AMIRUL AMIR ABD JAMIL NOV 2009

PERFORMANCE SIMULATION OF MODENAS 110cc 4-STROKE MOTORCYCLYE ENGINE

AMIRUL AMIR ABD JAMIL

Report submitted in fulfilment of the requirements
for the award of the degree of
Bachelor of Mechanical Engineering with Automotive

Faculty of Mechanical Engineering
UNIVERSITI MALAYSIA PAHANG

NOVEMBER 2009

SUPERVISOR'S DECLARATION

We hereby declare that we have checked this project report and in our opinion this project is satisfactory in terms of scope and quality for the award of the degree of Bachelor of Mechanical Engineering with Automotive.

Signature:
Name of Supervisor: NIK MOHD IZUAL BIN NIK IBRAHIM
Position: LECTURER
Date: 23 NOVEMBER 2009

Signature:
Name of Co-Supervisor: ISMAIL BIN ALI
Position: INSTRUCTOR ENGINEER
Date: 23 NOVEMBER 2009

STUDENT'S DECLARATION

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

Signature:

Name: AMIRUL AMIR ABD JAMIL

ID Number: MH06027

Date: 23 NOVEMBER 2009

DeDicateD to my beloveD parents, family, anD frienDs..

Thank you for all your supporT, ideas, and cooperaTion..

All of you AlwAys in my heArt forever..

ACKNOWLEDGEMENTS

First of all, I would like to express my grateful to ALLAH s.w.t. for the blessing given to me to complete my project.

In order to complete my work, I have engaged with many people in helping me especially my supervisor Mr Nik Mohd Izual B. Nik Ibrahim and Mr. Ismail B. Ali. I wish to express my sincere appreciation to them for their encouragement, guidance, advices and motivation. Without their continued support and interest, this thesis would not have been the same as presented here.

Not forgotten, I would like to express my gratitude to all university staff from Mechanical Engineering Faculty for being so cooperative and not forgetting all my friends for the support, advice and information sharing.

Last but not least, I acknowledge my sincere indebtedness and gratitude to my parents for their love, dream and sacrifice throughout my life. Not forgotten Farah Diana Ahmad Kamaluddin. I cannot find the appropriate words that could properly describe my appreciation for their devotion, support and faith in my ability to attain my goals. Without endless love and relentless support from my family, I would not have been here.

AMIRUL AMIR ABD JAMIL NOV 2009

ABSTRACT

This thesis present about the effect of compression ratio to the Modenas 110cc four stroke motorcycle engine. The objective of this project is to increase the performance of the engine. Using GT Power engine simulation software, the original parameter will be check and change to get the highest performance. The changes of the performance will be compared before it is choose. The percentages change will be calculated and compared with the original parameter. As the conclusion, the compression ratio will affect the engine performance and fuel consumption of the engine.

AMIRUL AMIR ABD JAMIL NOV 2009

ABSTRAK

Tesis ini membentangkan tentang perkara yang berkadar langsung apabila nisbah tekanan injin empat lejang Modenas Kriss 110cc diubah. Objektif projek ini adalah untuk meningkatkan prestasi enjin agar menjadi lebih tinggi. Dengan menggunakan perisian GT Power, ukuran asal injin akan dicatat dan ukuran itu akan diubah untuk mendapatkan ukuran yang menunjukkan prestasi paling tinggi. Perubahan prestasi akan dibanding terlebih dahulu sebelum di pilih. Peratus perubahan akan dikira dan dibandingkan dengan ukuran asal. Kesimpulannya, nisbah tekanan enjin akan mempengaruhi prestasi enjin dan penggunaan minyak di dalam engine.

TABLE OF CONTENTS

LIST OF TABLE

LIST OF FIGURE

LIST OF SYMBOLS

rpm	Revolution per minute
mm	Millimeter
kW	Kilowatt
N-m	Newton meter
g/kW-h	Gram over kilowatt hour
%	Percent

LIST OF ABBREVIATIONS

IV	Intake valve
EV	Exhaust valve
CR	Connecting rod
SP	Spark plug
PR	Piston ring
P	Piston
CS	Crank shaft
TDC	Top dead centre
BDC	Bottom dead centre
AFR	Air fuel ratio
IMEP	Indicated mean effective pressure
EVO	Exhaust valve open
SI	Spark ignition
BSFC	Brake specific fuel consumption
BMEP	Brake mean effective pressure
ISFC	Indicated specific fuel consumption

LIST OF FORMULA

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

Four stroke engines was created by Eugenio Barsanti and Felice Matteuci in 1854 and followed by first prototype in 1860 [3]. A French engineer, Alphonse Beau de Rochas was also made the conceptualized in 1862 [3]. But the first who develop a functioning four-stroke engine was Nicolaus Otto,the German engineer. That is why the four stroke principle known as Otto cycle and for four stroke engine using spark plugs called Otto engine.

The four stroke cycle is more fuel-efficient and clean burning than the two stroke cycle but it requires considerably more moving parts and manufacturing expertise and the resulting engine is larger and heavier than two stroke engine of comparable power output. The Otto cycle is characterized by four stroke or straight movement in a single direction which obtained intake (induction) stroke, compression stroke, power stroke and exhaust stroke. All the cycle must be repeated over and over for the engine to continue operation.

1.2 OBJECTIVE

- To investigate the performance of Modenas 110cc 4-stroke motorcycle engine using GT Power engine simulation software.
- To determine the best parameter for the engine part to achieve the highest performance.

1.3 SCOPE

The main operation of operating the engines are covered in this work is to improving the performance of Modenas 110cc 4-stroke motorcycle engine. An overview of the different methods, their effect on performance, emissions, and applicability of these methods to the current objective is presented. It consists of the experimental apparatus used for engine testing, and the development of a new setup consisting of a motorcycle engine before we used the simulation to compare the results. The results of two different ways on performance are also discussed. A model of the engine system is created using GT-Power software. The model is calibrated using experimental measurements, and is used to predict gain performance for change in different parameters. A model of this system is also created and compared with experimental predictions.

GT Power Simulation & Modeling

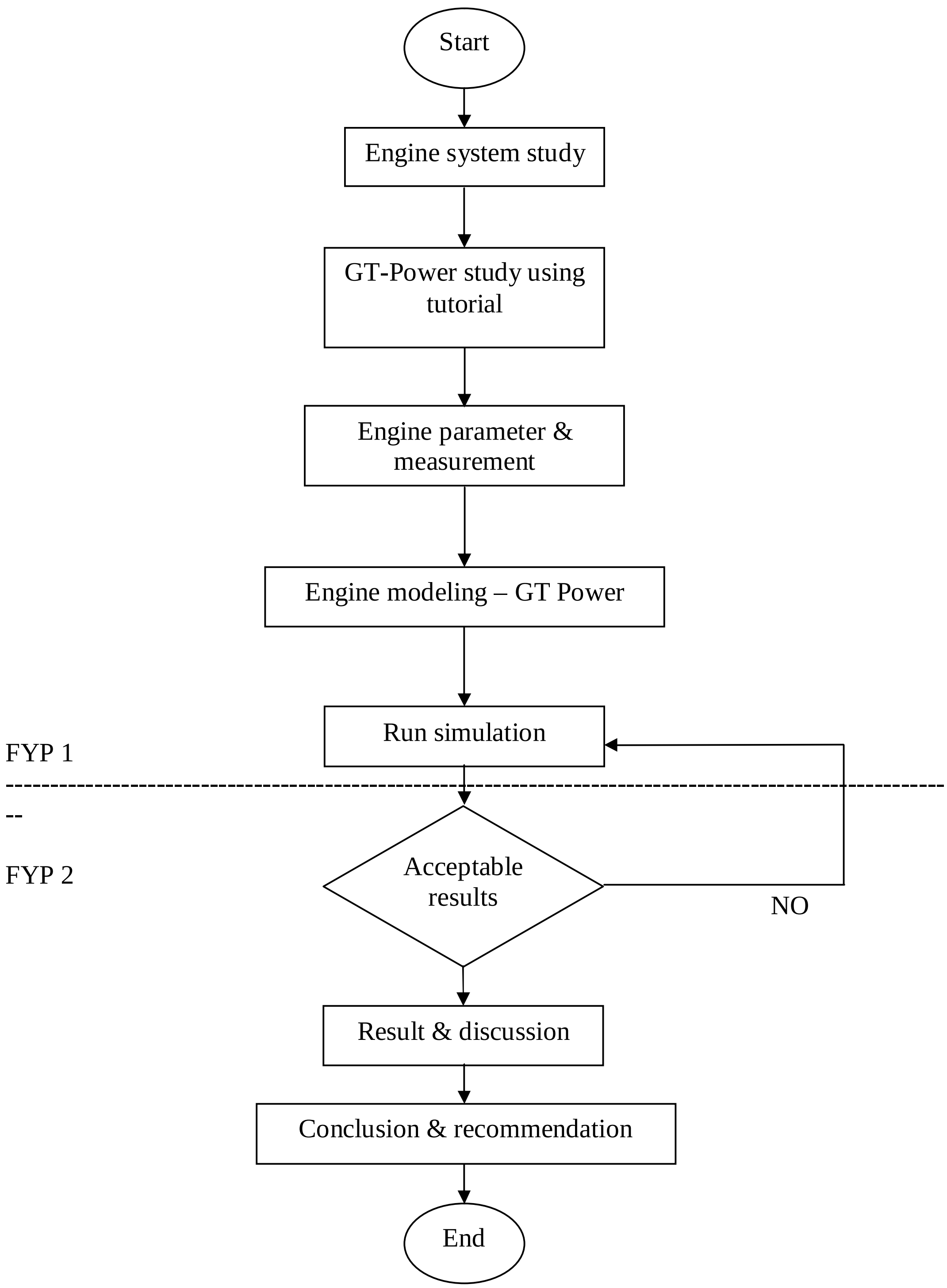


Figure 1.1: GT Power Simulation & Modeling Flow Chart

CHAPTER 2

LITERATURE REVIEW

2.1 FOUR STROKE ENGINE

The simple four stroke engine is shown below with several phases of the filling and emptying of the cylinder. The diagram shows the engine as a spark-ignited unit with access to the cylinder controlled by poppet valves actuated by cams, tappets and valve springs. The cylinder contains a piston with sealing affected by two piston rings and third lower ring which scrapes the excess lubrication oil off the cylinder walls back into the crankcase, which acts as an oil sump. The movement of the piston is controlled by a rotating crankshaft with the connecting rod linking the small-end bearing of the piston, through a wrist pin, to the big-end bearing on the crankpin. A flywheel is attached to the crankshaft to help smooth out the torque pulsations [1].

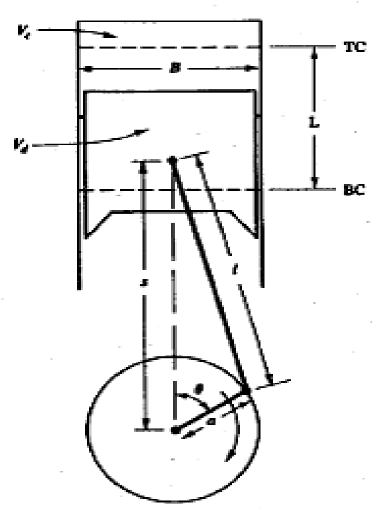


Figure 2.1: the parts of simple engine [8]

2.2 BASIC OPERATION OF FOUR STROKE ENGINE

The four-stroke or four–cycle engine is quite simply, regulated by a throttle plate. The throttle plate also called a butterfly valve controls the amount the amount of air entering the engine and controls the power output of the engine. When the throttle plate restricts intake air, the engine will operate at a lower output. During low or partial load operation, the engine has a low pressure.

When the throttle plate is held wide open, the engine will operate at its maximum output. During full throttle, or high load operation, pressure in the intake manifold is at or near atmospheric pressure. Therefore, a simple vacuum gauge can be used to accurately measure engine load [5].

The valves located in the cylinder head, control airflow into and out of the cylinders. These valves are opened by an eccentric camshaft that rotates at one half crankshaft speed. An ignition system provides the spark necessary for combustion, and a fuel management system provides the fuel/air mixture, smooth and controlled burn.

2.2.1 Four stroke cycle

Intake stroke

The Intake stroke begins with the piston at top dead center or TDC and moving downward toward bottom dead center or BDC. As the piston travels down the cylinder, it creates a low pressure or vacuum. The camshaft opens the intake valve at the beginning of the stroke and the air/fuel mixture is drawn into the cylinder past the intake valve. Air will always flow from an area of high pressure toward an area of lower pressure. Because the pressure is lower in the cylinder than in the intake manifold, it actually pushes the air-fuel mixture into the cylinder [4] [5].

