ENERGY BALANCE STUDY FOR 4 STROKE GASOLINE ENGINE ANALYSES

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JUDUL: **ENERGY BALANCE STUDY FOR 4 STROKE GASOLINE ENGINE ANALYSES**

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ENERGY BALANCE STUDY FOR 4 STROKE GASOLINE ENGINE ANALYSES

ABDULLAH SHARIFFUDIN BIN MOHAMAD

Thesis submitted in fulfillment of the requirements for the award of the degree of
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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 in candidate of any other degree.

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ABSTRACT

This thesis focuses on a comparative energy balance study based on a four cylinder gasoline engine data operated on different engine speed of wide open throttle. The key objective of conducting the energy balance study is to determine the most influence factor that contribute to the heat losses. The transfer of energy was measured for losses to the engine coolant and exhaust, usable power output, as well as friction losses. In a conventional internal combustion engine, approximately one-third of total fuel input energy is converted to useful work. A major part of the energy is lost with the exhaust gases. In addition, another major part of energy input is rejected in the form of heat via the cooling system. The importance of this study is to identify the key factor that contribute to the heat losses which further can be use to minimize the heat losses and at the same time improves the power output and mechanical efficiency. In present study, heat balance has been investigated theoretically for different engines speeds and load. To analyze energy balance, a zero-dimensional multi-zone thermodynamics model has been developed and used. The results showed that input energy at low engine speed was distributed 6.95%, 21.66%, 20.30%, and 0.88% to the major areas of coolant, exhaust, and power output, and friction, respectively. Differently the input energy at high engine speed was distributed 22.09%, 26.6%, 30.09%, and 5.68% to the major areas of coolant, exhaust, power output, and friction respectively. Energy loss increases with increasing engines speed and load and thus the mechanical efficiency decreases. Future improvements to obtain distinguishable results are outlined.

ABSTRAK

Tesis ini menfokuskan kajian ke atas keseimbangan tenaga secara bandingan di dalam enjin gasoline 4 silinder yang beroperasi pada kelajuan enjin yang berbeza dengan bukaan pendikit maksimum. Objektif utama dalam melakukan kajian keseimbangan tenaga adalah untuk mengenalpasti fator paling mempengaruhi yang menyumbang kepada kehilangan tenaga. Tenaga yang dipindahkan adalah ditukarkan kepada kerja yang berguna, kehilangan tenaga pada cecair penyejuk dan ekzos, serta sedikit kehilangan tenaga akibat beban pelbagai. Lebih kurang satu per tiga jumlah input tenaga minyak ditukarkan pada kerja yang berguna dalam pembakaran dalam enjin biasa. Bahagian paling utama kehilangan tenaga adalah gas ekzos. Tambahan lagi, bahagian utama lain tenaga input dibuang dalam bentuk haba melalui system penyejuk. Kepentingan kajian ini adalah untuk mengurangkan kehilangan tenaga dan dalam masa yang sama meningkatkan penghasilan kerja berguna dan kecekapan mekanikal. Dalam kajian terkini, keseimbangan tenaga pada kelajuan dan beban enjin yang berbeza telah diselidik secara teori. Untuk menganalisis keseimbangan tenaga, model sifar pelbagai zon termodinamik akan dibentuk dan digunakan. Keputusan menunjukan yang tenaga input pada kelajuan yang rendah ditukarkan kepada 6.95%, 21.66%, 20.30%, dan 0.88% kepada bahagian utama iaitu cecair penyejuk, ekzos, kerja berguna dan geseran. Manakala tenaga input pada kelajuan yang tinggi ditukarkan kepada 22.09%, 26.6%, 30.09%, dan 5.68% kepada bahagian utama iaitu cecair penyejuk, ekzos, kerja berguna dan geseran. Kehilangan tenaga meningkat dengan peningkatan kelajuan dan beban enjin seterusnya mengurangkan kecekapan mekanikal. Secara kasar, pembaikan pada masa hadapan adalah untuk mandapatkan keputusan yang boleh dibandingkan.

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LIST OF SYMBOLS

A	Independent Variable
B	Dependent Variable
C_d	Discharge Coefficient
C_P	Specific heat
d	Diameter
F_P	Friction Power
g	Gravitational Acceleration
h	Height
I_P	Indicated Power
k	Coefficient
$\dot{m}$	Mass Flow Rate
n_R	Number of Crank Revolution For Each Power Stroke per Cylinder
N	Engine Speed
P	Local Gauge Pressure
P_b	Brake Power
ΔP	Pressure Drop
Q	Volumetric Flow rate
R	Standard Universal Gas Constant
r_e	Expansion Compression Ratio

t	Time
T	Temperature
v	Volume
V	Flow Velocity
V_d	Displacement Value
$\%$	Percentage

LIST OF ABBREVIATIONS

ABDC	After Bottom Dead Centre
ATDC	After Top Dead Centre
BDC	Bottom Dead Centre
CA	Crank Angle
EVO	Exhaust Valve open
FMEP	Friction Mean Effective Pressure
ICE	Internal Combustion Engine
IMEP	Indicated Mean Effective Pressure
IP	Indicated Power
LHR	Low Heat Rejection
LPM	Liter per Minute
MEP	Mean Effective pressure
rpm	Revolution per Minute
SI	Spark Ignition
TDC	Top Dead Centre
WOT	Wide Open Throttle

CHAPTER 1

INTRODUCTION

1.1 BACKGROUND OF THE STUDY

Nowadays, the consumption of internal combustion engines in vehicle faced with some of problem. Increasing demand and emission to environment, depleting fossil fuel, and quest for increasing the efficiency of an internal combustion engine have lead to research and development on production of energy efficient engines. In order to improve that performance and efficiency of engine, many experimental and theoretical investigations have been carried out explaining the energy balance study of spark ignition engines. Minimizing the energy losses in the engine definitely improves the power output and efficiency of the energy losses is the key point why it is important in energy balance research. A cursory look at the internal combustion engine heat balance indicates that the input energy is divided into roughly three equal parts: energy converted to useful work, energy transferred to coolant and energy lost to exhaust. In analysis work, method of analyze data are important in order to conduct heat balance analysis.

1.2 PROBLEM STATEMENT

The main purpose of study is to determine exact amount of fuel energy converted to useful work, amount of fuel energy lost by exhaust gasses, cooling and friction. In order to fulfill the requirements, energy balance in spark ignition engine at different engine speed need must be evaluate and analyze.

1.3 OBJECTIVES OF THE STUDY

1. To conduct energy balance study on four stroke cylinder in-line Mitsubishi Magma (4G15) 1.5 liters spark ignition gasoline engine.
2. To identify the most influential factor that contributes the heat losses in engine operation.

1.4 SCOPES OF THE STUDY

Study of previous research and analyze previous data for energy balance studies in the internal combustion. Engine operating under consideration are the rate of air consumption, rate of fuel consumption, cylinder pressure, ignition timing, engine torque, and engine speed. Identify and determine the energy distribution during engine operation. The portion of the fuel's energy is transferred to the engine as useful work, and heat losses through the coolant, exhaust and friction. Making comparison between the results in this thesis with similar results found through the literature search. The results in this thesis are determined by the calculation analysis. Identify and determine the factor influence the heat losses as the final study.

