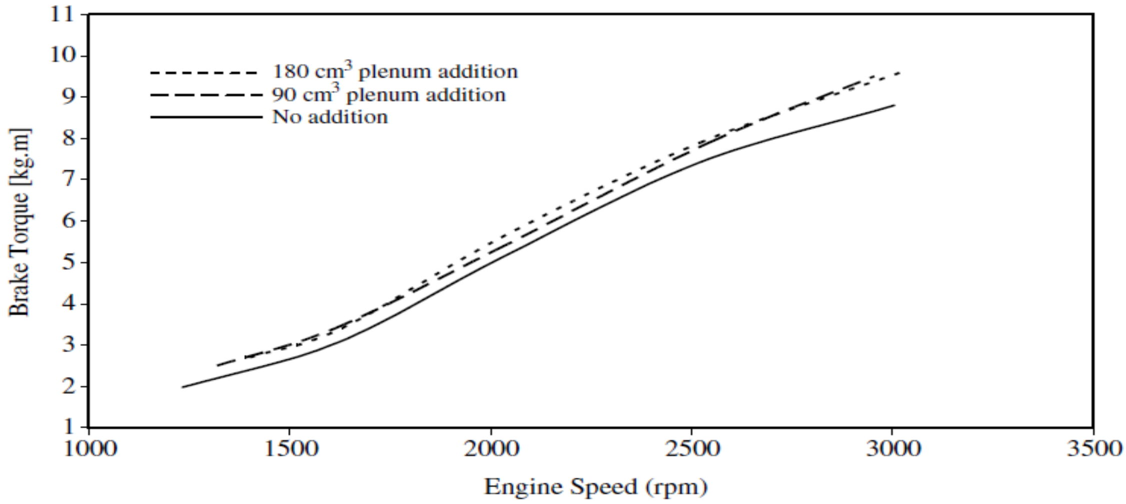


**Figure 2.17:** Brake torque graph

Source: (Ceviz, 2006)

Figure 2.17 is the graph of further research on the engine performance base on the intake plenums as the variables. This research is conduct by Ceviz in the 'Intake plenum volume and its influence on the engine performance, cyclic variability and emissions'. This graph show that the effect of three different plenum volume and length in the brake torque at different engine speed. Obviously from this graph pattern, the additional of plenum will create better brake torque. Increasing brake torque is mostly means that increasing in the engine performance. The plenum of with no addition can be neglected from this graph due to no better performance output compared to the plenum with additional length. We can see that the bigger plenum addition on the intake manifold gives high performance of the engine at 1700 rpm to 2600 rpm but it become worse after that. The smaller plenum addition gives better engine performance after 2600 rpm. From this data, we can see every different plenum gives different engine performance. By a good calculation and right selection, we can optimize the usage of this manifold to get better engine performance producing from it.

Figure 2.18 and Figure 2.19 show graphical representations of the effects of plenum size on torque at various engine speeds of 2002 Honda F4i Engine (Hamilton and Lee, 2008). At low RPM, the smaller plenum appears to produce higher torque than the large plenum although the effect is modest. However, above 7,000 RPM, it is clear that torque tends to increase significantly as plenum size increases. This effect is more pronounced at higher RPM. In fact at the maximum RPM tested, torque dropped from 48 Nm to 36 Nm when comparing the largest and smallest plenums; a 25% performance loss in torque. It is also evident that there are two distinct torque peaks; one at approximately 6500 RPM and the second near 10,000 RPM. It is also interesting to note that the torque peak has a very weak dependence on plenum size. For the 6.0 L plenum, torque peaks at 54.5 Nm at 10,000 RPM while the 1.2 L plenum results in a peak of 48.5 Nm at 9500 RPM. This is because runner length is the primary factor in determining the frequency (driven by engine RPM) for resonant supercharging.

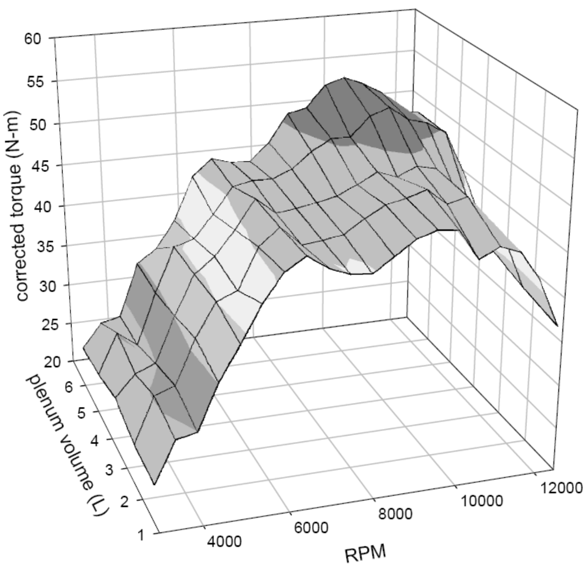


**Figure 2.18:** Torque – RPM and plenum volume sweep

Source: ( Hamilton and Lee, 2008)

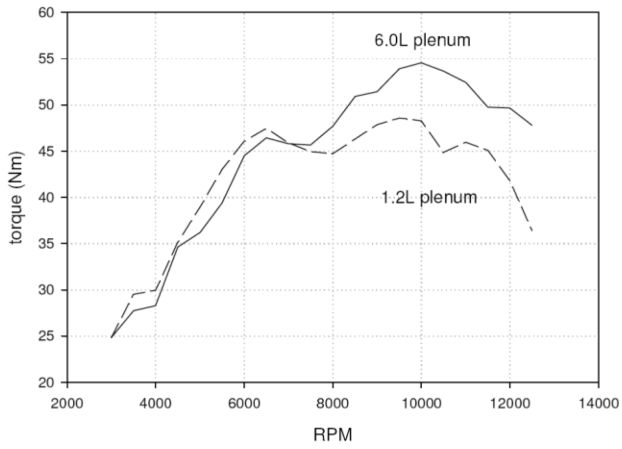


**Figure 2.19:** Comparison of torque between smallest and largest plenums tested

Source: (Hamilton and Lee, 2008)

## 2.4 THEORY OF DYNAMICS

### 2.4.1 WAVE THEORY

For further study on the pressure waves which occur in an intake manifold it is easiest to consider the application in pipe organs. The fluid (air) in the system is sloshing back and forth due to its inertia, bouncing against the resilience of the compressed gas in the resonant cavities; there are compression and expansion waves travelling through the gas, reflecting from closed and open ends, and from changes in cross-section (Varsos, 2010).The overarching principle in a pipe organ is the way the pressure waves inside the pipes reflect back along the pipe based upon whether they encounter an open or closed end of the pipe. To briefly explain what occurs and for future reference, two main types of waves form inside the pipe. These waves are known as rarefaction and compression waves. A rarefaction wave is a wave of less than atmospheric pressure and a compression wave is one greater than atmospheric pressure. When a wave reaches an open end of a pipe a wave of opposite form is reflected back down the pipe, if the wave reaches a closed end, a wave of the same form is reflected back along the pipe (Smith and Morrison, 2002).

In an intake manifold there are two significant events in the intake stroke of the engine, these are the opening and closing of the intake valve. When the intake valve closes, a compression wave forms, whereas when the intake valve opens a rarefaction wave is formed. These waves can be analyzed by the soon to be described method of characteristics, considering that if the incident wave sees a larger cross section, the reflected wave is of the other type (compression wave is reflected as expansion, and expansion wave as compression) and if the incident wave sees a smaller cross-section, the reflected wave is of the same type (Varsos, 2010).

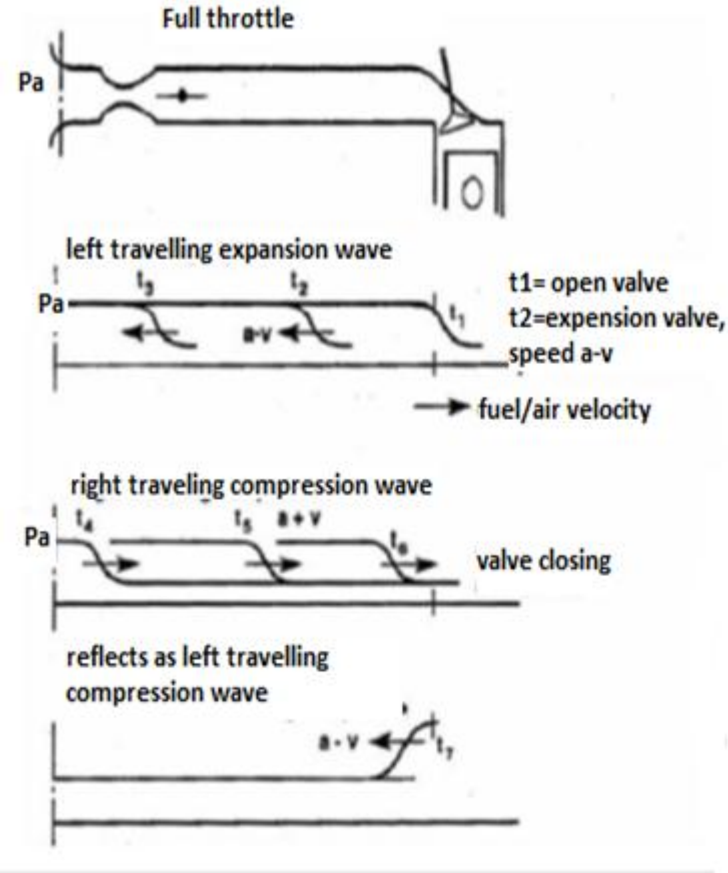


**Figure 2.20: S**chematic representation of wave action in the inlet manifold

Source: (Lumley, 1999)

These waves reflect up and down the intake runner, in the same fashion as in a pipe organ. As these waves are created and propagate they interact with each other in 6 a similar fashion to any other sounds waves, they sum together to form either a wave of higher amplitude, or even possibly diminish to a wave of zero amplitude. In the instance where a rarefaction wave is created upon valve opening it travels along the intake runner to the plenum where a compression wave is reflected back to the valve, when this compression wave hits the intake valve it propagates into the cylinder and increases the pressure in the cylinder. The amplitude and phase of theses pressure waves depend on intake manifold geometry, engine speed and valve timing (Engelman, 1953).

### 2.4.2 FLUID FLOW

In the intake manifold system, it often to be built in by combination of pipes and ducts that function as the guider to the air to be used into the engine to feed the combustion process. As such it is open to analysis and optimization as any network pipes and ducts may be (Porter, 2009). The mechanism that will effect on the fluid flow value of the intake system is involved the head loss, or pressure loss due to certain geometries within the flow, specifically for bends, valve, entrance and re-entrance flows. Besides that, that characteristic of state air flow such of velocity profiles for both turbulent and laminar flows.

Pressure losses in pipes can be categorized into two categories that are major and minor. Basically, major losses occur due to the physical length of the pipe and the viscous losses associated with the friction between the wall and the fluid. Then, minor losses occur due to variations in geometry through the piping such as bends, elbow, valves, entrances and re-entrance. The terms major and minor do not refer to the relative sizes of the losses necessarily, but in typical piping systems involving many long straight sections with few bends and valves the major losses are more substantial than the minor (Munson et. al, 2006). For intake manifold however, the ‘minor’ losses are far more significant, and typically dominate the pressure losses experienced.