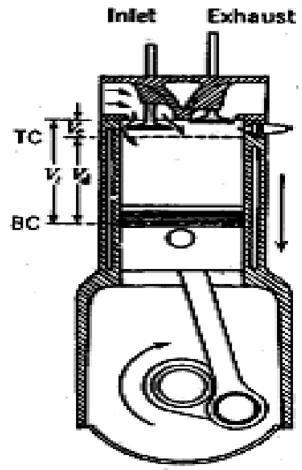


Figure 2.2: Intake stroke [8]

Fuel, is provided by the fuel management system (a carburetor or fuel injector), as a finely atomized mist. The combination of low pressure and engine heat in the intake manifold, help convert the small droplets of liquid fuel, into the fuel vapor necessary for efficient combustion [3].

It is very important that fuel be delivered in the correct ratio to the air entering the cylinder [3]. The correct air/fuel ratio for most conditions is about 14 parts air for every one part of fuel, measured in pounds. Expressed numerically, as: 14:1 AFR, the scientific name for this ratio is stoichiometric. The oxygen contained in the air is the element necessary for combustion. Since air contains about 21% oxygen, the ratio is 3.09 pounds of oxygen for every pound of fuel. Any more fuel would cause the mixture to be rich. Any less fuel and the mixture will go lean. Either one of these can cause misfire, fuel wasting, polluting, power robbing condition, that causes rough idle and can do serious, expensive damage to many different parts. Maintaining the proper mixture is critical to engine efficiency and is one of the leading causes of poor fuel economy and excessive emissions. Modern engines use a variety of computer managed feedback systems, which do an outstanding job of keeping this delicate fuel trim balancing act in check.

Compression stroke

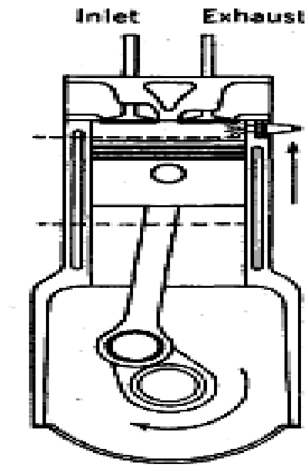


Figure 2.3: Compression stroke [8]

As the piston reaches BDC, the valve spring has already closed the intake valve, sealing the cylinder and the piston begins to compress the air/fuel mixture. Compressing the mixture causes it to increase in temperature. Rapid temperature increase completes the transformation of liquid fuel and air, to air/fuel vapor. This change is extremely important as the spark plug can only ignite fuel vapor efficiently. Liquid fuel, fuel droplets or fuel mist, require an open flame to rapidly burn and will not properly ignite with a spark type ignition. Compression also causes the air/fuel mixture density to greatly increase [4] [5] [3] [10]. This is important, because a complete, rapid, controlled burn can only happen, if the distance between the molecules of air and fuel, are close enough for the flame front to rapidly bridge the air gaps between it.

Notice the use of the word burn as opposed to explode. If the mixture fails to burn smoothly but instead explodes all at once, the mixture will suddenly explode or “ping”. The proper term is detonation or pre-ignition. When this happens in a gasoline-fueled engine, power is lost and worse, NOX (Oxides of Nitrogen) emissions will soar off the chart [3]. If used properly, in a controlled manner, this ping can efficiently produce power. In fact, this principle is what makes a diesel engine run. Diesel engines have no spark plugs. Instead, it use extremely high compression and lower octane fuels to produce detonation of torque. This type of propulsion is useful for low cost, low RPM and high power applications. Compression loss will happen if there is a breach anywhere in the cylinder or the combustion chamber. A burned valve, blown head gasket and or worn/broken piston rings are a few common causes of compression loss. Any one, or a combination of these, will cause; low power, misfire, poor idle, poor fuel economy and elevated emission levels. If not repaired, severe engine damage and a melted catalytic converter will almost certainly result.

Power stroke

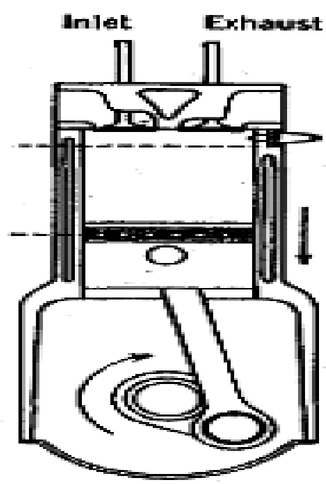


Figure 2.4: Power stroke [8]

When the piston gets back up to the top of the cylinder, near TDC, the mixture has been squished down to about 1 eighth of its original volume (8:1 compression ratio). The ignition system, then supplies a spark, which ignites the compressed mixture [4] [5] [3]. The mixture begins to burn at the area of the spark plug and a flame front travels across the combustion chamber. As the mixture burns and expands, pressure in the combustion chamber increases steadily. The timing of the spark is critical. The pressure need to happen at just the right moment, thrusting the piston down the cylinder with maximum force.

If ignited too soon, combustion will happen before the piston reaches top dead center (pre-ignition). This causes combustion pressure to push on the piston while the crankshaft is trying to push the piston up on the compression stroke. When this happens, the mixture will explode or detonate robbing power and causing engine damage. Power is lost, trying to move the piston back against the engines momentum.

Pre-ignition occurs when the mixture explodes rather than burning in a controlled manner. This explosion happens when the heat of compression and the heat of the advancing flame front combine to ignite the unburned mixture all at once. The pressure increase that happens with detonation is very rapid, very hot and very high, placing extra load on the piston, connecting rod, crank and engine block. The knocking or pinging noise heard, is the explosion, taking place in the cylinder. Detonation is usually caused by low octane fuel or by the spark plug firing prematurely. On the other hand, if the spark happens too late, the piston is already traveling down the cylinder; the pressure increase is not as great [5] [3]. Less cylinder pressure means less power produced, lower mileage and poor performance. With the correct air/fuel mixture, ignited at the optimum time, a smooth pressure increase will drive the piston down in the cylinder in an even powerful motion. The piston pushes against the connecting rod, the rod pushes against the crankshaft and the engine will produce efficient, clean power.

Exhaust stroke

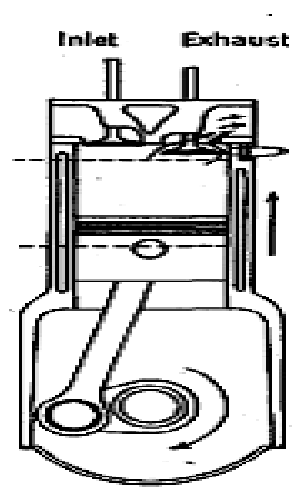


Figure 2.5: Exhaust stroke [8]

When the piston reaches bottom dead center on the power stroke and starts back up the cylinder, the exhaust stroke begins. Near bottom dead center the camshaft opens the exhaust valve and as the piston moves back up the cylinder bore, it pushes the burned exhaust gasses, out through the open valve. When the piston, once again, returns to top dead center, the exhaust valve closes, the burned exhaust, has been purged, and the engine is once again ready to begin the four stroke sequence [4][5].

2.2.2 Compression Ratio

The compression ratio is a value that represents the ratio of the volume of the combustion chamber, either in the internal-combustion chamber or external-combustion chamber, from its largest capacity to smallest capacity [10]. During the compression stroke of the four-stroke cycle: both the intake and exhaust valves are closed so no air can escape, and the piston moves upward from bottom dead center (BDC) to top dead center (TDC) so that the air/fuel mixture in the cylinder is compressed into the combustion chamber.

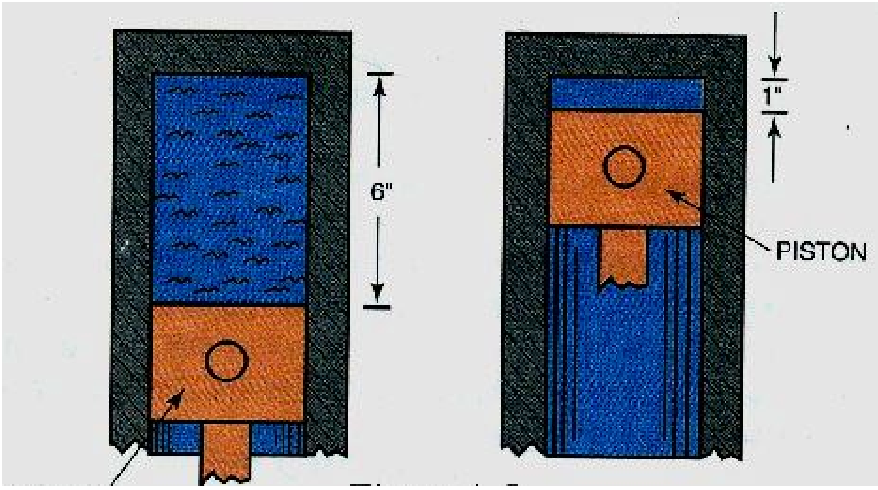


Figure 2.6: Compression ratio

Compression ratio is the relationship of cylinder volume (or displacement) with the piston at BDC to cylinder volume with the piston at TDC. If the volume of the cylinder with the piston at BDC is 10 times greater than the volume of the combustion area with the piston at TDC, then 10 units of volume get squeezed into 1 unit of space, and the compression ratio is 10.0:1. There are five factors that affect compression ratio which are;

- cylinder swept volume
- clearance volume, piston dome or dish
- head-gasket volume
- chamber volume

Cylinder Swept Volume

The swept volume of the cylinder indicates how much air the piston displaces as it moves from BDC to TDC. Increasing the cylinder volume without making any other changes will increase the compression ratio because it enlarges the cylinder volume without increasing the combustion chamber volume. In other words, the piston will have to cram more air into the same amount of space.

Cylinder volume is calculated using the bore and stroke of the engine with this formula [10]:

$$V_{sv} = A_{bo} \times L_{st}$$

$$V_{sv} = \frac{\pi {d^2}_{bo} L_{st}}{4} \tag{2.1}$$

Clearance Volume

Clearance volume is determined by the distance from the cylinder block deck to the top of the piston flat (not counting any dishes or domes) when the piston is at TDC. In many engines, the pistons don’t come all the way up to the height of the deck. They can be anywhere from 0.003 to 0.020 inch below it.

This amount is known as the piston deck height, and it affects compression ratio because it affects the volume of air in the combustion area when the piston is at TDC [11] [15]. If the piston is farther below the deck, then clearance volume is increased and the compression ratio is reduced. If the piston is closer to the deck, clearance volume is reduced and compression ratio is increased.

This is how to calculate the clearance volume once the piston deck height knows:

$$V_{cv} = \frac{V_{sv}}{CR-1} \tag{2.2}$$

Piston Dome

Note that clearance volume does not take into account any pop-up domes or sunken-in dishes on the head of the piston. These configurations also increase or decrease volume in the combustion chamber and affect the compression ratio [7]. The manufacturer's catalog will list the displacement in cubic centimeters of the dishes or domes on the piston, but that it's not consistent whether the cc's of a dish as a positive or a negative number.

Head-Gasket Volume

Head-gasket volume is determined by the compressed thickness of the gasket. A thicker gasket adds volume and reduces compression; a thinner gasket reduces volume and increases compression. A gasket's compressed thickness is listed in the manufacturer's catalog and ranges from 0.051 inch to 0.015 inch. Also, the gasket bore is often larger than the engine bore.

Chamber Volume

The volume of the combustion chambers is the final factor in determining compression ratio. The larger the chamber, the more volume is added to the cylinder and the lower the compression ratio; smaller chambers reduce volume and increase the compression ratio.

The volume of the chambers can vary greatly depending on the type of heads and valves used, the amount the heads may have been milled, the number of valve jobs that have been performed, and any custom chamber grinding that has been done.

Manufacturers of cylinder heads will tell the range of sizes of the chambers in their heads, but for any used or custom-machined heads, the only way to know the size of the chambers is to have a machine shop check [3].

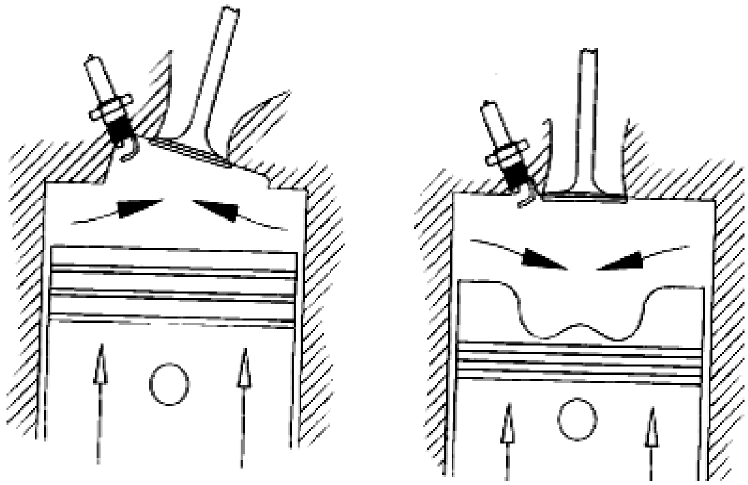


Figure 2.7: Combustion chamber [8]

2.2.3 Air and Fuel System

Air is sucked in through the throttle and carburetor via the air intake. Its flow is controlled by the throttle valve. The suction occurs during the intake stroke when the piston is moving down and the intake valve is open. The throttle valve is used for controlling the speed (power output) of the engine. It is a disc that can be rotated so as to either un-impede the air flow (full throttle) or block the airway preventing air flow.