1.5 FLOW CHART OF THE STUDY

The flow chart of the overall procedure of the study is shown in Figure 1.1

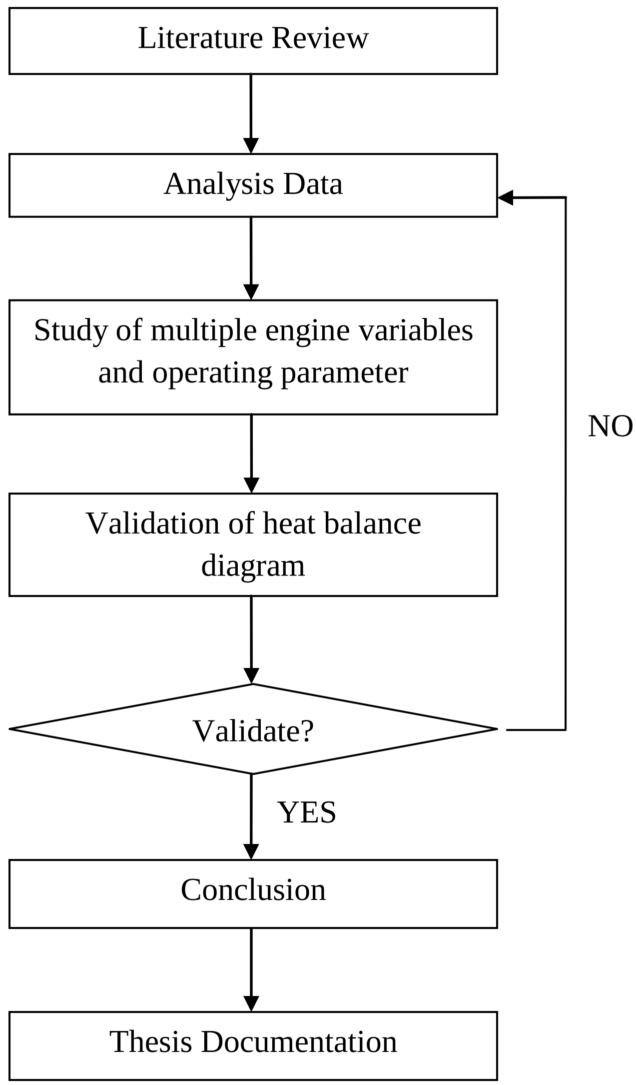

Figure 1.1: Flow chart of the overall procedure of the study

1.6 ORGANIZATION OF THESIS

This thesis is divided into five chapters including the introduction which is organized to present the sequence of logical thoughts. In chapter 1, there are five main items and the items are background of the study, problem statement, objectives of the study, scopes of the study, and structure of the thesis. Chapter 2 gives the significant importance of energy balance analysis, the foundation or theoretical of energy balance, and also the previous research that focus onto energy balance analysis. Chapter 3 presents heat balance analysis. This chapter also describe about the relationship of dependent variable and independent variable from heat balance analysis. Chapter 4 addresses the validation of the predictions results against data analyze results. The main focus is present the result of relationship between independent variable and dependent variable. Chapter 5 presents the significant finding or summary of the study and the suggestion for future works.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

This chapter presents previous studies and recent progress in energy balance study in four stroke gasoline engine that produced the framework of current study. The literatures are roughly composed of three outstanding subjects which are the importance of energy balance analysis, foundation or theoretical of energy balance and the previous research that focus onto the energy balance analysis.

2.2 THE IMPORTANCE OF ENERGY BALANCE ANALYSIS

The energy produced due to combustion of fuel in an engine is partly converted into work and the rest is lost. The knowledge of how the energy is lost will help in finding means to reduce the losses as well as to improve the performance of the engine in terms of efficiency and power output. Base on work of Kumar (2004), data experimental from single cylinder, naturally aspirated 4-stroke gasoline engine was used to determine the fuel flow rate, air flow rate, and exhaust gases temperature. Estimation of energy lost due to heat transfer to engine components was arrived at.

In a conventional internal combustion engine, approximately one-third of total fuel input energy is converted to useful work. Since the working gas in a practical engine cycle is not exhausted at ambient temperature, a major part of the energy is lost with the exhaust gases. In addition, another major part of energy input is rejected in the form of heat via the cooling system (Tymaz, 2003). If the energy normally rejected to the coolant could be recovered and reutilized instead on the crankshaft as useful work, then a substantial improvement in fuel economy would result. In the study of Tymaz (2003), four stroke cylinder in-line Mitsubishi Magma (4G15) 1.5 liters spark ignition gasoline engine was tested at different speeds and load conditions. The results indicate a reduction in fuel consumption and heat losses to engine cooling system of the gasoline engine (Tymaz, 2003). The quest for increasing the efficiency of an internal combustion engine has been going on since the invention of this reliable workhorse of the automotive world. In recent times, much attention has focused on achieving this goal by reducing energy lost to the coolant during the power stroke of the cycle.

The first law of thermodynamics is satisfied as long as energy is converted regardless of how that energy is apportioned between various categories (Tymaz, 2003). The second law stipulates that all the input energy cannot be converted into work; in other words, it is impossible to obtain 100% efficiency, so some heat has to be rejected, preferably at the lowest possible temperature to achieve highest possible efficiency (Tymaz, 2003). The reduction in the in-cylinder heat transfer to either the coolant and/or the environment does not violate the second law of thermodynamics and, moreover, according to the first law, has the potential of producing more work (Tymaz, 2003). Added to this, another important advantage of the concept is the great reduction in parasitic losses due to reduction of cooling system, thus increasing the brake horsepower of the engine (Tymaz, 2003). These prospects of improving the design and performance have generated impetus to active research on adiabatic or more appropriately, low heat rejection (LHR) or insulated engines (Tymaz, 2003).

2.3 FOUNDATION OR THEORETICAL OF ENERGY BALANCE

The knowledge and understanding about the theoretical of the energy balance is important in develop the mathematical formulation. The foundation of energy balance will be described onto three parts and there are the basic principles of energy balance, the concept of the open system, and the theoretical equation and formula.

2.3.1 Basic Principles of Energy Balance

The methods and techniques for performing the energy balances followed the same basic principles. “Energy balance for a direct injection diesel engine shows that about one-third of fuel energy input is lost to environment through heat transfer, another third is wasted as exhaust heat and only one-third is available as shaft work” (Sharma and Jindal, 1989). While the percentages vary between engines, this seems to be the accepted rule of thumb. Stated more precisely, energy enters the engine in the form of fuel and “leaves as energy in the exhaust, cooling water, brake power and heat transfer. Heat losses must be decreased to improve the engine efficiency. It is very important to know the fraction of the heat loss mechanisms (Yuksel and Ceviz, 2003). Aside from energy transfer to power output, exhaust, and engine components/coolant, all of these energy balance studies find the “unaccounted for” losses which is the remainder of what was not measured (found from energy conservation). Depending on the loading conditions, the percentage of the unaccounted for losses ranged between 1 and 23% (Kumar, et al. 2004; Taymaz, 2006; Ajav, Singh and Bhattacharya, 2000).

2.3.2 The Concept of Open System

An open system is a state of a system, in which a system continuously interacts with its environment. Open systems are those that maintain their state and exhibit the characteristics of openness previously mentioned (Ajav, Singh and Bhattacharya, 2000). Open systems contrast with closed system. Systems are rarely ever either open or closed but open to some and closed to other influences. Basic characteristics of an open system are environment, input, throughput and output. And some control systems with feedback. In internal combustion engine, it means the mixture of air and fuel as the input for the system. The other inputs item are coolant in and lubricating oil temperature. The power to crankshaft useful is the output item in this system.