Besides that, velocity profiles also being taking serious in the built of an intake manifold as they are one of the tools utilized in validating the results of various simulations. This character that flow in the pipe is a well-studied phenomena and has been noted to depend on a number of factors. The predominant effect on the final shape is the non-dimensional quantity Reynolds number, more specifically whether it is above or below the transition from laminar to turbulent flow (Porter, 2009). This factor determines whether the profile is parabolic or much flatter.

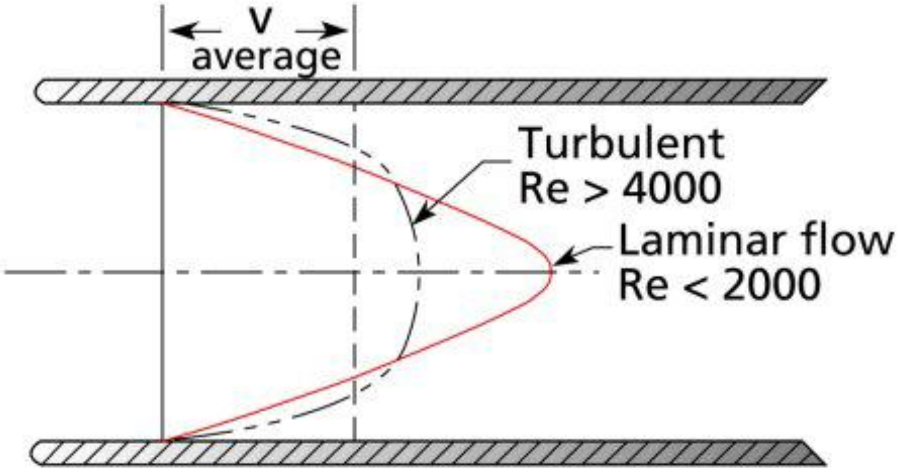


**Figure 2.21:** Velocity profiles within a pipe

Source: (Chaurette, 2008)

### 2.4.3 DESIGN EQUATION

In designing the manifold, some equation must be considered in determining some of operating parameter that involved in experiment.

Brake power ($P_b$),

$$P_b = 2\pi nT10^{-3} \tag{2.1}$$

Where $\pi$ is the crankshaft rotaional speed (rev $s^{-1}$) and T is the torque (Nm).

The thermal efficiency of the engine is

$$\eta_{th} = \frac{P_b}{Q} \tag{2.2}$$

Q is the total heat supplied by the fuel;

$$Q = CV\dot{m}_f \quad (2.3)$$

Specific fuel consumption (sfc*)* is the fuel flow rate per unit power output and calculated from;

$$\text{sfc} = \frac{\dot{m}_f}{P_b} \quad (2.4)$$

Where $m_f$ is the fuel consumption ($gh^{-1}$) and $P_b$ is the brake power (kW).

### 2.4.4 GAS DYNAMICS

The gas flow during HVOF thermal spraying is governed by three general conservation equations. The mass conservation can by expressed as:

$$\frac{\delta}{\delta x_j}\left(\rho u_j\right) = 0 \quad (2.5)$$

where Cartesian component of the instantaneous gas velocity, and ρ is the gas density. The repeated indices imply summation over all coordinate directions. For a viscous fluid the momentum conservation equation is

$$\frac{\delta}{\delta x_j}\left(\rho u_i u_j\right) = \frac{\delta p}{\delta x_i} + \frac{\delta}{\delta x_j}\left[\mu\left(\frac{\delta u_i}{\delta x_j} + \frac{\delta u_j}{\delta x_i} + \frac{2}{3}\frac{\delta u_k}{\delta x_k}\delta_{ij}\right)\right] \quad (2.6)$$

where ρ is the static pressure, μ is the molecular viscosity coefficient, and $\delta_{ij}$ is the Kronecker delta function. The energy conservation equation is

$$\frac{\delta}{\delta x_j}\left(\rho u_i H\right) = \frac{\delta}{\delta x_j}\left(k\frac{\delta T}{\delta x_j}\right) + \frac{\delta}{\delta x_j}\left[\mu\left(\frac{\delta u_i}{\delta x_j} + \frac{\delta u_j}{\delta x_i} + \frac{2}{3}\frac{\delta u_k}{\delta x_k}\delta_{ij}\right)\right] - \frac{\delta}{\delta x_j}\left(J_{ij}h_i\right) + S_a$$

(2.7)

where H is the total enthalpy, **κ** is the conductivity, $J_{ij}$ is the total diffusive mass flux, hi is the enthalpy for species i and $S_a$ is the additional sources, respectively. In the present study, the Favre (or density) approach is used to average the above conservation equations. The Favre averaging process introduces unknown terms called the Reynolds stresses. The Boussinesq eddy viscosity concept is also employed to treat these stress terms. In addition, the renormalization group (RNG) κ–ε model is used to estimate the turbulent eddy viscosity.

### 2.4.5 STEADY FLOW

This chapter is to study about the steady flow that will be used in the experiment of intake manifold testing. As we know that the flow bench (Superflow SF-1020) will be used in the experiment.is necessary to study about the steady flow as the flow bench will produce in steady flow state.

The flow parameters used to characterize the engine heads are the flow coefficient, $C_f$, the swirl coefficient, $C_s$, and the swirl ratio, $R_s$. The flow and swirl coefficients are measured at discrete valve lifts (Douglas Heim and Jaal Ghandi 2011). The flow coefficient is a measure of the actual mass flow rate to a theoretical mass flow rate and is defined as

$$C_f = \frac{\dot{m}}{\rho V_B A_v} \qquad (2.8)$$

Where $\dot{m}$ is the air mass flow rate, ρ is the air density, $A_V$ is the valve inner seat area, and $V_B$ is the Bernoulli velocity given by

$$V_B = \sqrt{\frac{2\Delta P}{\rho}} \qquad (2.9)$$

Where ΔP is the pressure drops across the test section. The flow coefficients are plotted against the engine crank angle degrees in Fig. 2.25. Comparing the ports with the non-shrouded valve, the performance port achieves higher flow coefficients at the crank angles corresponding to the range of highest valve lift. This result was expected since the performance port was designed to enhance the flow coefficient in this range. The flow coefficients of the ports with the shrouded valve compared to the non-shrouded valve are less since the flow path is partially restricted by the shroud. A mass-weighted average flow coefficient, $C_{f,avg}$ , was obtained by performing a mass-weighted average over the engine crank angle profile, defined as

$$C_{f,avg} = \frac{\sum_{\theta} \dot{m} C_f}{\sum_{\theta} \dot{m}} \tag{2.10}$$

The swirl coefficient, $C_s$, is a characteristic non-dimensional rotation rate and is calculated for impulse-type meters using

$$C_s = \frac{8T}{\dot{m} V_B B} \tag{2.11}$$

Where B is the cylinder bore and T is the torque measured by the meter. Figure 2.25 gives the swirl coefficients plotted against the non-dimensional valve lift, *l*/D, where *l* is the valve lift and D is the valve inner seat diameter. The shrouded valve directionally restricts the flow into the cylinder, tending to enhance the swirl compared to the non-shrouded valve, resulting in greater swirl coefficients. The swirl ratio, $R_s$, is a convenient single metric that takes into account the flow and swirl coefficients over the entire lift profile of the engine. The swirl ratio is calculated as

$$R_s = \frac{\pi B S \eta_v^2 \int_{\theta_{IVO}}^{\theta_{IVC}} C_s C_f \delta\theta}{4 A_v \left( \int_{\theta_{IVO}}^{\theta_{IVC}} C_f \delta\theta \right)^2} \tag{2.12}$$

Where $\eta_v$ is the volumetric efficiency (assumed equal to 1 for all calculations), $\theta_{IVC}$ and $\theta_{IVO}$ are the crank angle, inradians, at intake valve open and intake valve close,respectively, and S is the engine stroke.

### 2.4.6 TURBULENCE

The turbulence flow will be reveal on the characteristic and the specification of this type of flow in this chapter. This is because of the turbulent flow sometimes occurring on the air intake system for some supercharging system.

The Reynolds number of an flow determine the behaviour and characteristic of fluid motion which is a dimensionless parameter and defined as

$$Re = \frac{vL}{\upsilon} \tag{2.13}$$

Where *v* is the kinematic viscosity, *u* is the velocity and *L* is the characteristic length of the fluid. This Reynolds number equation is a ratio between the inertial and viscous forces in the fluid that determines the regime of the flow. Laminar flow is when viscous force dominates, the fluid travels smoothly in the domain and the force acting for rapid fluctuations is suppressed by the forces acting to keep the flow in a steady behaviour. For turbulent flow, this is when the inertial forces dominate over the viscous forces and the Reynolds number is sufficiently large, the flow will become unsteady and fast fluctuations in the flow will occur.

**Figure 2.22:** Transition between laminar and turbulent flow

Source: (Nylander, 2008)

### 2.4.7 IDEAL GAS LAW

In many research on the flow of air, the air is always consider as an ideal gas as measurement in an experiment. Thus, by using the ideal gas law the parameter of flow such as pressure, volume and temperature can be determine.

$$pV = nRT \tag{2.14}$$

Relationship between molar mass in chemical

$$m = nM \tag{2.15}$$

Definition of density

$$\rho = \frac{m}{v} \tag{2.16}$$

The ideal gas law conversion

$$pM = \rho RT \tag{2.17}$$

Transformation from $\rho$ gives the equation:

$$\rho_{air}\left[\frac{kg}{m^3}\right] = \frac{P_{manifold}\left[bar\right]M_{air}\left[\frac{g}{mol}\right]}{R\left[\frac{m^3bar}{mol.K}\right]T_{manifold}\left[K\right]} \tag{2.18}$$

Where $\rho$ is the density of the manifold air, p is the intake manifold pressure, M is the molar mass of air, R is the molar intensive universal constant and T is the temperature of the air in the manifold.

### 2.4.8 VOLUMETRIC EFFICIENCY

In studying about flow in the intake manifold, the volumetric efficiency is one of important parameters that define the ratio between the mass of air inducted into the cylinder and the volume the same mass of air would displace in the intake manifold. The volumetric efficiency rates effectiveness of the induction process hence it is desirable to maximize the outcome (Ferguson and Kirkpatrick, 2001). The parameters that effecting the volumetric efficiency is manifold design followed by the valve size, lift and timing.

$$e_v = \frac{\dot{m}_{air}\left[\frac{kg}{min}\right]}{\rho_{air}\left[\frac{kg}{m^3}\right] V_d\left[m^3\right]\frac{N\left[\frac{revolution}{min}\right]}{2}} \tag{2.19}$$

Where $\dot{m}_{air}$ is the mass flow of air, $\rho_{air}$ is the air density, $V_d$ is the engines total displacement volume and N is the engine speed.

