CYLINDER BY CYLINDER ENGINE MODELING OF SINGLE CYLINDER 4 STROKE ENGINE FOR CONTROL SYSTEM DEVELOPMENT

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Report submitted in partial fulfillment of the requirements for the award of Bachelor of Mechanical with Automotive Engineering

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FACULTY OF MECHANICAL ENGINEERING

I certify that the project entitled "*Cylinder by Cylinder Engine Modeling of Single Cylinder 4 Stroke Engine for Control System Development*" is written by Hasfazri bin Abdul Rahman. I have examined the final copy of this project and in my opinion; it is fully adequate in terms of scope and quality for the award of the degree of Bachelor of Engineering. I herewith recommend that it be accepted in partial fulfillment of the requirements for the degree of Bachelor of Mechanical Engineering.

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

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DEDICATION

I specially dedicate to my beloved parents
(Abdul Rahman bin Awang & Faridah bt. Awang), my siblings,
My supervisor and those who have guided
And motivated me for this project

ABSTRACT

The objective of this thesis is to design and simulate cylinder by cylinder engine model for control oriented study based on single cylinder four stroke engines, which combines both physical formulae, such as engine geometries, and empirical formulae. The engine performance, torque and power is calculated by integrating the pressure inside cylinder within one engine cycle. The importance of this study is to predict the engine performance parameters such as indicated work, brake power, and torque that provided with air fuel ratio data and detail geometrical specifications. The model of FZ150i full engine specifications is used for simulation in order to predict the engine performance. The model is simulated between 2000 to 6000 rpm of engine speed range. From this simulation, the result shows that it is almost same with the experimental data by Sitthiracha (2006). Without build the real engine, all the engine performance parameter can be calculated from this simulation, and reduced the time and cost.

ABSTRAK

Objektif tesis ini adalah untuk mereka bentuk dan simulasi silinder dengan menggunakan model enjin silinder bagi kajian kawalan yang berdasarkan enjin empat strok dimana ia mengabungkan kedua-dua formula fizikal, seperti geometrik enjin dan formula empirik. Prestasi enjin, tork dan kuasa boleh dihitung dengan mengintegrasikan tekanan dalam silinder bagi tempoh satu kitaran enjin. Kepentingan kajian ini ialah untuk meramal prestasi parameter enjin seperti kerja tertunjuk, kuasa brek dan tork dengan menggunakan data nisbah bahan api udara dan spesifikasi geometrik yang lebih terperinci. Spesifikasi enjin penuh Model FZ150i digunakan dalam simulasi supaya dapat meramal prestasi enjin. Model disimulasikan dengan kelajuan enjin pada kelajuan 2000 hingga 6000 rpm. Hasil keputusan simulasi ini menunjukkan ianya hampir sama dengan data eksperimen oleh Sitthiracha (2006). Tanpa membina enjin yang sebenar, kesemua prestasi parameter enjin boleh dihitung daripada program simulasi ini dimana ianya dapat menjimatkan masa dan kos.

ACKNOWLEDGEMENT

In the name of Allah S.W.T the most gracious and merciful, first and foremost, after a year of struggle and hard work, with His will, this thesis is completed. Thanks to Allah for giving me the strength to complete this project and the strength to keep on living. I would like to convey heartiest appreciation to my supervisor, Mr. Mohd Fadzil Abdul Rahim for his consistency, advising and giving ideas throughout this thesis. I appreciate his consistent support from the first day I start doing the thesis. I also sincerely thank him for spent time to make the simulation running.

To my parent, Abdul Rahman Awang and Faridah Awang, thank you for all the support and sacrifice that both of you gave me. Your sacrifice is too great to be measured. It will be never forgotten. And also to all my siblings that me never forget.

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

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LIST OF SYMBOL

a	Crank Radius
A	Exposed Combustion Chamber Surface Area
AR	Reference Area
b	Cylinder Bore
CD	Discharge Coefficient
Cf	Frictional Loss Factor
Cheat	Charge Heating Factor
Cm	Mean Piston Speed
Div	Inlet Valve Diameter
Dv	Valve Diameter
f	Fraction of Heat Added
h	Convection Heat Transfer Coefficient
HV	Heating Value of Fuel
IVC	Inlet Valve Close Angle After BDC
IVO	Inlet Valve Open angle Before TDC
k	Specific Heat Ratio
l	Connecting Rod Length
Liv,max	Maximum Inlet Valve Lift
Lv	Valve Lift Function
mair,stoich	Theoretical Amount of Air Requirement
m	Mass Flow Rate
ma	Air Mass
N	Engine Speed
P	Pressure Inside Cylinder

Pe	Effective Power
Qin	Overall Heat Input
Q	Heat Addition
Qloss	Heat Transfer
R	Gas Constant
s	Stroke
Tg	Temperature of Cylinder Gas
Texh	Exhaust Gas Temperature
Tw	Cylinder Wall Temperature
pf	Friction Mean Effective Pressure
pme	Brake Mean Effective Pressure
T0	Stagnation Temperature
V	Cylinder Volume
Vd	Displacement Volume
$\Delta\theta$	Duration of Heat Addition
ε	Compression Ratio
ηv	Thermal Efficiency
θ	Crank Angle
$\theta 0$	Angle of Start of Heat Addition
ρa	Air Density

LIST OF ABBREVIATION

SYMBOL	SPECIFICATION
BDC	Bottom Dead Center
MVEM	Mean Value Engine Model
CCEM	Cylinder-by-cylinder Engine Model
EGR	Exhaust Gas Recirculation
TDC	Top Dead Center
WOT	Wide Open Throttle
HC	Hydrocarbon
rpm	Round per Minute
CO	Carbon Monoxide

LIST OF APPENDICES

CHAPTER 1

INTRODUCTION

1.0 PROJECT TITLE

Cylinder by Cylinder Engine Modeling of Single Cylinder 4 Stroke Engine For Control System Development.

1.1 PROJECT BACKGROUND

The cylinder by cylinder engine model (CCEM) is a mathematical model derived from basic physical principles such as conservation of mass and energy equations. CCEM can predict an engine's main external variables such as crankshaft speed and manifold pressure, and important internal variables, such as volumetric and thermal efficiencies.

The model consists of three main components: the throttle model, exhaust model, and crank-slider model. Adjusting the parameters allow the model to be used for new engines of interest. The importance of this study is to predict the engine performance parameters such as indicated work, brake power, and torque that provided with air fuel ratio data and detail geometrical specifications.

The method that has been chosen for this project is using simulation to predict the engine performance parameters. This is because this method can save time and reduce the spending of money to build engine test-rig for experimental work.

1.2 PROBLEM STATEMENT

In order to develop a new electronic control system for an engine, a control oriented model need to be proposed to assist the initial design and development work. The control oriented model shall be based on cylinder-to-cylinder approach which include the throttle body, exhaust and crank-slider model.

1.3 PROJECT OBJECTIVES

The objective of this study is to design and simulate cylinder by cylinder engine model for control oriented study based on single cylinder four stroke engines.

1.4 PROJECT SCOPE

Basically, this analysis based on:

1. The model is based on single cylinder four strokes SI Engine.
2. The model consisted of dynamics model of air-flow, fuel-flow and crankshaft dynamics.
3. The model is limited to physical/geometrical representation of the engine only. The fuel injection with it associated controller is not part of current scope.
4. The model used to predict engine performance parameters such as indicated power, brake power, torque and provided with air fuel ratio data and detail geometrical specifications.
5. The model used to simulate engine operating range from idle speed condition until achievable maximum speed and load.

1.5 FLOW CHART OF STUDY

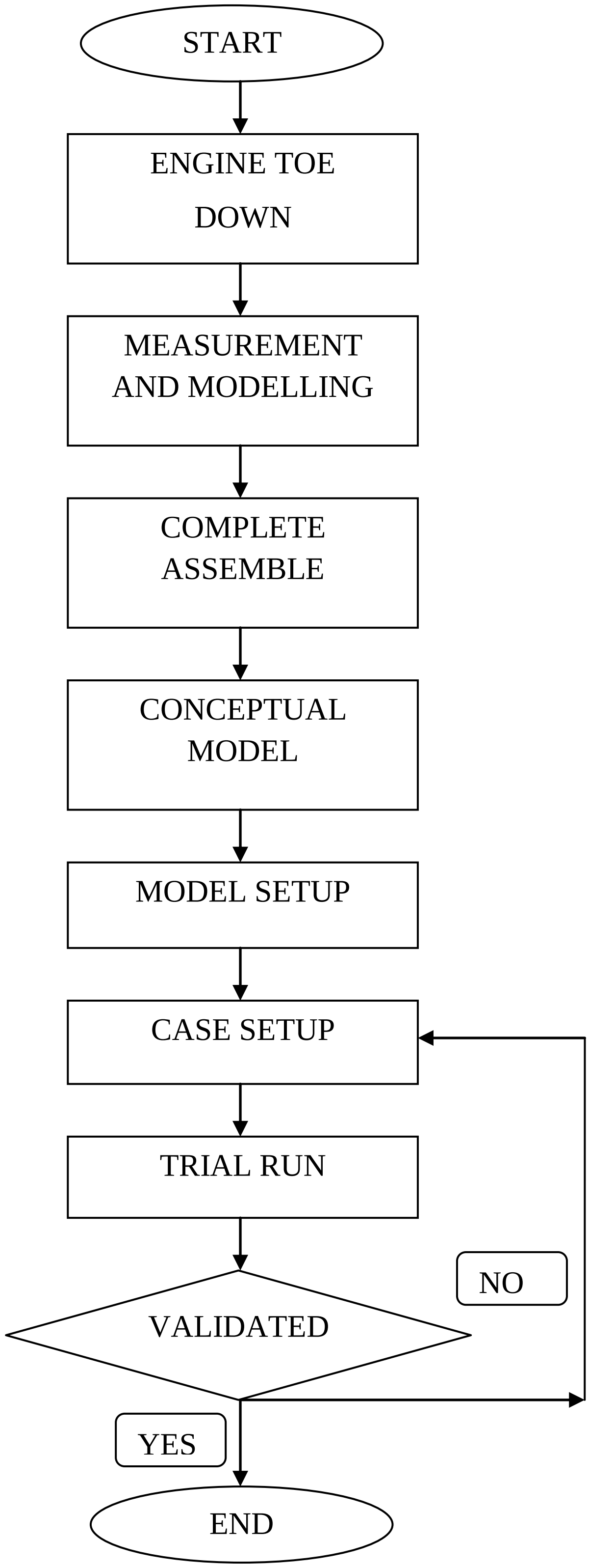


Figure 1.0: Flow chart of final year project 1

1.6 SUMMARY

The CCEM is a mathematical model derived from basic physical principles such as conservation of mass and energy equations. The importance of this study is to predict the engine performance parameters such as indicated work, brake power, and torque. The model is provided with air fuel ratio data and detail geometrical specifications to perform the simulation.

CHAPTER 2

LITERATURE REVIEW

2.0 INTRODUCTION

In this chapter, introductions about cylinder by cylinder engine modeling for control system development of a single cylinder-four-stroke engine is discussed. It is used mainly for design and simulating the system of engine without spending much time on experimental bench. From that the engine parameters from the output such as torque, brake power, and indicated work can be determined. A CCEM is describing each cylinder individually, could prove a useful and also needed to model cylinder individual phenomena such as misfire when developing systems for diagnostics. This simulation method allows the engine designed to test different parameters without building real parts or even real engines.

2.1 INTERNAL COMBUSTION ENGINE PRINCIPLE OPERATION

2.1.1 Spark Ignition Engine

The spark ignition engine is used in variety application such as cars, motorcycle, small engine generator and etc.