The concept of the open system was used in the development of thermodynamics theory for energy balance. This is exactly method and can be very helpful in considering the total character of an engine testing in this research. The great advantages of this concept is that once one has identified all the mass and energy flows into and out of the system it is essential to know exactly what is going on inside the system in order to draw up a 'balance sheet' of inflows and outflows. The multifarious inflows and outflows to and from a test are as follows:

Table 2.1: The Flows and Outflows of Heat Balance

In	Out
Fuel	Ventilation air
Ventilation air	Exhaust
Charge air	Engine cooling water
Cooling water	Dynamometer cooling water
Electricity for services	Electricity from dynamometer
	Losses through walls and ceiling

Source: Martyr and Plint, 2008

Balance sheets may be drawn up for fuel, air, water and electricity, but by far the most important is the energy balance, since every one of these quantities has associated with it a certain quantity of energy. The same concept may be applied to the engine within the cell. This may be picture as surrounded by its own control surface, through which the following flows take place (Martyr and Plint, 2008).

Energy supplied to an engine is the heat value of the fuel consumed. Only a part of this energy is transformed into useful work. The rest of it is either wasted or utilized in special application like turbo compounding. The two main parts of the heat not available for work are the heat carried away by the exhaust gases and the cooling medium. Figure 3.1 illustrates the same for spark ignition engines (Pulkabek, 1997).

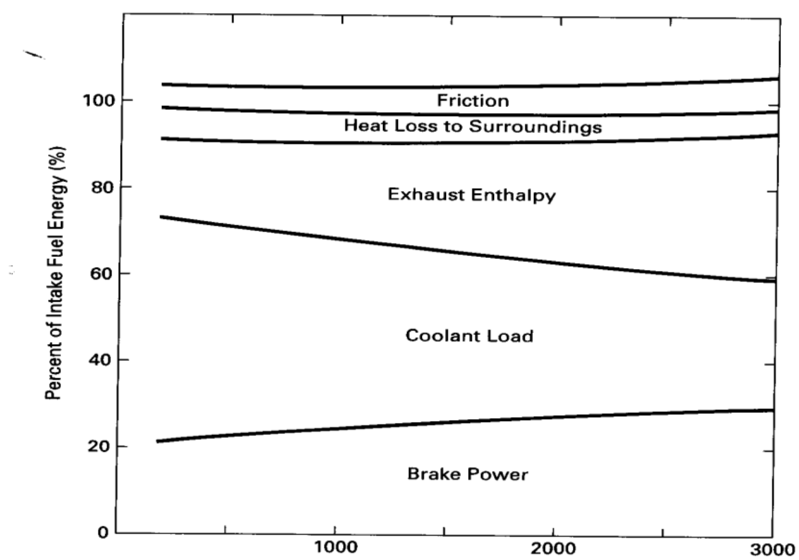

Figure 2.1: Distribution of energy in a typical SI engines as a function of engine speed

Source: Pulkrabek 1997

To give sufficient result of heat balance analysis, an analysis should include a method of finding the fuel flow rate, air flow rate, and exhaust gases temperature. The small losses, such as radiation and incomplete combustion, the above enumerated data makes it possible to account for the heat supplied by the fuel and indicates its distribution.

It may be argued the same amount of frictional power, is accounted in the rise of cooling water temperature. However, it is taken into account here to show that the frictional losses also include blow down and pumping losses and therefore it is not appropriate to put in the heat balance. Since there are always certain losses which cannot be accounted for, by including friction forces in the heat balance, the accounted balance is shown in Figure 3.2. Usually the amount of heat carried by lubricating oil is comparatively small and is normally not included.

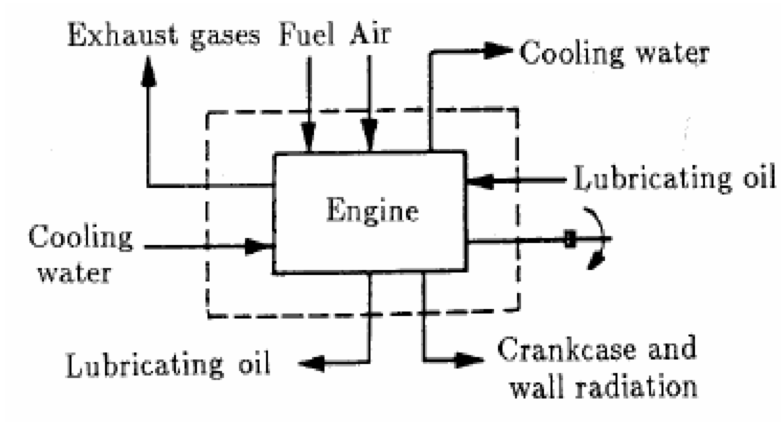

Figure 2.2: External Heat Balance

Source: Heywood 1994

2.3.3 The Theoretical Equation and Formula

The steady state flow energy equation gives the relationship between these quantities, and is usually expressed in kilowatts (Martyr and Plint, 2008):

$$H_1 = P_S + (H_2 - H_3) + Q_1 + Q_2 \qquad (2.1)$$

Where:

H_1 = combustion energy of fuel = $\dot{m}_f \times C_L \times 10^3$

P_S = power output of engine

H_2 = enthalpy of exhaust gases * = $(\dot{m}_f + \dot{m}_a) \times C_P \times T_e$

H_3 = enthalpy of inlet air = $\dot{m}_a \times C_P \times T_a$

Q_1 = heat to cooling water = $\dot{m}_w \times C_w \times (T_{2w} - T_{1w})$
Q_2 = convection and radiation

This assumes that the specific heat of the exhaust gas, the mass of which is the sum of the masses of air and fuel supplied to the engine, is equal to that of air. This is no strictly true, but permits an approximate calculation to be made if the temperature of the exhaust gas is measured (Martyr and Plint, 2008).

Note that it is not possible to show the indicated power directly in this energy balance since the different between it and the power output, representing friction and other losses, appears elsewhere as part of the heat to the cooling water and other losses (Martyr and Plint, 2008).

2.4 ENERGY BALANCE ANALYSIS FROM PREVIOUS STUDY

The heat transfer to the cooling water is found from the temperature rise in the coolant as it passes through the engine and the mass flow rate of coolant (Taymaz, 2003).

$$\dot{Q}_{water} = \left(\dot{m} \times c_p\right)_{water} \times (T_{out} - T_{in}) \tag{2.2}$$

The mass flow rate of the exhaust is known in terms of the measured mass flow rates of air and fuel since (Taymaz, 2003)

$$\dot{m}_{exhaust} = \dot{m}_{air} + \dot{m}_{fuel} \tag{2.3}$$

In doing these energy balances, it is common practice to evaluate the maximum heat that can be recovered from the exhaust gas. This is computed from an energy balance on the exhaust where it is cooled to ambient temperature (Taymaz, 2003)

$$\dot{Q}_{exhaust} = \dot{m}_{ex}\left[h_{ex}(T_{exhaust}) - h_{ex}(T_{ambient})\right] \tag{2.4}$$

An overall first law energy balance for an engine provides useful information on the disposition of the initial fuel energy. For a control volume, which surrounds the engine, the steady-flow energy conservation equation is

$$\dot{m}_{fuel}\,h_{fuel} + \dot{m}_{air}\,h_{air} = P_b + \dot{Q}_{cool} + \dot{Q}_{misc} + \left(\dot{m}_{air} + \dot{m}_{fuel}\right)h_{ex} \tag{2.5}$$

Where, h is enthalpy, P_b is the brake power, $\dot{Q}_{cool}$ is the heat transfer rate to the cooling medium, $\dot{Q}_{misc}$ is the heat rejected to the oil (if separately) plus convection and radiation from the engine's external surface (Taymaz, 2003).

The indicated power is the sum of the brake power and the friction power. A substantial part of the friction power (about half) is dissipated between the piston and piston rings and cylinder wall and is transferred as thermal energy to the cooling medium. The remainder of the friction power is dissipated in the bearings, valve mechanism, or drives auxiliary devices, and is transferred as thermal energy to the oil or surrounding environment (Taymaz, 2003).