### 2.4.9 INTAKE MANIFOLD PRESSURE

This equation of intake manifold pressure was obtained from the combination of the volumetric efficiency equation with the ideal gas law equation.

$$P_{manifold}\left[bar\right] = \frac{\dot{m}_{air}\left[\frac{g}{min}\right] R\left[\frac{m^3 bar}{mol.K}\right] T_{manifold}\left[K\right]}{V_d\left[m^3\right]\frac{N\left[\frac{revolution}{min}\right]}{2} M\left[\frac{g}{mol}\right] e_v} \tag{2.20}$$

### 2.4.10 CONSERVATION OF ENERGY

As we know that intake manifold component is majorly content of piping section, the similar method by using conservation of energy theory on the mass also can be applied on a flow. The volumes of intake manifold are controlled and we can investigate the rate of increasing in velocity of the flow inside the manifold. Thus, the effectiveness of an intake manifold can be determined. The control volume in the intake manifold can be considered for conservation of energy analyses are the pipe sections and the nozzles shape of intake manifold. The steady-state head form of the conservation of energy equation, which accounts for minor losses and frictional head loss, is used to compute the velocity through each segment (Heim, 2010).

$$\frac{P_{in}}{\gamma}+\frac{v_{in}^2}{2g}+z_1=\frac{P_{out}}{\gamma}+\frac{v_{out}^2}{2g}+z_2+h_f+\sum h_m \tag{2.21}$$

Where z is the elevation relative to some reference datum, $\gamma$ is the specific gravity (i. e. density of the fluid multiplied by the acceleration due to gravity), $h_f$ is the frictional head loss, and $\sum h_m$ is the sum of the minor head losses. The equation can be simplified into

$$P_{in}-P_{out}=\frac{1}{2}\rho\left(v_{out}^2-v_{in}^2\right)+\left(h_f+\sum h_m\right)\rho g \tag{2.22}$$

## 2.5 PREVIOUS SYSTEMS BENCHMARKING

This section is benchmarking the air intake systems that have been built by University of Toronto FSAE team since this team has been designing and building cars for eleven years. It is observation from different systems that have been designed and tested data that gathered in this section. Three of common intake systems styles from 2004 until 2007 is observed and analyzed. The qualitative performance of each of these systems was determined from previous driver according from their experiences. All of the data are combined with theoretical information on intake system design, allowed for the effects of changes to different systems parameter to be hypothesized.

**Table 2.2:** Intake and exhaust characteristic of Formula SAE of Toronto Racing car.

| | **2004** | **2005** | **2006** | **2007** |
|---|---|---|---|---|
| | | **Intake** | | |
| Runner length | 8.5” | 10.25” | 9.5” | 9.625” |
| Plenum Volume | 3L | 1L | 1.5L | 2L |
| Intake style | Trapezoid | Cone | Log | Log |
| | | **Exhaust** | | |
| Primary length | 22” | 23” | 20” | 26” |

Source: (Damji, 2008)

**Table 2.3:** Summary of previous Driver’s experience with past intake system

| **Intake system** | **Driver experience** |
|---|---|
| 2004 | Poor engine output. Resulted in poor on-track performance regardless of the suspension setup |
| 2005 | Relied heavily on resonant tuning. Had peak performance at 7,500 rpm and 11,500 rpm. Lacked transient response due to fuel mapping issues caused by the peaky performance |
| 2006 | Good performance from 6,500 rpm to 12,000 rpm. Below 6,500 rpm, there was a large section where the engine had a negative pressure wave at intake valve close that reduced performance. This made it difficult for corner exiting |
| 2007 | Made good low and midrange power, from 5500 rpm to 9500 rpm. Very easy to drive around the track, compared to the previous years. Lacked power above 9,500rpm |

Source: (Damji, 2008)

In the past few years the design of the University of Toronto has been evolutionary in nature, instead of revolutionary – meaning that the car has been designed based on experience from previous years. Once a system has been optimized to its greatest potential, a new design is created. The 2007 race car is currently the most optimized car and will therefore be used as a performance benchmark for UT2008. This means that changes to the system parameters will be evaluated based on their performance relative to the 2007 system. This thesis project will be successful if the performance of the 2008 system is superior to that of the 2007 system (Damji, 2008).

## 2.6 AIR INTAKE SYSTEM LEAKAGES

The air intake system development, the fault that must be concern in this process is the leakages of the system. This concern is more on the intake manifold in the air intake system because it placed after the air flow is through air regulator or throttle body. Intake manifold leakage often occurs on the fabrication process that includes the bonding process. The assembled component by using the welding method can create cavity that will create malfunction to the engine. It is because when air leakages are happen on the intake manifold, it will increase the amount of uncontrolled oxygen for combustion process. It may cause in high engine rotational speed and unpredictable manifold pressure. This malfunction may let end-user experience (Ahmed, 2011):

i. Hissing noise
ii. Engine stumbling
iii. Rough/fast idling or stalling
iv. Poor gas mileage
v. Hesitation/poor pick up

Whereas, in actual air leakages in manifold absolute pressure sensor to generate false measurements that can result in AFR deviation. Because of that, the mechanism will cause the emissions to increase. An inevitable consequence of manifold leakages is the loss of generated power due to lean air fuel mixture.

## 2.7 SWIRL EFFECT

Swirl effect is occurred on the air flow in certain design and dimension of manifold. This kind of mechanism is very useful in order to help the combustion process for an engine especially in spark ignition engine. Swirl pattern can be measured by using swirl flow that gives the number of swirl occurred. To build a manifold that can produced swirl is by select a good shape of the bend design and also the numbered of degree of the bending. A good swirl flow can optimized the usage of a manifold by mixing the air-fuel mixture to gain a good combustion process or called homogeneous combustion to avoid the engine knocking. Swirl also has some kind of different pattern such as spiral-spiral, spiral-tangential and tangential-tangential. Figure 2.23 is showing the spiral-tangential concept of swirl.

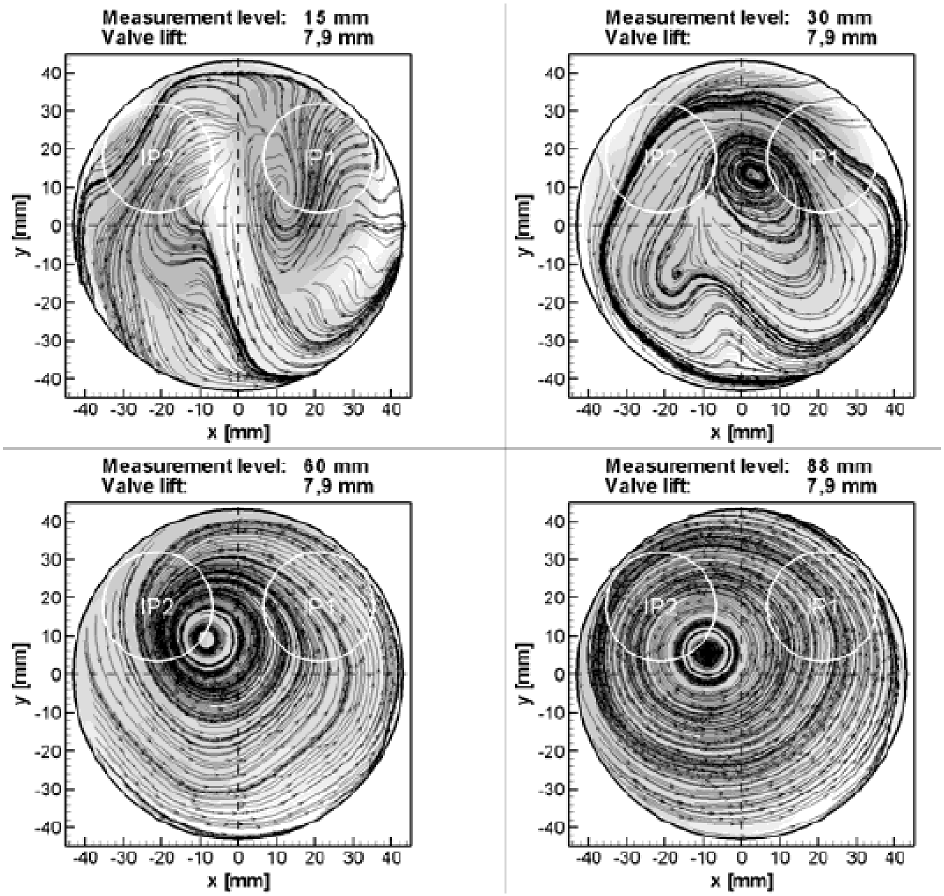


**Figure 2.23:** Spiral-tangential swirl pattern

Source: (Blechstein et. al, 2007)

The main swirl flow agent is in the valve lifting. Opened and closing valve at in certain opening gave certain swirl pattern. The effect of in-cylinder flow on the course and result of combustion is not defined by the swirl number and flow coefficient alone. The level of swirl must be tailored to the injection system and combustion chamber geometry. However, the flow field structure and symmetry also glay significant part. Flow field characteristic that have a positive effect is more pronounced swirl profile of valve lift.

## 2.8 3-D CFD ANALYSES

Nowadays, much kind of technologies were introduced to facilitating people in solving some sort of problem. One of a kind is using 3-D CFD analysis. In building an intake manifold, by using this method also can help in reduced cost and time to conduct a test. For this test, by using this simulation we can obtain results of steady and unsteady simulations. For example, we can know about pressure drop through this simulation that maybe affected by the sharp connection of a runner of an intake that make it difficult to make flow. We also can know how the curve of the runner is creating swirl pattern whether it is a good pattern or not. Besides that, the position of throttle body also can be determined in order to optimize the intake manifolds.

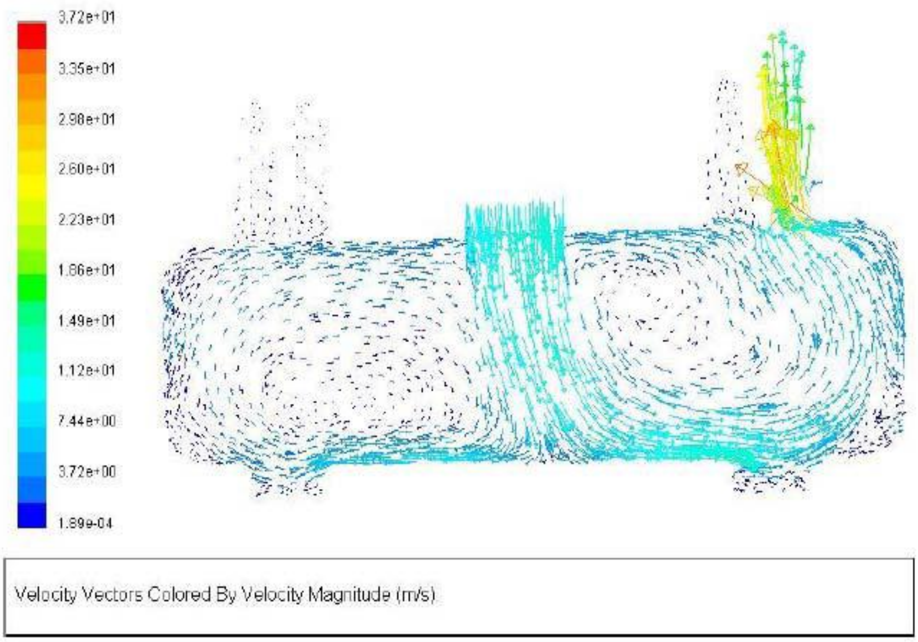


**Figure 2.24:**3-D CFD simulation (swirl pattern)

Source: (Safari et. al, 2003)

## 2.9 SUMMARY

From this chapter, the basic operation of intake manifold and its function can be learn and can be used as the guidence to understand the mechanism of this system. Besides that, something new also can be learn from this project in order to build a good design of an intake manifold. This chapter also contain the characteristic of an intake manifold an its system to give better understanding for this project.