The throttle valve is operated by hand through the use of a control lever. After passing through the throttle valve the air passes through the carburetor. The carburetor has a narrow air passage that causes the air flow to increase in speed. At the point where air flow reached its maximum speed a hollow needle feeds liquid fuel into the air stream. This causes the liquid fuel to break up into microscopically fine droplets producing an aerosol. This mixture of air and fuel is highly combustible.

2.2.4 Piston, Cylinder, Cams, Valves and Spark Plug

The engine block is a metal casting. The piston moves, up and down within the cylinder and drives the crank via the piston rod. Piston rings prevent gasses from escaping. The cavity below the cylinder is the crankcase which contains oil that provides lubrication. There are two valves: intake valve and the exhaust valve. These valves are operated by the two cams. The cams are geared to the drive shaft so that the drive shaft must make two complete revolutions for each revolution of a cam [2].

2.2.5 The Electrical System

The spark plug has a narrow gap through which an ionizing current passes resulting in a high temperature spark that causes the fuel-air mixture to ignite. Like the cams the contactor is driven at the half the speed of the drive shaft. The contactor completes the circuit to the ignition coil just after the intake valve has closed and the piston is starting to move downwards. The ignition coil is actually a type of electrical transformer. The low voltage current in the primary coil (few turns) results in a very high voltage in the secondary coil (many turns).

2.3 Engine Performance

The criteria for assessing the engine performance include the determination of engine power, engine efficiency, fuel consumption, engine stroke and bore, engine speed and mean effective pressure. All this criteria will show the engine performance and it maximum power.

There were many research related to the four stroke engine performance. Every part in the engine component got value itself to improve the performance of the engine. It has the advantages and disadvantages according to their modification.

2.3.1 Variable stroke-length and compression ratio

The engine power is based on the power cycle of the piston inside the cylinder. For the fixed-stroke engines, by throttling of the intake fuel-air mixture, the load variation is balanced out [12] [9]. Since the stroke remained constant, the pumping and frictional losses unchanged approach but in variable-stroke engines these loses are reduces. For different stroke lengths, the different power and efficiency are corresponding to the compression-ratios [14] [1].

The effect on engine performance based on the variable stroke-length is engine’s indicated power increases as the bore-to-stroke ratio decreases [11] [15][12]. If the stroke length increases, the compression ratio will decrease. This will cause the maximum pressure to decrease [10].

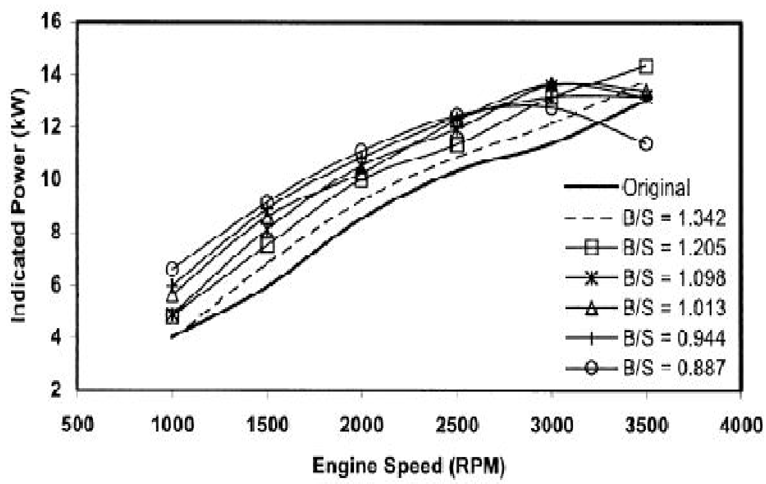


Figure 2.8: Indicated Power vs RPM [12]

For the indicated mean effective pressure (IMEP), it adversely affected by the engine’s displacement-volume [15] [12]. This is due to the reduced volumetric efficiency and the compression ratio at a larger stroke length. This will cause the cylinder pressure to decrease and it will lower the mean pressure inside the cylinder [11].

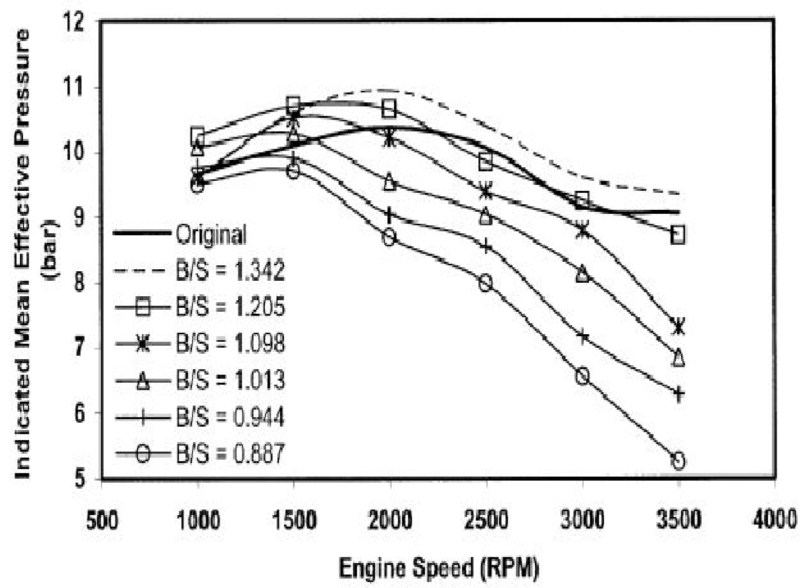


Figure 2.9: IMEP vs RPM [12]

For the larger stroke-length, it allows heat to be wasted and lost to the cylinder walls. This means that less heat is liberated by the combustion of the fuel which would be used to develop useful work. By increasing the stroke-length and reduces the compression ratio, it will cause the cylinder temperature at EVO to increase [15] [12].

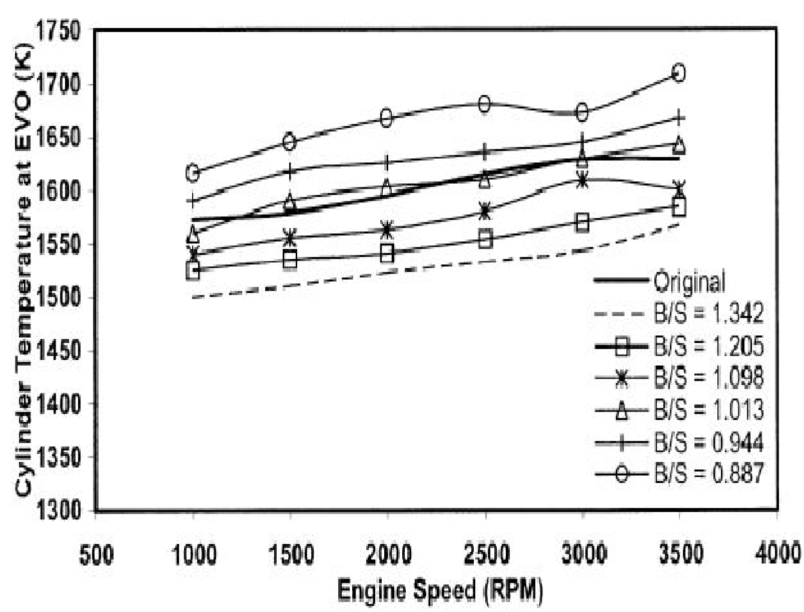


Figure 2.10: Cylinder Temperature vs RPM [12]

This is due to the lower expansion ratio. The effect of this factor is reflected in the decrease of indicated thermal efficiency at larger stroke length.

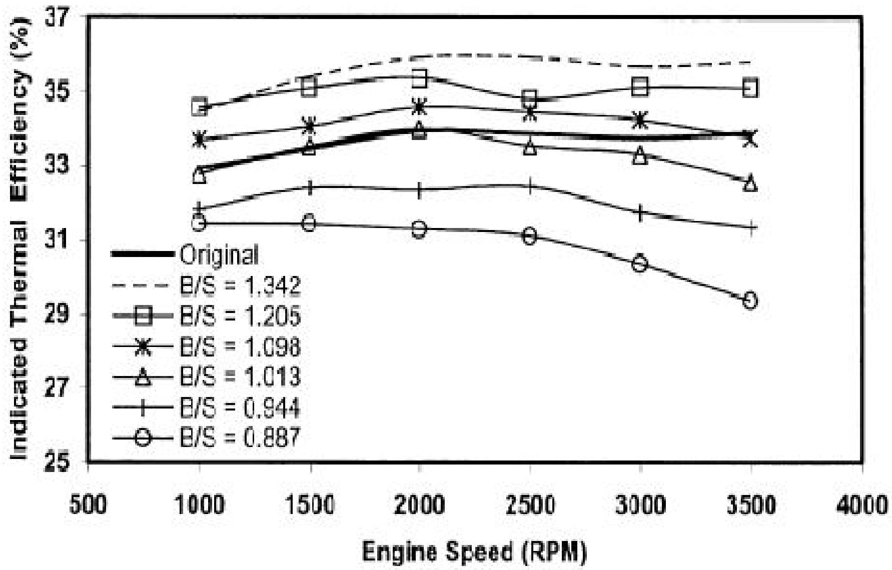


Figure 2.11: Indicated Thermal Efficiency vs RPM [12]

2.3.2 Spark Ignition Engines at Part Load

The four stroke spark ignition (SI) engine pressure-volume diagram (p-V) contains two main parts which are the compression combustion expansion (high pressure loop) and the exhaust-intake (low pressure or gas exchange loop) parts. The main reason for efficiency decrease at part load conditions for these types of engines is the flow restriction at the cross sectional area of the intake system by partially closing the throttle valve, which leads to increased pumping losses and to increased low pressure loop area on the p-V diagram.

Meanwhile, the poorer combustion quality, influence these pressure loop areas. This investigation shows that the potential for increasing the efficiency of SI engines at part load conditions is not yet exhausted. The most promising methods to decrease the fuel consumption at part load conditions are stratified charge and variable displacement engines.

2.3.3 Variable Valve Timing

When the intake opening time and intake closing time are optimized, the exhaust valve will closes at top center [2]. An early opening will shorten the expansion stroke for a given speed which will reduce the work done on the piston and the engine torque [6]. A high work will be required to displace the exhaust gas during the exhaust stroke which the late exhaust opening will not allow enough time for the in-cylinder pressure to reach the ambient pressure.

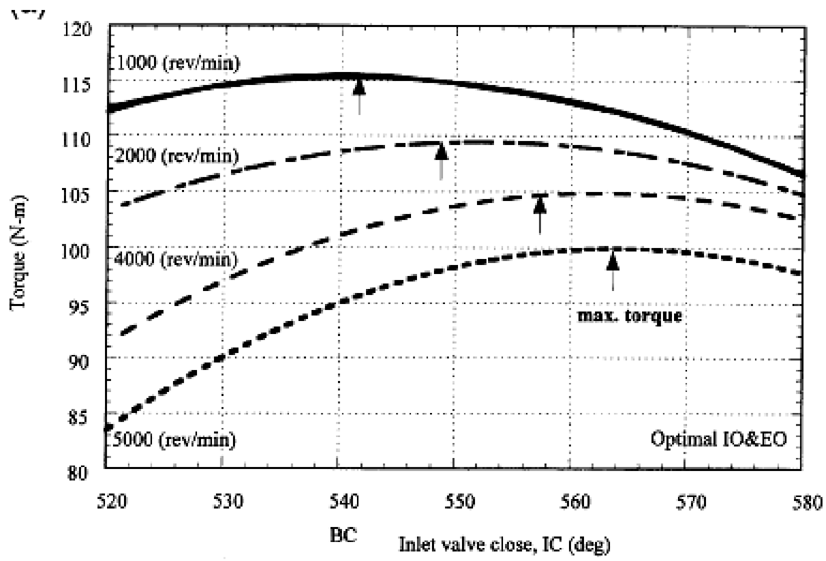


Figure 2.12: Torque vs Inlet Valve Close [1

2.3.4 Octane number higher than engine requirement

The effect of higher octane gasoline on the brake specific fuel consumption (BSFC) in variable load is the lower octane number used is better than the higher octane number. The higher octane number will delay the ignition longer and the speed of the flame will be shorter. This will cause the reduction of the maximum pressure in the engine output power and fuel consumption per output power will increase [13].