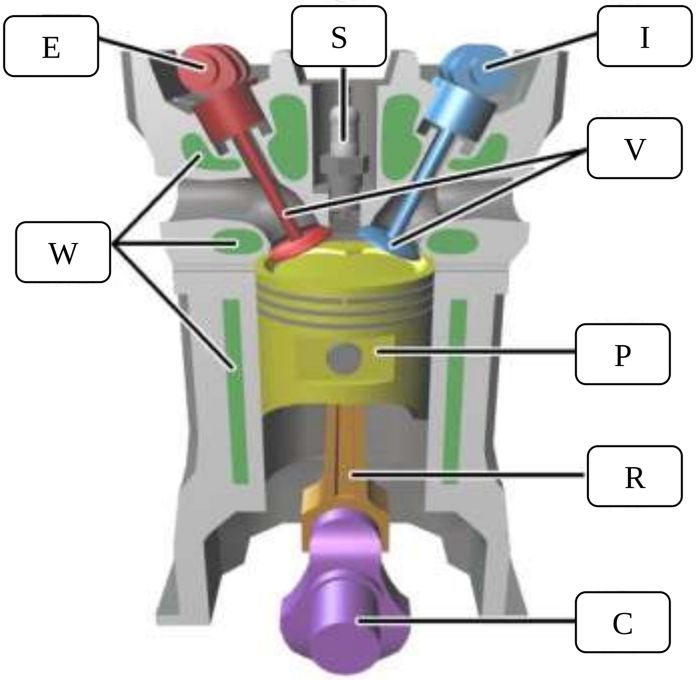


Figure 2.1: Schematic of spark ignition engine

Source: Stephen (2006)

Where:

E	-	Exhaust camshaft
I	-	Intake camshaft
S	-	Spark plug
V	-	Valves
P	-	Piston
R	-	Connecting rod
C	-	Crankshaft
W	-	Water jackets

2.1.2 Basics of Four Stroke Engine System

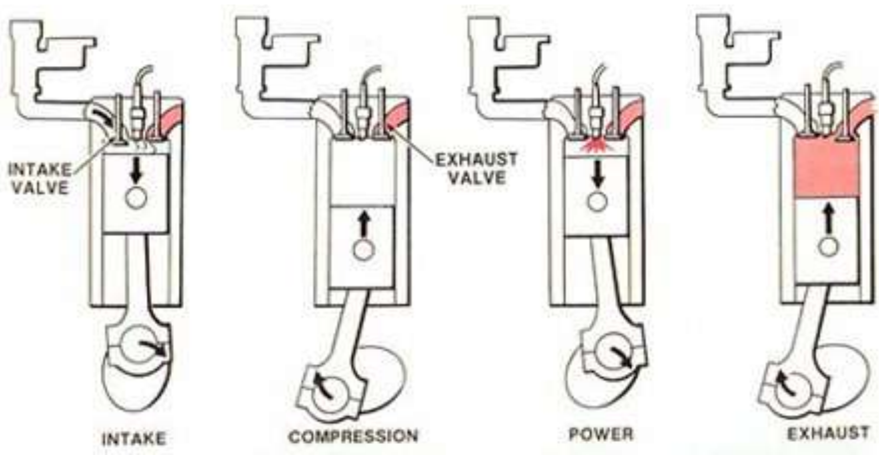


Figure 2.2: Four-Stroke Engine System

Source: Sitthiracha (2006)

I. **Intake Stroke** – The inlet valve is opened and the fuel/air mixture is drawn in as the piston travels down.

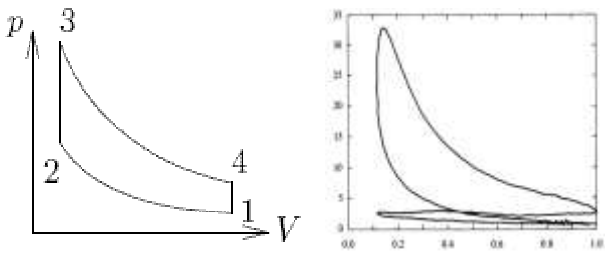
II. **Compression Stroke** – The inlet valve is closed and the piston travels back up the cylinder compressing the fuel/air mixture. And before the piston reaches the top of its compression stroke a spark plug emits a spark to combust the fuel/air mixture and the number of degrees before the top its stroke is the ignition advance which is called at 'top dead center' (TDC).

III. **Combustion Stroke** – The piston is now forced down by the pressure wave of the combustion at top dead center and it is called as the power stroke. The engine power is derived from this stroke.

IV. **Exhaust Stroke** – The exhaust valve is opened and the piston travels back up expelling the exhaust gases through the exhaust valve and at the top of this stroke the exhaust valve is closed. Then, this process was repeated.

The above is the cycle of operation of one cylinder of a 4-stroke engine which has two or more cylinders acting in concert with each other to produce the engine power. It is interesting to note that one complete engine cycle takes two revolutions but that individual valves and spark plugs only operate once in this time when their timing needs to be taken from a half engine speed signal, or we call it as the camshafts speed.

With models for each of these processes, the simulation of complete engine cycle can be built up and be analyzed to provide information on engine performances. These ideal models that describe characteristic of each process are proposed. However the calculation needs information from each state as shown in Fig.2.3a and Fig.2.3b.



(a) (b)

Figure 2.3: Pressure-volume diagram of Otto cycle:

(a) Ideal, (b) real

Source: Zeng et. al,(2004)

Overall engine work can be determined by integrating the area under the pressure-volume diagram and there are so many previous works concerned mainly prediction the pressure inside the combustion chamber (Zeng et. al, 2004). But the pressure and volume are influenced by engine geometries during variation of crank angle. Then the pressure and displacement volume are needed to convert as functions of crank angle. Then Kirkpatrick et. al (2005) proposed the method that can calculate the pressure and volume at any crank angle. And the combustion process can be described by the simple correlation, (Heywood, 1988).

The results from Zeng et. al (2004), indicated that heat transfer from inside the cylinder to engine cooling water had much influences on the pressure inside the cylinder and the heat transfer function is needed to take into account in the model.

Many researchers reported that the mass of mixture that flows into the cylinder during intake stroke is a very importance parameter (Andersson, 2002), by Heywood, (1988), because it affects amount of fuel which mixes with the air. By combining the ideal gas law and volumetric efficiency, this mass can be determined and it is very difficult to evaluate because they are affected by many factors, and for examples the manifold geometries and valve timing, (Heywood, 1988). So Kuo et. al (1996) assumed that the pressure inside manifold and inside the cylinder is the same value, and the effect of volumetric efficiency can be neglected. But Kuo (1996), used corrective equation from real experiment to compensate the errors. While Zeng and Assanis (2004), took the effect of volumetric efficiency into account.

However the data were obtained from the real experiment and stored in a 3-dimension table by relation between engine speed and intake manifold pressure. The combining of those methods that are mentioned above can predict the engine performances precisely if some testing data are known, mainly the volumetric efficiency.

2.2 CONTROL ORIENTED MODELING

There are numerous ways of describing reality through a model (Ramstedt and Silverlind, 2001). Some are more complex than others and the different approaches may differ in both structure and accuracy.

From previous study, a model of a four-cylinder spark ignition engine and capability to model an internal combustion engine from the throttle to the crankshaft output by Crossley and Cook (1991). It is used well-defined physical principles supplemented, where appropriate, with empirical relationships that describe the system's dynamic behavior without introducing unnecessary complexity, and it have some relation with this study.

2.2.1 Physical sub-model

This example describes the concepts and details surrounding the creation of engine models with emphasis on important control oriented modeling techniques. The basic model uses the enhanced capabilities of control oriented modeling to capture time-based events with high accuracy. During this simulation, a triggered subsystem models the transfer of the air-fuel mixture from the intake manifold to the cylinders via discrete valve events. The places takes the concurrently with the continuous-time processes of intake flow, torque generation and acceleration.

The second model adds an additional triggered subsystem that provides closed-loop engine speed control via a throttle actuator. This model can be used as standalone engine simulations and also can be used within a larger system model, such as an integrated vehicle and power train simulation in the development of a traction control system. This model is based on published results by Sitthiracha (2006). It describes the simulation of a four-cylinder spark ignition internal combustion engine. They work also shows how a simulation based on this model was validated against dynamometer test data. The following sections are analyzing the key elements of the engine model that were identified by them:

I. Throttle
II. Intake manifold
III. Mass flow rate
IV. Compression stroke
V. Torque generation and acceleration

I. Throttle

The first element of the model is the throttle body. The control input is the angle of the throttle plate. The rate at which the model introduces air into the intake manifold can be expressed as the product of two functions which is an empirical function of the throttle plate angle only and as the function of the atmospheric and manifold pressures. And in cases of lower manifold pressure (high pressure), the flow rate that is through the throttle body is sonic and is only as a function of the throttle angle. This model accounts for this low pressure behavior with a switching condition in the compressibility equations shown in Equation 1.

II. Intake Manifold

The simulation models the intake manifold as a differential equation for the manifold pressure. The difference in the incoming and outgoing mass flow rates represents the net rate of change of air mass with respect to time. This quantity, according to the ideal gas law, is proportional to the time derivative of the manifold pressure. Note that, unlike the model of Crossley and Cook (1991), although this can easily be added, this model doesn't incorporate exhaust gas recirculation (EGR).

III. Intake Mass Flow Rate

The mass flow rate is a function of the manifold pressure and the engine speed in order to determine the total air charge pumped into the cylinders, simulation integrates the mass flow rate from the intake manifold and samples it at the end of each intake stroke process. This is determines the total air mass that is present in each cylinder after the intake stroke and before compression.

IV. Compression Stroke

In an inline four-cylinder four-stroke engine, the 180° of crankshaft revolution separate the ignition of each successive cylinder and the results in each cylinder firing on every other crank revolution. From this model, the intake, the compression, the combustion, and the exhaust strokes occur simultaneously, which means that it is occur at any given time, one cylinder is in each phase. To account for compression, the combustion of each intake charge is delayed by 180° of crank rotation from the end of the intake stroke.

V. Torque Generation and Acceleration

The final element of the simulation describes the torque developed by the engine combustions. The empirical relationship is dependent to the mass of the air charge, the air-fuel mixture ratio, the spark advance, and the engine speed is used for the torque computation.

2.2.2 Mean Value Engine Model (MVEM)

The mean value engine model (MVEM) is a mathematical model derived from basic physical principles such as conservation of mass and energy equations. The MVEM is based on some simplified assumptions and time averaged combustion of the engine parameters, it is also the models of the engine with a reasonable approximation and gives a satisfactory amount of information about the physics of the fluid energy passing through an engine system.

MVEM can predict an engine's main external variables such as crankshaft speed and manifold pressure, then the important of internal variables of the engine such as volumetric and thermal efficiencies. Normally, the differential equations used in MVEM will predict the fuel film flow, manifold pressure and crankshaft speed. The MVEM is widely used for engine control development because of its simplicity and short simulation time.

Mean value engine models (MVEMs) also attempts to capture dynamics in a time-scale spanning over several combustion cycles, or a few tenths of them. The faster events are not of interest other than their effects on a larger time-scale and the pulsating flow through the inlet port is modeled by its average value over a cycle, for example. The speed and torque output of the engine, dynamics of the turbocharger and the pressure build-up in the inlet and exhaust manifolds are the aspects of most interest in MVEMs.

As a contrast, the in-cycle engine model is designed to capture details in the combustion-cycle like the pressure inside the cylinders over the strokes and effects of different valve-timings. Modeling principles and approximations will vary between models depending on their intended use. Some of the common approximations and some of the choices made in this project are discussed in this chapter, (Ken, 1994-1997).

2.2.3 Cylinder to Cylinder engine model

Cylinder-by-Cylinder Engine Model (CCEM), describing each cylinder individually and CCEM is also needed to model cylinder individual phenomenon such as misfire when developing systems for diagnostics. Unlike the mean method, the cylinder-by-cylinder method is better because this method derives from engine geometries, which is very useful for improving and optimizing the engine in the future.

2.2.4 Comparison between (MVEM) and (CCEM)

When considering the cylinder, two main approaches can be found. The most commonly use is mean value engine method. The mean value method defines number of cylinders as one which occupies whole displacement volume. The fluctuating flow through the inlet port is modeled by average value over a cycle. The dynamics of speed, the engine torque, the pressure build-up in the inlet and the exhaust manifolds are the aspects of most interest in this approach.

Another method to the mean method is the cylinder-by-cylinder engine approach. Unlike the mean method, it describes each cylinder individually and generates for example a torque signal with each individual combustion pulse present.

Normally the mean model is sufficient enough for use in processes such as control system design in simulation, but from the aspect of detailness, the cylinder-by-cylinder method is better because this method derives from engine geometries, which is very useful for improving and optimizing the engine in the future.

2.3 CYLINDER TO CYLINDER MODEL

There are a number of approaches available when deciding on the basis of a model. Physical equation theoretically describing the system is the most common method since it creates a general model working for many operating areas. Its drawback is that reality might be difficult to describe correctly in theory. Another common approach bases on the model entirely on measured data which is stored as a table of two or more dimensions in a so called black box depending on input signals data. This approach often provides an accurate result since it is based directly on empirical formulation that is only defined for a limited region. A combination of both approaches is commonly used. The main basis of the model rests on physical equations and empirical equations are used to model the certain complexities.

The chosen model is bases on the pressure inside the cylinder prediction. There are two main approaches to be achieved, which is mean value and cylinder-by-cylinder models. Since the objective of this project intends to develop the model which can describe effects of each parameter on the engine performance, the cylinder-by-cylinder method is used in order to achieve this goal. Another consideration for model selection is limited by physical properties. Since there are no perfect equations which can describe phenomena in the engine, both physical and empirical formulae are used in the model.