# CHAPTER 3

# METHODOLOGY

## 3.1 INTRODUCTION

This section is briefly discussed about what method will be used in this project. Base on the scope of project, to obtain the result for this project, the design of the manifold must be invent in order to get a model for the next process. Then this design will be approved first by supervisor to keep this project is on the right track. After that, this design must be transform to real component through the fabrication process. The complete intake manifold must being through some sort of inspection to make sure that there is no malfunction cases when the experiment is done to the intake manifold. The data produced from the experiment will be analyze and discussed later on.

## 3.2 METHOD FLOW CHART

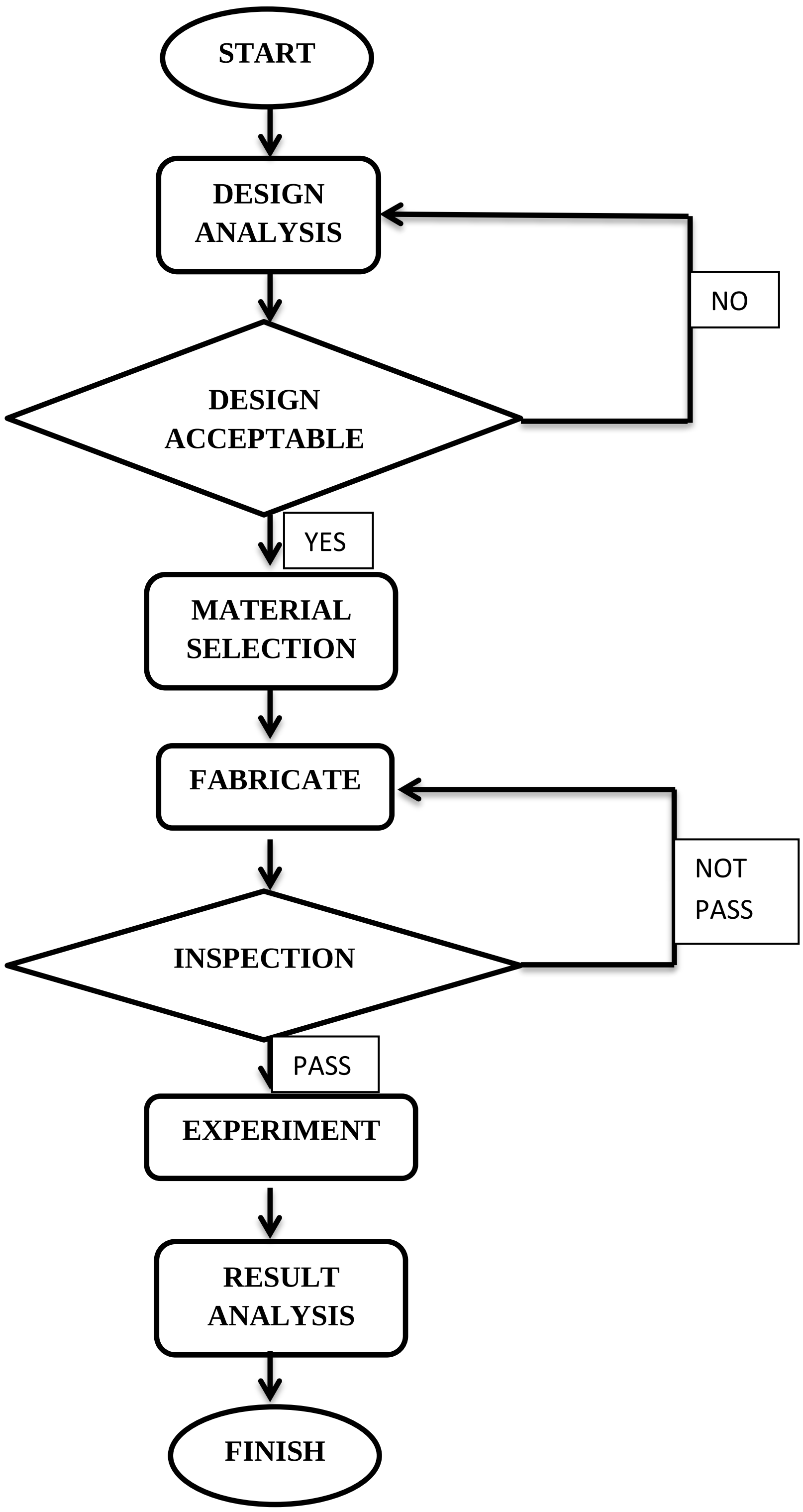


**Figure 3.1**: Method flow chart

## 3.3 MANIFOLD DESIGN

The design process of this material is based on the journal that had being studied. All important input that effect the performance of the intake manifold must be investigates through the journals and book that have been studied. The design should have right dimension and shape to keep the good result to be achieved. Base on journal have been discuss before, the swirl pattern and the flow rate of intake manifold is put in the highest consideration.

Basically the model that had fabricated was combined between two models of intake manifold in the market place with some dimension modification. The type of model of intake manifold is performance type and the standard intake manifold for Proton Waja 1.6 SOCH intake manifold. The usage of those two different models because it had different in advantage and disadvantage that can be used to optimized the usage of the intake manifold.

**Figure 3.2**: Performance intake manifold

Source: Fiber dynamic incorporated patent product

**Figure 3.3**: Proton Waja 1.6 SOCH intake manifold

## 3.4 MATERIAL SELECTION

In the intake manifold fabrication, the material selection section is the basic priority in order to making a functional intake manifold. Today, a lot of manufacturer of intake manifold have taking a step on the material selection in intake manifold. In the earlier century of making the intake manifold, the material that choose always more on the mild steel as it easy to be fabricated and low in costing. Then, they have moved on using the aluminum as it lighter and good surface finish. Nowadays, the technology have found that plastic material is the most suitable material in making intake manifold. But the plastic material must be reinforced that will have increment in costing. Material that had being selected for intake manifold fabrication of this project was according to the suitability and the performance and also the cost of the product. In order to get this project done, the smallest value of cost was being chosen as the factor of the material selection. Basic mild steel is used as it strong and easy to be fabricated. The material was used to make the plenum, custom valve, steel elbow and the intake manifold runner. The mild steel tube of several dimension is choose as it very suitable in guiding air flow. Other material such as plastic and aluminum must be terminate from material selection because of high in costing and complex to be fabricated. Appendix B2 is the figure of the tube that had been chosen.

## 3.5 FABRICATION

The fabrication process in this project was by using arc welding method in assembling task. This is because steel pipe is very suitable to be assembled with the used of this type of welding. Bonding process is very important to create zero leakage of the air flow that will make problem in the experimental test or in the real car testing. First step in the fabrication process after the material selection process is to cutting the steel pipe into several parts according dimension that have being discussed in the design process. The steel pipe consists of several parts to be cut. It is involved in making the plenum and runners. Dimension of a plenum always bigger than a runner. The fabrication of runner is more complex than the plenum fabrication process. This is because runners consist of curve pipe that required bending process for the pipe to get in curve shape. After all part have being completely fabricated. All of them are being assembled by using the arc welding process. The current of the arc welding machine must be regulated at the suitable value to avoid the bonding to have leakage. The pipe then completely assembled as refer to appendix B3.

The design of the intake manifold of this project consists of two path of air flow. For that reason, a custom valve is fabricated to be fit into the intake plenum. The valve had the same shape with the plenum that function to close either of paths that would not be used. The valve is moved by using manual human force.

For precaution step, all the outer part of intake manifold is give with thin coating to keep from rusting and in order to avoid leakage through small hole at the bonding part. This is one of the method that avoiding leakage besides to avoid corrosion on the intake manifold. All of the figures of fabrication process can be obtain from the appendix B3.

## 3.6 EXPERIMENT

### 3.6.1 REFERENCE ENGINE

For this experiment, a reference engine must be select to be converts from real engine into an experimental set up. The Proton Waja 1.6 SOCH or Mitsubishi 4G18 was selected as this experiment was to identify the flow rate of an intake manifold that build on that engine specification. Table below are showing the consideration of that engine for this experiment.

**Table 3.1**: Mitsubishi 4G18 engine specification

| Description | Specifications |
|---|---|
| Type | Gasoline Engine |
| Number of cylinder | 4 in line |
| Bore (mm) | 76.0 |

### 3.6.2 ACRYLIC CYLINDER BLOCK

The acrylic material is used to duplicate the real block cylinder to be match with the head cylinder during the experiment. The measurement of the acrylic head cylinder is depending on the bore size of the 4G18 block cylinder.


**Figure 3.4**: Acrylic block cylinder for 1600cc

### 3.6.3 EXPERIMENTAL SET UP

This sub-chapter is showing that this experimental set up. All of above component that has being discussed before is involved in this experimental set up. This experiment is using the flow bench machine or recognized as SuperFlow SF-1020. It is to measure the flow rate that can be produced by the intake manifold that have been fabricated and will be compared to the standard intake manifold as the base line of this project. This type of flow bench is choose because the SF-1020 measures and records air flow at OEM engineering accuracy, faster than any other flow bench on the market. It can test up to 240 hp (179 kW) per cylinder at test pressure up to 65” (165 cm) of water. The unique variable orifice adjust flow range between 25 cfm and 1,000 cfm (12-472 l\s), based on FlowCom™ input, to fit the valve size. The SF-1020 comes standard with the FlowCom™ digital airflow measurement system for accurate, repeatable and fast testing (SuperFlow Technologies Group 2012).


**Figure 3.5**: Flow bench


**Figure 3.6**: Flow bench data screen

The data that was collected between the two intake manifold was in every sequence of valve lift. The range of data collected is from 0 mm until 8 mm of valve lift. Every mm of valve lift is set up by using dial gauge as it consists of precise measurement. A bold was positions align the valve lift and the dial gauge is put on top the bold to take the measurement of the valve lift.

**Figure 3.7**: Dial gauge position

### 3.6.4 EXPERIMENT PROCESS

This sub-chapter will show the mechanism of this experiment and how it will operates on the intake manifold. The data was collected was from the standard intake manifold and fabricated intake manifold. Firstly, the flow bench will operates as the engine in this experiment. It will create the vacuum situation as the real engine creating vacuum on top of the piston. The acrylic block will operates as the real engine cylinder block as it given the total volume to create vacuum in the real engine. Then, when the flow bench is running it will do the suction of air from the outer space. The air will flow pass the intake manifold and going into the valve lift. Then the air will flow through the head cylinder and acrylic block and will be received at the flow bench. The pressure drop from outer space onto the flow bench will be taken by flow bench as it converting the measurement from pressure drop to the flow rate. This experiment was conducted in different test pressure. Test pressure means the pressure that maintains by the flow bench to perform suction process.

