Analysis of Air Flow in Intake System Using Computational Fluid Dynamics

ROSLI ABU BAKAR & KWAN TECK HENG

ABSTRACT

Air motion within the intake system is one of the important factors which govern the performance of an engine. Hence the flow phenomenon inside the intake system should be understood. The whole intake system of 1.3L twin venturi carburetor engine from air induct until manifold is modelled by SolidWorks and 3-D simulation is presented by using COSMOS FloWorks. Flow field velocity varying to engine speed is observed and is compared with theoretical result for validation. CFD simulation is also done in the filtering which the flow in a section of an intake system is considered resisted by a porous media. Besides that, steady state simulations have been accomplished to obtain the flow behavior due to the pressure loss for variation operating condition of idle speed, wide open throttle and sudden acceleration in the intake system. At last, comparison between the flow behavior in the first runner and second runner are observed. Finally, according to the simulation result, some suggestions are recommended to improve the performance of intake system.

INTRODUCTION

Air motion inside the intake system is one of the important factors, which governs the engine performance of multi cylinder SI engines. An intake system conveys drawn air from atmosphere into engine and mix with the fuel for combustion. The purpose of air intake system is to filter and measure the air into engine. Air is filtered by air filter and passed into the intake manifold in varying volume. The amount of air entering the engine is a function of throttle valve opening angle and engine rpm. Air velocity is increased as it passes through the long, narrow intake manifold runners and resulting in improved engine volumetric efficiency. The correct amount of air is very important for correct engine operation to provide maximum power. Hence the flow phenomenon inside the intake system should be fully understood in order to consider the current requirement of higher engine efficiency.

INTERNAL COMBUSTION ENGINE

Classification Engine

The simulation will be conducted on Proton Saga Magma 12 valve 4 Stroke engine and the engine classification is show in **Table 2.1**.

Table 2.1: Classification of Engine

Type of Classification	Specification
Cycle of operation	Otto cycle engine
Type of fuel used	Petrol engines
Method of charging	Naturally aspirated engines
Type of ignition	Spark-ignition engine
Number of cylinder	Four cylinder
Methods of fuel injection	Carburetor
Engine Capacity	1300cc

Intake System Parts

Air Induct

Air induct as shown in **Figure 2.2** is used to draw the air from atmospheric and direct the air into the intake manifold.

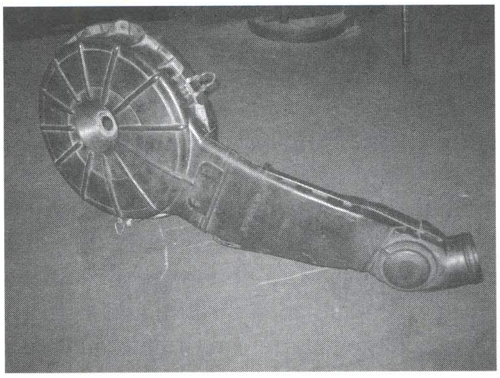

Figure 2.2: Air induct

Air Filter

The intake air must be clean. Airborne contaminants can shorten engine life or even cause premature failure. Air filters as shows in **Figure 2.3** are used on engines to trap contaminants, yet provide a free flow of air into the engine.

Carburetor

This carburetor model is a downdraught or downdraft progressive twin venturi instrument with vacuum-controlled secondary throttle as shown in **Figure 2.4** and **2.5**. The choke control is automatic in operation and is controlled by a coolant-heated wax capsule. [3] The air-fuel mixture burns in the combustion chamber of the engine. The carburetor is a device which is used to supply the engine with a combustible air-fuel mixture in the proper ratio for the purpose of combustion. The quantity of fuel and air can be mixed in different ratios inside the carburetor. The speed of the engine changes according to the quantity of the fuel in the air-fuel mixture.

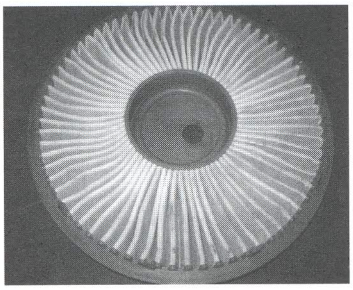

Figure 2.3: Ring type air filter.

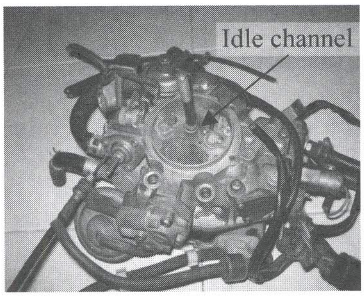

Figure 2.4: Down draft carburetor.

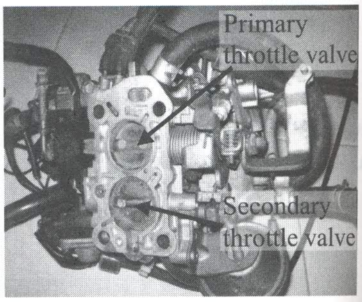

Figure 2.5: Twin venturi with primary and secondary throttle valve.

Internal System of Fixed-Venturi Carburator

The fixed venturi carburetor system needs several special systems or circuits which change the air-fuel ratio to suit varying operating conditions.

Intake Manifold

Throttle body is connected to the intake manifold with the intake ports in the cylinder head. The manifold has a set of passages or runners through which air-fuel mixture flows as shown in **Figure 2.9**. Fuel mixes with the air in the carburetor or throttle body injection as it enters the intake manifold. The intake manifold is a one piece casting of iron or aluminum alloy.

The passages or runners carry air-fuel mixture. They are as short as possible and designed to avoid sharp corners.

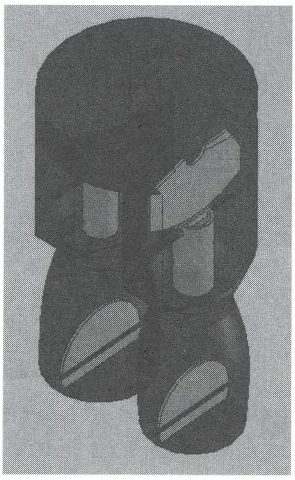

Figure 2.8: Acceleration condition for carburetor

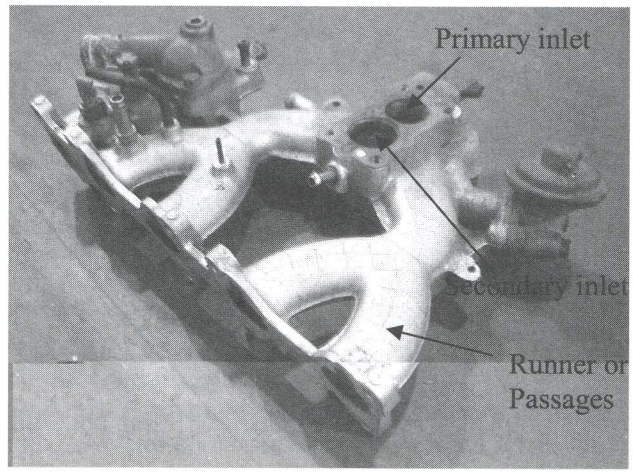

Figure 2.9: Primary and secondary inlet through intake manifold.