2.3.1 Intake dynamics

The difference in the incoming and outgoing mass flow rates represents the net rate of change of air mass with related to time. And according to the ideal gas law, it is proportional to the time derivative of the manifold pressure.

$$\dot{P}_m = \frac{RT}{V_m}(\dot{m}_{ai} - \dot{m}_{ao}) \tag{2.1}$$

Where:

$\dot{P}_m$	:	Rate of change of manifold pressure (bar/s)
R	:	specific gas constant
T	:	temperature (K)
V_m	:	Manifold volume (m^3)
$\dot{m}_{ao}$	:	Mass flow rate of air out of the manifold (g/s)

2.3.2 Torque dynamics

To describe the torque developed by the engine, an empirical relationship dependent upon the mass of the air charge, the air/fuel mixture ratio, the spark advance, and the engine speed is used for the torque computation.

$$Torque_{eng} = -181.3 + 379.36\,(\dot{m}) + 21.91\left(\frac{A}{F}\right) - 0.85\left(\frac{A}{F}\right)^2$$
$$+0.26(\sigma) - 0.0028(\sigma^3) + 0.027(N) - 0.000107(N^2)$$
$$+\,0.00048(N)(\sigma) + 2.55(\sigma)(m_a) - 0.05(\sigma^2)(m_a) \qquad (2.2)$$

Where:

m_a	:	Mass of air in cylinder for combustion (g)
$\left(\frac{A}{F}\right)$	:	air to fuel ratio
σ	:	spark advance (degree before top dead center)
$Torque_{eng}$	:	torque produced by the engine (Nm)

The engine angular acceleration was calculated using Equation 2.3:

$$J\dot{N} = Torque_{eng} - Torque_{load} \qquad (2.3)$$

Where:

J	:	Engine rotational moment of inertia ($kg.m^2$)
$\dot{N}$	:	Engine angular acceleration (rad/s^2)

2.3.3 Exhaust dynamics

During the normal engine operation, when the piston compressed the air/fuel mixture is ignited during the power stroke, the resultant of combustion exerts pressure on the top of the piston. Since this pressure ultimately results in the movement of the piston itself, it can be consider it as being correlated to the power output of the engine. Mean Effective Pressure can be calculated using the following formula:

$$MEP = 150.8 \text{ x } (Torque / CID) \tag{2.4}$$

Where:

CID : the cubic-inch displacement of the engine

BMEP is the average (mean) pressure which, if imposed on the pistons uniformly from the top to the bottom of each power stroke, would produce the measured (brake) power output.

$$BMEP\ (psi) = 150.8 \text{ x } TORQUE\ (Nm) / DISPLACEMENT\ (mm) \tag{2.5}$$

2.4 SUMMARY

Details of mean value and cylinder by cylinder models are discussed. The results can be produced by running the simulation only and will decrease time and cost. The cylinder by cylinder method has been chosen because this method derives from engine geometries, which provide detail description of the engine.

CHAPTER 3

METHODOLOGY

3.0 INTRODUCTION

This section describes the engine model developed in SIMULINK. The SIMULINK graphical block diagram language allows models to be written in a modular format. The SIMULINK engine model is simulated within a larger system model that also includes all the engine system such as throttle model, crank-slider model and exhaust model. So, all the data from the engine has been embedded in into the simulation to run the system, and engine operating parameters are adjusted for parameters study.

3.1 FLOW CHART FOR FINAL YEAR PROJECT:

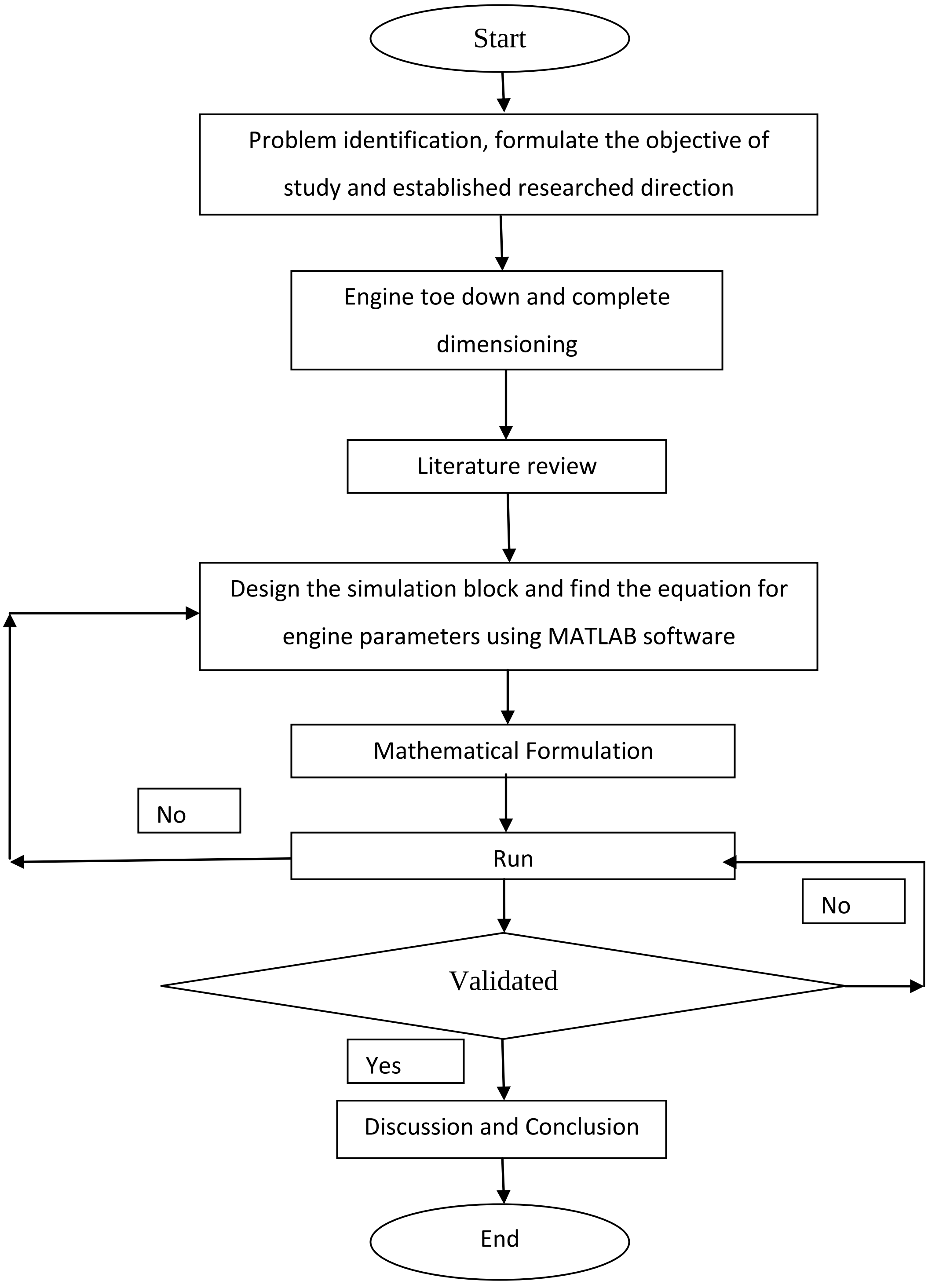


Figure 3.1: Flow chart of overall Final Year Project

3.2 ENGINE MODELING USING SIMULINK

This section will describe the details of the engine and control system modeling. The basics of building and running a Simulink model is the parameters dialog box, which provides extensive options for selecting and configuring the differential equation solver used to perform model simulation.

Therefore, the equations leading the model are made of basic differential equation and generally, mathematical engine simulations are practice in two ways, which are fluid dynamic based and thermodynamic based models. Thermodynamic cycle models are based on the thermodynamic analysis of the content of cylinder during the engine cycle. In these models, the 1st law of thermodynamics is applied to open system of air fuel and residual gas mixture into the manifolds and cylinders and it is zero dimensioned.

The purpose of this chapter is to explain the engine modeling by using Simulation, drawing and editing the model and run the model.

3.2.1 Main Engine Model

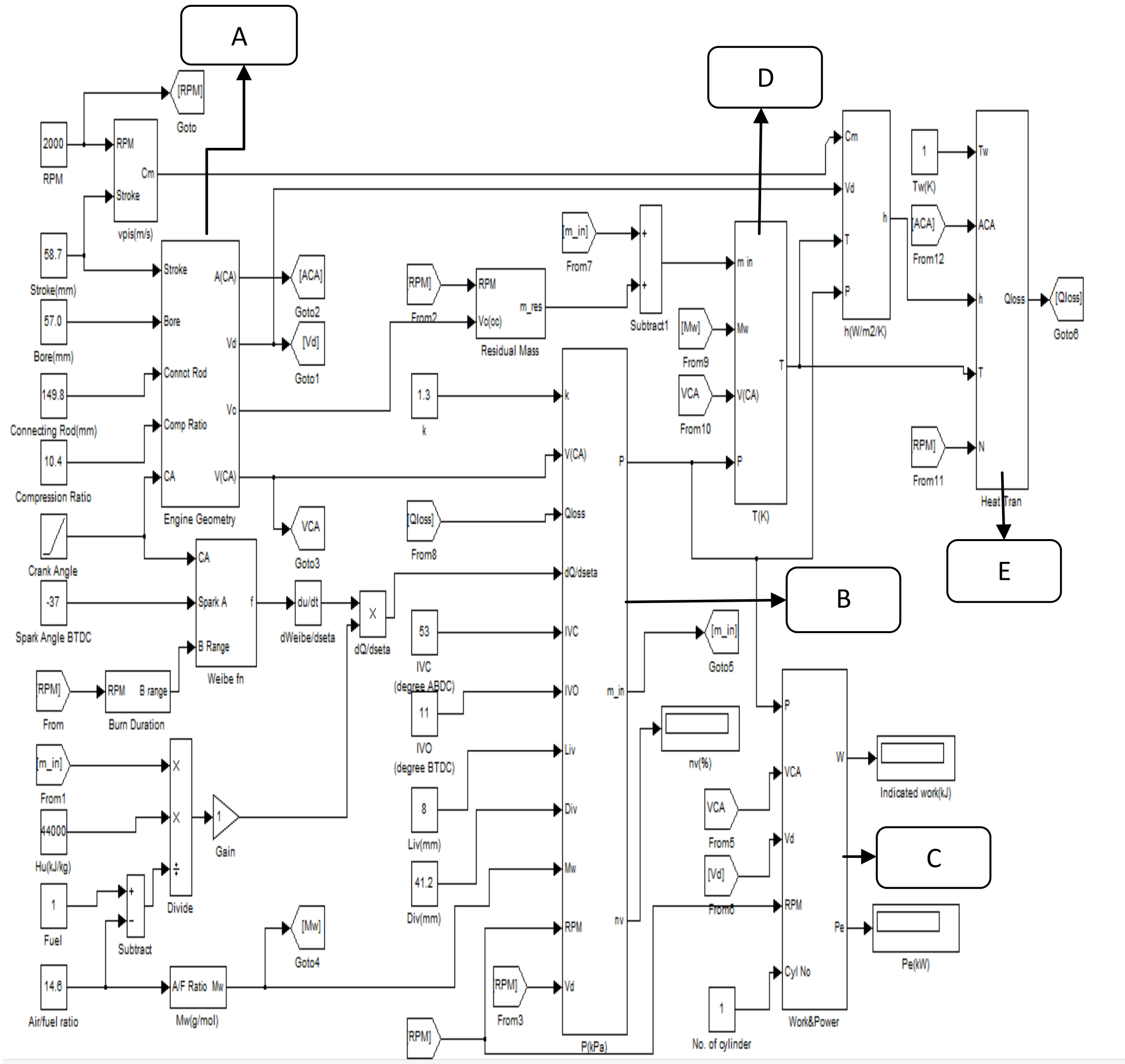


Figure 3.2: Main model block diagram

A represents the engine geometries block details that contains all the input data. B is representing the power block details that are containing the output of thermal efficiency, $\eta_{v.}$ C is representing the work and power block detail, in order to get the output of indicated work and power. Then, D is representing the Temperature block details that are as input temperature data of engine. And, E is representing the heat block details whereas this block will calculate the heat loss from this engine modeling.

3.2.2 Engine Model Block Detail

$$Cm = (0.166)(Stroke)(RPM) \qquad (3.1)$$

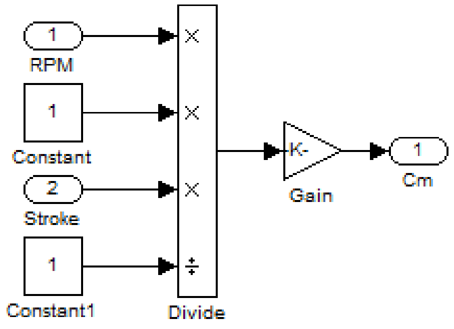


Figure 3.3: Cm block details

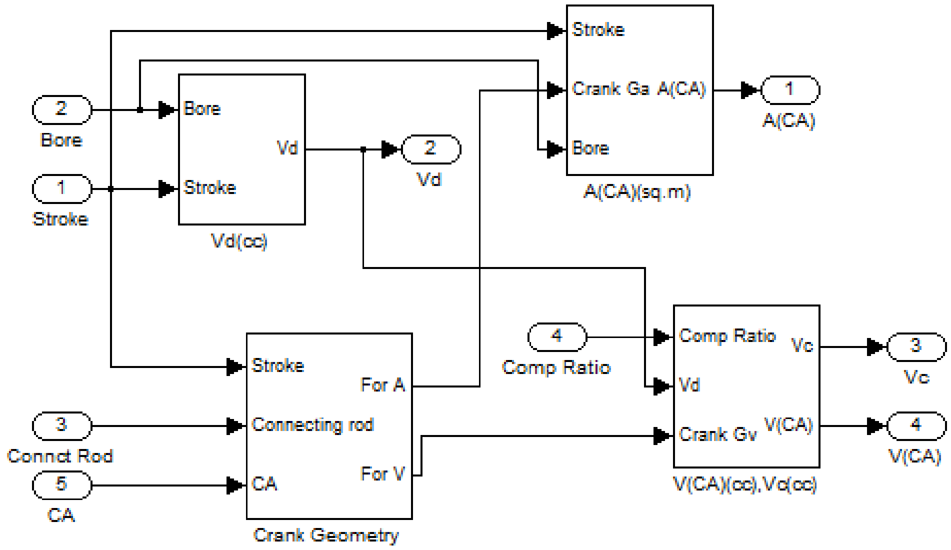


Figure 3.4: Engine geometry block details

$$A(CA) = \frac{\pi}{2}b^2 + \pi b\frac{S}{2}\left(R + 1 - \cos\theta + (R^2 - \sin\theta)^{1/2}\right) \tag{3.2}$$