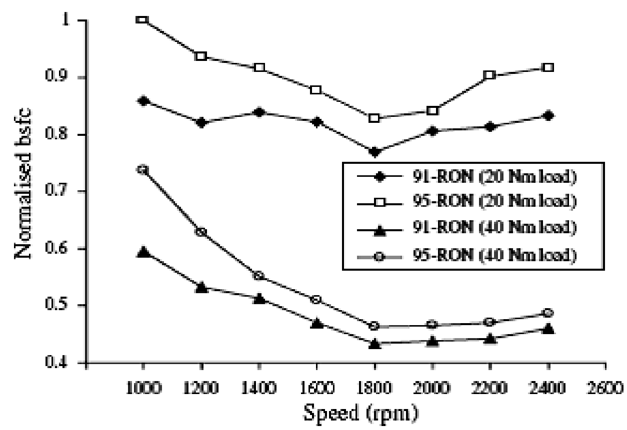


Figure 2.13: Normalised BSFC vs RPM [13]

CHAPTER 3

METHODOLOGY

3.1 GT-POWER SIMULATION

As the industry-standard engine simulation tool, GT-POWER is used by all the leading engine and vehicle makers and their suppliers. Besides that, it's also used for ship, power-generation engines, small two and four stroke engines and racing engines such as NASCAR and F1. To model any advanced concept, it provides the users with many components. Among it advantage are it ease of use and its tight integration with the rest of the GT-SUITE, which give GT-POWER a virtual engine perspective. GT-POWER also provides with a proven set of high-productivity features for pre- and post-processing, DOE/optimization, neutral networks and control modeling which is in the GT-SUITE environment. GT-POWER is available as a standalone tool or coupled with GT-DRIVE, GT-FUEL and GT-COOL as the GT-SUITE flow product.

Among it application are

- Torque curve and fuel consumption
- Manifold design and tuning
- Transient performance and response
- Valve profile and timing optimization
- Combustion and emissions
- Turbocharger response and matching
- EGR system design
- Full vehicle performance
- Design analysis

- Coupled one-dimensional and three-dimensional simulations

3.2 MODENAS KRISS 110cc

The Modenas 110cc engine has been using GT-POWER as the simulation program by predictive one-dimensional flow simulations. This chapter will give some general information on one-dimensional flow simulations and specific information on some simulation areas in need of special attention. All the experimental data needed for validation purposes will also be discussed.

3.3 ENGINE PARAMETER AND MEASUREMENT

Specifications of Modenas Kriss 110cc Engine

Table 3.1: Engine Base Parameter

Engine Parameter (unit)			Value
Bore (mm)			53.00
Stroke (mm)			50.60
Connecting rod length (mm)			100.25
Displacement (cc)			111.00
Compression ratio			9.3:1
Max horsepower			6.6 kW @ 8500 rpm
Max torque			9.3 N-m @ 4000 rpm
Valve timing :	Inlet	Open	20^{o} BTDC
		Close	60^{o} ABDC
	Exhaust	Open	55^{o} BBDC
		Close	25^{o} ATDC

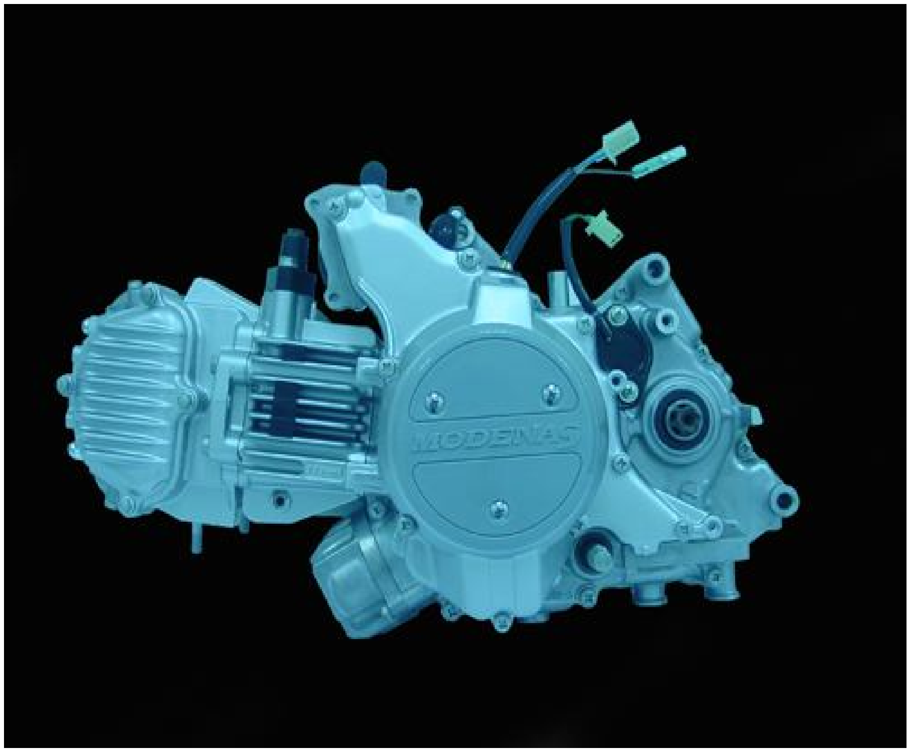


Figure 3.1: Modenas 110cc engine

3.4 ONE-DIMENSIONAL COMPUTER SIMULATION

By analyzing the mass and energy flows between the individual engine components and the heat and work transfer within each component, the engine performance can be studied. Each pipe needs to be represented in the actual simulation model which contains information of dimensions, surface roughness, temperature and etc.

To develop the GT-POWER of single-cylinder four stroke engine modeling is step by step and in general the system representation is built into designated components which are pipes, flow splits, cylinder, and environment and connected to by one another by connections model such as orifices and valves. The GT-POWER library of the all engine components data will be input the engine components size data. Select Window and then Tile with Template Library from the menu will create the GT-POWER model. All the available templates in GT-POWER contains in the Template

Library. Before used to created objects and parts, some of these templates need to be copied into the project. Click on the icons listed and drag them from the template library into the project library for the purpose of this model. Some of these objects and templates have already been defined and included in the GT-POWER template library. For first case of the simulation, all of the parameter in the model listed automatically in the case setup and each one must be defined.

GT-SUITE produces several output files that contain simulation results in various formats, whenever a simulation runs. The GT-POST application is the post-processing for the most output available and it used to view animation and order analysis output. Report tables that summarize the simulation produced after the simulation was finished. All the important information about the simulation and simulation results in tabular form contains in these reports. The engine result performance is informed by the computational simulation of the engine model.

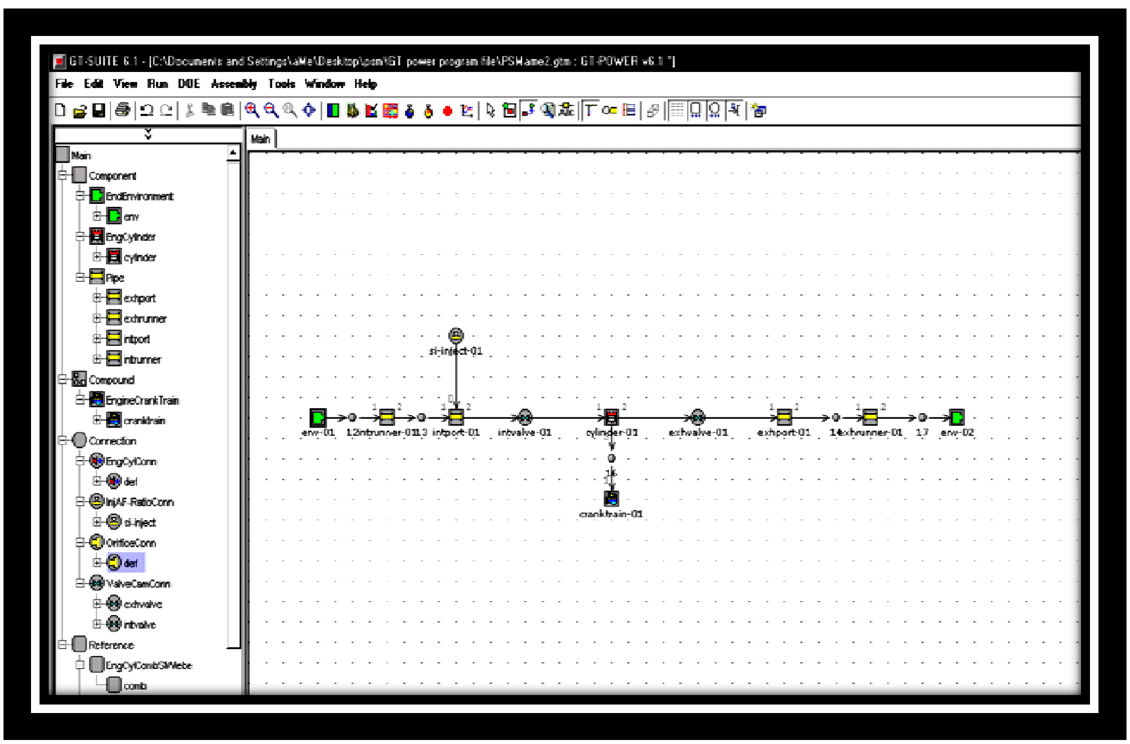


Figure 3.2: GT-POWER system layout

- The ambient condition designate ‘Env’ will start the intake system. For the intake and exhaust system, the ‘Env’ is used to describe the boundary condition of pressure, temperature and the mixture composition. Then it is connect to the default orifice to an inlet pipe.

- To model the intake and exhaust systems, ‘Pipe’ template will be used. The pipe is assumed round and straight. The effect of other physical geometries such as the friction multiplier, heat transfer multiplier and pressure loss coefficient are adjusted in this account.

- ‘InjAF-RatioConn’ Connection is modeled as carburetor. It describes as an injector that injects fluid at a prescribed fuel-to-air mixture entering the main pipe.

- ‘EngineCrankTrain’ is modeled as the cranktrain. The engine friction needs to define in this sector. This can show the piston cylinder moving in the cylinder.

- 'EngCylinder' is modeled as the cylinder. The values of the burn rate are stored and this does not indicate a change in actual values or the method of calculation. It is used to identify all the engine parameter such as connecting rod, bore, stroke, heat transfer and combustion model.

Table 3.2: summary of input data in one-dimensional engine model

Engine part	Type of information
Engine characteristics	Compression ratio
Cylinder geometry	Bore, stroke, connecting rod length
Intake and exhaust system	Geometry of manifolds
Fuel injectors	Location, fuel/air ratio
Intake and exhaust valve	Valve diameter, lift profile, discharge coefficient
Ambient	Pressure, temperature and humidity

3.5 Analysis of the flow

Simulation of one-dimensional flow involves the solution of conversation equation; mass-, energy- and momentum, in the direction of the mean flow. Mass conservation states that the rate of change in mass within a sub system is equal to the sum of mass flowing into and out from the system:

$$\frac{dm}{dt} = \sum_i \dot{m}_\iota - \sum_e \dot{m}_e \tag{3.1}$$

where *i* denotes inlet and *e* for exit. In one-dimensional flow the mass flow rate, *m* is defined by

$$\dot{m} = \rho A U \tag{3.2}$$

Where ρ is the density

A is the cross-sectional area

U is the fluid velocity

Energy conservation states that the rate of change of energy in a sub system is equal to the sum of energy transfer in and out of the system. The means of energy transfer are work, the energy accompanying the mass flow and heat:

$$\frac{d(me)}{dt} = -p\frac{dV}{dt} + \sum_i \dot{m}_\iota H - \sum_e \dot{m}_e H - h_g A\left(T_{gas} - T_{wall}\right) + \frac{dQ_{ch}}{dt} \qquad (3.3)$$

where *e* is the internal energy, *H* the total enthalpy, h_g the heat transfer coefficient, T_{gas} and T_{wall} the temperature of the gas and wall respectively and Q_{ch} the energy released during combustion. The heat transfer from internal fluids to pipe and flowsplit walls is dependent on the heat transfer coefficient, the predicted fluid temperature and the internal wall temperature. The heat transfer coefficient, which is calculated at every time step, is a function of fluid velocity, thermo-physical properties and the wall surface finish. The internal wall temperature is defined by the user.