Flows in Intake System

Intake system typically consists of an air filter, a carburetor and throttle or fuel injector or throttle with individual fuel injectors in each intake port and an intake manifold for a spark ignition engine. Pressure losses occur as the mixture passes through or by each of these components during induction process. The flows in intake system are pulsating. Many aspects of these flows can be analyzed on a quasi-steady basis, and the pressure indicated in the intake system represents time averaged values for a multi cylinder engine. The drop in pressure along the intake system depends on engine speed, the flow resistance of the elements in the intake system, the cross sectional area through which the fresh charge moves, and the charge density.

Theoretical Flow

Ideal Gas

We presume that such a gas has no viscosity. This is an idealized situation that does not exist, however there are instances in engineering problems where the assumption of an ideal fluid is helpful. When we refer to the flow of an ideal gas, the effects of viscosity are introduced into the problem. This results in the development of shear stress between neighboring gas particles when they are moving at different velocities. In the case of an ideal gas flowing in a straight conduit, all particles move in parallel lines with equal velocity show as **Figure 2.10**. In the flow of a real gas the velocity adjacent to the wall and produce a velocity profile such as shown in **Figure 2.11**.

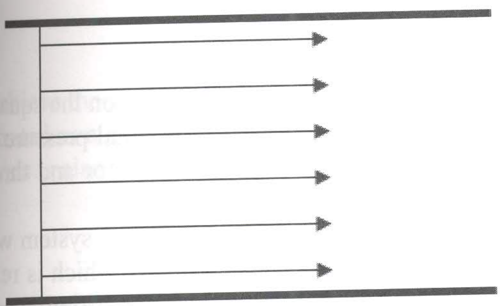

Figure 2.10: Typical velocity profile for ideal gas

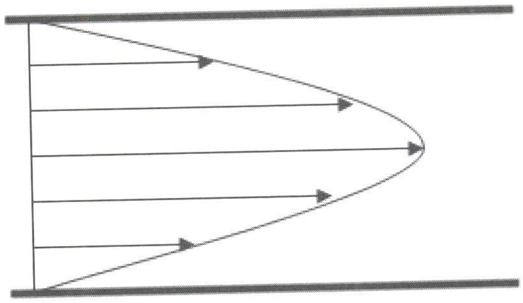

Figure 2.11: Typical velocity profile for real gas

Incompressible Flow

Incompressible gas flow assumes the gas has constant density. Though gas is slightly compressible we usually assume them to be incompressible.

Steady State Flow

Steady flow means steady with respect to time. Thus all properties of the flow at every point remain constant with respect to time.

Flow Rate

Flow rate is the quantity of fluid flowing per unit time across any section.

$$Q = \int_A u dA = AV \qquad (2.1)$$

Where Q is the volume flow rate (m^3/s); u is the time mean velocity through an infinitesimal is dA, while V is the mean velocity (m/s) over the entire sectional area A (m^2).

Control Volume

The concept of a free body diagram as used in the static of rigid bodies and in fluids static, is usually inadequate for the analysis of moving fluids. Instead, the concept of control volume is used in the analysis of fluid mechanics. Control volume refer to a fixed region in space which does not move or change shape. It is usually chosen for region that fluid flows into and out of. It is called as closed boundaries of control surface.

General Equation of Continuity

The general equation of continuity is used for flow through regions with fixed boundaries. [4]

Steady flow: $\rho_1 A_1 V_1 =: \rho_2 A_2 V_2$ (2.2)

If the fluid is incompressible, ρ = constant, so

$\rho_1 = \rho_2$ and $\partial\rho / \partial t = 0$

Incompressible flow: $A_1V_1 = A_2V_2 = Q$ (2.3)

Frictional Loss

The pressure in the manifold is less than the atmospheric pressure by an amount dependent on the square of the speed due to the friction in each part of the intake system during intake stroke. The total pressure drop is the sum of the pressure loss in each component of the intake system: air filter, carburetor and throttle, manifold, inlet port and inlet valve.

Bernoulli's equation shows that the resistance coefficient for each component in intake system which depends on its geometric details and the flow velocity. The flow is assuming quasi-steady which is related to the mean piston speed. [7] **Figure 2.12** shows an example of the pressure losses due to the friction across the air cleaner, carburetor, throttle and manifold plenum of a standard four cylinder automobile engine intake system.

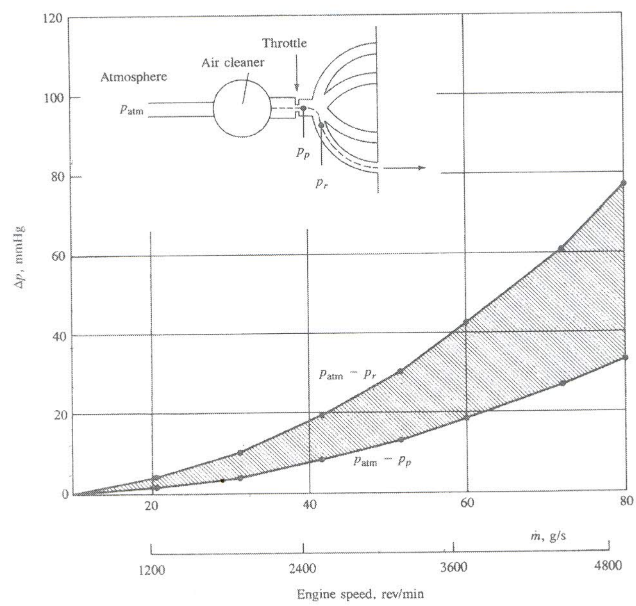

Figure 2.12: Pressure losses in the intake system of a four stroke cycle spark-ignition engine determined under steady flow conditions. [7]

Mean Piston Speed

Mean piston speed is often a more appropriate parameter than crank rotational speed for correlating engine behavior as a function of speed. For example, gas-flow velocities in the intake and the cylinder all scale with mean piston speed, *Sp*. Resistance to gas flow into the engine or stresses due to the inertia of the moving parts limit the maximum mean piston speed to within the range 8 to 15 m/s [7].

Mean piston speed, $Sp = 2LN$ (2.4)

Where;

L = Piston stroke (m)

N = Rotational speed of crankshaft (revolution per second)

METHODOLOGY

To achieve the goals and objectives of this project, a structure of overall methodology has been planned and illustrated as a guideline. The introduction and general procedures of Computational Fluid Dynamics (CFD) analysis is also explained in methodology.