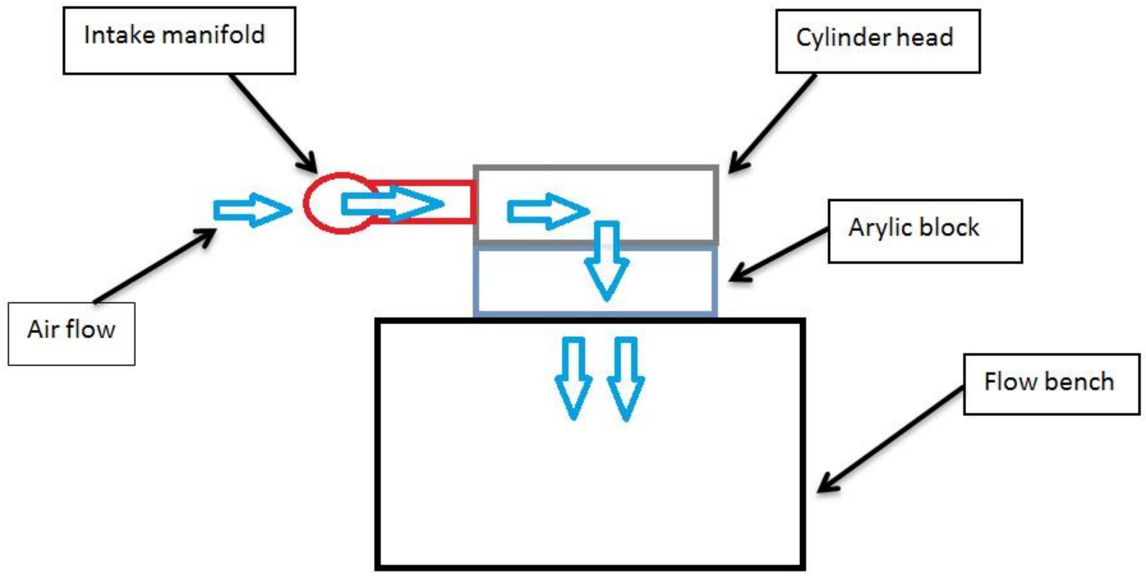


**Figure 3.8**: Experiment diagram

**Figure 3.9**: Standard intake manifold test

**Figure 3.10**: Variable intake manifold test

### 3.6.5 EXPERIMENT PROCEDURE

This is test on two different intake manifold that is standard intake manifold and custom variable intake manifold. All procedure are shown as below

1. All apparatus such as head cylinder, intake manifolds, acrylic block and are set up on the flow bench as figure 3.8. For the first experiment, the test is on the head cylinder without the presents of intake manifolds.

2. All set of apparatus is well placed and sealed by using silicone paste. It is to create zero leakage during experiment.

3. The flow bench is now set on the configuration panel. The set is on the range that is set on range 2. Test pressure is put on data requirement such as test pressure = 50cm$H_2O$.

4. Both valve lift set at fully closed condition.

5. Then, the start button is on to generate the steady pressure by the flow bench.

6. The reading of flow rate at zero valve lift is taken.

7. The experiment data is taken by increasing a valve lift at sequence of 1mm until maximum valve lift at 8mm.

8. After all data of the single valve lift is taken. The valve lift is set on zero again. The experiment now is continued by using double valve lift that repeat the procedure no. 7.

9. After the data of head cylinder is taken. Second experiment is put on other test product with different test pressure set depends on the required data. The experiment is now repeating the procedure no. 4 until no. 8.

# CHAPTER 4

# RESULT AND ANALYSIS

## 4.1 INTRODUCTION

The flow rate that produced from both intake manifolds is representing the data of this experiment. All of the data that can be converts into graph will be discussed by comparing with the base line data that is the flow rate of the standard intake manifold. Every unusual shape of graph and improvement of the graph will be discussed by referring with literature review and facts that suitable with the situation in this chapter.

## 4.2 RESULT

All of the data collected is taken in different condition. The variable that used to be manipulated in this experiment was test pressure, type of intake manifold (type of path of the air flow such as long or short manifold) and the valve lift.

**Table 4.1:** Runner characteristic

| Type | Shape | Swirl |
|---|---|---|
| Standard | Medium curve | Medium |
| Long | Sharp curve | High |
| Short | Straight | Low |

### 4.2.1 FLOW TEST RESULT

Table 4.2 is showing the result of flow rate at single and double valve lift for head cylinder at constant test pressure equal to 50 $cmH_2O$.

**Table 4.2:** Flow tests result for head cylinder

| **Test pressure** = 50 $cmH_2O$ | | |
|---|---|---|
| | **Flow rate (liter/second)** | |
| **Valve lift(mm)** | Single valve lift | Double valve lift |
| 0 | 1 | 1 |
| 1 | 3 | 4 |
| 2 | 5 | 10 |
| 3 | 9 | 17 |
| 4 | 12 | 24 |
| 5 | 15 | 30 |
| 6 | 17 | 36 |
| 7 | 19 | 39 |
| 8 | 20 | 39 |

Table 4.3 show the result of flow rate for set of head cylinder with standard intake manifold at single and double valve lift at constant test pressure of 10 $cmH_2O$.

**Table 4.3:** Flow test 1 results for head cylinder with standard intake manifold

| **Test pressure** = 10 $cmH_2O$ | | |
|---|---|---|
| | **Flow rate (liter/second)** | |
| **Valve lift(mm)** | Single valve lift | Double valve lift |
| 0 | 0 | 0 |
| 1 | 1 | 2 |
| 2 | 3 | 6 |
| 3 | 5 | 10 |
| 4 | 7 | 14 |
| 5 | 9 | 17 |
| 6 | 10 | 19 |
| 7 | 11 | 20 |
| 8 | 12 | 21 |

Table 4.4 show the result of flow rate for set of head cylinder with standard intake manifold at single and double valve lift at constant test pressure of 25 cm$H_2O$.

**Table 4.4:** Flow test 2 results for head cylinder with standard intake manifold

| **Test pressure** = 25 cm$H_2O$ | | |
|---|---|---|
| | **Flow rate (liter/second)** | |
| **Valve lift(mm)** | Single valve lift | Double valve lift |
| 0 | 1 | 1 |
| 1 | 2 | 4 |
| 2 | 5 | 9 |
| 3 | 8 | 16 |
| 4 | 11 | 22 |
| 5 | 14 | 27 |
| 6 | 16 | 30 |
| 7 | 18 | 33 |
| 8 | 19 | 34 |

Table 4.5 show the result of flow rate for set of head cylinder with standard intake manifold at single and double valve lift at constant test pressure of 50 cm$H_2O$.

**Table 4.5:** Flow test 3 results for head cylinder with standard intake manifold

| **Test pressure** = 50 cm$H_2O$ | | |
|---|---|---|
| | **Flow rate (liter/second)** | |
| **Valve lift(mm)** | Single valve lift | Double valve lift |
| 0 | 1 | 1 |
| 1 | 3 | 5 |
| 2 | 7 | 13 |
| 3 | 11 | 23 |
| 4 | 16 | 32 |
| 5 | 20 | 38 |
| 6 | 23 | 38 |
| 7 | 26 | 38 |
| 8 | 27 | 38 |

Table 4.6 show the result of flow rate for set of head cylinder with variable intake manifold at single and double valve lift at constant test pressure of 10 $cmH_2O$.

**Table 4.6:** Flow test 1 results for head cylinder with variable intake manifold

| **Test pressure** = 10 $cmH_2O$ | | | | |
|---|---|---|---|---|
| | **Flow rate (liter/second)** | | | |
| | Single valve lift | | Double valve lift | |
| **Valve lift(mm)** | Short path | Long path | Short path | Long path |
| 0 | 0 | 0 | 0 | 0 |
| 1 | 1 | 1 | 2 | 2 |
| 2 | 3 | 3 | 6 | 5 |
| 3 | 5 | 5 | 10 | 9 |
| 4 | 7 | 7 | 14 | 12 |
| 5 | 9 | 8 | 18 | 14 |
| 6 | 10 | 9 | 21 | 15 |
| 7 | 12 | 10 | 23 | 16 |
| 8 | 12 | 11 | 24 | 17 |

Table 4.7 show the result of flow rate for set of head cylinder with variable intake manifold at single and double valve lift at constant test pressure of 25 $cmH_2O$.

**Table 4.7:** Flow test 2 results for head cylinder with variable intake manifold

| **Test pressure** = 25 $cmH_2O$ | | | | |
|---|---|---|---|---|
| | **Flow rate (liter/second)** | | | |
| | Single valve lift | | Double valve lift | |
| **Valve lift(mm)** | Short path | Long path | Short path | Long path |
| 0 | 1 | 1 | 1 | 1 |
| 1 | 2 | 2 | 4 | 4 |
| 2 | 5 | 5 | 9 | 10 |
| 3 | 8 | 8 | 17 | 15 |
| 4 | 11 | 10 | 23 | 20 |
| 5 | 14 | 13 | 29 | 23 |
| 6 | 16 | 15 | 34 | 25 |
| 7 | 19 | 16 | 37 | 26 |
| 8 | 20 | 17 | 39 | 27 |

Table 4.8 show the result of flow rate for set of head cylinder with variable intake manifold at single and double valve lift at constant test pressure of 50 cm$H_2O$.

**Table 4.8:** Flow test 3 results for head cylinder with variable intake manifold

| **Test pressure** = 50 cm$H_2O$ | | | | |
|---|---|---|---|---|
| | **Flow rate (liter/second)** | | | |
| | Single valve lift | | Double valve lift | |
| **Valve lift(mm)** | Short path | Long path | Short path | Long path |
| 0 | 1 | 1 | 1 | 1 |
| 1 | 3 | 3 | 6 | 5 |
| 2 | 6 | 6 | 13 | 12 |
| 3 | 11 | 11 | 24 | 21 |
| 4 | 16 | 15 | 33 | 28 |
| 5 | 20 | 19 | 38 | 32 |
| 6 | 24 | 22 | 47 | 35 |
| 7 | 27 | 24 | 51 | 37 |
| 8 | 28 | 25 | 52 | 38 |

## 4.3 ANALYSIS

All of the data that set on the table is show as in a graph method in this sub-chapter. All the graph will be analyze according to the shape of graph and data comparison in different condition. The analysis of the data according to performance must be taking two elements as consideration. As we know that the intermittent or pulsating nature of the airflow through the intake manifold into each cylinder may develop resonances in the airflow at certain speeds. These may increase the engine performance characteristics at certain engine speeds, but may reduce at other speeds, depending on manifold dimensions and shape. (Ceviz and Akin, 2010) The number flow rate is put on the priority of determining the performance. Secondly, the swirl pattern value that produced can be used as a catalyst method in this system which means as an agent in helping to increase the performance. This analysis also used to determine the best runner that will be used for an engine at different speed of engine.