Momentum conservation states that the net pressure forces and wall shear forces acting on a sub system are equal to the rate of change of momentum in the system plus the net flow of momentum out of the system:

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum_i \dot{m}_\iota u - \sum_e \dot{m}_e u - 4C_f \frac{\rho u^2}{2}\frac{dxA}{D} - C_p \frac{1}{2}\rho u^2 A}{dx} \qquad (3.4)$$

where *u* is fluid velocity, C_f the friction loss coefficient, *D* the equivalent diameter, C_p the pressure loss coefficient and *dx* the element length. In order to obtain the correct pressure and friction loss coefficients the software uses empirical correlations to account for pipe curvature, surface roughness etc.

The amount of chemical energy released during combustion, Q_{ch}, needs to be simulated. By expressing the change in internal energy with thermodynamic relationships and neglecting the crevice effects, the heat release rate can be expressed as a function of cylinder pressure:

$$\frac{dQ_{ch}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} + Ah_c(T - T_w) \qquad (3.5)$$

where *A* is the combustion chamber surface area and Q_{hc} the heat transfer coefficient. The heat transfer coefficient is estimated according to a modified Woschni's correlation.

The deduced heat release rate can be expressed in a parametric form suggested by Wiebe:

$$x_b = 1 - exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1}\right] \quad (3.6)$$

where x_b is the percentage of burned mass at crank angle θ, θ_0 is the crank angle at start of combustion, Δθ the total combustion duration, *a* can be expressed as a function of the defined combustion duration and *m* is an adjustable parameter. The parameters θ *0,* Δθ and *m* can be adjusted so that the Wiebe function, resembles the accumulated heat release rate, which is based on measured cylinder pressure and effects of changing *m* and Δθ can be seen.

The cylinder pressure needs to be measured or in some way estimated, in order to calculate the heat release. The implication is that 1-D simulation using a heat release analysis for combustion simulation can never become truly predictive.

CHAPTER 4

RESULT AND DISCUSSION

4.1 PRELIMINARY RESULT

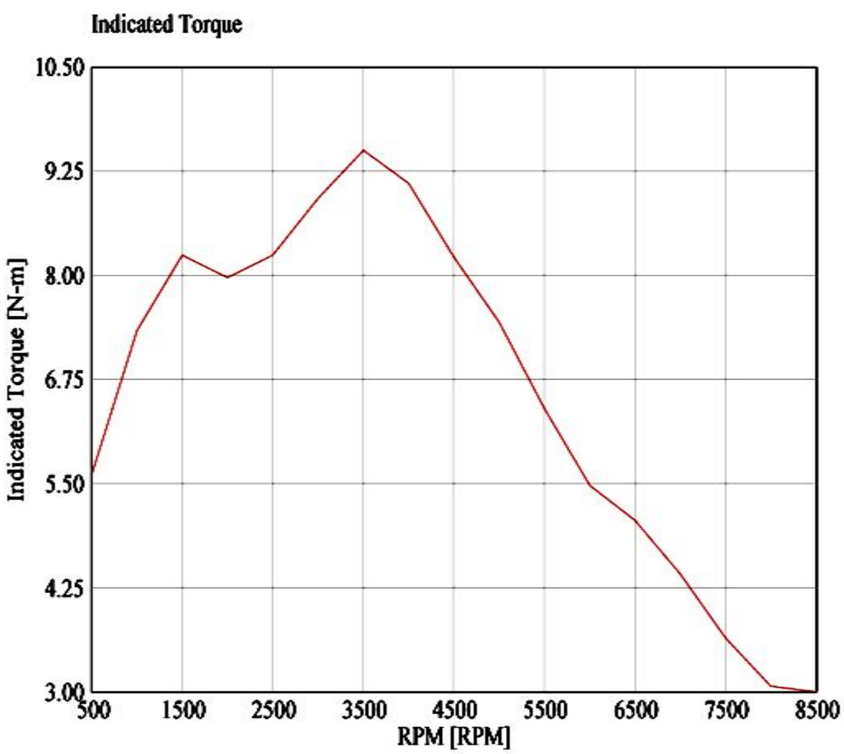


Figure 4.1: Indicated torque vs rpm

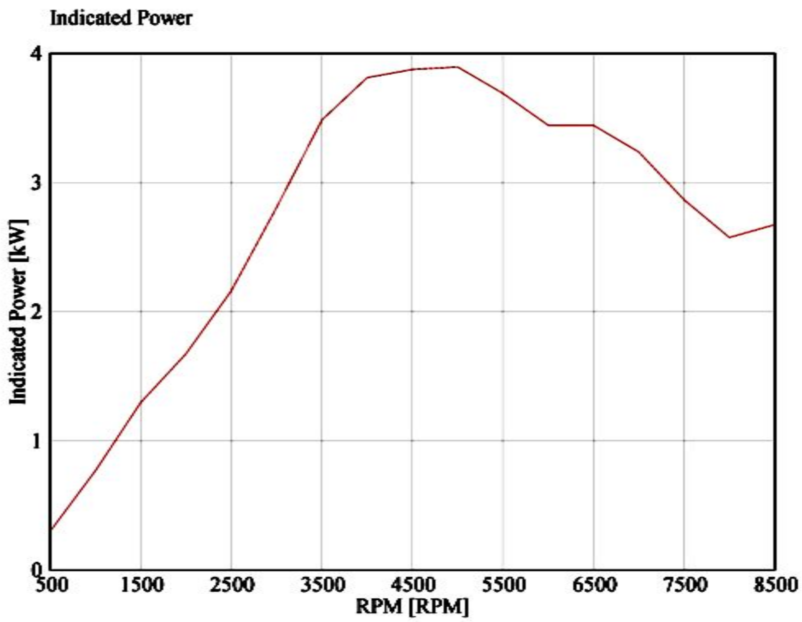


Figure 4.2: Indicated power vs rpm

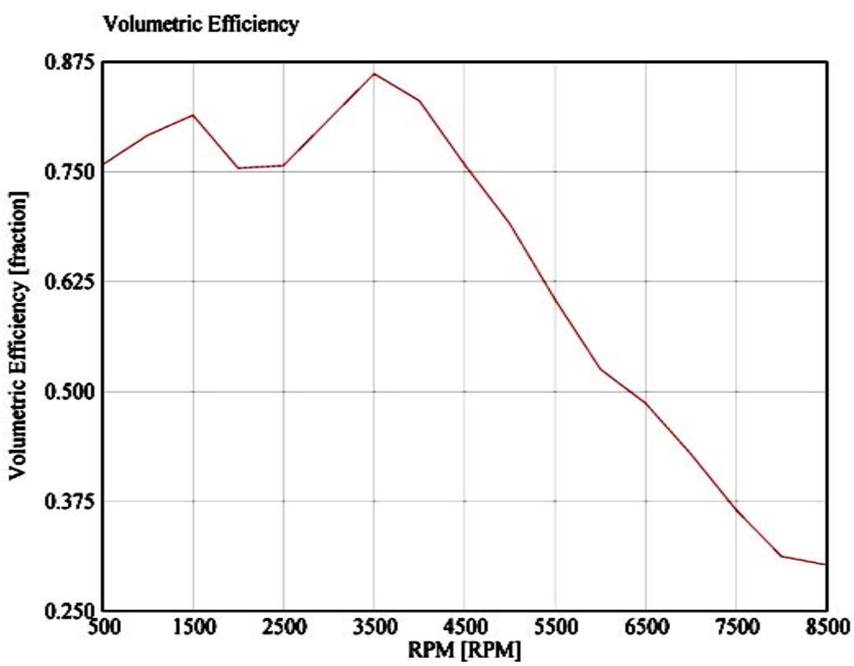


Figure 4.3: Volumetric efficiency vs rpm

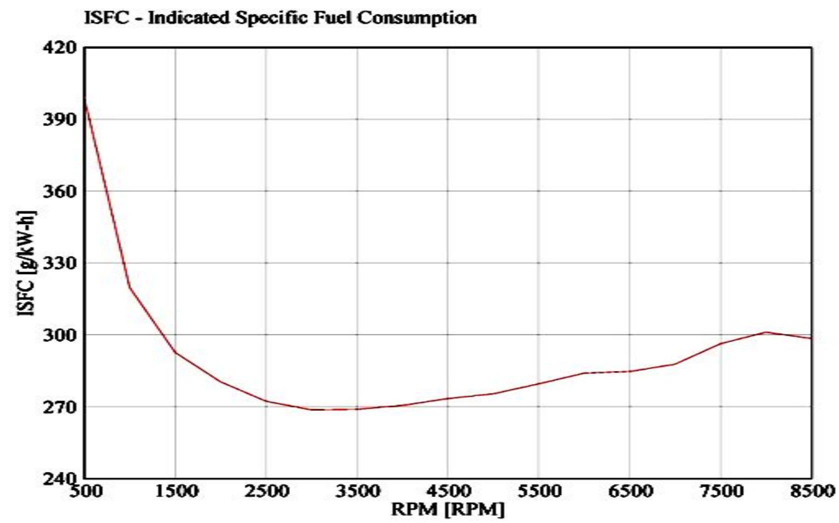


Figure 4.4: Indicated specific fuel consumption vs rpm

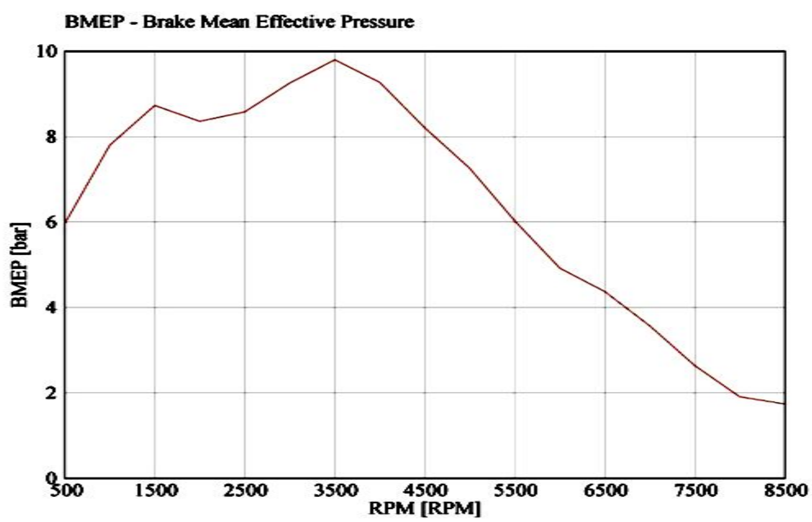


Figure 4.5: Brake mean effective pressure vs rpm

The **Figure 4.1** – **Figure 4.5** shows the original performance of the Modenas 110cc four stroke motorcycle engine using the original parameter. The peak value for indicated torque is 9.498 N-m at 3500 rpm. The peak value for indicated power is 3.893 kW at 5000 rpm. For ISFC, lowest value is 268.733 g/kW-h at 3000 rpm and the highest value is 399.231 g/kW-h at 500 rpm. The maximum BMEP for this engine is 9.793 bar which occur at 3500 rpm.

This is the preliminary results before the parameter is changed. The change makes on the bore and stroke size but the displacement of the engine is still the same. The effect of the changes will be discussed.

4.2 ENGINE SPECIFICATION

For the original bore and stroke size for Modenas 110cc motorcycle engine cylinder is 53mm for bore and 50.6 for stroke. As the improvement in the performance of the engine, the size of bore and stoke will change and the result will be plot. The displacement of the engine remains the same.

4.3 VARIABLE BORE AND STROKE LENGTH

Table 4.1 shows the different in size of bore and stroke for the engine cylinder. As the displacement remains constant, the compression ratios for the engine are variable. The effect of the variable in compression ratio will be shown in graph plotted after the GT-Power simulation is run.