Simulation Procedure

a. 3D CAD drawing of intake system model
- 3D CAD drawing was created for air induct, air filter, carburetor and intake manifold using SolidWorks.

b. Mesh generation
- The assembled intake system was automatic generated in COSMOSFloWorks.

c. Defined boundary conditions
- Total pressure is applied for intake system inlet and outlet due to different operating conditions.
- Porous condition for air filter is also defined.

d. COSMOSFloWorks solver is used to calculate and obtained the flow field result.

e. Visualization and study of the results.

RESULT AND DISCUSSION

Initial Condition

To start the COSMOSFloWorks, initial conditions of the intake system flow field have to be defined which show as **Table 4.1**

Table 4.1: Initial conditions manually defined in wizard setting.

Initial Conditions	User Defined
Units	SI system
Fluid type	Gas
Analysis flow type	Internal
Roughness	0 micrometer
Fluid substances	Air
Default wall conditions	Adiabatic wall
Ambient pressure	101325 Pa
Ambient temperature	293.2 K
Result resolutions	Level 7

Mesh Generation

Figure 4.1 shows the computational mesh of intake system model which is automatically generated in COSMOSFloWorks. The mesh of the model is created by dividing the computational domain into slices, which are further subdivided into rectangular cells. The initial mesh is fully defined by the generated basic mesh and the refinement settings. Each refinement has a criterion and level for refinement which level 7 are defined for intake system model. The refinement criterion denotes which cells have to be split, and the refinement level denotes the smallest size to which the cells can be split. The higher the level of refinement, the coarser the mesh generated. **Figure 4.2** show that the different between basic mesh cell and refinement cell in manifold runner. The computational mesh is adjusted to solve problem's features to resolve them better.

Figure 4.1: Automatic computational mesh of intake system.

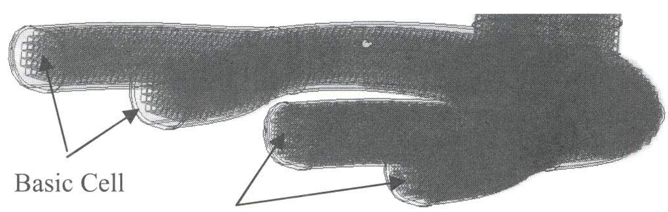

Figure 4.2: Comparison basic mesh sells size and refinement cell for intake manifold

Velocity of Flow varying to rpm

The velocity of flow at each point which has been defined in the intake system is obtained from the CFD and is compared with the theoretical result.

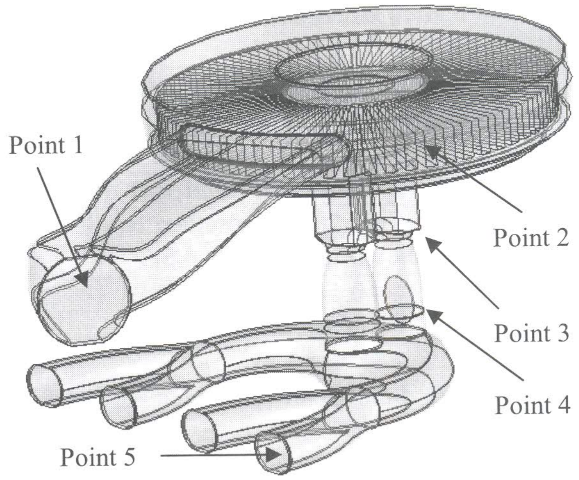

Figure 4.3: Point defined for velocity profile.

CFD Modeling

The CFD modeling of intake system is done by setting up the initial conditions of the model and defined the boundaries condition for inlet and outlet of volume region. The air pressure is set to total atmospheric pressure which is 101325 Pa at the opening air that the airs flowing into the air induct. Total pressure at the exit of air is set according to the manifold pressure that has been defined in the **Table 4.2**. Velocity of each point in the intake system as shown in **Figure 4.3** is defined and shown in **Table 4.3**.

Point 1: Inlet of air induction
Point 2: Outlet of air filter box
Point 3: Venturi in carburetor
Point 4: Outlet of carburetor
Point 5: Outlet of intake manifold

Table 4.2: Manifold pressure after pressure loss in intake system

rpm	Pressure drop Δp (mmHg)	Pressure drop Δp (Pa)	Manifold pressure (Pa)
2000	14.7	1959.8	99365.2
2500	18.7	2493.1	98831.9
3000	24.7	3293.1	98031.9
3500	32.7	4359.6	96965.4

Table 4.3: Velocity flow obtained by CFD.

rpm	Mean velocity (m/s)				
	Point 1	Point 2	Point 3	Point 4	Point 5
2000	10.0	7.0	60.0	30.0	40.0
2500	12.0	8.0	68.0	33.0	46.0
3000	13.5	10.0	78.0	40.0	52.0
3500	15.0	11.0	90.0	42.0	60.0

Theoretical Results

The velocities of flow are all scaled with the mean piston speed. The mean piston speed is varying to the engine rotational speed, rpm. The intake system is a control volume which refer to a fixed region in space which does not move or change shape. Volume flow rate in the cylinder of each rpm have been calculated and are all scale with the cross sectional area of each point that defined in the intake system. **Table 4.4** shows the volume flow rate varying to the rpm. The theoretical results of flow's velocity of each point are calculated by theoretical formula and are shown in the **table 4.5**.

Calculation

The calculation below is an example to find the mean velocity at point 1 for rpm 2000.

a. Cross sectional area for cylinder

$A = \Pi D^2/4$
$= \Pi \times 0.071^2 / 4$
$= 0.004\ m^2$

Where;

D = Bore of cylinder (m)

b. Mean piston speed (m/s)

$Sp = 2LN$
$= 2 \times 0.082 \times 2000/60$
$= 5.47$ m/s

c. Volume flow rate (m^3/s)

$Sp = V_1$
$Q = A_1 V_1$
$= 0.004 \times 5.47$
$= 0.02188\ m^3/s$

d. General equation of continuity for incompressible flow

$Q = A_1V_1 = A_2V_2$

Therefore;

$Q = A_2V_2$
$0.02188 = 2.41 \times 10^{-3} \times V_2$
$V_2 = 9.08$ m/s

Where;

A_1 = Cross sectional area of piston (m^2)
V_1 = Mean piston speed (m/s)
A_2 =Cross sectional area of each point (m^2)
V_2 = Mean velocity of each point (m/s)