4.3.1 Constant test pressure equal to 50 cm$H_2O$ and single valve lift

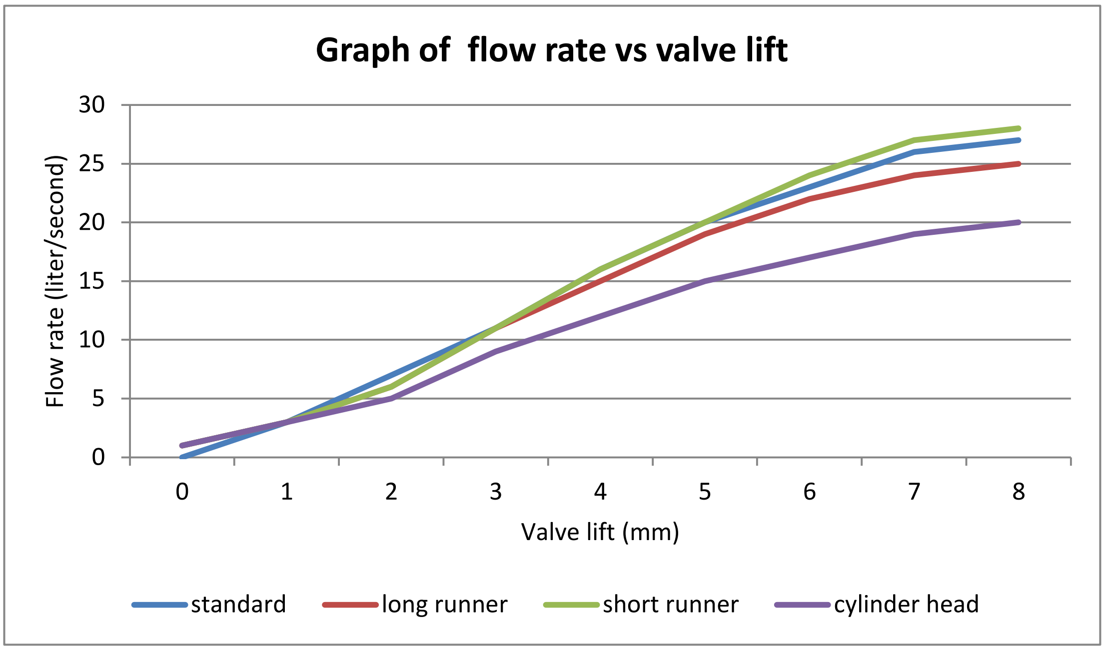


**Figure 4.1**: Comparison between intake manifold flow rate with the head cylinder

This graph is showing the flow rate within four different conditions. The flow rate measured is through standard intake manifold, long runner intake manifold, short runner intake manifold and lastly by using cylinder head alone without intake manifold. Those graphs are constant in test pressure and valve lift condition is to study about the result in different shape and diameter for flow rate producing. Logically, the flow rate that produced from the cylinder head alone must be higher than the flow rate that through the intake manifold because intake manifold will make the flow became slower as it will pass through the curve inside the intake manifold that can cause air friction. But this experiment proved that cylinder head alone cannot achieve higher flow rate although it consist lower in air friction. Those intake manifold produced higher flow rate because of it is functioning as a nozzle in the system. The intake manifold is just like a throat in a nozzle system that will increase the velocity of the air flow thus increase the flow rate itself. The smaller capacity of intake manifold compared to the air surrounding will create pressure different that effecting the flow to be higher in velocity. Further study on nozzle effect can be refers to thesis Baik and Kim. This mechanism also can be support by using the theory of conservation of energy that depends on the pressure drop. The increase of velocity will be used to help in pressure replacement in the intake manifold that will create better flow rate. That is why the intake manifold is really important in engine system as it determined the air mixture flow rate of the engine received.

4.3.2 Constant valve lift condition (double)

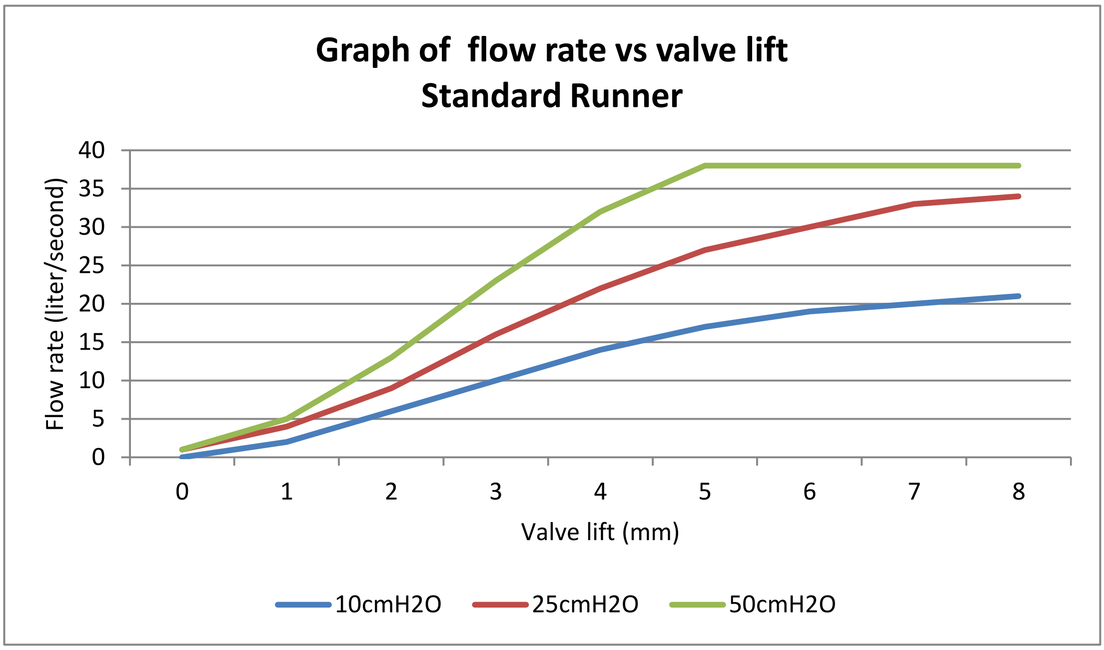


**Figure 4.2:** Graph of flow rate for standard runner at different test pressure

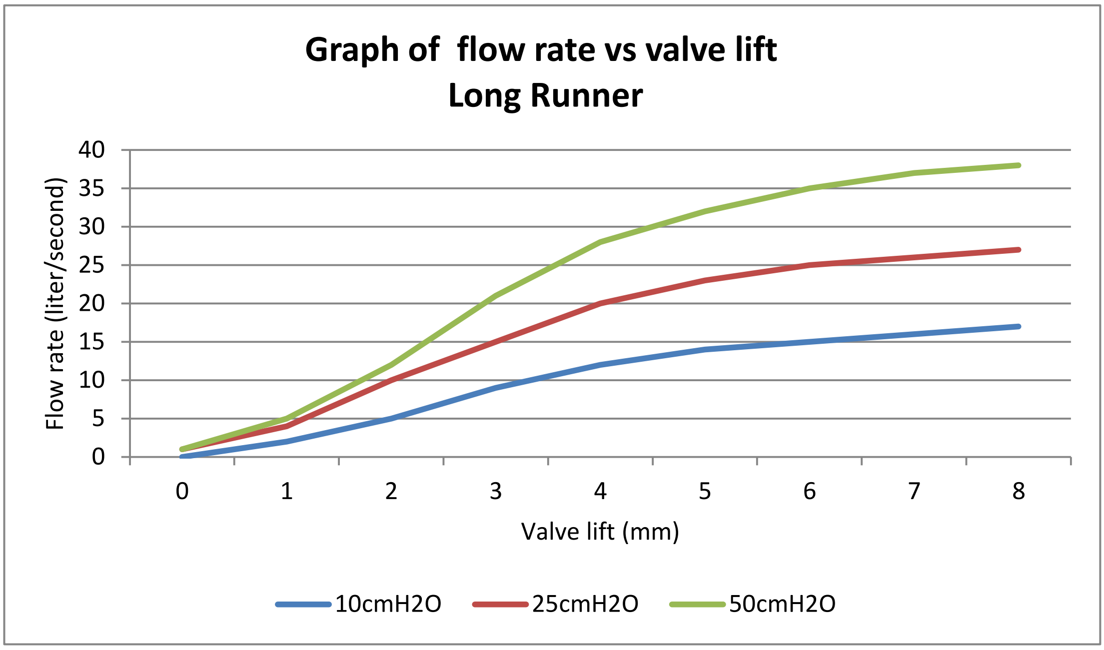


**Figure 4.3:** Graph of flow rate for long runner at different test pressure

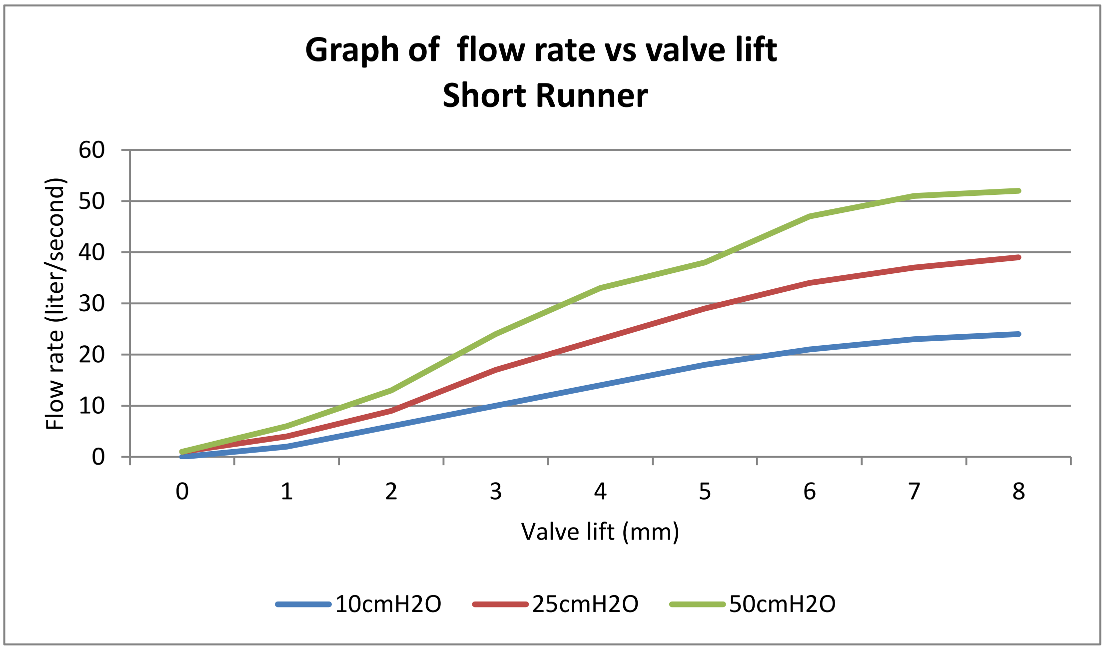


**Figure 4.4:** Graph of flow rate for short runner at different test pressure

Those three graphs are taken in three different condition of test pressure that is test pressure of 10 $cmH_2O$, 25 $cmH_2O$ and 50 $cmH_2O$. All of those graphs are constant in valve lift condition that is at double valve lift. Each graph is representing the type of runner that is used. Those three graphs are to show that the test pressure value is varied with the flow rate. At all type of runner, the higher test pressure create higher flow rate. The test pressure that is used is representing the value of engine speed in real situation. Thus, from this graph we can say that the increase of engine speed will used to have larger number of flow rate.