In the analysis of Ajav (1999), data from the experimental was analyzed at five load levels: no load, 25, 50, and 75% full load and full load. The heat losses through the various loads were calculated as follows: The total heat (Q) supplied by the fuel given as)

$$Q = \frac{CV \times \dot{M}_f}{3600} \tag{2.6}$$

Where: $\dot{M}_f$ is the fuel consumption (kg/h) and CV is the calorific value of fuel (kJ/kg). The percentage of heat supplied per second which is converted to useful work q_1 is

$$q_1 = \frac{Bhp \times h_e}{Q} \times 100 \tag{2.7}$$

Where: Bhp is the brake horse power, h_e is the heat equivalence of Bhp (0.7344) and Q is the total heat supplied by fuel (kJ/s).

The percentage of heat taken from the engine by the cooling water (q_2) was determined by measuring the flow rate of water $\left(\dot{M}_w\right)$ entering the engine as well as the temperature difference of inlet and outlet water.

$$q_2 = \frac{\dot{M}_w}{3.6} \times C_w \times (T_2 - T_1) \tag{2.8}$$

Where: C_w is the specific heat of water (kJ/kg K), T_1 is the inlet water temperature (K) and T_2 is the outlet water temperature (K) (Ajav, 1999).

The percentage of heat lost through the exhaust gases q_3 was calculated considering the heat necessary to increase the temperature of the total mass $\dot{M}_g$ (kg/ h) from outside conditions T_a (K) to the temperature of the exhaust T_g (K). This heat loss is also known as `sensible heat', and to calculate it, it is necessary to estimate the mean specific heat $\left(C_g\right)$ of the gases which, in this case, was assumed to be the value for air with a mean temperature of the exhaust (Ajav, 1999).

$$q_3 = \frac{\dot{M}_g}{3600} \times C_g \times \left(T_g - T_a\right) + \frac{\dot{M}_g}{3512} \times \left(T_g - T_a\right) \tag{2.9}$$

The unaccounted percentage of heat losses (q_4) is given as

$$q_4 = 100 - (q_1 + q_2 + q_3) \tag{2.10}$$

From energy conversation the following energy balance can be written by using the above mentioned energy terms (Durgun and Sahin, 2008):

$$Q_{in,a} + Q_f = W_i + Q_{exh} + Q_h \tag{2.11}$$

Where the left hand side terms show total input energy and the right hand side terms show total output energy, $Q_{in,a}$ [kJ] is enthalpy of the intake air, Q_f [kJ] is chemical energy of the fuel, W_i [kJ] is indicated useful work and Q_{exh} [kJ] is heat lost through the exhaust gases, Q_h [kJ] is heat lost through the walls which is determined by using well known Annand's correlation (Durgun and Sahin, 2008).

$$Q_{in,a} = m_a \times C_{p,a} \times T_0 \tag{2.12}$$

$$Q_{exh} = m_{exh} \times C_{p,exh} \times T_{exh} \tag{2.13}$$

$$Q_f = m_f \times Q_{UHV} \tag{2.14}$$

$$W_i = \varphi_i \times W_{gr} \tag{2.15}$$

where m_f, m_a, and m_{exh} [kg] are the masses of the fuel, air and exhaust gases, respectively, Q_{UHV} [MJ/kg] is the upper heating value of the fuel calculated from Mendeleyev's formula, $C_{p,}$ [kJ/kg K] is specific heat capacity under constant pressure, T_{exh} [K] and T_0 [K] are the exhaust and ambient air temperatures (Durgun and Sahin, 2008).

2.5 SUMMARY

This chapter dedicated the literature findings relevant to this study. There is reviewed the importance of energy balance. Next, the foundation or theoretical of energy balance is describe. The items that highlight are the basic principle of energy balance, the concept of open system, and the theoretical equation.

CHAPTER 3

METHODOLOGY

3.1 INTRODUCTION

This chapter is dedicated to the description of major approach for theoretical calculation and heat balance analysis. Some parameter that was determined in heat balance analysis is in important to identify the dependent variable as factor for heat losses. The method of correlation between the heat losses and the factor that contribute to the losses are also described in this chapter.

3.2 PROCEES OF ENERGY DISTRIBUTION

Here heat balance has been determined by virtue of the assumption that there is a closed system in the engine cylinder from the intake stroke to the exhaust process (Durgun and Sahin, 2008). Input energy is taken equal to the sum of chemical energy of the fuel and enthalpy of the intake air. Kinetic energies of intake and exhaust gases were not taken into account. Intake air properties have been computed by using Ferguson's functions in respect of ambient air temperature and pressure (Durgun and Sahin, 2008). The output energy of the engine is taken equal to the sum of useful work, heat lost to the walls and heat lost through the exhaust gases. Base on work of Durgun and Sahin, (2008), the indicated work has been separated to the brake work (useful work) and the work spent to the mechanical losses. Heat losses to the walls have been computed during complete engine cycle by using Annand's correlation (Durgun and Sahin, 2008). Also, heat lost through the exhaust gases has been calculated by using exhaust gas temperature and specific heat at constant pressure of the exhaust gases. The composition of the exhaust gases and thermodynamic properties of the working fluid have been determined by using Olikara and Borman's method (Durgun and Sahin, 2008). The steady state flow energy equation gives the relationship between these quantities, and is usually expressed in kilowatts:

The combustion energy of fuel is defined as:

Combustion energy of fuel = Brake power + Cooling losses + Exhaust losses + Friction

$$\dot{m}_{fuel} = P_b + Q_{cooling} + Q_{Exhaust} + F_P \tag{3.1}$$

The amount of energy (power) available for use in engine is defined as:

$$Q_{fuel} = \dot{m}_{fuel} \times Q_{LCV} \tag{3.2}$$

$$\dot{m}_{fuel} = \frac{\left(\frac{v_{fuel}}{1000}\right)}{t} \times \rho_{fuel} \tag{3.3}$$

Where:

ρ_{fuel} = Fuel density

t = Fuel consumption time

v_{fuel} = Fuel volume

Q_{LCV} = Low calorific value

$\dot{m}_{fuel}$ = Fuel flow rate

The power P_b delivered by the engine and absorbed by the dynamometer is defined as:

$$P_b = \frac{2\pi NT}{60 \times 10^3} \tag{3.4}$$

Where:

N = Engine speed

T = Engine torque

The heat transfer to the cooling water is found from the temperature rise in the coolant as it passes through the engine and the mass flow rate of coolant mathematically defined as:

$$Q_{cooling} = \dot{m}_{water} \times C_P \times (T_{Out} - T_{In}) \tag{3.5}$$

Where:

$\dot{m}_{water}$ = Coolant flow rate

C_P = Specific heat of water

T_{In} = Inlet coolant temperature

T_{Out} = Outlet coolant temperature

A heat loss through the exhaust system is defined as:

$$Q_{Exhaust} = \dot{m}_{Exhaust} \times C_P \times T_{Exhaust} \tag{3.6}$$

$$\dot{m}_{Exhaust} = (\dot{m}_{Fuel} + \dot{m}_{Air}) \tag{3.7}$$

The mass air flow rate is determined by using pressure drop across the orifice:

$$\Delta P = \rho \times g \times h \tag{3.8}$$

The flow velocity is defined as:

$$V = \sqrt{\frac{2\Delta P}{\rho_{Air}}} \tag{3.9}$$

The volumetric flow rate is defined as:

$$Q = C_d \times \pi \times \frac{d^2}{4} \times V \tag{3.10}$$

The mass flow rate is defined as:

$$\dot{m}_{Air} = \rho_{fuel} \times Q \tag{3.11}$$

Where:

ΔP = Pressure drop across orifice meter

ρ = Density of coolant liquid in manometer

h = High difference of manometer

V = Flow velocity

ρ_{Air} = Density of air

Q = Volumetric flow rate

C_d = Coefficient of discharge for digital anemometer

d = Diameter of orifice plate

The temperature in the exhaust system is calculated using the ideal gas isentropic expansion relationship between pressure and temperature:

$$T_{Exhaust} = T_{EVO} \times \left(\frac{P_{Exhaust}}{P_{EVO}}\right)^{(k-1)/k} \tag{3.12}$$

$$P_{EVO} = P_{Max} \times \left(\frac{1}{r_e}\right)^{k} \tag{3.13}$$

$$T_{EVO} = \frac{P_{EVO}}{\rho \times R} \tag{3.14}$$

Where:

P_{EVO} = Pressure when exhaust valve open

P_{Max} = Pressure maximum

r_e = Expansion compression ratio

$k = 1.35$

T_{EVO} = Temperature when exhaust valve open

ρ = Density of air

R = Gas constant

Engine friction is taken as the difference between the indicated powers, $\mathbf{I_P}$ and the brake power, $\mathbf{B_P}$. The engine friction is mathematically expressed in terms of frictional power, $\mathbf{F_P}$:

$$I_P = B_P + F_P \tag{3.15}$$

$$I_P = \frac{IMEP \times V_d \times N}{n_R \times 60} \tag{3.16}$$

Where:

B_P = Brake power

V_d = Displacement value

IMEP = Indicated mean effective pressure

N = Engine speed

n_R = Number of crank revolution for each power stroke per cylinder

3.3 SAMPLE OF CALCULATION

Table 3.1 shows the result from the experiment. Sample calculation of energy balance is shown at 4000 rpm.

Table 3.1: Experimental Result

Engine Speed (RPM)	Engine Torque (Nm)	Fuel Consumption Time (Sec)	Pressure drop (mm)	Coolant Flow rate (LPM)	Coolant in Temp (Deg C)	Coolant out Temp (Deg C)
1500	101	19.66	128.64	3	34	60
2000	103	16.72	165.29	5	38	66
2500	110	16.44	179.21	5	39	70
3000	106	14.5	253.04	5	40	77
3500	108	12.35	306.34	9	50	82
4000	107	12.03	380.22	15	62	89

Source: Fadzil, 2008

Combustion energy of fuel, Q_{fuel}

$$Q_{fuel} = \dot{m}_{fuel} \times Q_{LCV}$$

$$\dot{m}_{fuel} = \frac{\left(\frac{50 \times 10^{-3}}{1000}\right)}{t} \times \rho_{fuel}$$

$$\dot{m}_{fuel} = \frac{\left(\frac{50 \times 10^{-3}(m^3)}{1000}\right)}{12.03(s)} \times 768.27(kg/m^3)$$

$$\dot{m}_{fuel} = 0.003193142 \ kg/s$$

$$Q_{fuel} = 0.003193142 \ (kg/s) \times 40 \times 10^3 (J/kg)$$

$$Q_{fuel} = \mathbf{127.7256858\ kW}$$

Energy Converted to Useful Work

Brake Power, P_b

$$P_b = \frac{2\pi NT}{60 \times 10^3}$$

$$P_b = \frac{2\pi \times 4000(rev/min) \times 107(Nm)}{60 \times 10^3}$$

$$P_b = \mathbf{44.82005519\ kW}$$

Percentage of energy loss converted to useful work:

$$Percentage\ (\%) = \frac{P_b}{Q_{fuel}} \times 100\%$$

$$Percentage\ (\%) = \left(\frac{44.82005519\ (kW)}{127.7256858\ (kW)}\right) \times 100\%$$

$$Percentage\ (\%) = \mathbf{35.09087065}\%$$

Energy losses through the cooling system, $Q_{cooling}$ is:

$$Q_{cooling} = \dot{m}_{water} \times C_P \times (T_{Out} - T_{In})$$

$$Q_{cooling} = \frac{15.5}{60}(kg/s) \times 4.18(kJ/kg.K) \times (360 - 331)(K)$$

$$Q_{cooling} = \mathbf{31.3151667\ kW}$$

Percentage of energy loss to the cooling system:

$$Percentage\ (\%) = \frac{Q_{cooling}}{Q_{fuel}} \times 100\%$$

$$Percentage\ (\%) = \left(\frac{31.3151667\ (kW)}{127.7256858\ (kW)}\right) \times 100\%$$

$$Percentage\ (\%) = \mathbf{24.51751695}\%$$

Energy losses to the friction, F_P

$$\text{Indicated Power} = \text{Brake Power} + \text{Friction Power}$$

$$I_P = B_P + F_P$$

$$I_P = \frac{IMEP \times V_d \times N}{n_R \times 60}$$

$$I_P = \frac{1100(kPa) \times 1488 \times 10^{-6}(m^3) \times 4000(rev/min)}{2 \times 60}$$

$$I_P = 54.56\ kW$$

$$F_P = 54.56(kW) - 44.82005519(kW)$$

$$F_P = \mathbf{9.73994481\ kW}$$

Where:

$B_P = 44.82005519 kW$

$V_d = 1488 cm^3$

$IMEP = 1100 kPa$

$N = 4000\ rev/min$

$n_R = 2$

Percentage of energy loss to the friction:

$$Percentage\ (\%) = \frac{F_P}{Q_{fuel}} \times 100\%$$

$$Percentage\ (\%) = \left(\frac{9.73994481\ (kW)}{127.7256858\ (kW)}\right) \times 100\%$$

$$Percentage\ (\%) = \mathbf{7.625674311}\%$$

Energy losses through the exhaust system, $Q_{Exhaust}$

$$Q_{Exhaust} = \dot{m}_{Exhaust} \times C_P \times T_{Exhaust}$$

$$Q_{Exhaust} = (\dot{m}_{Fuel} + \dot{m}_{Air}) \times C_P \times T_{Exhaust}$$

$$\Delta P = \rho \times g \times h$$

The pressure drop is calculated as:

$$\Delta P = 721(kg/m^3) \times 9.81(m/s^2) \times \left(\frac{430}{1000}\right)(m)$$

$$\Delta P = 3041.3943\ Pa$$

The velocity is calculated as:

$$V = \sqrt{\frac{2\Delta P}{\rho_{Air}}}$$

$$V = \sqrt{\frac{2 \times 3041.3943(Pa)}{1.15(kg/m^3)}}$$

$$V = 72.72813342\, m/s$$

The volume flow rate is calculated as:

$$Q = C_d \times \pi \times \frac{d^2}{4} \times V$$

$$Q = 0.686 \times \pi \times \left(\frac{0.03^2}{4}\right)(m^2) \times 72.72813342(m/s)$$

$$Q = 0.035266223\ m^3/s$$

The mass air flow rate is calculated as:

$$\dot{m}_{Air} = \rho_{fuel} \times Q$$

$$\dot{m}_{Air} = 1.15(kg/m^3) \times 0.035266223(m^3/s)$$

$$\dot{m}_{Air} = \mathbf{0.040556156\ kg/s}$$

Where:

ΔP = Pressure drop across orifice meter (Pa)

$\rho = 721\,\mathrm{kg/m^3}$

$h = 400\mathrm{mm}$

V = Flow velocity (m/s)

$\rho_{Air} = 1.15\,\mathrm{kg/m^3}$

Q = Volumetric Flow Rate $(\mathrm{m^3/s})$

$C_d = 0.686$

$d = 30\mathrm{mm}$

The exhaust gases temperature $T_{Exhaust}$ is estimated as:

$$T_{Exhaust} = T_{EVO} \times \left(\frac{P_{Exhaust}}{P_{EVO}}\right)^{(k-1)/k}$$