Table 4.4: Volume flow rate in cylinder varying to rpm

rpm	Mean piston speed, V_1 (m/s)	Area, A_1 (m^2)	Volume flow rate, Q (m^3/s)
2000	5.47	0.004	0.02188
2500	6.83	0.004	0.02732
3000	8.20	0.004	0.03280
3500	9.57	0.004	0.03828

Table 4.5: Theoretical mean velocity

Mean velocity, V_2 (m/s)	Area, A_2 (m^2)	rpm 2000	2500	3000	3500
Point 1	2.41×10^{-3}	9.08	11.34	13.61	15.88
Point 2	3.40×10^{-3}	6.44	8.04	9.65	11.26
Point 3	4.16×10^{-4}	52.60	65.67	78.85	92.02
Point 4	9.62×10^{-4}	22.74	28.40	34.10	39.79
Point 5	7.05×10^{-4}	31.04	38.75	46.52	54.30

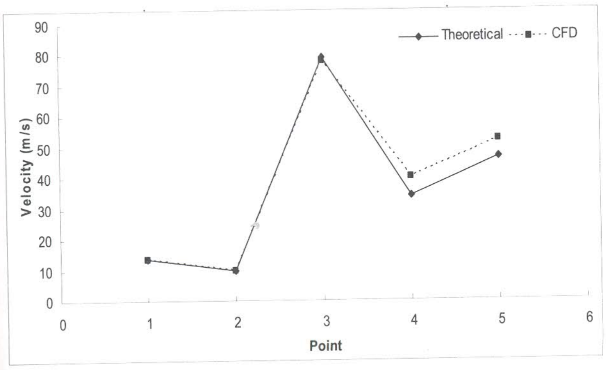

Figure 4.4: Comparison flow velocity between theoretical and CFD result at rpm 2000.

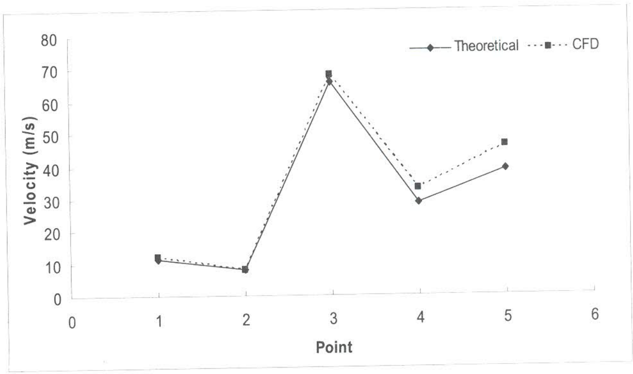

Figure 4.5: Comparison flow velocity between theoretical and CFD result at rpm 2500.

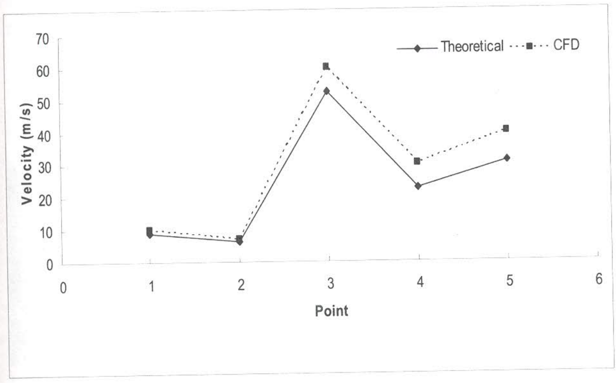

Figure 4.6: Comparison flow velocity between theoretical and CFD result at rpm 3000.

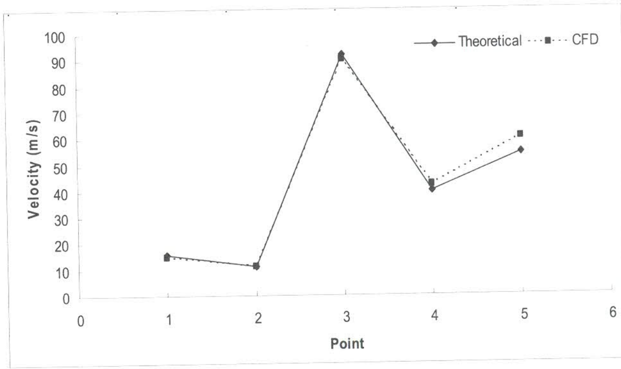

Figure 4.7: Comparison flow velocity between theoretical and CFD result at rpm 3500.

Comparison between Theoretical and CFD Result

The flow's velocity depends on the cross sectional area. The pressure becomes lower when it passes through small cross sectional area, thus the flow velocities is higher. The smallest area for air flow is at the venturi of carburetor where the fuel is sprayed due to the difference pressure. Since the difference pressure is higher at point 3, the metering system will spray more fuel. Thus, higher velocity is required to provide good atomization and vaporization of air-fuel mixture.

Theoretical air velocity is calculated to validate the result that obtained from CFD results. Overall the air velocity not much difference at point 1, 2 and 3 except point 4 and 5. Air velocity at point 4 and 5 are slightly higher than theoretical result because the frictional losses of the wall are not included in CFD analysis. The roughness of the wall is defined as zero because it is so difficult to specify the wall roughness.

Besides that, the pressure drop is affected by the geometric of the twin venturi carburetor. The pressure drops when the air flows through the separation point of the twin venturi where the air chooses to flow into primary or secondary venturi. Seem the higher the pressure drop will tend to high velocity, thus the CFD results are slightly higher than theoretical results at point 4. Pulsating flow at the manifold also causes the higher pressure drop at point 5; indirectly the velocity is higher at point 5.

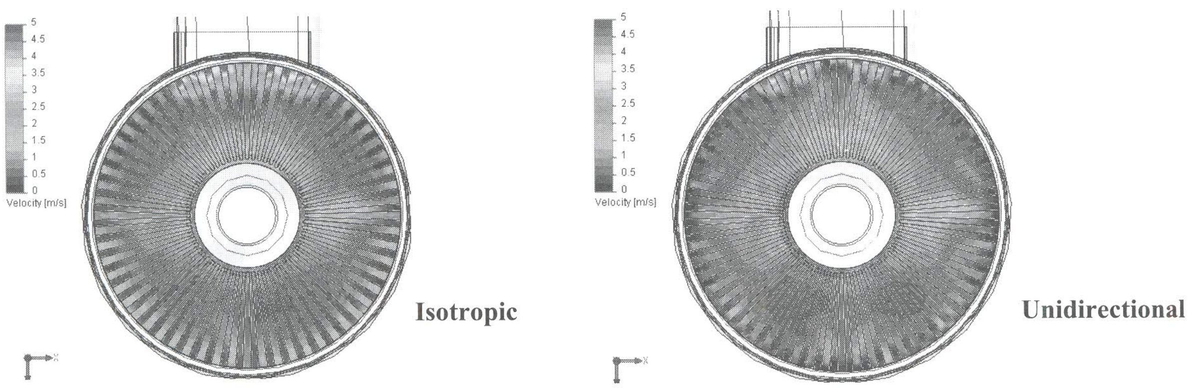

Figure 4.8: Porous surface plot for isotropic and unidirectional permeability type with porosity 0.5.