### 4.3.3 Constant test pressure equal to 10 $cmH_2O$ and single valve lift

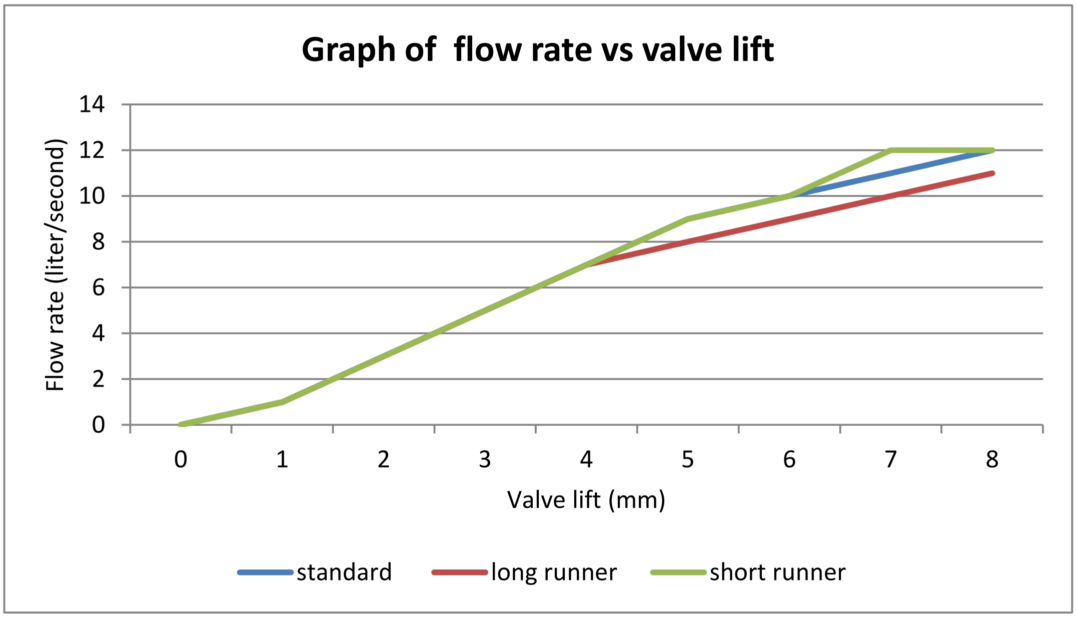


**Figure 4.5:** Graph of flow rate for different flow path at constant test pressure equal to 10 $cmH_2O$

This graph is showing the flow rate produced by three different type of runner at constant test pressure equal to10 $cmH_2O$ and in single valve lift. This comparison is between standard, long and short runner. This data are taken at the lowest test pressure of all data that means it is in low engine speed in the real situation. From the graph, we can see that almost in all valve lift variable the standard intake manifold flow rate has same value with the short intake manifold. In real situation, by using the standard runner will gave more value in performance. This is because although it has the same value with the short intake manifold, the standard runner will create more performance is because the swirl pattern that produced from the curve of the runner that will use to help in increasing the performance of an engine that discussed more in the thesis of The Influence of Swirl Control Strategies on the Intake Flow in Four Valve HSDI Diesel

Engine by E. Mattarelli, M. Borghi, D. Balestrazzi and S. Fontanesi of University of Modena and Reggio E.,Italy. The long runner is considered as the lowest performance produced is because of the low flow rate produced. This is because long runner is fabricated by using higher value angle of curve compared to the standard runner. Thus, at low engine speed the standard runner has the higher value in performance.

### 4.3.4 Constant test pressure equal to 50 cm$H_2O$ and double valve lift

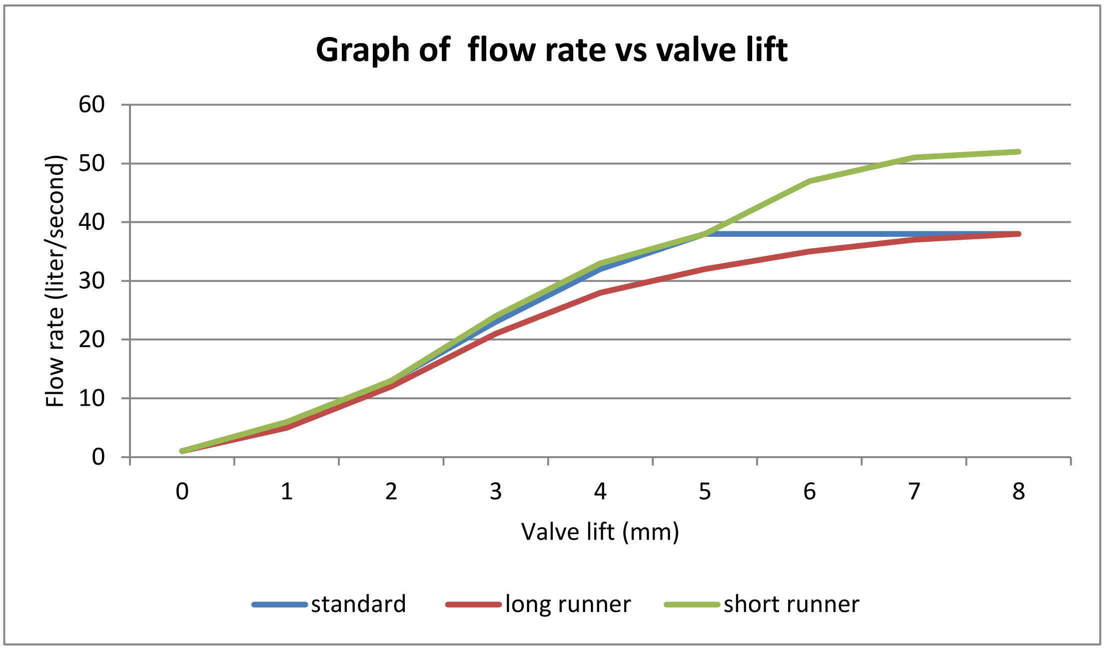


**Figure 4.6:** Graph of flow rate for different flow path at constant test pressure equal to 50 cm$H_2O$

This graph consists of three different data of flow rate within different condition of air flow path. The comparison is between standard and variable intake manifold that consist of two different paths that is long and short path. This graph was taken at the maximum test pressure and maximum flow rate of both intake manifolds. By the result, we can see that short path give major different of flow rate compared to the standard and short path. This is because short path has the lowest air resistance because it has shorter in length of path and also straight in shape compared to the standard and long path that consist of curve shape that slower the velocity of air at maximum test pressure. Test pressure so related to the engine speed in real situation. Thus, the higher flow rate produced by short runner can gives more engine speed that very related to the performance of an engine compared to the standard and short runner. From this data, it is proved that shorter and straight runner can gives more engine performance at the top end power of an engine.

4.3.5 Constant test pressure equal to 50 cm$H_2O$ and short runner

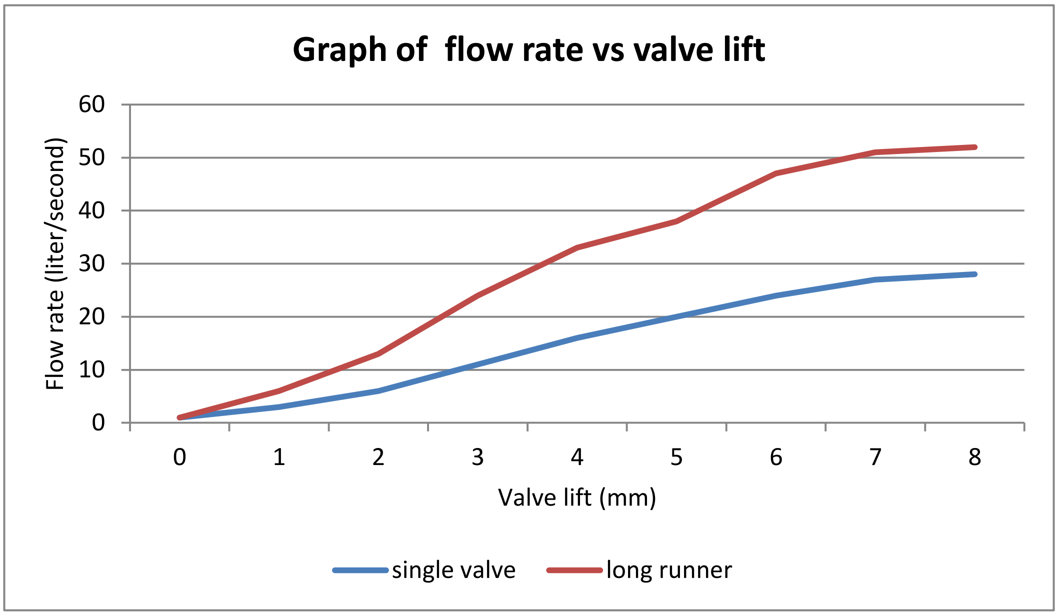


**Figure 4.7:** Graph of flow rate for short runner at single and double valve lift.

This graph is creates from data of flow rate between two different condition that are in single valve and double valve condition. This comparison is using same path of air flow and same test pressure equal to 50 cm$H_2O$. This result is to study about the swirl that can be builds at this condition. This is because the short runner will be used at this level of test pressure and this is categorized as high rpm in real engine situation. We know that short runner create the lowest number of swirl according to the shape of flow that only pass through the straight runner that have no swirl agent compared to the standard and long runner that have curvy shape at their manifold that function as a swirl agent. By this data, we can see that single valve and double valve opening will create large different number of flow rate. Thus, by controlling the valve opening the different number of flow rate between these two conditions can be used as a swirl agent for short intake manifold to increase the performance of the engine. ( Mattarelli et. al, 2003).

# CHAPTER 5

# CONCLUSIONS AND RECOMMENDATIONS

## 5.1 Introduction

This chapter is concluding this entire project task. Consideration is put on the design, fabrication and data analysis.

## 5.2 Conclusions

From the fabrication process, the material that used can be considered as suitable with this project because of minor leakage were occur on the intake manifold.

The design of this project succeeds to be used in the data analysis as we can used to compare with the standard intake manifold. Besides that, the flow rate that produced by the short runner succeed to overcome the flow rate of the standard of intake manifold.

The data analysis was by obtaining the graph converted from the data collected. The value of the test pressure in the experiment is indicating the engine speed in real situation. The data proved that higher engine speed will used to have higher flow rate through the intake manifold. The flow rate in the intake manifold used to show the level of performance of an engine because increasing flow rate will cause increasing in performance.

Thus, the data set shows that the standard runner is performing at early stage of engine speed. Although the value of flow rate is same but the standard runner has a better swirl value than short runner. For the short runner, it perform maximum at the end stage of engine speed because the number of flow rate has big different compared to the long and standard runner.

Then, the data also show the different flow rate between three different intake manifold runners. Those intake manifold runners have different maximum performance in a different phase of engine speed. Because of that reason, the variable intake system is used to switch the type of runner in order to optimize the performance of the intake manifold.

As a conclusion, the short runner is very suitable at the end of engine speed or high engine speed while the standard runner is suitable for early stage of the engine speed. In designing an intake manifold, the shape of the curve also can’t simply being applied by increasing the value of curve angle. An example, the long runner supposed to have good in swirl pattern because of the narrower shape of curve but still not good in performance compared to the standard runner that have better flow rate. The narrower curve of intake manifold cause the decrease in flow rate of the air. Thus, good design is obtain from the suitable angle of curve instead of value of the angle.

## 5.3 Recommendations

For this project, it is good to prove the flow rate of the design by using the CFD software as it can determined the losses that cause by the fabricated intake manifold through rough surface and also leakage in the fabricated model.

Besides that, the experiment also can be conduct in the real engine to study the performance of an engine. The experiment can be conduct by using a car that is measured on a Dynamometer machine for its performance.

Other than that, we also can used different size of runner diameter as to study the flow rate of an intake manifold for different diameter of runner.

Lastly, the study of shape intake manifold also can be determined by using the swirl flow meter as to measure the true value of a swirl pattern according to the different shape of an intake manifold.

**REFERENCES**


Ali A. Alsaeed. 2005. Dynamic stability evaluation of an automotive turbocharger rotor bearing system, Virginia Polytechnic Institute and State University.

ArcelorMittals' DOM http://www.arcelormittal.com/tubular

Bell, C. 1997. Maximum Boost. Cambridge, USA: Bently Publishers

Chaurette, J. 2008. Pump and Pump system glossary. Light My Pump

Desmond E. Winterbone and Richard J Pearson. 1999. Design Techniques for Engine Manifolds. Professional engineering publishing.

Douglas Heim and Jaal Ghandi. 2011 A detailed study of in-cylinder flow and turbulence using PIV. University of Wisconsin Madison.