Pressure of exhaust gases is calculated as:

$$P_{EVO} = P_{Max} \times \left(\frac{1}{r_e}\right)^{k}$$

$$P_{EVO} = 6705(kPa) \times \left(\frac{1}{9.2}\right)^{1.35}$$

$$P_{EVO} = \mathbf{335.1853941}\ \boldsymbol{kPa}$$

$$P_{EVO} = \rho \times R \times T_{EVO}$$

Temperature of exhaust gas is calculated as:

$$T_{EVO} = \frac{P_{EVO}}{\rho \times R}$$

$$T_{EVO} = \frac{335.1853941(kPa)}{1.15(kg/m^3) \times 0.287(kJ/kg.K)}$$

$$T_{EVO} = \mathbf{1015.559746\ K}$$

$$T_{Exhaust} = 1015.559746(K) \times \left(\frac{101(kPa)}{335.1853941(kPa)}\right)^{(1.35-1)/1.35}$$

$$T_{Exhaust} = \mathbf{744.68863}\ K$$

The heat loss through exhaust is then become:

$$Q_{Exhaust} = (\dot{m}_{Fuel} + \dot{m}_{Air}) \times C_P \times T_{Exhaust}$$

$$Q_{Exhaust} = \left(0.00319(kg/s) + 0.04055(kg/s)\right) \times 1.08(kJ/kg.K) \times 744.68863(K)$$

$$Q_{Exhaust} = \mathbf{35.37250036\ kW}$$

Where:

P_{EVO} = Pressure when exhaust valve open (kPa)

P_{Max} = 6705kPa

$r_e = 9.2$

$k = 1.35$

$\rho = 1.15\ kg/m^3$

$R = 0.287\ kJ/kg.K$

The percentage of heat losses through exhaust:

$$Percentage\ (\%) = \frac{Q_{Exhaust}}{Q_{fuel}} \times 100\%$$

$$Percentage\ (\%) = \left(\frac{35.37250036\ (kW)}{127.7256858\ (kW)}\right) \times 100\%$$

$$Percentage\ (\%) = \mathbf{27.69411661}\%$$

3.4 METHOD OF IDENTIFYING INFLUENTIAL FACTOR

In chapter one, the second objective to achieve in this study is to identify the most influential factor that contributes to the heat losses in engine operation. The relationship between A and B means the relationship between the amount of energy losses and the factor of energy losses. The known independent parameter are the engine speed, ignition timing, fuel flow rate, air flow rate, coolant flow rate, and air fuel ratio and outlet coolant temperature. The amount of energy losses through the cooling system, exhaust system and friction are selected as the dependent variable. The plot of graph dependent variable versus independent variable can show the trend and relationship between each other. Graph plot A-B which has proportional or inversely relationship have selected as the factor influence the heat losses.

3.5 SUMMARY

This chapter has unveiled the underlying approach to analyze the data from the previous experimental result. In the first part of the contents, (a) heat balance concept, (b) process of energy distribution, and (c) sample of calculation. The second part is described the method of influential factor. The understanding heat balance concept is important in order to know how to calculate and analyze the data.

CHAPTER 4

RESULT AND DISCUSSION

4.1 INTRODUCTION

This chapter presents the data analysis of collected results from the previous experimental work. Analysis composed of calculation of the mass fuel flow rate, air fuel flow rate, exhaust gases temperature, indicated power, brake power, friction power, heat losses through the exhaust system and cooling system. Result of heat balance analysis at different engine speed was presented as a bar chart. In the next section, the results for identification of influential factor of heat losses are also presented.

4.2 ANALYSIS OF EXPERIMENTAL DATA

The previous study was carried out data at engine speed of 1500 rpm to 4000 rpm with speed interval of 500 rpm. The engine was run at wide-open throttle condition and maximum engine torque. Controlled variable is the throttle opening; independent variable is engine speed while torque and other parameters are set as dependent variable. Important parameter for heat balance analysis are engine torque, fuel consumption time, coolant flow rate, inlet coolant temperature and coolant outlet temperature.

4.2.1 Calculated Results

Table 4.1, 4.2, 4.3, 4.4, 4.5, 4.6, and 4.7 present the results of data analysis using the equation such as in the sample of calculation at 4000 rpm.

Table 4.1: Heat of Combustion at Different Engine Speed

Engine Speed (RPM)	Fuel Consumption Time (Sec)	Rate of Fuel Consumption (kg/s)	Heat of Combustion (kW)
1500	19.66	0.001953891	78.15564598
2000	16.72	0.002297458	91.89832536
2500	16.44	0.002336588	93.46350365
3000	14.5	0.002649207	105.9682759
3500	12.35	0.003110405	124.4161943
4000	12.03	0.003193142	127.7256858

Table 4.1 present the result for fuel consumption time, rate of fuel consumption, and heat of combustion at different engine speed. Based on the table, trend of fuel consumption time is decrease as engine speed increase. From calculation work, result for rate of fuel consumption and heat of combustion are increase as engine speed increased.

Table 4.2: Brake Power at Different Engine Speed

Engine Speed (RPM)	Engine Torque (Nm)	Brake Power (kW)	Percentage of Brake Power (%)
1500	101	15.8650429	20.29929214
2000	103	21.57226955	23.47406166
2500	110	28.79793266	30.81195497
3000	106	33.30088213	31.42533164
3500	108	39.58406744	31.81584813
4000	107	44.82005519	35.09087065

Table 4.2 present the result for engine torque and brake power at different engine speed. Based on the table, engine torque increase from 1500 rpm to 2500 rpm and then decrease at 3000 rpm. At 3500 rpm, engine torque become increase and decrease again at 4000 rpm. The overall trend of brake power and percentage of brake power almost increase as engine speed increased.

Table 4.3: Coolant Losses at Different Engine Speed

Engine Speed (RPM)	Coolant Flow rate (LPM)	Coolant In Temp (Deg C)	Coolant Out Temp (Deg C)	Coolant Losses (kW)	Percentage of Coolant (%)
1500	3	34	60	5.434	6.952792638
2000	5	38	66	9.753333333	10.61317853
2500	5	39	70	10.79833333	11.55352936
3000	5	40	77	12.88833333	12.16244506
3500	9	50	82	20.064	16.12651802
4000	15	62	89	28.215	22.09031005

Table 4.3 present the result for coolant flow rate, inlet coolant temperature, outlet coolant temperature, and coolant losses at different engine speed. The coolant flow rate start increase from 1500 rpm to 2000 rpm and then start constant until 3000 rpm. At 3000 rpm, the coolant flow rate increase again to 4000 rpm. The overall trend of inlet coolant temperature, outlet coolant temperature, coolant losses, and percentage of coolant losses almost increase as engine speed increase.

Table 4.4: Friction Losses at Different Engine Speed

Engine Speed (RPM)	Indicated Mean Effective Pressure (kPa)	Indicated Work (kW)	Friction Losses (kW)	Percentage of Friction (%)
1500	890	16.554	0.688957	0.88152
2000	900	22.32	0.74773	0.81365
2500	960	29.76	0.962067	1.02935
3000	970	36.084	2.783118	2.62637
3500	1000	43.4	3.815933	3.06707
4000	1050	52.08	7.259945	5.68401

Table 4.4 present the result for indicated mean effective pressure, indicated work, and friction losses at different engine speed. Based on the table, indicated mean effective increase as engine speed increased. The overall trend of indicated work, friction losses and percentage of friction loss are increase from 1500 rpm to 4000 rpm.