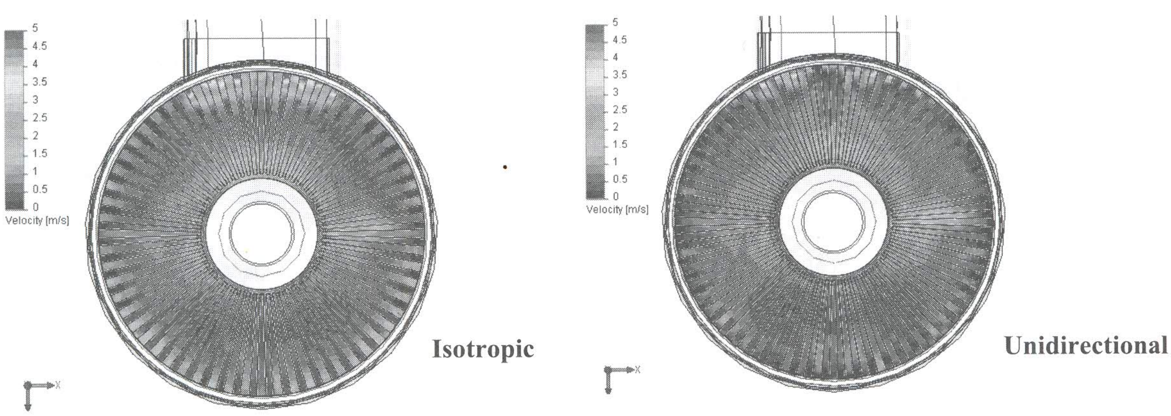

Figure 4.9: Porous surface plot for isotropic and unidirectional permeability type with porosity 0.6.

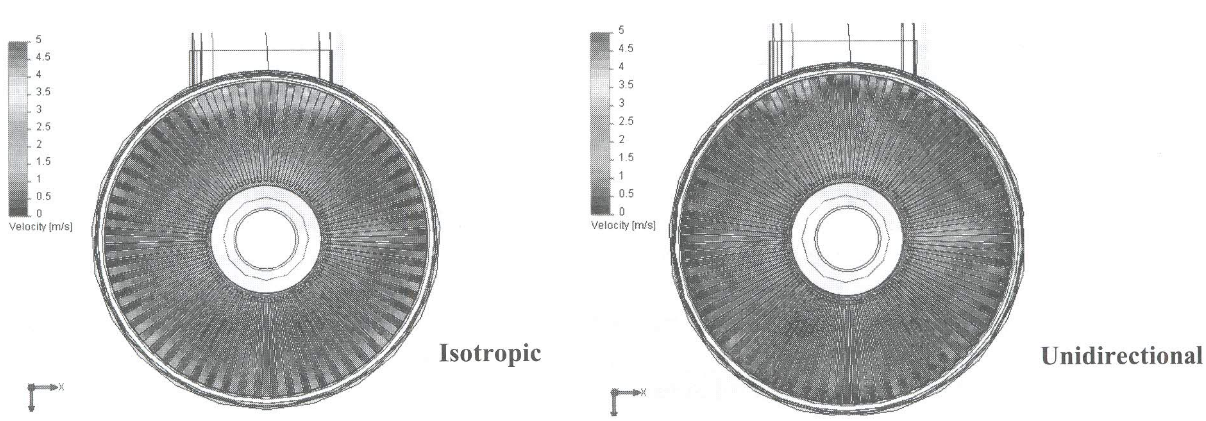

Figure 4.10: Porous surface plot for isotropic and unidirectional permeability type with porosity 0.7.

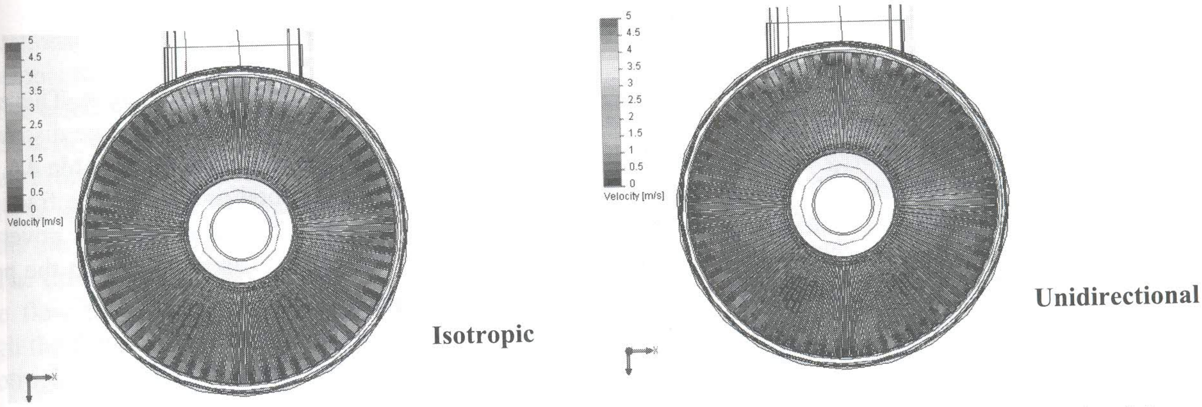

Figure 4.11: Porous surface plot for isotropic and unidirectional permeability type with porosity 0.8.

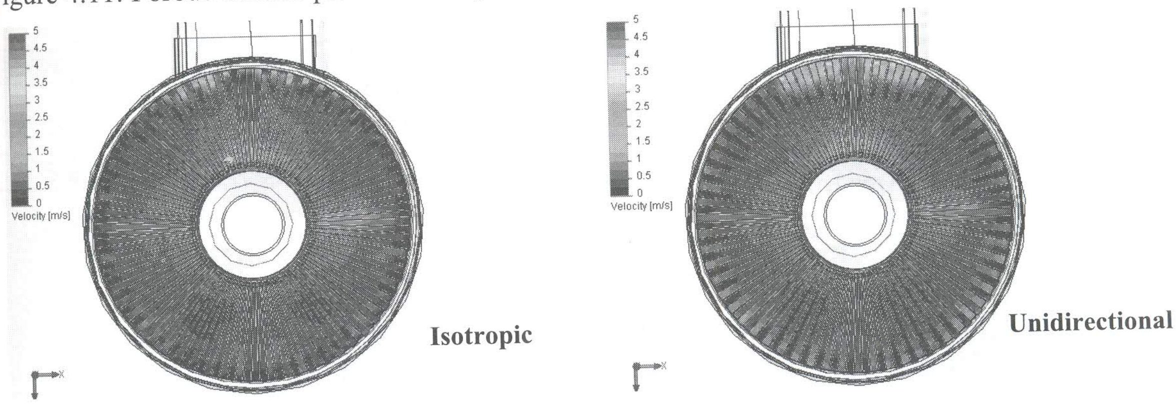

Figure 4.12: Porous surface plot for isotropic and unidirectional permeability type with porosity 0.9.