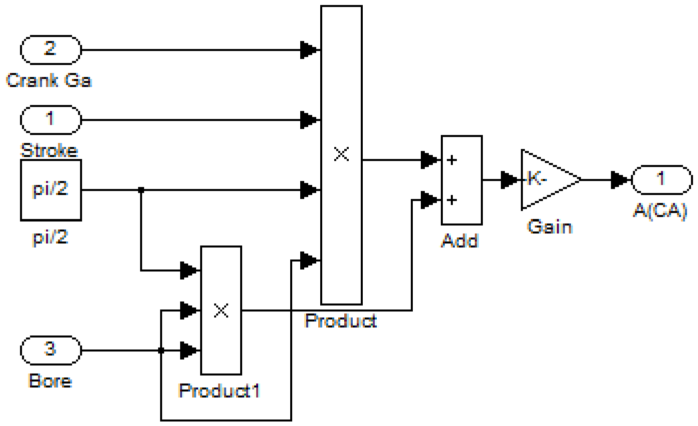


Figure 3.5: Engine geometry/A(CA) block details

$$V(CA) = \frac{V}{V_d} = \frac{1}{r-1} + \frac{1}{2}\left(1 + R - \cos\theta - (R^2 - \sin^2\theta)^{1/2}\right) \tag{3.3}$$

$$V_c = \frac{V_d}{r-1} \tag{3.4}$$

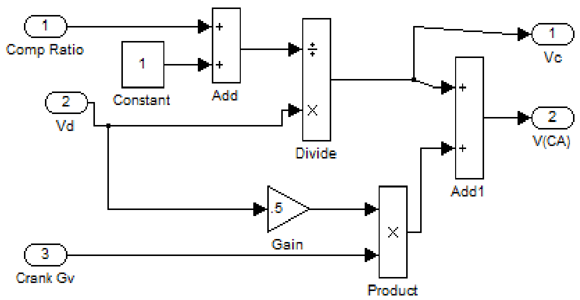


Figure 3.6: Engine geometry/V(CA), Vc block details

$$V_d = \left(\frac{\pi}{2}\right) b^2 s \tag{3.5}$$

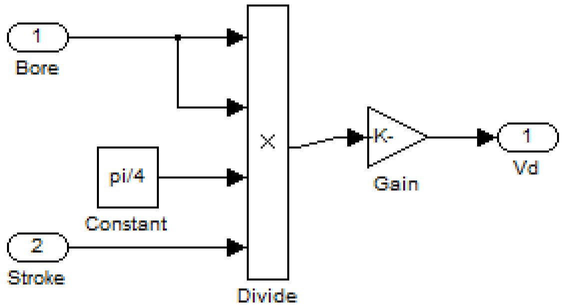


Figure 3.7: Engine geometry/Vd block details

$$ForV = \frac{V_d}{r-1} + \frac{V_d}{2}\left(1 + R - \cos\theta - (R^2 - \sin^2\theta)^{1/2}\right) \tag{3.6}$$

$$ForA = \frac{\pi}{2} b^2 + \pi b \frac{s}{2}\left(R + 1 - \cos\theta + (R^2 - \sin\theta)^{1/2}\right) \tag{3.7}$$

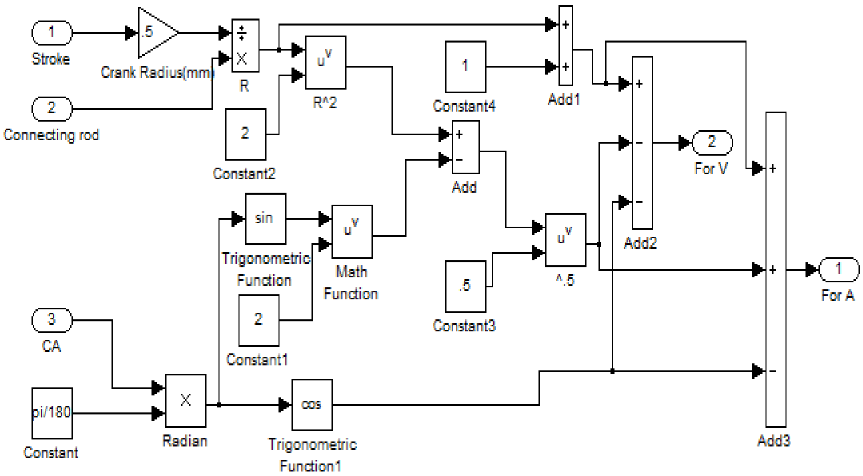


Figure 3.8: Engine geometry/Crank Geometry block details

$$T_{exh} = 3.3955\left(\frac{N}{1000}\right)^3 - 51.9\left(\frac{N}{1000}\right)^2 + 279.49\left(\frac{N}{1000}\right) + 676.21 \qquad (3.8)$$

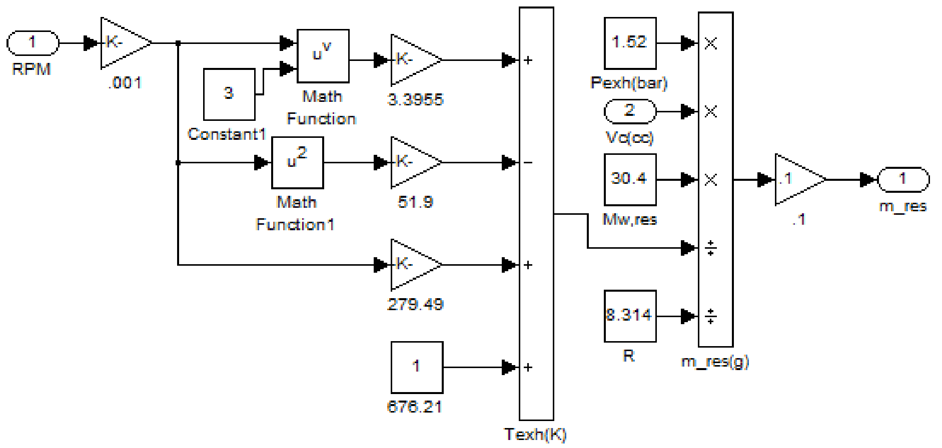


Figure 3.9: Residual mass block details

$$T = \left(\frac{PV}{mR}\right), V = V(CA) \qquad (3.9)$$

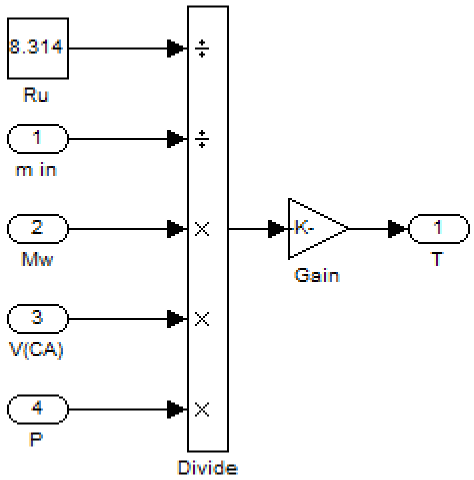


Figure 3.10: T block details

$$Q_{loss} = hA(T - T_w) \tag{3.10}$$

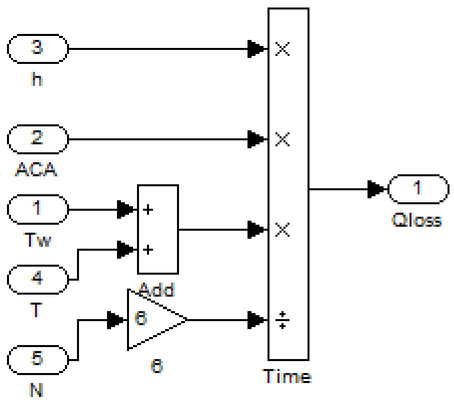


Figure 3.11: Heat Transfer block details

$$\Delta\theta = -1.6189\left(\frac{N}{1000}\right)^2 + 19.886\left(\frac{N}{1000}\right) + 39.951 \tag{3.11}$$

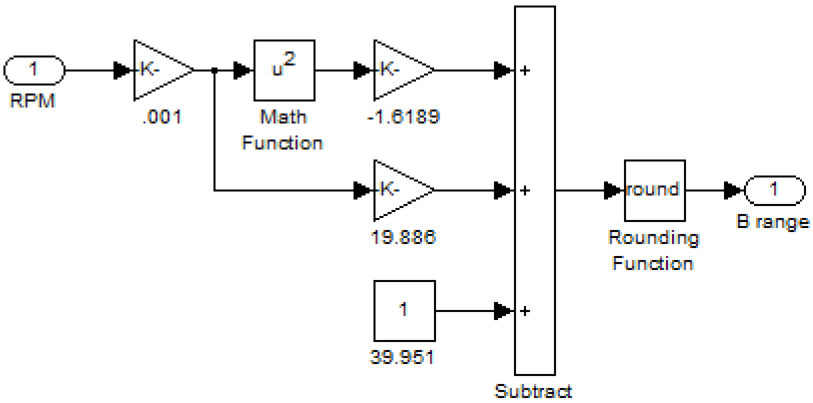


Figure 3.12: Burn duration block details

Hohenberg correlation, $h = 130V^{-0.06}P^{0.8}T_g^{-0.4}(C_m + 1.4)^{0.8}$ (3.12)

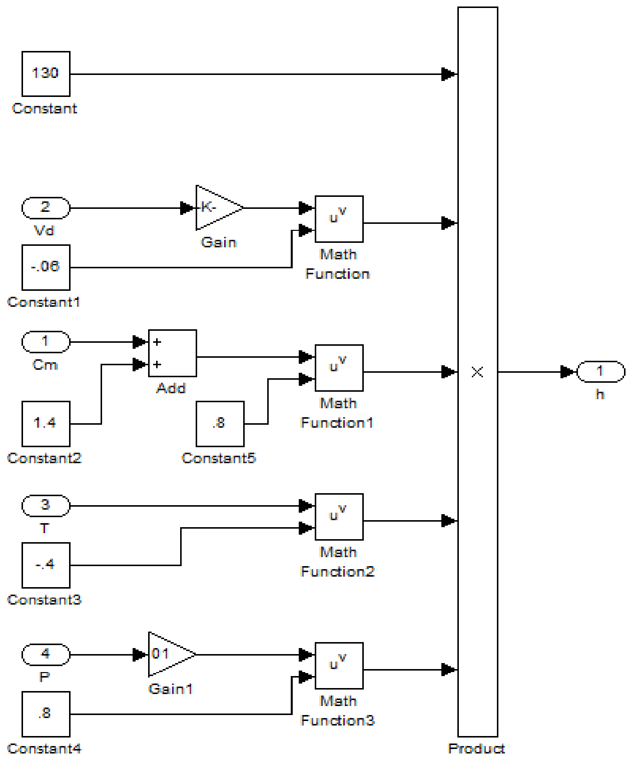


Figure 3.13: h block details

$$f(\theta) = 1 - \exp\left[-5(\frac{\theta - \theta_0}{\Delta\theta})^3\right]$$ (3.13)