Table 4.5: Mass Air Flow Rate at Different Engine Speed

Engine Speed (RPM)	Height Difference of manometer (mm)	Pressure Drop Across Orifice (Pa)	Flow Velocity (m/s)	Volumetric Flow Rate (m^3/s)	Mass Air Flow Rate (kg/s)
1500	128.64	909.872	39.77922	0.01929	0.02218
2000	165.29	1169.098	45.09117	0.02186	0.02514
2500	179.21	1267.554	46.95149	0.02277	0.02618
3000	253.04	1789.754	55.79083	0.02705	0.03111
3500	306.34	2166.746	61.3861	0.02977	0.03423
4000	380.22	2689.3	68.38891	0.03316	0.03814

Table 4.5 present the result for flow velocity, volumetric flow rate, and mass air flow rate. Result of height difference in manometer and pressure drop across orifice also show in the table. Based on the table, height difference in manometer and pressure drop across orifice are increases as engine speed increased. The overall trend of flow velocity, volumetric flow rate, and mass air flow rate are increases from 1500 rpm to 4000 rpm.

Table 4.6: Exhaust Losses at Different Engine Speed

Engine Speed (RPM)	Pressure Max (kPa)	Pressure when EVO (kPa)	Temperature when EVO (K)	Exhaust Temperature (K)
1500	5646	282.2	855.16	655.651
2000	6119	305.9	926.802	695.911
2500	6374	318.6	965.425	717.279
3000	6227	311.3	943.16	704.989
3500	6254	312.6	947.25	707.252
4000	6350	317.4	961.79	715.278

Table 4.6 present the result for pressure maximum and exhaust temperature at different engine speed. Result of pressure and temperature when exhaust valve open also show in the table. Based on the table, pressure maximum and exhaust temperature increase as engine speed increased. The overall trend of pressure and temperature when exhaust valve open are also increase as engine speed increased from 1500 rpm to 4000 rpm.

Table 4.7: Exhaust Gases Temperature at Different Engine Speed

Engine Speed (RPM)	Exhaust Flow Rate (kg/s)	Exhaust Temp (K)	Exhaust Losses (kW)	Percentage of Exhaust (%)
1500	0.024	655.651	16.925098	21.65563
2000	0.028	695.911	20.776717	22.60837
2500	0.029	717.279	22.119466	23.66642
3000	0.034	704.989	25.936701	24.47591
3500	0.038	707.252	28.59372	22.98231
4000	0.044	715.278	33.975489	26.60036

Table 4.7 present the result for exhaust flow rate and losses through the exhaust system at different engine speed. Result of exhaust temperature also shows in the table. Based on the table, exhaust temperature increase as engine speed increased. The overall trend of exhaust flow rate and losses through the exhaust system are also increase as engine speed increased from 1500 rpm to 4000 rpm.

4.2.2 Heat Balance Diagram

In present study, heat balance analysis has been performed theoretically for gasoline engines at different engine speed of wide open throttle condition. From the various numerical applications, it is determined that the portion of available fuel energy is converted to useful work, and the rest of total fuel energy is lost by exhaust gases or lost by heat transfer. For example; for gasoline engine given by Ganesan (2000), approximately 20% of the fuel energy is converted to useful indicated work, approximately 33% of the fuel energy is carried away by exhaust gases approximately 33% of the fuel energy are spent for coolant system and approximately 10 % loss to the friction. Figure 4.1, 4.2, 4.3, 4.4, 4.5, and 4.6 present the results of heat balance analysis at different engine speed.

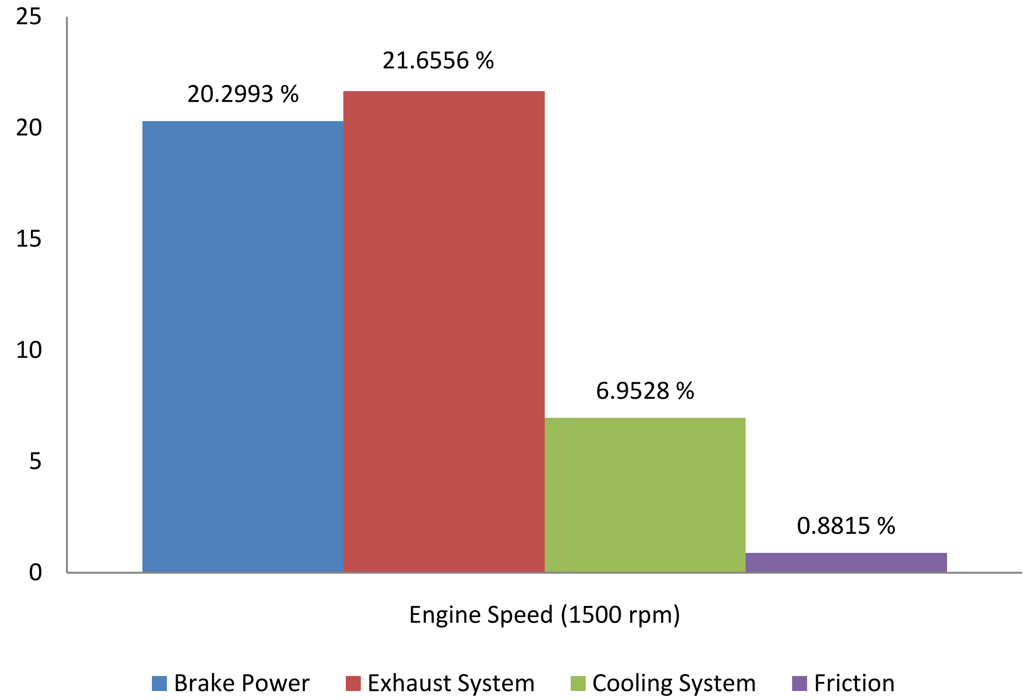

Figure 4.1: Heat Balance Bar Chart at 1500 rpm

Figure 4.1 present the heat balance bar chart at 1500 rpm. The percentage of fuel combustion that converts to useful work as brake power is about 20.3%. The rest was loss through the exhaust system (21.7%), cooling system (7.0%) and friction (0.9%). The bar chart presents the highest heat losses in engine operation is exhaust system. Besides that, the cooling system also contributes intermediate percentage of heat losses while a friction loss presents lowest percentage of heat losses.