Duk-Sang Kim, Yeun-Jun Yoo and Yong-Seok Cho. Numerical analysis of flow characteristic for intake valve design, Kookmin University, Seoul, Korea and Korea National University of Technology, Seoul, Korea.

E. Mattarelli, M. Borghi, D. Balestrazzi and S. Fontanesi. The Influence of Swirl Control Strategies on the Intake Flow in Four Valve HSDI Diesel Engine. University of Modena and Reggio E.,Italy.

Engelman, H. W. 1953. The Tuned Manifold. Supercharging without a blower. ASME paper 53-DGP-4.

Ferguson, C. R. and Kirkpatrick, A. T. 2001. Internal combustion engines. New York: John Wiley & Sons.

Fiber dynamic incorporated patent product

Fredrik Karlsson. 2001. Modelling the intake manifold Dynamics in a diesel engine. Linkôpings Universitiet.

Georgios V. Varsos. 2010. 1-D Simulation of the intake manifold of a single-cylinder Engine. Imperial College London.

Heywood, John B. 1987. Internal Combustion Engine Fundamentals, McGraw-Hill Book Company.

Jae-Sang Baik and Youn-Jea Kim. 2008. Effect of nozzle shape on the performance of high velocity oxygen-fuel thermal spray system. School of Mechanical Engineering, Center for Advanced Plasma Surface and Technology, Sungkyunkwan University.

L. J. Hamilton and J. E. Lee. 2008. The effect of intake plenum volume on the performance of a small normally aspirated restricted engine. US Naval Academy.

Lumley. 1999. Engines: an introduction. Cambridge University.

M. A. Cevis and M. Akin. 2010. Design of a new SI engine intake manifold with variable plenum, Department of Mechanical Engineering, Faculty of Engineering, University of Ataturk, Ezurum 25240, Turkey.

M. A. Cevis, .Intake plenum volume and its influence on the engine performance, cyclic variability and emissions. Department of Mechanical Engineering, Faculty of Engineering, University of Ataturk, Ezurum 25240, Turkey.

M. Safari and M. Ghamari. 2003. Intake Manifold Optimization by Using 3-D CFD Analysis Iran Khodro Powertrain Company (IPCO).A. Nasiritosi ,Iran University of Science and Technology (IUST).

Matthew A. Porter. 2009. Intake manifold design using computational fluid Dynamics. University of New South Wales at the Australian Force Academy.

Munson, B. R., Young, D. F., & Okiishi, T. H. 2006. Fundamentals of Fluid Mechanics. Hoboken: Jown Wiley & Sons.

Nilufar Damji. 2008. Optimization of the intake system for the 2008 Formula SAE race car. Department of Mechanical And Industrial Engineering, University of Toronto.

Pär Nylander. 2008. CFD modeling of water ingestion in air intake system. Lulea University of Technology.

Qadeer Ahmed. 2011. Fault Diagnosis Methodologies for Automotive Engine Air Intake System. Mohammad Ali Jinnah University, Islamabad.

Ronald Douglas Matthews, General Motors Foundation Combustion Sciences and Automotive Research Laboratories.

Samantha Rochelle Heim. 2010. Conservation of Mass in a Piping Network. Oregon State University.

Smith, P., & Morrison, J. 2002. Scientific Design of Exhaust and Intake Systems. Cambridge: Bentley Publishers.

SuperFlow Technologies Group. 2012.

Ugur Kesgin. 2004. Effect of turbocharging system on the performance of a natural gas engine. Department of Naval Architecture, Mechanical Engineering Faculty, Yildiz Technical University.

Volkswagen of America Manual Book. 2006.

Wanjie Huang, Xin Gong, Xiaolei Guo, Zhenghua Dai, Haifeng Liu, Zhuwei Cao and Chuanhong Wang. Study of the pressure drop phase gas-solid flow through nozzle. Key Laboratory of Cool Gasification of Ministry of Education, Institute of Clean Cool Technology, east China University of Science and Technology, Shanghai 200237, PR China.

APPENDIX A1

**Table 2.1:** 2002 Honda F4i Engine Specifications

| **Description** | **Specification** |
|---|---|
| Engine configuration | Inline four, 4 stroke, DOCH |
| Displacement | 600 cc |
| Fuel system | EFI |
| Cooling system | Water |
| Lubricant system | Wet sump |
| Compression ratio | 12:1 |
| Valves per cylinder | 4 |
| Bore | 67.0 mm |
| Stroke | 42.5 mm |
| Valve timing | |
| IVO | $30^{o}$ BTC |
| IVC | $60^{o}$ ABC |
| EVO | $50^{o}$ BBC |
| EVC | $10^{o}$ ATC |

Source: (Hamilton and Lee, 2008)

APPENDIX A2

**Table 3.2:** Minimum mechanical properties* of steel pipes

| **Tube sizes up to and including 2.750" OD with a Maximum Wall of .125"** | | | | | | | |
|---|---|---|---|---|---|---|---|
| SAE Steel Grade | Yield Strength | | Tensile Strength | | Elongation % in 2" (Strip Section) | | Minimum Hardness |
| | KSI | N\mm$^2$ | KSI | N\mm$^2$ | As-Drawn | Stress Relieved | Rb |
| 1008 | 55 | 379 | 65 | 448 | 10 | 15 | 74 |
| 1010 | 60 | 414 | 70 | 483 | 10 | 15 | 79 |
| 1015 | 65 | 448 | 75 | 517 | 10 | 15 | 81 |
| 1018 | 70 | 483 | 80 | 552 | 10 | 15 | 83 |
| 1020 | 65 | 483 | 75 | 586 | 10 | 15 | 83 |
| 1021 | 75 | 517 | 85 | 586 | 8 | 15 | 87 |
| 1025 | 75 | 517 | 85 | 586 | 8 | 15 | 87 |
| 1026 | 75 | 552 | 85 | 621 | 8 | 15 | 90 |
| **TuffDOM 520 | 75 | 552 | 85 | 621 | - | 15 | 90 |
| 1030 | 80 | 552 | 90 | 621 | 8 | 12 | 91 |
| **1035 | 85 | 586 | 95 | 655 | - | 10 | 93 |
| **TuffDOM 620 | 90 | 665 | 100 | 724 | - | 15 | 96 |

Source: ArcelorMittals' DOM

APPENDIX A3

**Table 3.3:**SuperFlow SF-1020 specification

| **Descriptions** | **Specification** |
|---|---|
| Calibration Test Pressure | 25” of water |
| Range | 0-1,100 cfm |
| Exhaust Capacity | 1,000 cfm @ 25” of water |
| Power | 240 VAC, 75A, single phase |
| Weight | 480 lbs (281 kg) |
| Dimension | 48 x 33 x 43 in. (122 x 84 x 110 cm) |

Source: SuperFlow Technologies Group (2012)

APPENDIX B1

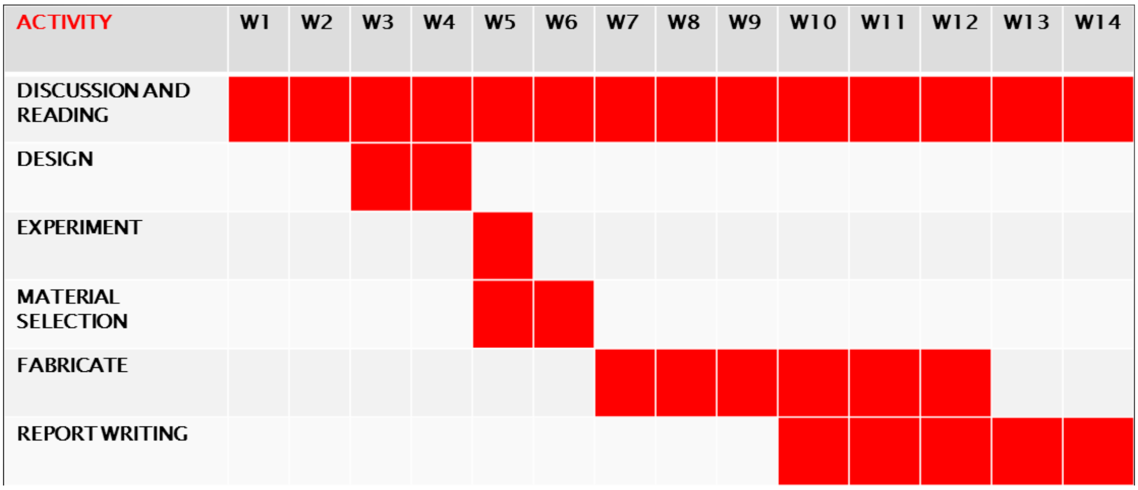


**Figure 1.1**: Gantt chart for semester 1


**Figure 1.2**: Gantt chart for semester 2

APPENDIX B2

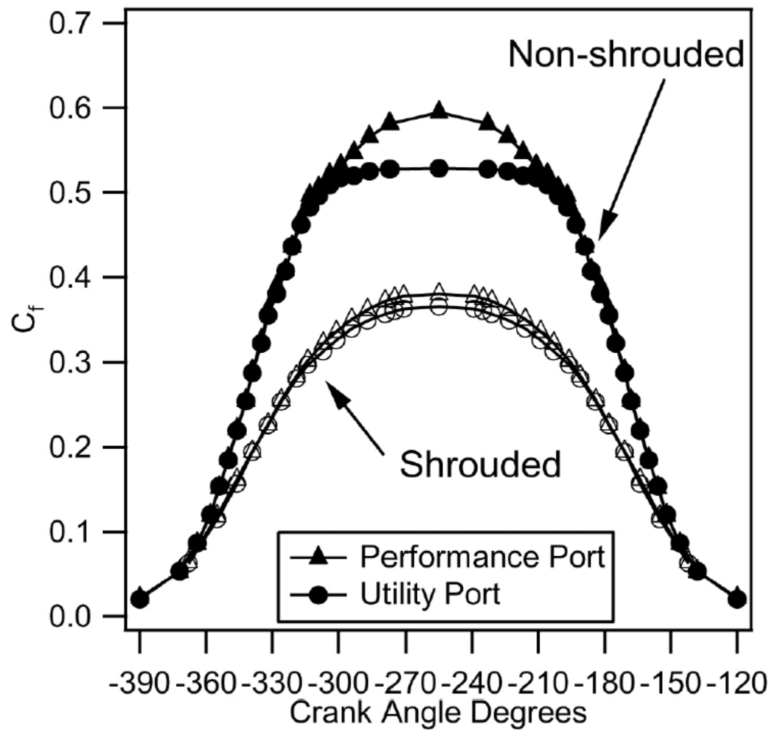


**Figure 2.25:** Flow coefficient for different port geometry and valve type at the 0 degree orientation. 0 crank angle degrees refers to TDC at end of compression

Source: (Heim and Ghandi, 2011)

APPENDIX B3

**Figure 3.11**: Round hollow mild steel

(Source: Chryssafidis S.A.)

APPENDIX B4

**Figure 3.12**: Arc welding machine

**Figure 3.13**: Runners welded to the plenum

**Figure 3.14**:plenum without valve

**Figure 3.15**: Two path runner

**Figure 3.16**: Custom valve inside the plenum

**Figure 3.17**: Valve opening on both path

**Figure 3.18**: Complete welded manifold

**Figure 3.19**: Human force valve switch

**Figure 3.20**: Complete coated manifold