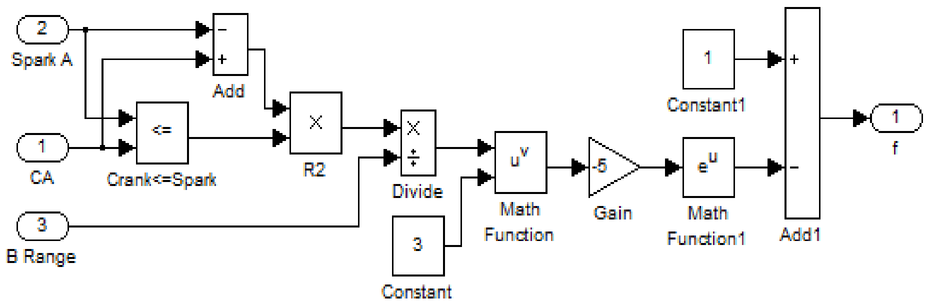


Figure 3.14: Weibe fn block details

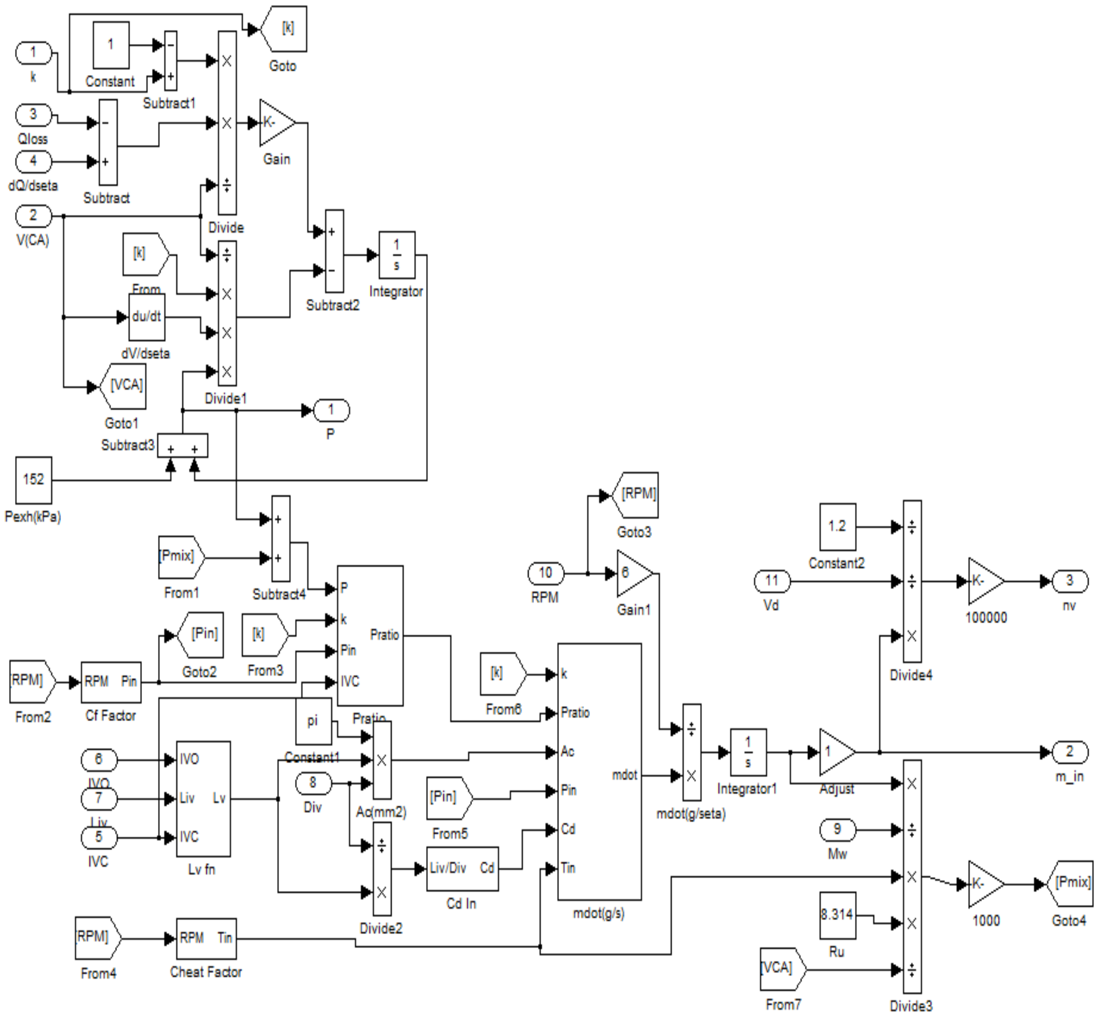


Figure 3.15: P block

$$CR = \frac{V_1}{V_2}, PR = \frac{P_1}{P_2}$$
$$where: \qquad (3.14)$$
$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2} \Rightarrow \frac{V_1}{V_2} = \frac{T_1}{T_2}\frac{P_2}{P_1} \Leftrightarrow CR = \frac{T_1}{T_2}PR$$

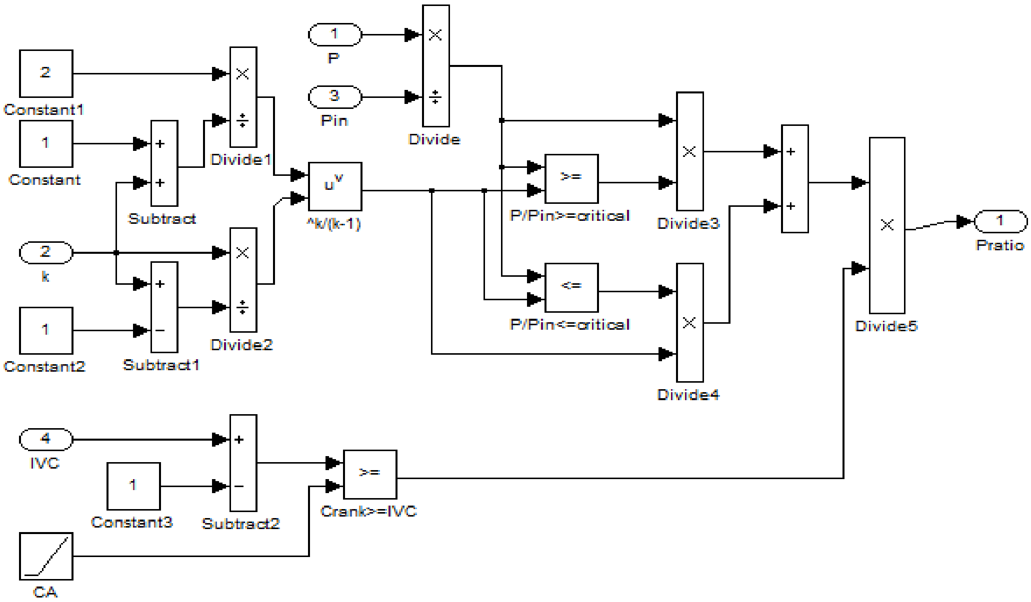


FIGURE 3.16: P/Pratio block details

$$L_v = \frac{L_{iv,\max}(1+\cos\theta)}{2} \qquad (3.15)$$
$$where, \theta = \frac{\pi(IVO - IVC + 2\theta + 540)}{\text{IVO+IVC+180}}$$

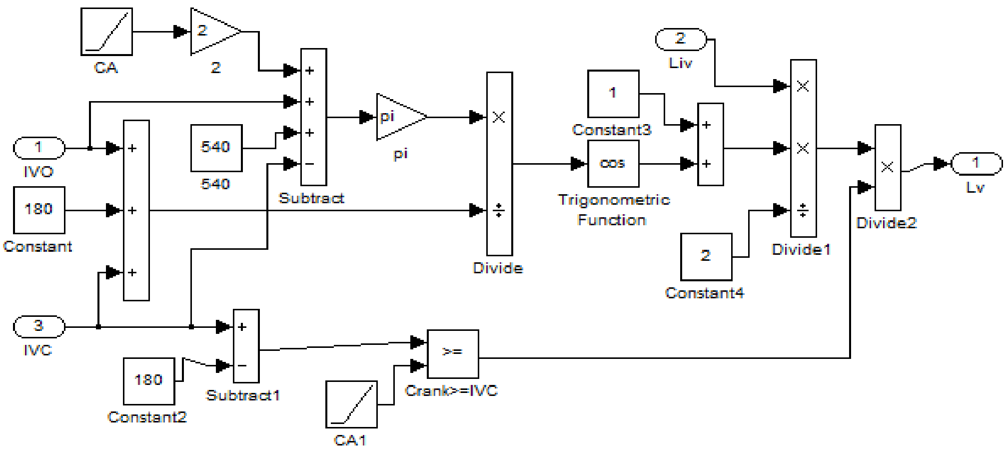


Figure 3.17: P/Lv block details

$$C_d = 190.47\left(\frac{L_v}{D_{iv}}\right)^4 - 143.13\left(\frac{L_v}{D_{iv}}\right)^3 + 31.248\left(\frac{L_v}{D_{iv}}\right)^2 - 2.5999\left(\frac{L_v}{D_{iv}}\right) + 0.6913 \qquad (3.16)$$

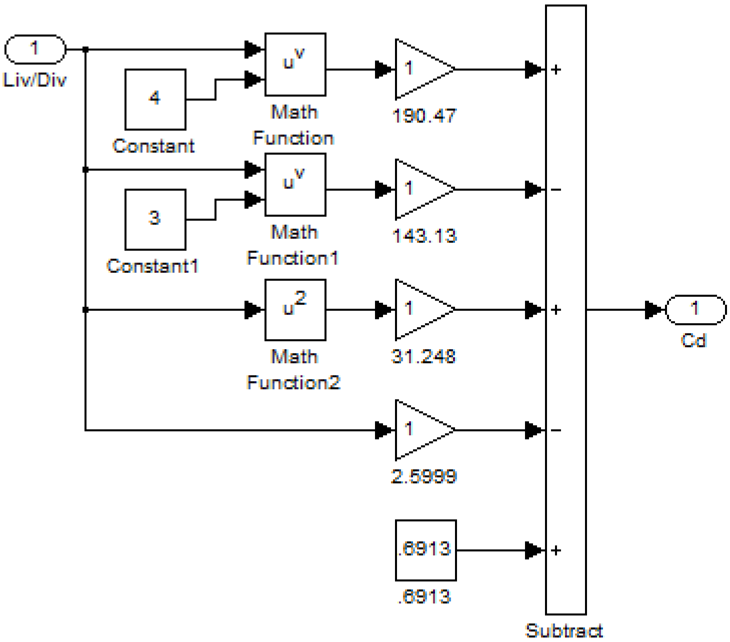


Figure 3.18: P/Cd block details

$$m_{dot} = \frac{C_d A_R P_o}{(\mathrm{R}T_o)^{0.5}} \left(\frac{P_T}{P_o}\right)^{\frac{1}{k}} \left\{ \frac{2k}{k-1} \left[1 - \left(\frac{P_T}{P_o}\right)^{\frac{k}{k-1}} \right] \right\}^{0.5} \tag{3.17}$$

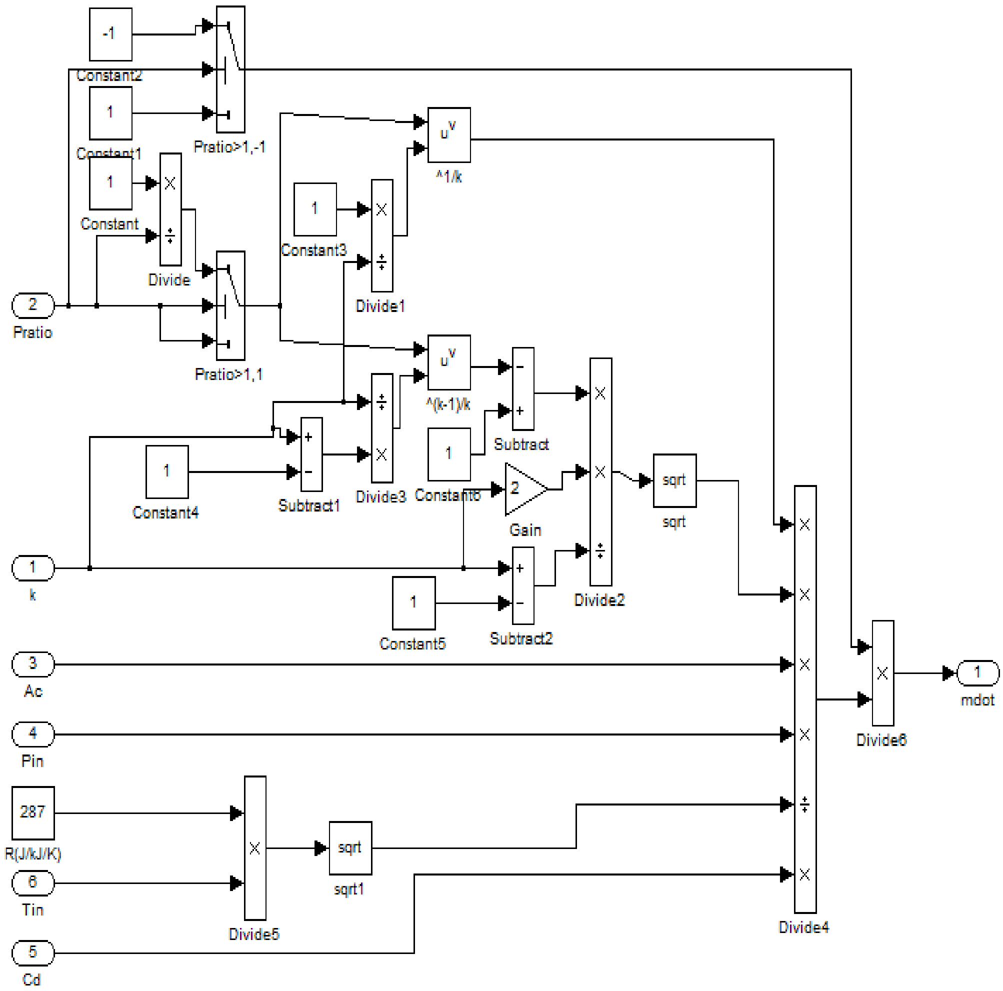


Figure 3.19: P/mdot block details

$$C_{heat} = -0.043624\left(\frac{N}{1000}\right) + 1.2953 \tag{3.18}$$

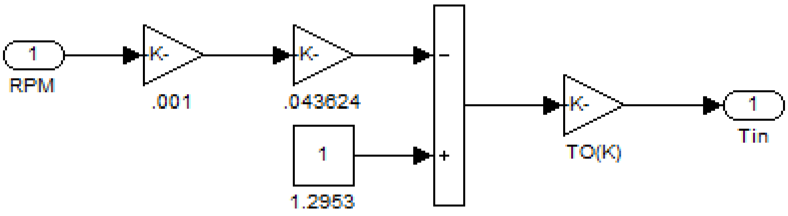


Figure 3.20: P/Cheat factor block details

$$C_f = -0.019738\left(\frac{N}{1000}\right) + 0.986923 \tag{3.19}$$

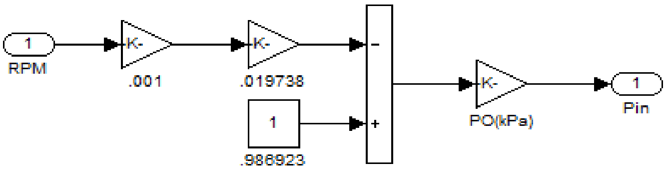


Figure 3.21: P/Cf factor block details

Molecular weight equation for gasoline/octane, C8H18:

$$= \left[C(8) + H(18)\right],\ \text{C=12, H=1} \tag{3.20}$$

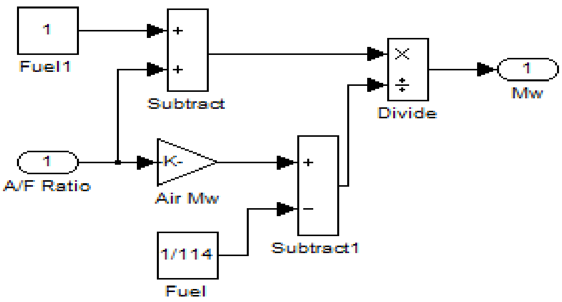


Figure 3.22: Mw block details

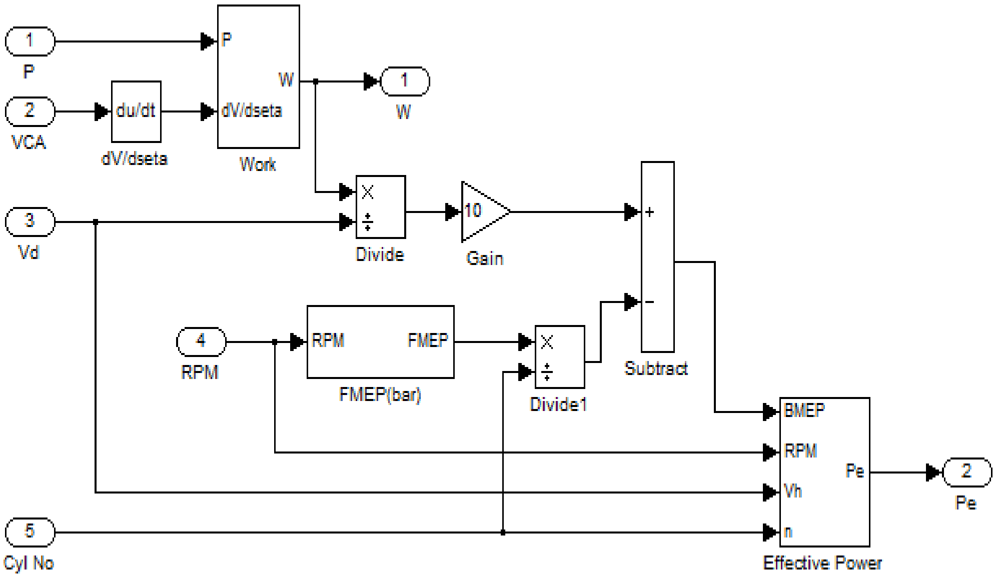


Figure 3.23: Work & Power block details

$$W = \int_{t_o}^{t} PdV \qquad 3.21$$

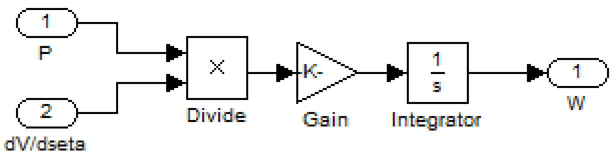


Figure 3.24: Work & Power/Work block details

$$\text{Frictional loss, } P_f = 0.05\left(\frac{N}{1000}\right)^2 + 0.15\left(\frac{N}{1000}\right) + 0.97 \qquad (3.22)$$

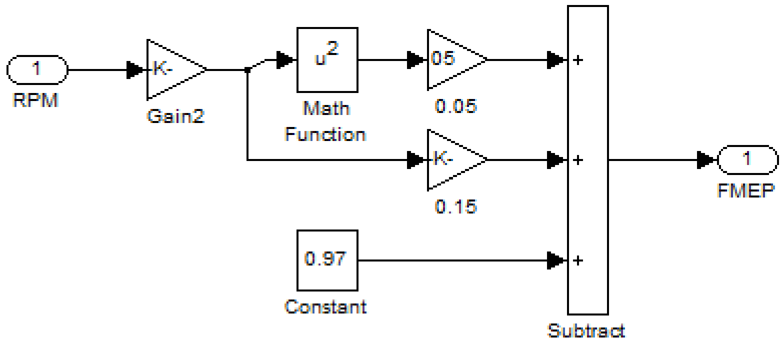


Figure 3.25: Work & Power/FMEP block details

$$\begin{aligned} P_e &= 0.5N.P_{me}.V_d \\ P_{me} &= P_{mi} - P_f \\ P_{mi} &= W = \int_{t_o}^{t} PdV \end{aligned} \qquad (3.23)$$

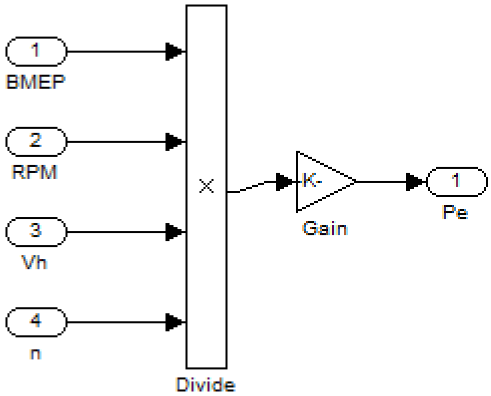


Figure 3.26: Work & Power/Effective power block details

3.3 THROTTLE MODEL

The valve is usually the most important flow restriction in the intake system of four-stroke cycle engines. This thesis only considers the intake valve in order to determine η*v*. The mass flow rate through a poppet valve is usually described by the equation for compressible flow through a flow restriction, equation 3.24. This equation is derived from a one-dimensional isentropic flow analysis, and real gas flow effects are included by means of an experimentally determined discharge coefficient (*CD*).

$$\dot{m} = \frac{C_d A_R P_o}{(\mathrm{R}T_o)^{0.5}} \left(\frac{P_T}{P_o}\right)^{\frac{1}{k}} \left\{ \frac{2k}{k-1} \left[1 - \left(\frac{P_T}{P_o}\right)^{\frac{k}{k-1}} \right] \right\}^{0.5} \tag{3.24}$$

Where:

$$\left(\frac{P_T}{P_o}\right) = \left(\frac{2k}{k-1}\right)^{k/k-1}$$

p0	:	ambient pressure, and
P_T	:	the cylinder pressure.
T_0	:	ambient temperature.

For A_R, the most convenient reference area in practice is the so called valve curtain area since it varies linearly with valve lift and is simple to determine.

$$A_R = \Pi D_V L_v \tag{3.25}$$

Eq.3.24 should be converted into a function of crank angle also by dividing with *6N* same as Eq.3.25.

$$\frac{dm}{d\theta} = \frac{C_d A_R P_o}{6\mathrm{N}(\mathrm{R}T_o)^{0.5}} \left(\frac{P_T}{P_o}\right)^{\frac{1}{k}} \left\{ \frac{2k}{k-1} \left[1 - \left(\frac{P_T}{P_o}\right)^{\frac{k}{k-1}} \right] \right\}^{0.5} \tag{3.26}$$

3.4 CRANK-SLIDER MODEL

The volume of the piston cylinder can be determined as a function of crank angle from the compression ratio (ε), the stroke (*s*), bore (*b*) and connecting rod length (*a*). The geometric parameters of the piston cylinder can be described by the crank slider model which is represented in Fig.3.27.

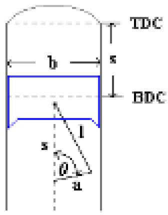


Figure 3.27: Piston cylinder and geometries

The equations of volume and area that relate to crank angle are described as following equation:

$$V(\theta)=\frac{V_d}{\varepsilon-1}+\frac{V_d}{2}\left[\frac{1}{a}+1-\cos\theta-(\left(\frac{1}{a}\right)^2-\sin^2\theta)^{1/2}\right] \quad (3.27)$$

$$A(\theta)=\frac{\pi}{2}b^2+\pi b\frac{s}{2}\left(R+1-\cos\theta+(R^2-\sin\theta)^{1/2}\right) \quad (3.28)$$

3.5 EXHAUST MODEL

The valve is usually the most important flow restriction in the exhaust system of four-stroke cycle engines. The mass flow rate through a poppet valve is usually described by the equation for compressible flow through a flow restriction, Eq. below. This equation is derived from a one-dimensional isentropic flow analysis, and real gas flow effects are included by means of an experimentally determined discharge coefficient (*CD*).

$$\dot{m} = \frac{C_d A_R P_o}{(\mathrm{R}T_o)^{0.5}} \left(\frac{P_T}{P_o}\right)^{\frac{1}{k}} \left\{ \frac{2k}{k-1} \left[1 - \left(\frac{P_T}{P_o}\right)^{\frac{k}{k-1}} \right] \right\}^{0.5} \tag{3.24}$$

Where:

$$\left(\frac{P_T}{P_o}\right) = \left(\frac{2k}{k-1}\right)^{k/k-1}$$

p0	:	ambient pressure, and
P_T	:	the cylinder pressure.
T_0	:	ambient temperature.

$$A_R = \Pi D_V L_v \tag{3.25}$$

Eq.3.24 should be converted into a function of crank angle also by dividing with *6N* same as Eq.3.25.

$$\frac{dm}{d\theta} = \frac{C_d A_R P_o}{6\mathrm{N}(\mathrm{R}T_o)^{0.5}} \left(\frac{P_T}{P_o}\right)^{\frac{1}{k}} \left\{ \frac{2k}{k-1} \left[1 - \left(\frac{P_T}{P_o}\right)^{\frac{k}{k-1}} \right] \right\}^{0.5} \tag{3.26}$$

This equation is similar to intake system, but the values of C_D and A_R for intake and exhaust system are totally different.

3.6 PARAMETER OF STUDY

The parameters of this study are to predict the engine performance such as indicated work, brake power, and torque. The model that be used in this project is FZ150i engine, and it will be used to simulate engine operating range from idle speed condition until achievable the maximum speed and load.

These models consist of dynamics model of air-flow, fuel-flow, and crankshaft dynamics which is receipt to the geometrical representation of this model only.

3.7 MODEL BUILDING STEP

1. Engine modeling using solid work.
2. Created blog diagram of engine by selecting icon in SIMULINK.
3. Determine the mathematical equation.
4. Control system complete design.
5. Trial run.

3.8 BASELINE ENGINE SPECIFICATION

Table 3.1: Yamaha FZ150i engine details

TYPES	SPECIFICATIONS
Engine type	Liquid-cooled 4-stroke, SOHC, 4-valve
Cylinders	Single cylinder
Displacement	149.6 cm3
Bore × Stroke	57.0 × 58.7 mm
Compression ratio	10.4:1
Max. output	11.1kW/8500r/min
Max. torque	13.1Nm/7600r/min
Transmission Oil volume	1.15 L
Fuel tank volume	12.0 L
Carburetion	3C1(EFI)
Ignition	C.D.I

3.8.1 Case setup

Table 3.2: Simulation case setups

CASE	Engine Speed (rpm)	Air/fuel ratio	Parameter of interest
1	2000	8-18	1. Brake Power/ bmep
2	3000	8-18	2. Torque
3	4000	8-18	3. Indicated Work
4	5000	8-18	4. Thermal Efficiency
5	6000	8-18	

3.9 SUMMARY

For this chapter, all the problem setup completed with the right parameters. By selecting the suitable icon in SIMULINK and design a complete control system model, this analysis will be a trial run. With proper design of engine that is use for this project, all the input data must be correct based on Yamaha FZ150i model specification.

CHAPTER 4

RESULT AND DISCUSSION

4.0 INTRODUCTION

This chapter presents and describes results acquired through simulations made with the model. The results are compared to data which appear in some articles in order to prove the accuracy. The model is simulated between 2000 to 6000 rpm of engine speed range, with the detail of engine geometry based on the engine FZ150i full specification.