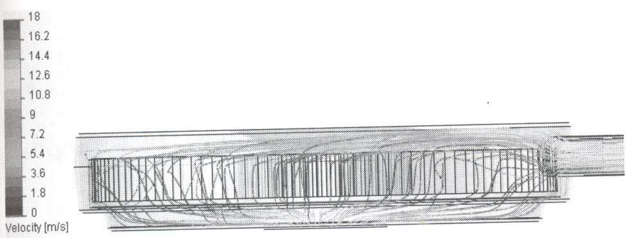

Figure 4.13: Side view of flow trajectories in intake system through air filter of isotropic permeability type.

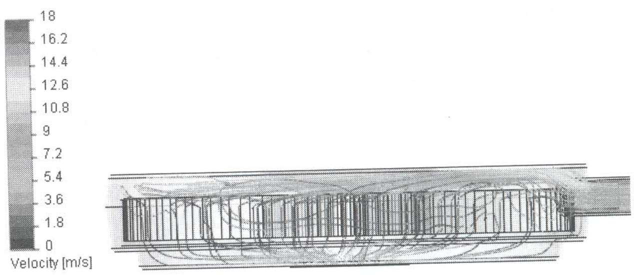

Figure 4.14: Side view of flow trajectories in intake system through air filter of unidirectional permeability type.

Porous Media

In this project, the flow in a section of an intake system is considered when intake flow is resisted by a porous bodies serving as air filter to remove dust particles from entering into engine. When designing an air filter, the engineer faces a compromise between minimizing the air filter resistance to the intake flow while maximizing the air filter internal surface area and duration that intake gases are in contact with that surface area. Therefore, a more uniform distribution of the intake mass flow rate over the air filter cross section favors its serviceability. The porous media capabilities of COSMOS FloWorks are used to simulate the air filter, which porous boundary conditions is defined to model the volume that the air filter occupies as a distributed resistance instead of discretely modeling all of individual passages within the air filter.

Porous Boundary Conditions

In the simulation, we considered the influence of the air filter porous medium permeability type. This means that isotropic and unidirectional media of the same resistance flow on the intake mass flow rate distribution over the air filer cross sections. The porous medium property of filter is defined as shown in **Table 4.6**.

Porosity is the effective of the air flow through the porous medium, defined as the volume fraction of the interconnected pores with respect to the total porous medium volume. The porosity will govern the intake flow velocity in the porous medium channels which governs the intake gas residence in the porous filter. Difference porosity will be modeling with CFD from low to high.

Table 4.6: User defined of air filter's properties for porous boundary condition.

Properties of air filter	User Define
Porosity	0.5 - 0.9
Permeability Type	Isotropic / unidirectional
Resistance calculation formula	Pressure drop, flow rate, dimension
Pressure drop vs flow rate	Mass flow rate
Length (m)	0.025
Area (m^2)	0.040
Pressure drop (Pa)	100
Mass flow Rate (kg/s)	0.01

Porous Medium Resistance

A pressure drop, flow rate and dimensions medium resistance to flow are selected to specify the porous medium resistance for air filter's media. The porous medium resistance is calculated using formula below according to the air filter properties that are listed in the **Table 4.6.**

$k = \Delta P \times S / (m \times L)$
$= 100 \text{ x } 0.040 / (0.01 \text{ x } 0.025)$
$= 16000 \text{ s}^{-1}$

Where;

ΔP = Pressure drop (Pa)
S = Cross sectional area (m^2)
m = mass flow rate (kg/s)
L = Length (m)

Comparing the flow through the Isotropic and Unidirectional Porous Filter

The analysis of the air filter is done by comparing the isotropic and unidirectional permeability filter with difference porosity of air filter from 0.5 to 0.9. The air filter is considered an isotropic permeability which the medium's permeability not depending on the direction within the medium. Otherwise the unidirectional permeability is which the medium being permeable in one direction only. The direction of the flow through the porous media is in Y axis for unidirectional case.

The porous filter is installed in the filter housing which the air flow from the inlet of air induction, the incoming flow is non-uniform because of the curvature and difference cross sectional area of induction system. Since the incoming flow is non-uniform, the flow inside the porous filter is non-uniform. **Figure 4.13**, **4.14** is compared and the flow for isotropic type is more uniform than unidirectional type.

It seems that the isotropic and unidirectional type of filter affects both the incoming flow. For isotropic case, the gas flows into the filter closer to the wall of the filter housing than unidirectional case. Nevertheless, due to the isotropic permeability, the main gas stream expands in the isotropic filter and occupies a larger volume in the next part of the body than in the unidirectional filter, which, due to its unidirectional permeability, prevents the stream from expanding. The expanding area of gas stream is larger for isotropic than the unidirectional type.

The effectiveness of the air flow through the air filter depends on the porosity of the air filter. **Figure 4.8, 4.9, 4.10, 4.11, 4.12** shows the comparison of flow behavior for different porosity and permeability type of filter. From the CFD results, the air flow velocity through the air filter is getting lower when the value of porosity is increased for both isotropic and unidirectional permeability type of air filter. The high porosity prevents the gas stream from expanding, thus the area of air flow through the filter become lower.

The flow velocity inside the air filter is analyzed by viewing the flow velocity value which is indicated by the flow trajectories colors accordance with the specified palette. The average velocity of air flow through the filter is in the range from 0 m/s to 1.5 m/s for both cases. On the whole, the flow velocities in the isotropic and unidirectional type of filter are practically the same.

Table 4.7: Volume and mass flow rate for variable conditions in intake system.

Conditions	Volume flow rate (m^3/s)	Mass flow rate (kg/s)
Idle speed	0.000828	0.000295
Wide open throttle	0.0322	0.0372
Sudden acceleration	0.0514	0.0476

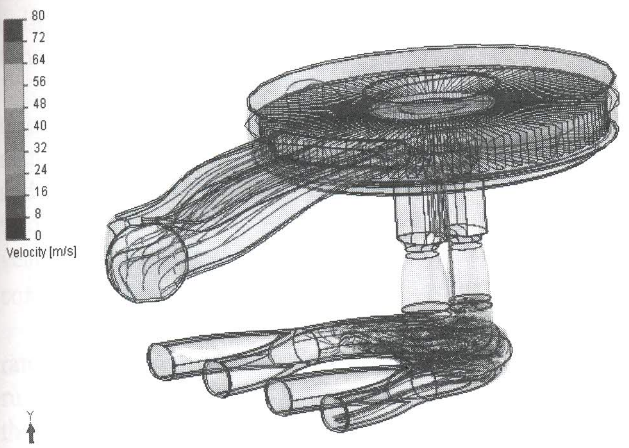

Figure 4.15: Flow trajectories of air during idle speed condition.