Table 4.1: Variable bore and stroke

Set of Modification	Bore x stroke (mm)	Bore/stroke	Compression ratio
Original	53 x 50.6 (Original)	1.047	9.3
1	54.7 x 47.7	1.147	8.79
2	53.8 x 49.2	1.093	9.058
3	52.2 x 52.2	1.000	9.559
4	51.2 x 54.4	0.941	9.897
5	50.6 x 55.7	0.908	10.108

4.4 DATA INTERPRET

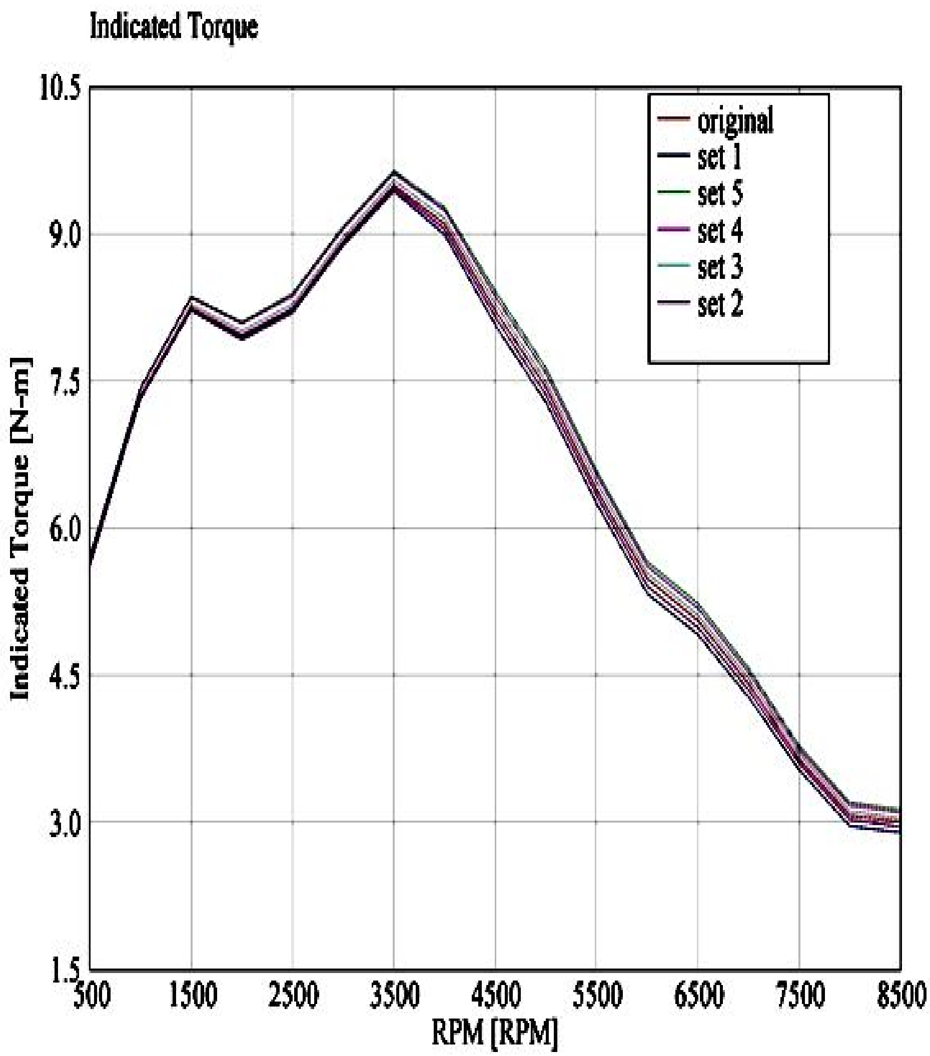


Figure 4.6: Indicated Torque vs Engine Speed

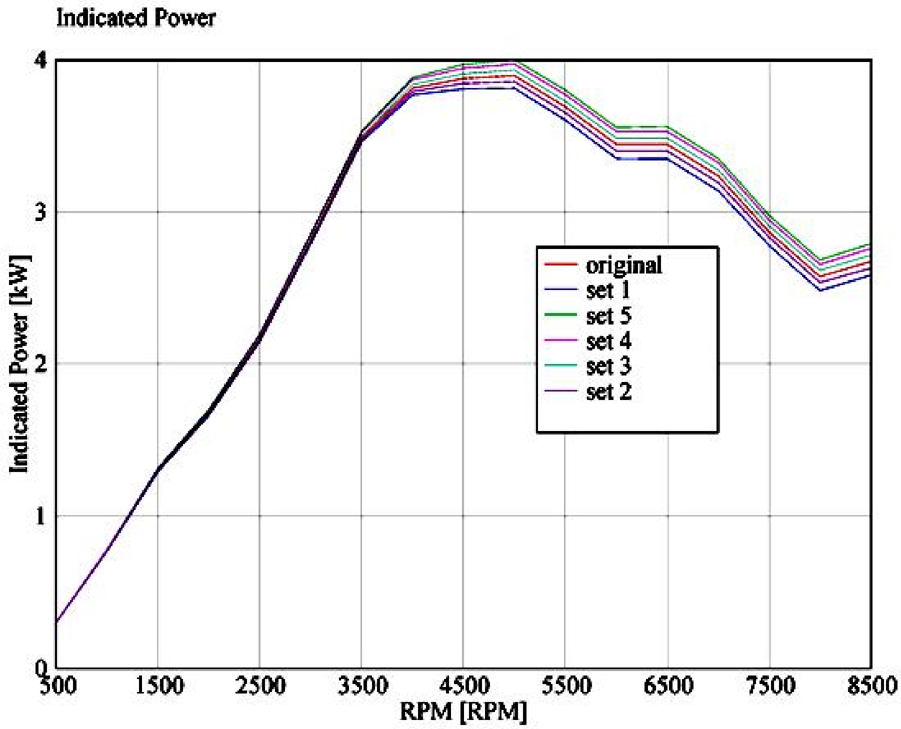


Figure 4.7: Indicated Power vs Engine Speed

For both **Figure 4.6** and **Figure 4.7**, increasing the compression ratio, the value of the indicated power and indicated torque is increased. For **Figure 4.7**, the maximum engine power as a function of engine speed for compression ratio variation. The results are obtained at wide-open–throttle with manufacturer setting ignition timing. During the operation, the power is seen reduced at high engine speed. This power loss is due to the displacement of air and low burning velocity. At high engine speed, the engine power and torque are further decreased.

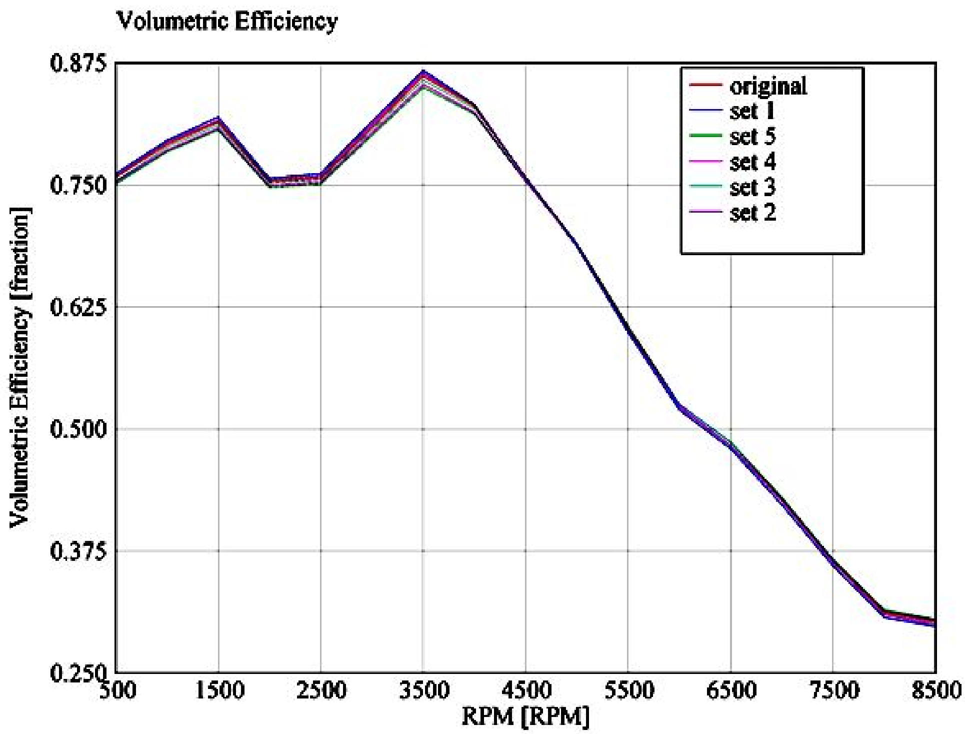


Figure 4.8: Volumetric Efficiency vs Engine Speed

As can be seen from **Figure 4.8**, volumetric efficiency (ηVOL) increases are caused primarily by the rising the compression ratio. However, trends from compression ratio variation show minor increases in ηVOL as the compression ratio is increased. The improved cylinder filling is caused by the reduced heating effect from the lower residual gas fraction, which increases the trapped charge density. This change in ηVOL is only minor, corresponding to a decrease in the residual gas fraction over the test compression ratio variation. Air consumption rates are highly dependent on intake resonant waves, which increase in intensity to become most dominant at wide open throttle. This resonant wave, when correctly timed in the induction process, improves air consumption and reduces the residual gas fraction within the cylinder.

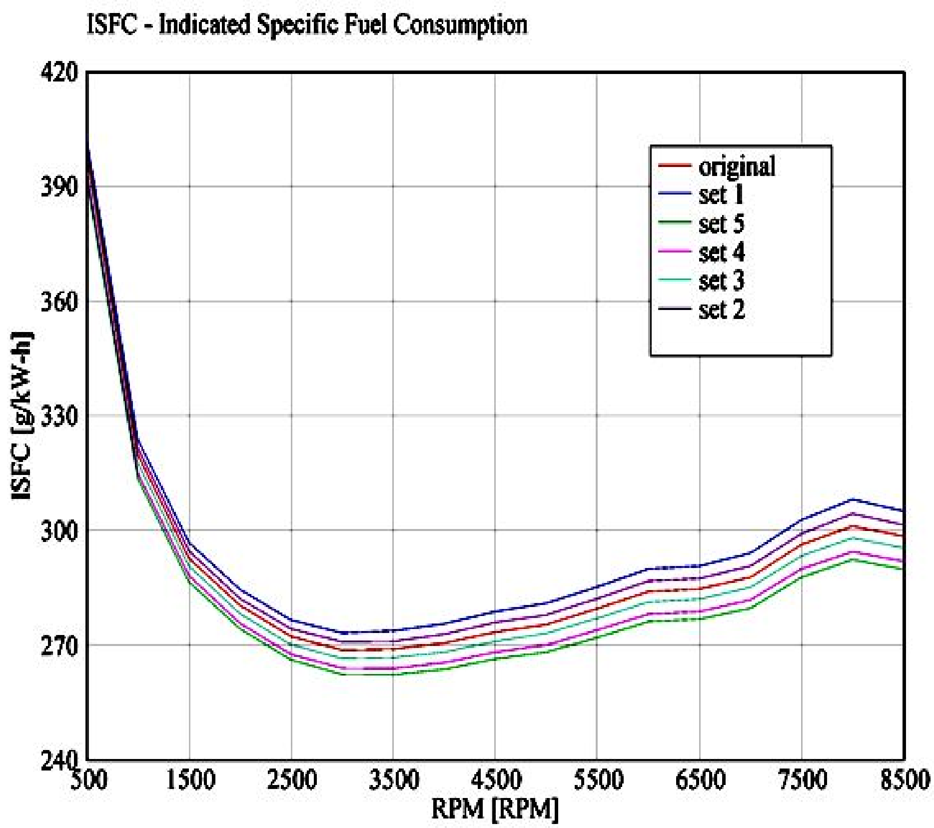


Figure 4.9: ISFC vs Engine Speed

Figure 4.9 shows Indicated Specific Fuel Consumption (ISFC) trends due to compression ratio changes show efficiency improvements for increasing compression ratio. ISFC graph corrected to stoichiometric fuel conditions to allow efficiency comparisons to other engines irrespective of the fuel mixture. Smaller bore and stroke engines inherently suffer from increased heat losses when compared to larger bore and stroke size due to the higher surface-to-volume ratio which results in reducing small engine efficiencies.