4.1 RESULT DATA AND GRAPH

4.1.1 Result of Brake Power

Table 4.1: Tabulated data for Brake Power

Engine Speed(rpm)	Brake Power/ bmep (kW)
2000	0.1609
3000	0.5243
4000	1.2210
5000	2.3590
6000	4.0460

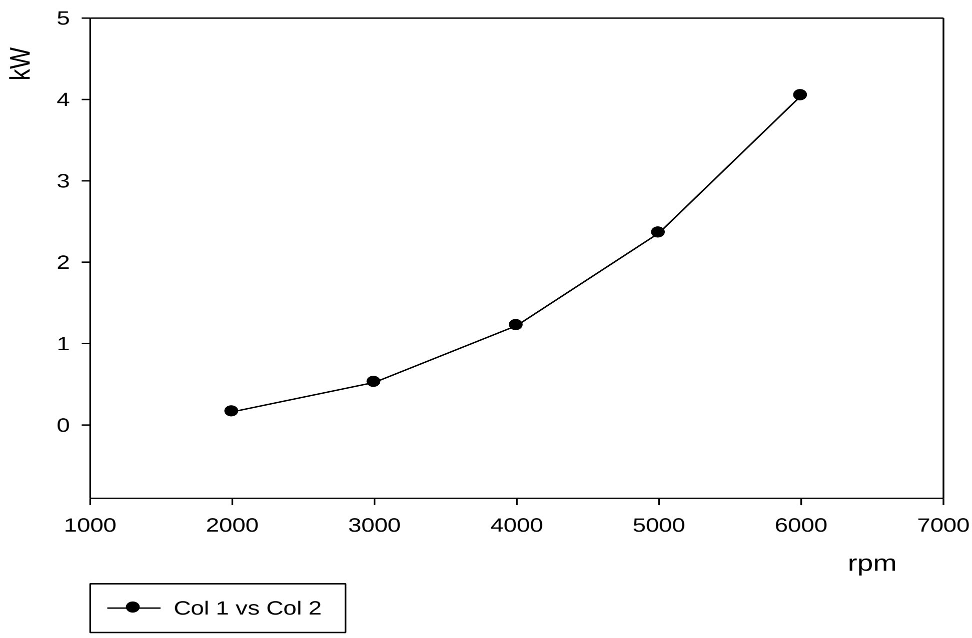


Figure 4.1: Brake Power versus Engine Speed

The term brake power, W_b , is used to specify that the power is measured at the output shaft, this is the usable power delivered by the engine to the load. The brake power is less than the power generated by the gas in the cylinders due to mechanical friction and parasitic loads. From the graph above, the result from simulation show that the brake power increases with engine speed increscent. But in the real engine, the value of brake power has a peak value and will decrease at the maximum engine speed. The differences are because the frictional model diagrams not included. Its means that there is no friction loss in this simulation and the brake power will increase continuously.

4.1.2 Result of Torque

Torque is a measure of an engine's ability to do work and power is the rate at which work is done.

Table 4.2: Tabulated data of Torque

Engine Speed(rpm)	Torque(Nm)
2000	0.081
3000	0.167
4000	2.915
5000	4.505
6000	6.439

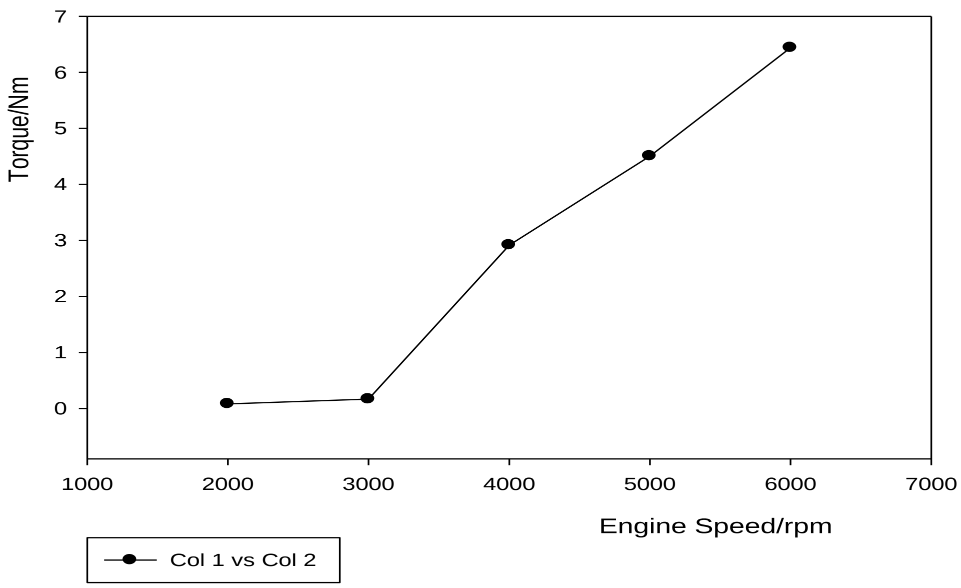


Figure 4.2: Torque versus Engine Speed

From the graph of torque versus engine speed, the torque of engine increase with the engine speed is increased. At 2000 to 3000 rpm, the torque increase in small value, but after that the torque increase significantly. But this is not the trend of real torque versus engine speed because the effect of friction in simulation is not occurring when crank slider model block diagram not included in this simulation. The model is responsible to make the friction effect in the engine. So, when the friction is doesn't exist, the torque will increase because the engine is not need to overcome the frictional loss in the engine.

4.1.3 Trend comparison graph of brake power and torque

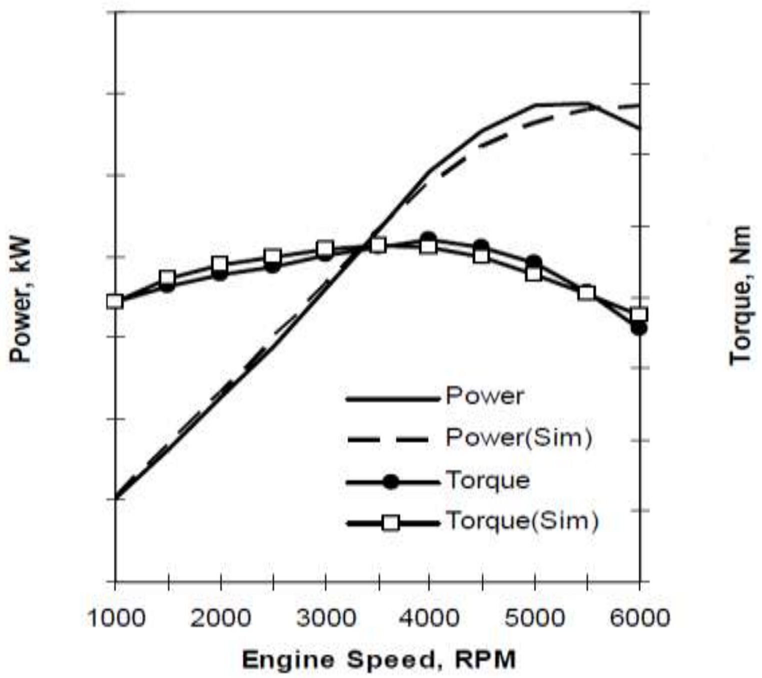


Figure 4.3: Brake Power and Torque versus Engine Speed Simulation (Sitthiracha, 2006)

From this graph, we can see that the curve of the graph is continuously increasing throughout the engine speed. But when it reached the maximum engine speed, the curve is decreasing and this is not happen in result of simulation by Sitthiracha, (2006). This happen because of frictional model is not included in the simulation where this model is uses to cause the friction loss in the engine. From Sitthiracha, (2006), the power and torque are decreasing at the last because this power used to overcome the friction loss in the engine.

4.1.4 Result of Thermal Efficiency

Table 4.3: Tabulated data for Thermal Efficiency

Engine Speed(rpm)	Thermal Efficiency (From 100 %)
2000	-41.67
3000	-27.78
4000	-20.83
5000	-16.67
6000	-13.09

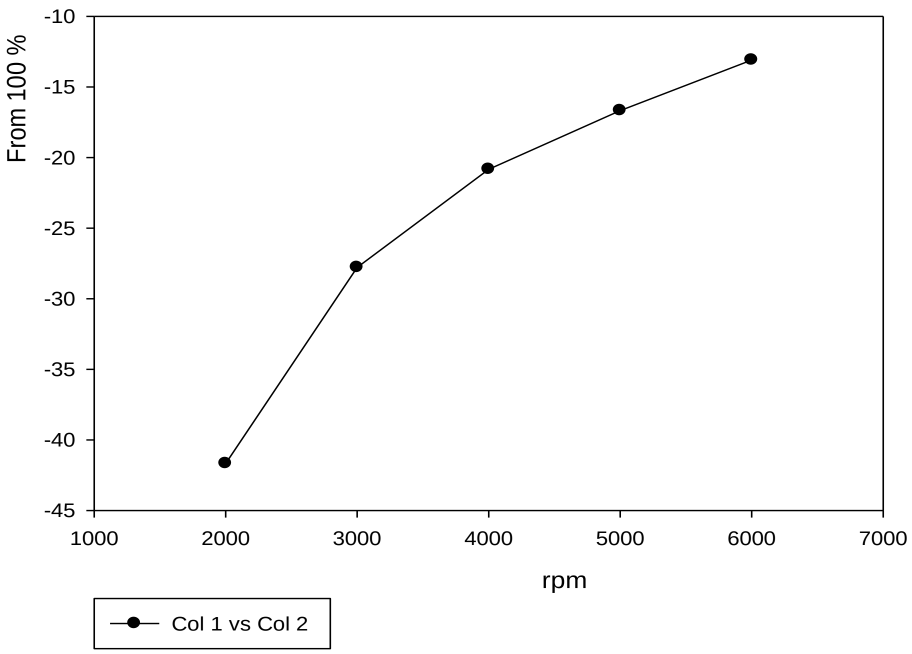


Figure 4.4: Thermal Efficiency verses Engine Speed

From figure4.4, thermal efficiency is the efficiency of a heat engine measured by the ratio of the work done by it to the heat supplied to it. From the graph above, it shows that the thermal efficiency increased when engine speed are increased. It is because, from theoretically the amount of work done by the engine normally increase when the engine speed increase, and the work done are related by thermal efficiency of the engine, then thermal efficiency will definitely increase until maximum point which is

86.91 % at 6000 rpm. It is not the right point of thermal efficiency because it may effect from no frictional model from this simulation.

4.1.5 Graph for comparison of Thermal Efficiency:

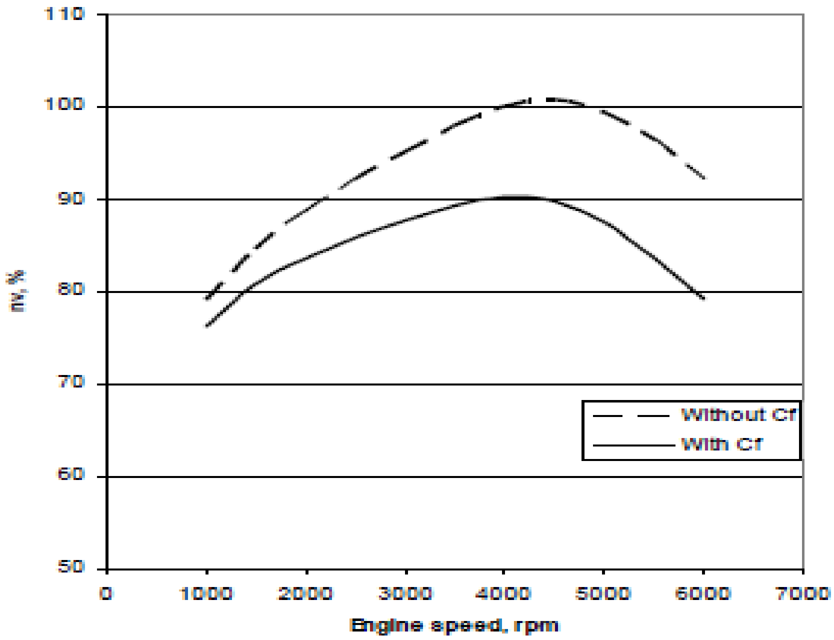


Figure 4.5: Thermal Efficiency versus Simulation (Sitthiracha, 2006)

According to Figure4.5, the curve of thermal efficiency increase with engine speed increasing, while Figure4.4 show the decreasing curve at the end of engine speed. This is because from thermal efficiency equation, $\eta_{th} = \frac{W_{out}}{Q_{in}}$, if work decrease then the thermal efficiency will decrease. The work decrease is related to the friction that occurs in frictional model.

4.1.6 Result of Indicated Work

Table 4.4: Tabulated data for Indicated Work

Engine Speed(rpm)	Indicated Work(kJ)
2000	0.2332
3000	0.9309
4000	2.0870
5000	3.6910
6000	5.7300

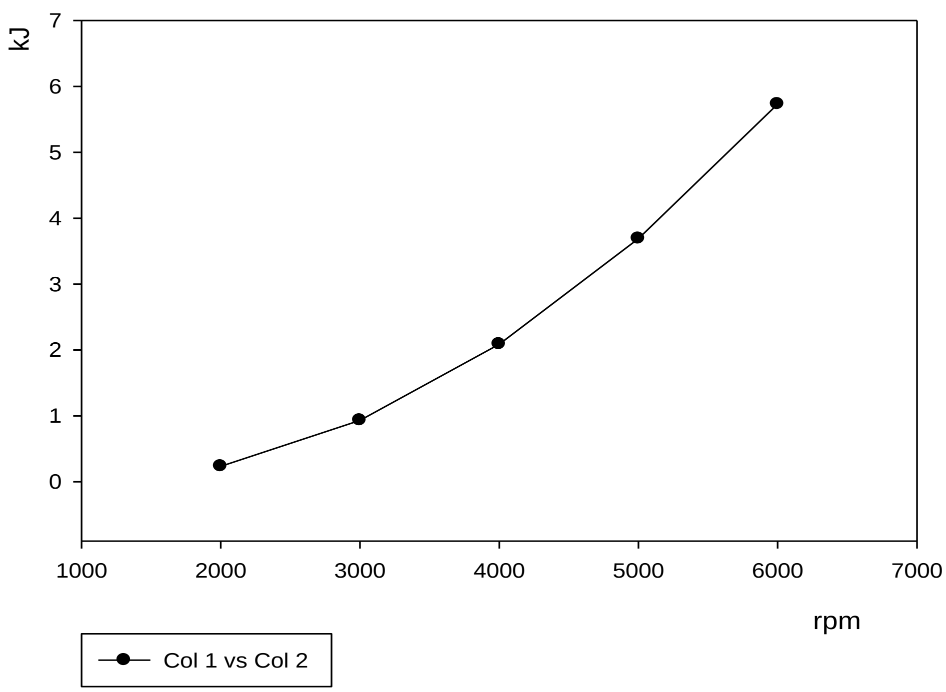


Figure 4.6: Indicated Work verses Engine Speed

From the figure4.6, the increasing of indicated work can be obtained; these phenomena may due to the pressure inside the engine. Indicated work is work delivered to the piston over the entire four-stroke cycle. Its means the increasing of engine speed, the pressure will increase and continued by indicated work increased. Further study for these phenomena can be observed in the thesis of reconstructing by Zeng et. al, (2004).