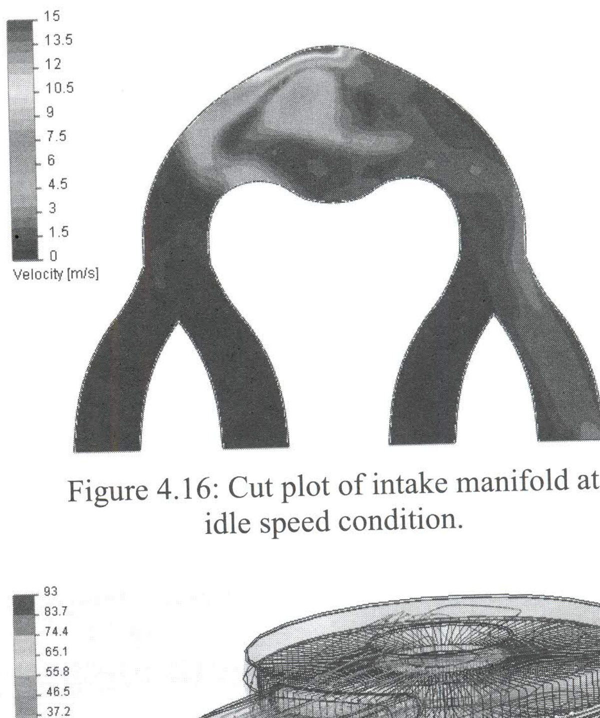

Figure 4.16: Cut plot of intake manifold at idle speed condition.

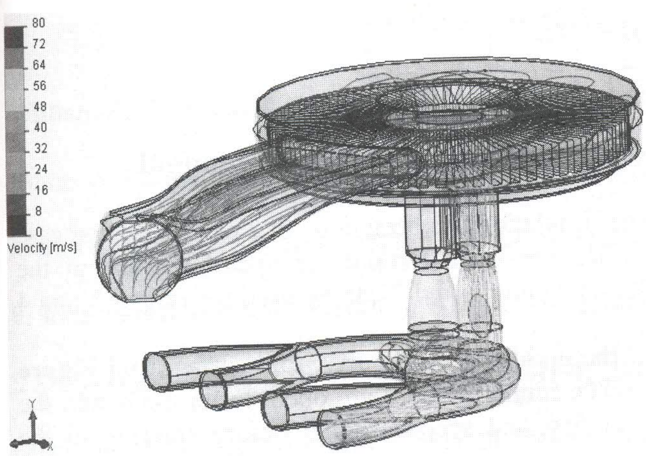

Figure 4.17: Flow trajectories of air during wide open throttle condition.

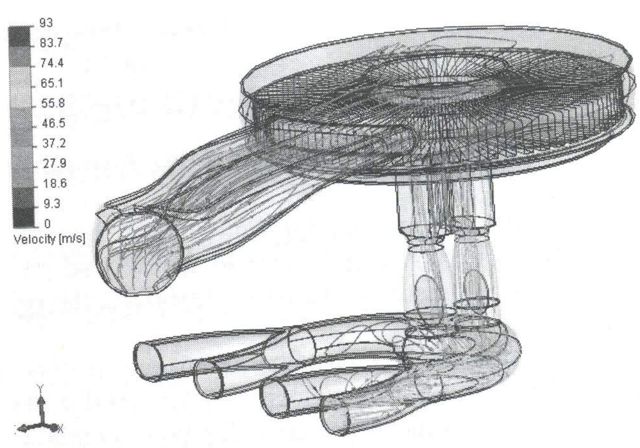

Figure 4.18: Flow trajectories of air during acceleration condition.

Variable Operating Conditions of Intake

During idle speed condition, very little air passes through the intake manifold because both of the throttle valves are close as shown in **Figure 4.15**. The volume and mass flow rate at the outlet of manifold is very low compared to wide open throttle and sudden acceleration conditions as shown in **Table 4.7**. The pressure in the manifold is about 50000 Pa, this cause a higher suction pressure to drawn the air through the idle channel, thus the velocity in the channel is very high.

The outlet air from the idle channel creates a pulsating flow in the manifold as shown in **Figure 4.15**. The flow in manifold is much lower than the idle channel because of sudden flow to the high cross sectional area in intake manifold. According to the **Figure 4.16**, the flow is more concentrated at the other side of the manifold and then flows into the opening port with low velocity. This is because the pulsating flow create pressure drop tends to suck the air.

The volume and mass flow rate of wide open throttle are lower than sudden acceleration condition. During sudden acceleration condition, both of the throttle valves are open to increase the volumetric of air-fuel mixture into the cylinder. Thus, the air flow through both venturi in the carburetor and is needed to supply additional fuel for quick acceleration.

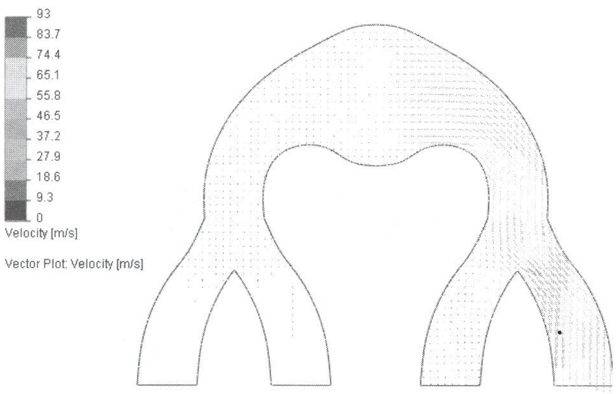

Figure 4.19: Vector plot at the plane of intake manifold which the air flow through first runner.

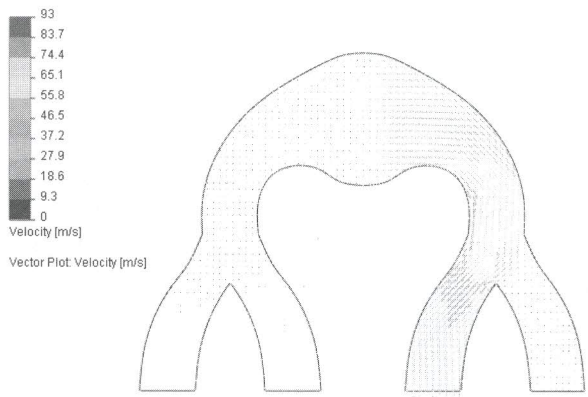

Figure 4.20: Vector plot at the plane of intake manifold which the air flow througn second runner.