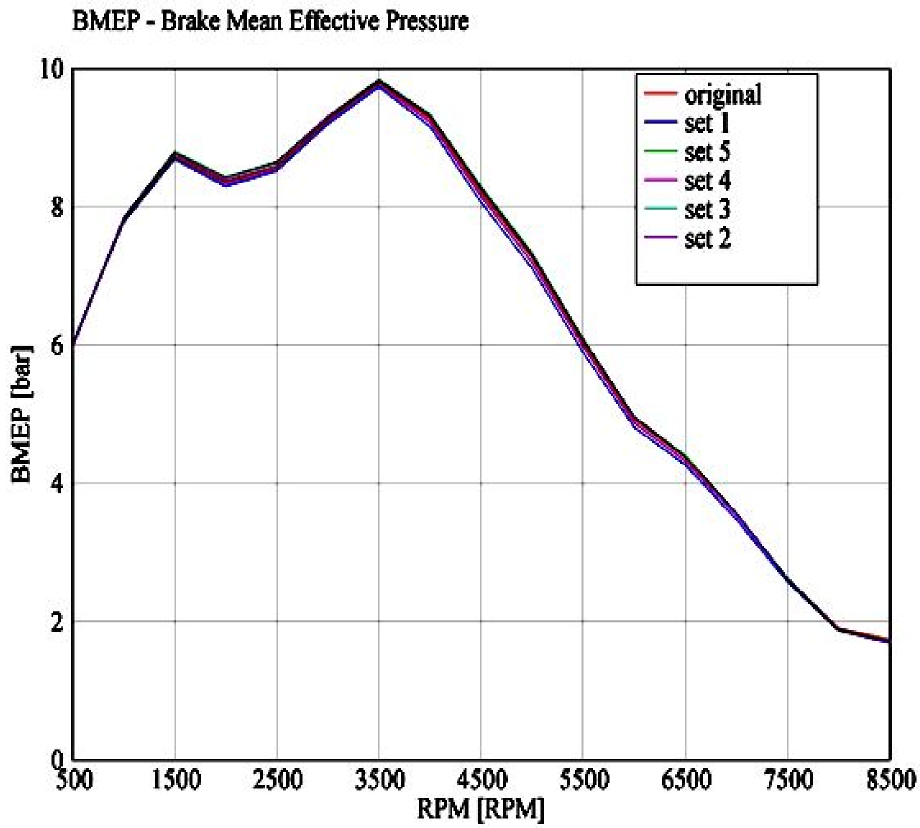


Figure 4.10: BMEP vs Engine Speed

Figure 4.10 displays BMEP in different compression ratio and different engine speed. When compared to different bore and stroke size, trends show matching BMEP effects for varying the compression ratio. Increases in BMEP are directly proportional to increases in air consumption. Increases in compression ratio are shown to directly correlate to increases in BMEP for all speeds. The increases in BMEP are attributed to a combination of both increased air consumption and improved combustion. As the compression ratio is increased, the residual gas fraction within the cylinder decreases. Reducing the hot products within the cylinder minimizes the warm up of the incoming charge during the induction process, improving charge density and air consumption. A bulk increase in charge density also enhances the combustion process with improved burning rates.

4.5 COMPARISON

In Table 4.2, the result from the simulation is listed down. The improved of the engine performance is indicated. For the best bore-stroke size to replace the original size is chosen.

Table 4.2: Result of the Simulation

Bore x stroke (mm)	Maximum Indicated torque (N-m)	Maximum Indicated power (kW)	Brake Mean Effective Pressure (bar)	Indicated Specific Fuel Consumption (g/kW-h)	Volumetric efficiency
original	9.498	3.893	9.793	399.231	0.8609
54.7 x 47.7	9.435	3.812	9.726	402.311	0.8673
53.8 x 49.2	9.468	3.855	9.761	400.828	0.8641
52.2 x 52.2	9.540	3.928	9.807	396.985	0.8573
51.2 x 54.4	9.612	3.971	9.825	393.639	0.8528
50.6 x 55.7	9.638	3.998	9.833	391.900	0.8499

For bore 50.6mm and stroke 55.7mm size is choose. According to the result in the GT Power simulation, this size can improve a lot in performance of Modenas 110cc 4-stroke motorcycle engine.

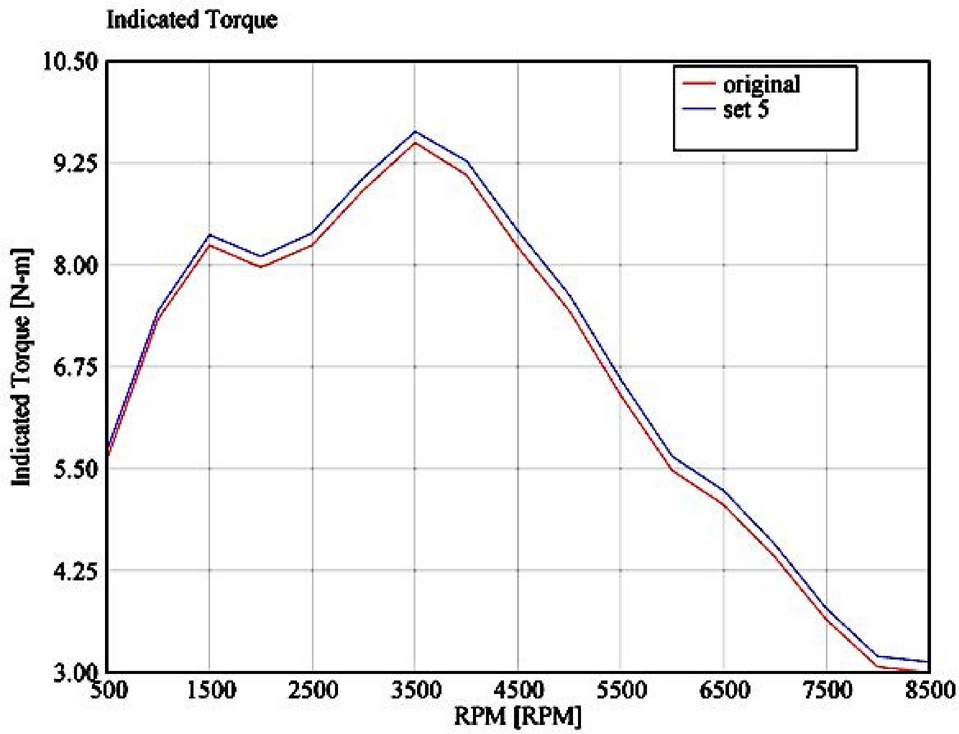


Figure 4.11: Indicated Torque Increasing

$$\frac{9.638-9.498}{9.498}\times 100\%$$
$$=1.47\%$$

In indicated torque, the improvement is 1.47% from the original size.

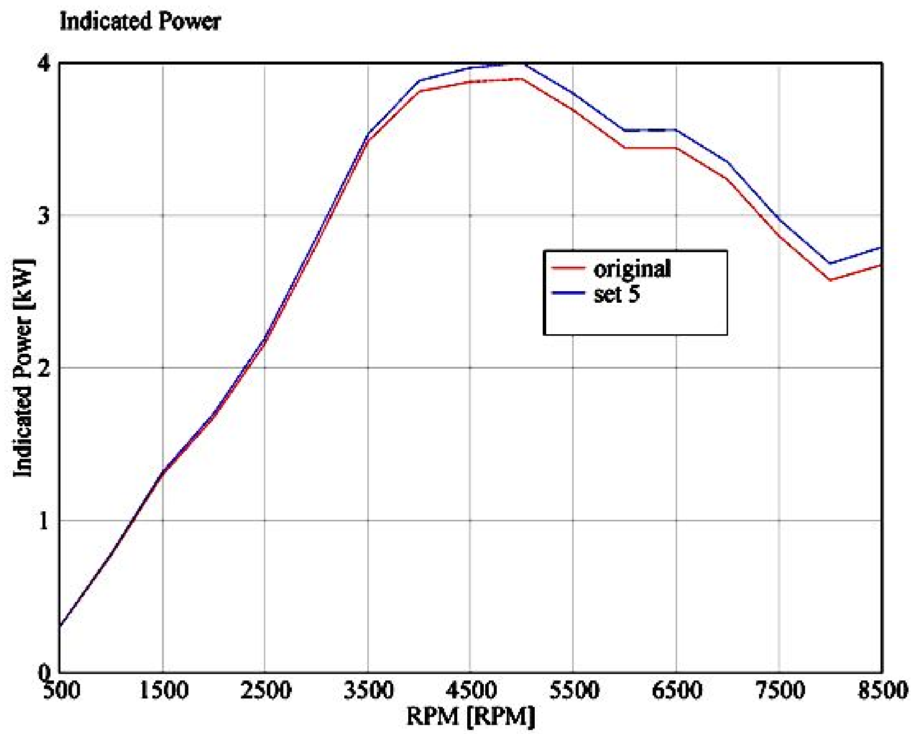


Figure 4.12: Indicated Power Improvement

$$\frac{3.998 - 3.893}{3.893} \times 100\%$$
$$= 2.7\%$$

The indicated power improve 2.7% from the original size

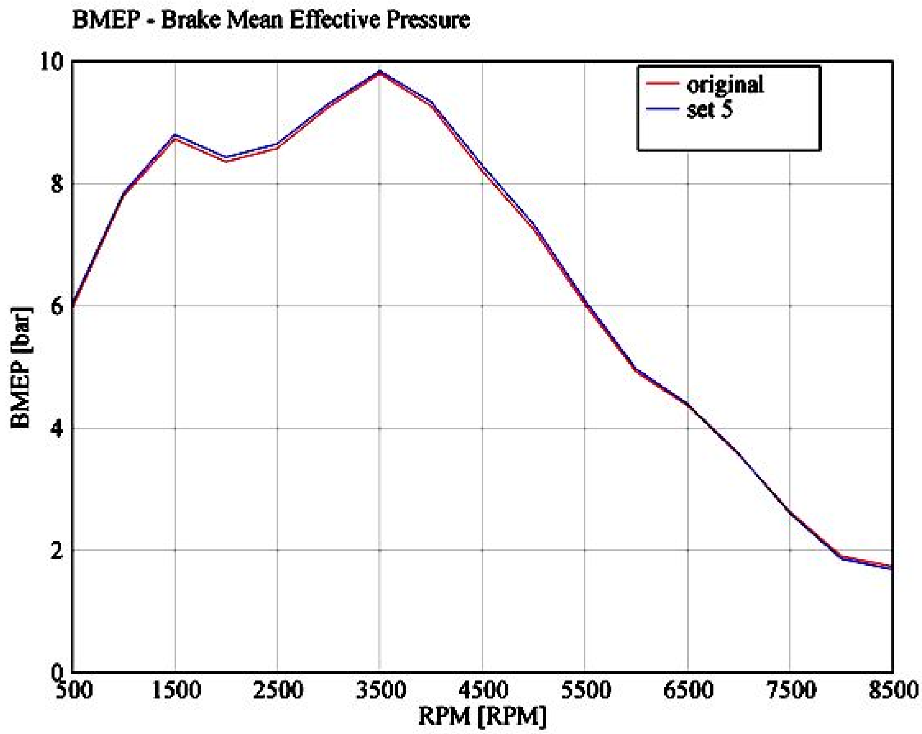


Figure 4.13: Brake Mean Effective Pressure

$$\frac{9.833-9.793}{9.793}\times 100\%$$
$$=0.408\%$$

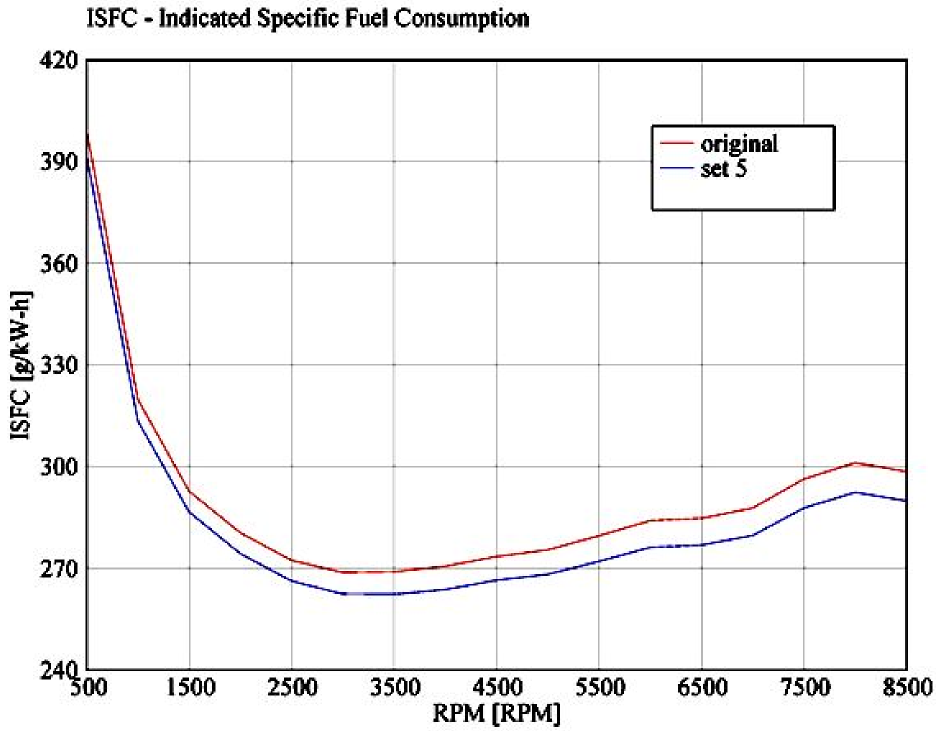


Figure 4.14: Indicated Specific Fuel Consumption Decreasing

$$\frac{[391.9 - 399.231]}{399.231} \times 100\%$$
$$= 1.84\%$$

The fuel consumption decreased about 1.84% from the original bore and stroke size.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

This investigation shows that the Modenas 110cc four stroke motorcycle engine can achieved the new scalar of the engine performance. In increasing about 1.48 % for indicated torque and 2.7 % in indicated power make the engine in it own class. The 50.6 mm bore and 55.7 mm stroke is very suitable to replace the original size. Otherwise, it also increased in BMEP and decreased in ISFC.