4.1.7 Result of Temperature

Table 4.5: Tabulated data for Temperature

Engine Speed(rpm)	Temperature(K)
2000	1856
3000	1873
4000	1881
5000	1884
6000	1995

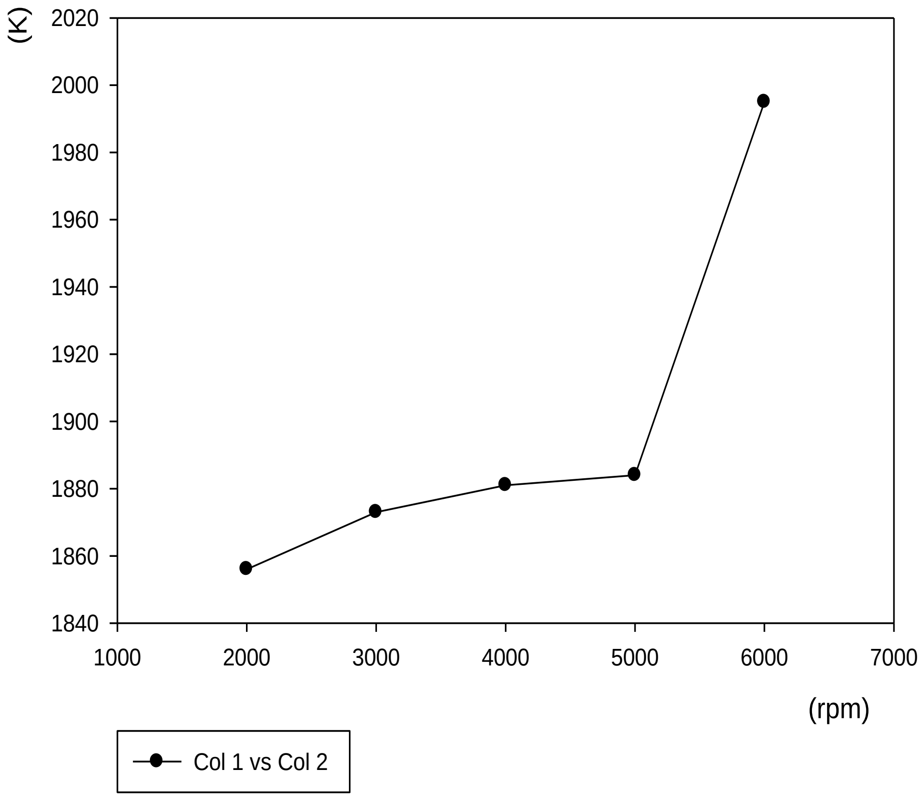


Figure 4.7: Temperature versus Engine Speed

From the graph above, at 2000 rpm to 5000 rpm of engine speed, the temperature was increased slowly, but after 5000 rpm of engine speed, the temperature was increased very faster. It is because in the system, the model diagram of cooling system doesn’t exist, and the temperature cannot be control.

4.2 SUMMARY

From this simulation, the result shows that it is almost same with the experimental data from sitthiracha, (2006). Without build the real engine, all the engine performance parameter can be calculated from this simulation, and will reduce the time and cost.

CHAPTER 5

CONCLUSION AND RECOMMENDATION

5.1 CONCLUSION

An analytical model of spark ignition engine has been constructed based on cylinder-by-cylinder engine model which combines both physical formulae such as engine geometries, and empirical formulae such as burning duration. The engine performance, torque and power calculated by integrating the pressure inside cylinder within one engine cycle.

In engine modeling, the model needs design parameters from real engine. It is the same as 3-D engine model which needs the 3-D geometry of combustion chamber, valves, and ports in order to achieve the accuracy.

The model is verified by data from FZ150i engine models. It can capture torque and power characteristics very well, without need to build the real engine.

5.2 RECOMMENDATION

For part load condition, the throttle body model is needed to integrate into the engine model. However, the real geometries of the throttle body such as close angle, maximum angle, inside throttle body diameter, are needed also in order to calculate the mass flow through the throttle itself.

All equations which describe the combustion range of the alternative fuels should be verified by experiment in order to prove model prediction accuracy.
However, the results from experiments may deviate from the predictions caused by proportion of fuel mixture and also testing conditions.

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APPENDIX A1

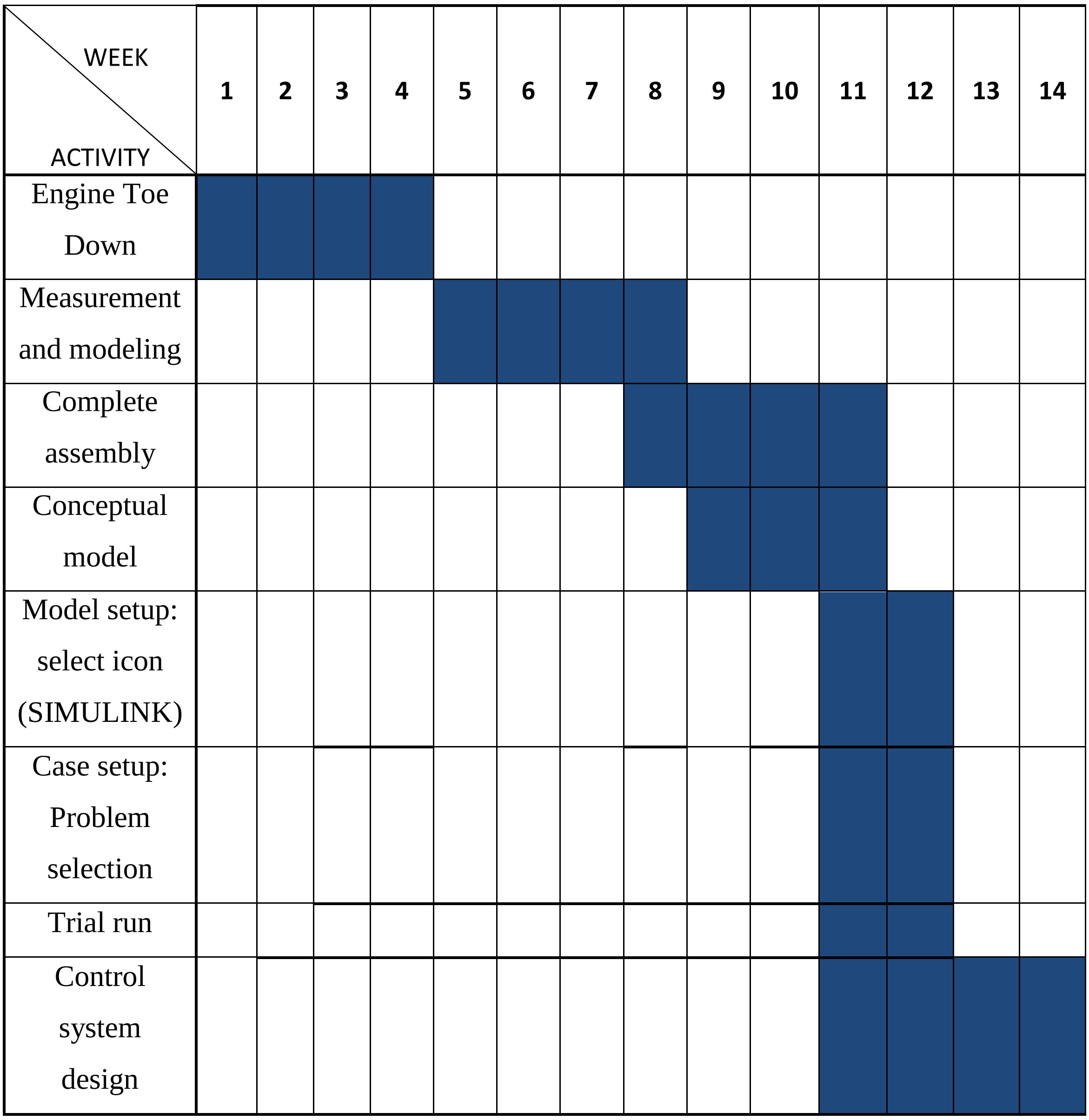

Figure A1 Gantt chart for Final Year Project 1

APPENDIX A2

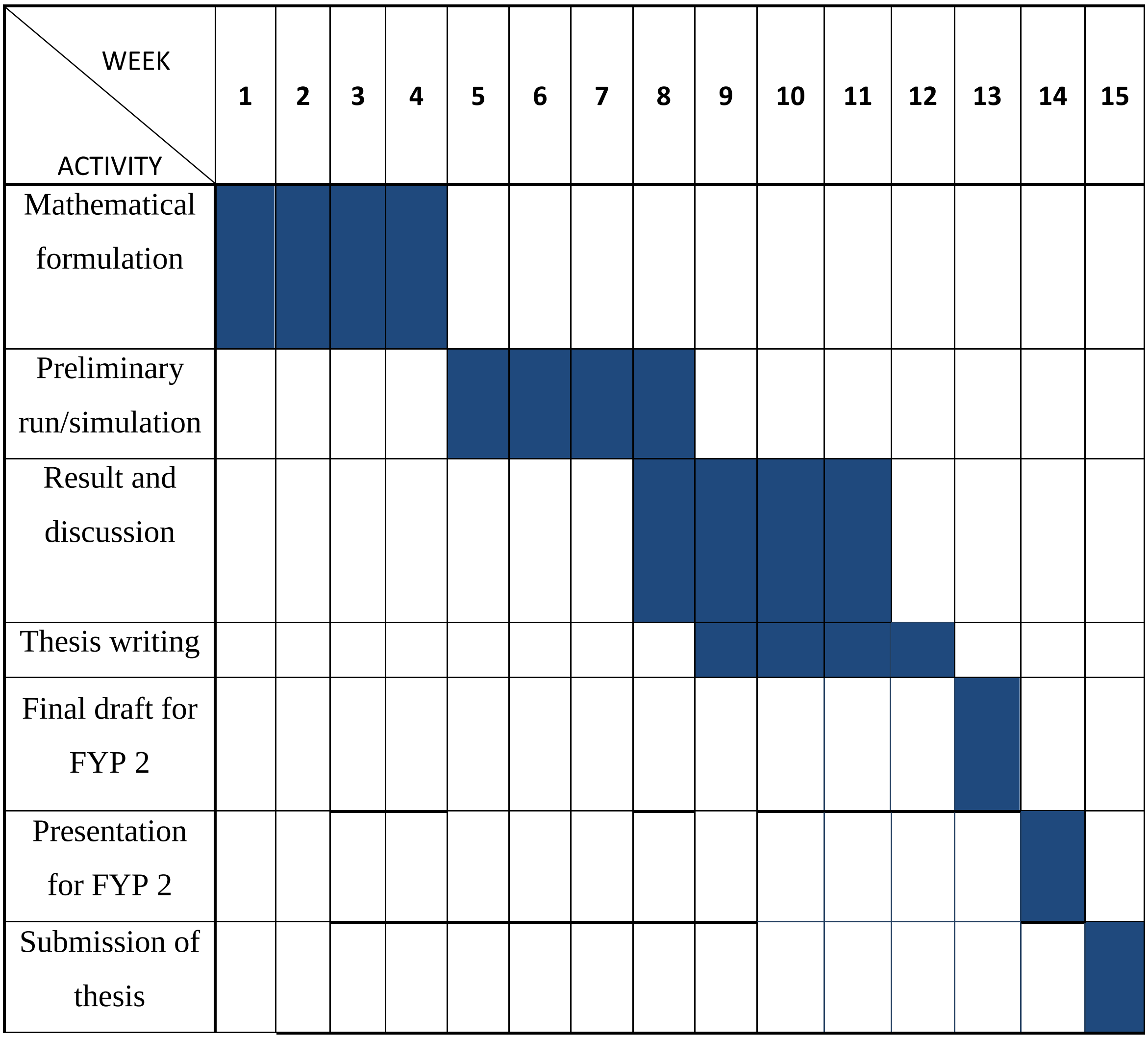

Figure A2 Gantt chart for Final Year Project 2

APPENDIX B

Table B1 Table of data result

Engine Speed (rpm)	**Brake Power/ bmep (kW)**	**Torque (Nm)**	**Indicated Work (kJ)**	**Thermal Efficiency (from 100%)**
2000	0.1609	0.081	0.2332	-41.67
3000	0.5243	0.167	0.9309	-27.78
4000	1.2210	2.915	2.0870	-20.83
5000	2.3590	4.505	3.6910	-16.67
6000	4.0460	6.439	5.7300	-13.09