Table 4.8: Velocity, mass and volume flow rate enter to the cylinder in one cycle.

Runners	1	2
Mass flow rate (kg/s)	0.0476	0.0415
Volume flow rate (m^3/s)	0.0514	0.0449
Average velocity (m/s)	63.74	59.43

Comparison between First Runner and Second Runner of Intake Manifold

Steady state conditions are applied to this simulation. The simulation has been done in 2 cases: the first case when the first runner is open and the second case when the second runner is open. Because of the symmetrical geometry of the intake manifold, the results for runner 1 and 2 can be used for runner 3 and 4 as well.

To analyze flow field better, a cut plot is done on the middle of intake manifold surface and **Figure 4.19**, **4.20** show the flow velocity vectors on this plate when runner 1 and 2 are open. From the result, the flow pattern is almost the same for both simulations in first and second runner before entering to the manifold's runner. When the runner 1 and 2 are open, there are some pulsating flow occurrence before reaching the runner. The pulsating flow will cause pressure to drop.

The pressure slightly dropped at runner 2 compared to runner 1 because of the curvature and sharp point before entering to the runner. The pressure drop tends to increase the flow velocity, thus the flow velocity is higher at the curvature surface. From the **Table 4.8**, the average velocity is lower for second runner due to the difficulties or resistance of air flow through the curvature and sharp edge.

The distribution of air to the cylinder is not uniform because the volume and mass flow rate in first runner is higher than the second runner. This condition will cause insufficient combustion in the chamber. High fuel consumption and emission may occur during this condition.

CONCLUSION

The flow's velocity varying to the rpm is obtained from CFD and is validated with the theoretical result. The flow velocities are obtained by defining the pressure losses or frictional losses in intake system using steady state simulation which when the pressure is indicated in the intake system represents the time averaged values for engine speed. Air velocity is increased as it passes through intake duct to the long and narrow intake manifold runners. From the result, the data obtained from CFD and theoretical are not much different so the simulation result is reliable.

The air filter in intake system is served as a porous media which the air flow is restricted when flow through it. From the result, the isotropic filter is more effective than the unidirectional filter due to the more uniform flow in the filter. Besides that, the expansion gas stream for isotropic filter minimized the resistance flow of the intake.

Variable operating conditions for intake system has been carried out to study the flow behavior through each component of the intake system. The volume and mass flow rate obtained from the outlet of manifold's runner stated that the sudden acceleration condition is highest followed by wide open throttle and idle speed condition. The volume and mass flow rate of idle condition is much lower than wide open throttle and sudden acceleration condition because both throttle valves is closed during intake stroke and a very little air is passed through the small cross sectional area of idle channel. During sudden acceleration condition, there is air flow through the secondary throttle valve to demands additional fuel.

The curvature and sharp point cause pressure drop in second runner, thus the volume and mass flow rate is lower than the first runner. The curvature and sharp point increase the difficulty of flow through the runner. The mass flow rate in runner 1 and 2 are different about 13 % to each other and this can demonstrate the insufficient design of the intake manifold to distribute the uniform air to the cylinders.

In conclusion, this project has presented a powerful method to study the flow behavior in an intake system. The procedures and the results of steady state simulations are explained.

REFERENCES

William H. Crouse and Donald L. Anglin, *Automotive Mechanics* Tenth Edition, McGraw- Hill, 1993.

Yunus A. Cengel and Michael A. Boles, *Thermodynamics* Fourth Edition, McGraw-Hill, 2002.

Mark Coombs and Spencer Drayton, *Proton Service and Repair Manual*, Haynes Publishing, 2003.

E. John Finnemore and Joseph B. Franzini, *Fluid Mechanics* Tenth Edition, McGraw-Hill, 2002.

S Srinivasan, *Automotive Engines*, McGraw-Hill, 2001.

Diego A. Arias and Timothy A. Shedd, "Numerical and Experimental Study of Fuel and Air Flow in Carburetors for Small Engines", SAE 2004-32-0053 / 20044340

John B. Heywood, *Inernal Combustion Engine Fundamentals* International Edition, McGraw-Hill, 1988.

M. Safari and M. Ghamari, "Intake Manifold Optimization by Using 3-D CFD Analysis", SAE 2003-32-0073.

Rakesh D., "Studies on the Airflow Behaviors in a Intake Manifold of a Multi Cylinder SI Engine", thesis sample, 2007

Barry Hollembeak, *Automotive Fuels and Emissions*,Thomson, 2005.

M. Al-Hasan, "Evaluation of Fuel Consumption and Exhaust Emissions during Engine Warm-up", American Journal of Applied Sciences 4 (3): 106-111, 2007.

Warren M.Farnell, *Fuel System and Emission Control* Fifth Edition, Pearson Prentice Hall, 2005.

C. Brehm and J. H. Whitelaw, "Air and Fuel Characteristics in the Intake Port of a SI Engine", SAE 1999-01-1491.

FLUENT, www.fluentusers.com, "*Introductory FLUENT Notes*", Feb 2003

Katlego Webster Makgata, "Computational Analysis and optimization of the Inlet System of a High-Performance Rally Engine", thesis 2005.

J.H.Ferziger / Milovan Peric, Computational Methods for Fluid Dynamics Third Edition, Springer, 2002.

De Winterbone and RJ Pearson, *Theory of Engine Manifold Design*, Professional Engineering Publishing, 2000.

Wolf Bauer, Pairote Balun, and John B. Heywood, "Heat Transfer and Mixture Vaporization in Intake Port of Spark-Ignition Engine", SAE 972983.

Shiyuji Hatakeyama, Mikiro Kondo, Yoshio Sekiya and Tadashi Murayama, "An Attempt at Lean Burn of a 4 Stroke Gasoline Engine by the Aid of Low Pressure Air Assisted In-Cylinder Injection", SAE 982698.

Georg Maier and Sigmar Wittig, "Influence of Air Assisted Fuel Injection on the Mixing Preparation within the Intake Ports of SI-Engines", SAE 982523.

Edwin Itano, Anthony J. Shakal, and Jay K. Martin, "Carburetor Exit Flow Characteristics", SAE 961730.

Kumpulan Fokus Automatif
Fakulti Kejuruteraan Mekanikal
Universiti Malaysia Pahang
Karung Berkunci 12, 25000 Kuantan
Pahang, Malaysia
Tel: 012-983 2631
E-mail: rosli@ump.edu.my