Higher compression ratios make it possible to improve power output, though there is a limit imposed by knocking. The simulation of GT Power were completed over a range of 8.79 until 10.108 by remains the displacement as the same which is 0.110 Liter.

In increasing in compression ratio, the fast burn occurs in the combustion chamber and this resulted in engine speed increased together with end-gas volume reductions around the periphery of the chamber. This effect the performance, efficiency and emission of the engine. The compression ratio should be optimize, which compensated for the higher levels of dissociation, friction and heat losses.

The main objective of this project achieved which is to investigate the performance of Modenas 110cc four stroke motorcycle engine using GT Power engine simulation software and determine the best parameter for engine part for the highest performance.

5.2 RECOMMENDATIONS

In this project, there are several analyses and work can be done to improve the performance of Modenas 110cc four stroke motorcycle engine. It is important to make sure that the project can operate in the internal combustion engine with the fewer defects and high performance.

Study on valve timing.

The valve timing will affect the engine performance. The study is on the different type of the valve timing will make something useful.

Study and analysis on the material selected

The study on material selected and analysis on it will avoid defect of the component. It is important to know the range of pressure and temperature of the component.

Design the Combustion Chamber

The shape of the combustion chamber is more important than many realize. It is far more important than what is done to the ports, or how big carburetors are. Current combustion chamber technology has resulted in very flat combustion chambers. The reduced volume of this shape results in a shorter flame period, which means ignition timing doesn't have to be advanced as far. The ignition advance required to get maximum power is in fact an indicator of how well the combustion chamber does its job.

Study about the Valve to Piston Clearance

The relationship between the valve and the piston necessarily changes in an engine modified with a different piston or camshaft, or when the cylinder deck clearance or cylinder head surface are modified. This relationship must change for maximum performance.

REFERENCES

1. Al-Baghdadi M.A.R.S., "Effect of compression ratio, equivalence ratio and engine speed on the performance and emission characteristics of a spark ignition engine using hydrogen as a fuel", Renewable Energy 29 (2004) 2245–2260

2. Ceviz M.A., "Intake plenum volume and its influence on the engine performance, cyclic variability and emissions", Energy Conversion and Management 48 (2007) 961–966

3. Ferguson C.R, Kirkpatrick A.T.(2001) " Internal Combustion Engines-Applied Thermosciences" (2nd Edition) New York: John Wiley & Sons, Inc

4. Gordon P. Blair (1999). " Design and Simulation of Four Stroke Engines" Warrendale, PA : Society of Automotive Engineer,Inc

5. Graham A Bell (2006). "Four-Stroke Performance Tuning" (3rd Edition) USA : Haynes Publishing

6. Harrison M.F, Dunkley A., "The acoustics of racing engine intake system", Journal of Sound and Vibration 271 (2004) 959-984

7. Huang R.F, Yang H.S, Yeh C.-N., " In-cylinder flows of a motored four-stroke engine with flat-crown and slightly concave-crown pistons", Experimental Thermal and Fluid Science 32 (2008) 1156–1167

8. John B Heywood (1988). " Internal Combustion Engine Fundamental "

9. Kutlar O.A, Arslan H,Calik A.T., “Methods to improve efficiency of four stroke, spark ignition engines at part load”, Energy Conversion and Management 46(2005) 3202–3220

10. Klein M, Eriksson L, Aslund J, “Compression ratio estimation based on cylinder pressure data”, Control Engineering Practice 14 (2006) 197–211

11. Mirani A.A, Khan J, Solangi S.A, Channar A.A, “Performance Evaluation of a Single Cylinder Four Stroke Petrol Engine”, ARPN Journal of Engineering and Applied Sciences (ts)

12. Ozcan H, Yamin J.A.A, “Performance and emission characteristics of LPG powered four stroke SI engine under variable stroke length and compression ratio”, Energy Conversion and Management 49 (2008) 1193–1201

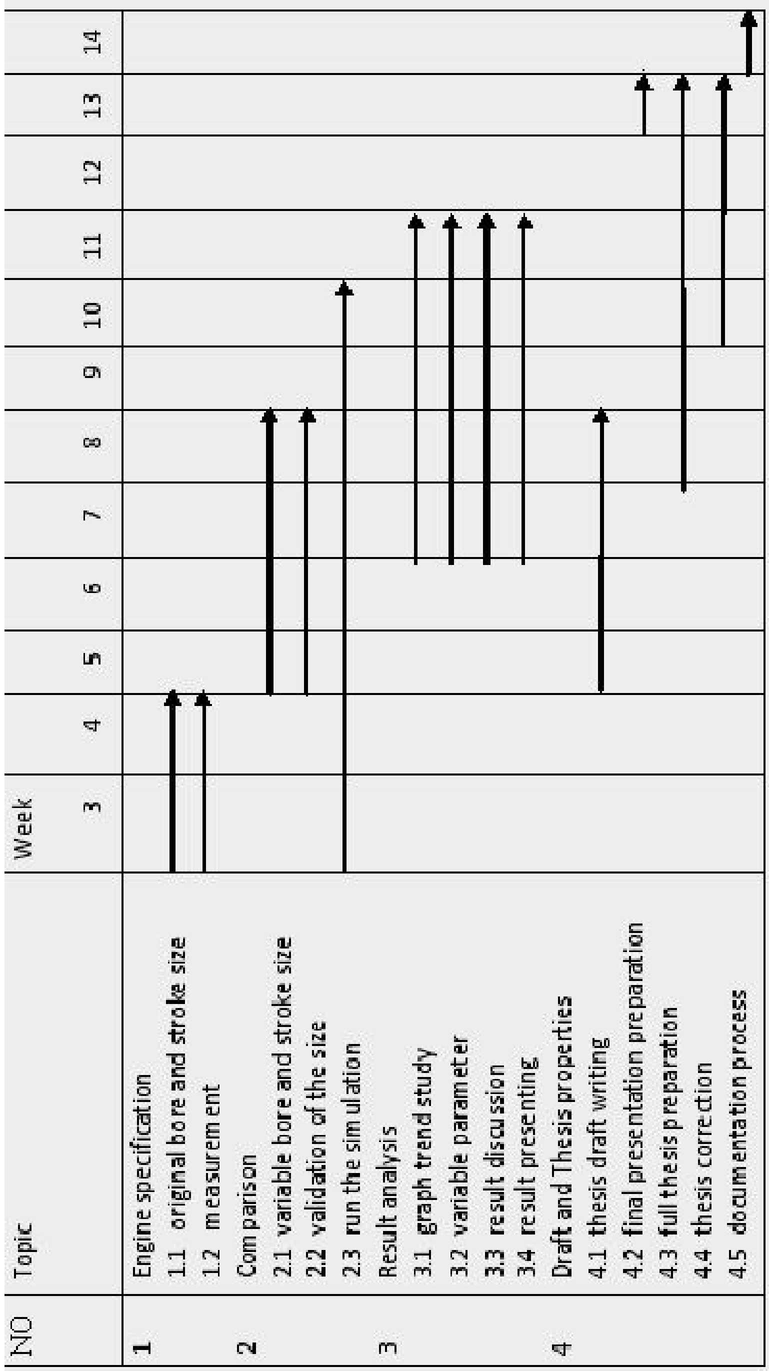
13. Sayin C, Kilicaslan I, Canakci M, Ozsezen N, ” An experimental study of the effect of octane number higher than engine requirement on the engine performance and emissions”, Applied Thermal Engineering 25 (2005) 1315–1324

14. Sher E, Kohany T.B., “Optimization of variable valve timing for maximizing performance of an unthrottled SI engine—a theoretical study”, Energy 27 (2002) 757–775

15. Yamin J.A.A, Dado M.H. ,” Performance simulation of a four-stroke engine with variable stroke-length and compression ratio”, Applied Energy 77 (2004) 447–463

APPENDIX A
GANTT CHART

NO	Topic	Week 3	4 5	6	7	8	9	10 11	12	13	14
1	proposal confirmation										
2	2.1 define problem statement										
	2.2 background writing										
	2.3 introduction writing										
	2.4 literature review										
3	METHODOLOGY										
	Computational Stage										
	3.1 one dimensional analysis										
	3.2 GT suite training										
	3.3 run simple analysis										
	3.4 design single engine										
	3.5 define engine geometries										
	3.6 re-design engine										
	3.7 model 1-D engine simulation										
	3.8 simulate the engine										
	3.9 analysis 1-D result										
4	PRELIMINARY RESULT										
	4.1 interprete and analysis all results										
	4.2 result documentation										
5	PRESENTATION PREPARATION										
6	PSM 1 PRESENTATION										

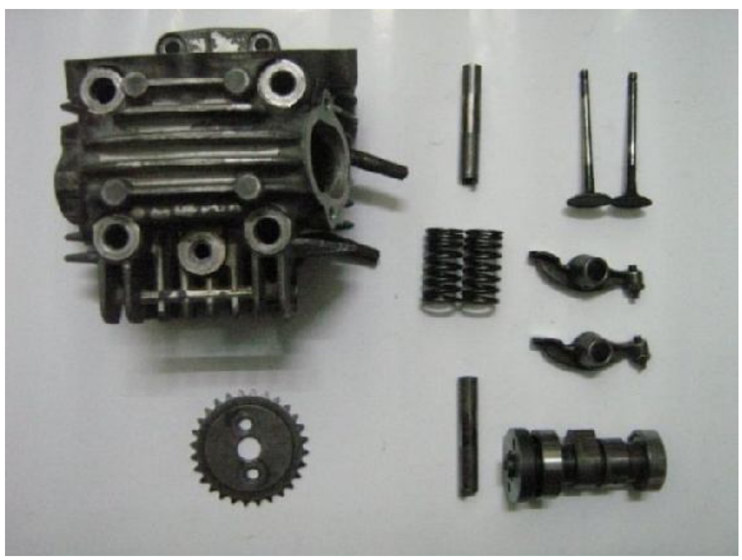


NO
Topic
Week
3
4
5
6
7
8
9
10
11
12
13
14
1
Engine specification
1.1 original bore and stroke size
1.2 measurement
2
Comparison
2.1 variable bore and stroke size
2.2 validation of the size
2.3 run the simulation
3
Result analysis
3.1 graph trend study
3.2 variable parameter
3.3 result discussion
3.4 result presenting
4
Draft and Thesis properties
4.1 thesis draft writing
4.2 final presentation preparation
4.3 full thesis preparation
4.4 thesis correction
4.5 documentation process

APPENDIX B

ENGINE BASE COMPONENT

Modenas 110cc engine component



Engine head component

Engine piston

Intake and exhaust poppet valve

APPENDIX C

TYPICAL DESIGN AND OPERATING DATA FOR INTERNAL COMBUSTION ENGINE

TABLE 2.1
Typical design and operating data for internal combustion engines

	Operating cycle	Compression ratio	Bore, m	Stroke/bore	Rated maximum: Speed, rev/min	Rated maximum: bmep, atm	Rated maximum: Power per unit volume kW/dm^3	Weight/power ratio, kg/kW	Approx. best bsfc, g/kW·h
Spark-ignition engines:									
Small (e.g., motorcycles)	2S,4S	6–11	0.05–0.085	1.2–0.9	4500–7500	4–10	20–60	5.5–2.5	350
Passenger cars	4S	8–10	0.07–0.1	1.1–0.9	4500–6500	7–10	20–50	4–2	270
Trucks	4S	7–9	0.09–0.13	1.2–0.7	3600–5000	6.5–7	25–30	6.5–2.5	300
Large gas engines	2S,4S	8–12	0.22–0.45	1.1–1.4	300–900	6.8–12	3–7	23–35	200
Wankel engines	4S	≈9	0.57 dm^3 per chamber		6000–8000	9.5–10.5	35–45	1.6–0.9	300
Diesel engines:									
Passenger cars	4S	17–23	0.075–0.1	1.2–0.9	4000–5000	5–7.5	18–22	5–2.5	250
Trucks (NA)	4S	16–22	0.1–0.15	1.3–0.8	2100–4000	6–9	15–22	7–4	210
Trucks (TC)	4S	14–20	0.1–0.15	1.3–0.8	2100–4000	12–18	18–26	7–3.5	200
Locomotive, industrial, marine	4S,2S	12–18	0.15–0.4	1.1–1.3	425–1800	7–23	5–20	6–18	190
Large engines, marine and stationary	2S	10–12	0.4–1	1.2–3	110–400	9–17	2–8	12–50	180