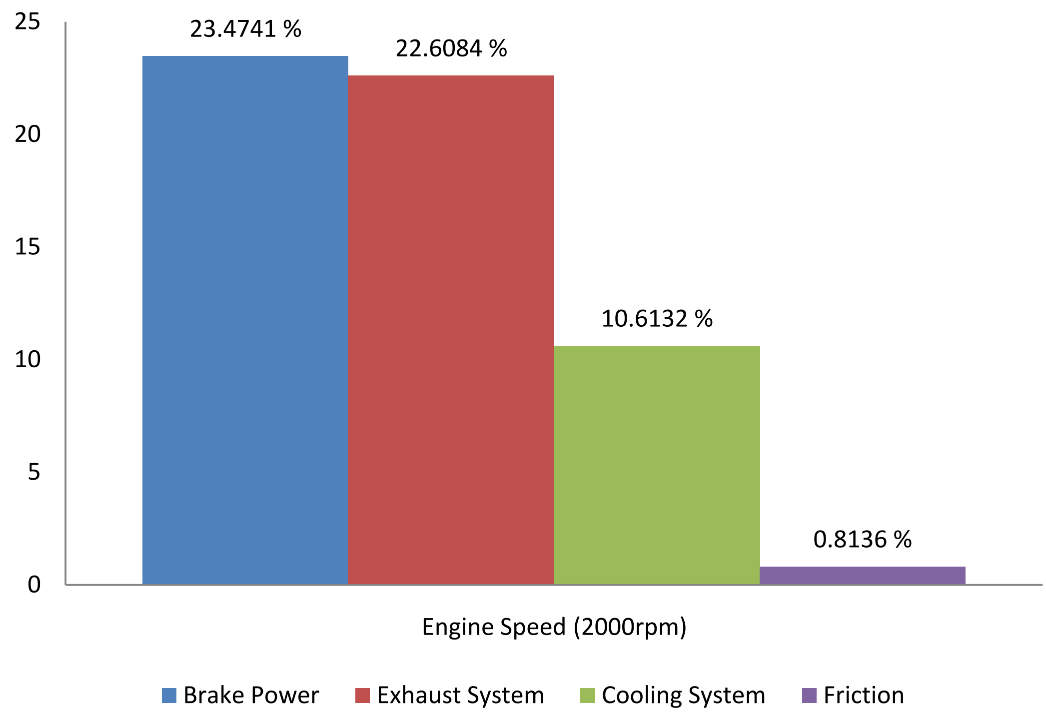

Figure 4.2: Heat Balance Bar Chart at 2000 rpm

Figure 4.2 present the heat balance bar chart at 2000 rpm. The percentage of fuel combustion that converts to useful work as brake power is about 23.5%. The rest was loss through the exhaust system (22.6%), cooling system (10.6%) and friction (0.8%). Based on the chart, the overall trend of energy balance for brake power, exhaust losses, cooling losses, and friction losses almost increase as engine speed increased. The percentage of brake power increase is about 3.2%. In exhaust system, there is increase in 0.9% while in cooling system; there is increase about 3.6%. The percentage of friction losses is decrease for 0.1%.

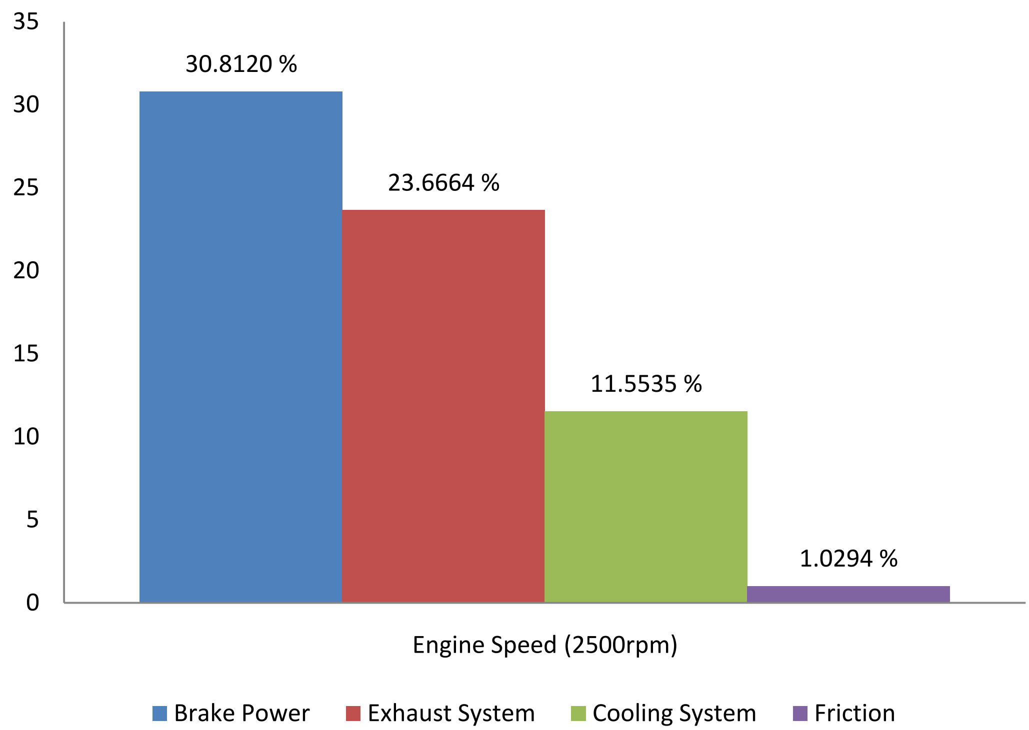

Figure 4.3: Heat Balance Bar Chart at 2500 rpm

Figure 4.3 present the heat balance bar chart at 2500 rpm. The percentage of fuel combustion that converts to useful work as brake power is about 30.8%. The rest was loss through the exhaust system (27.3%), cooling system (11.6%) and friction (1.0%). From the bar chart, the trend presents the percentage of energy balance for brake power, exhaust losses, cooling losses, and friction losses almost increase as engine speed increased. The percentage of brake power increase is about 7.3%. In exhaust system, there is increase in 1.1% while in cooling system; there is increase about 1.0%. The percentage of friction losses is also increase for 0.2%.

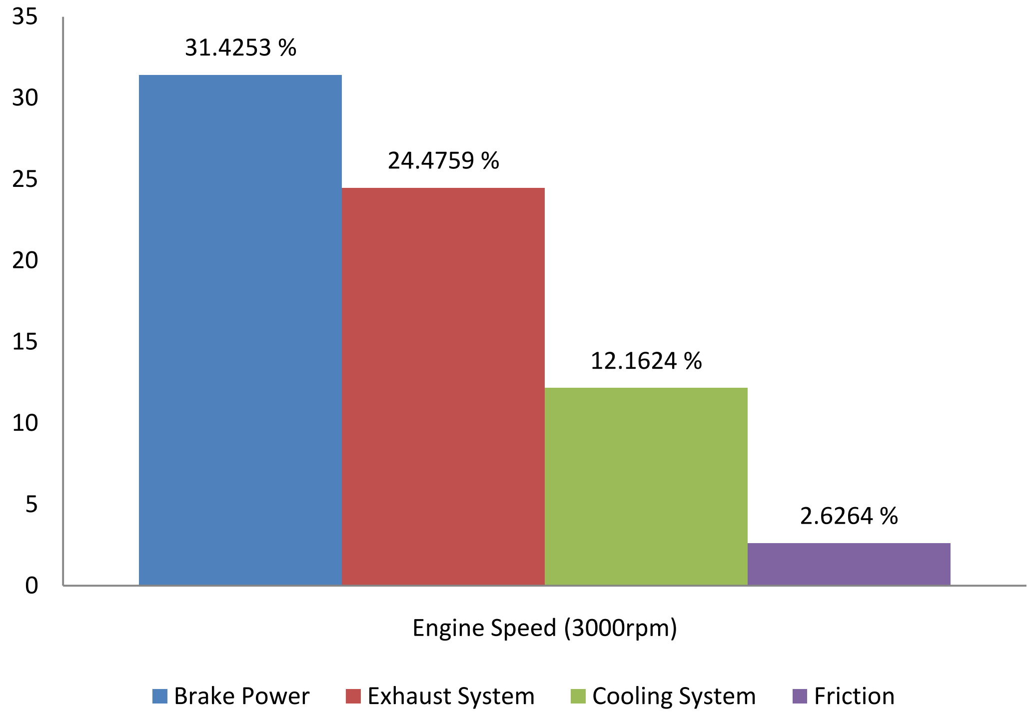

Figure 4.4: Heat Balance Bar Chart at 3000 rpm

Figure 4.4 present the heat balance bar chart at 3000 rpm. The percentage of fuel combustion that converts to useful work as brake power is about 31.4%. The rest was loss through the exhaust system (24.5%), cooling system (12.2%) and friction (2.6%). From the bar chart, the trend presents the percentage of energy balance for brake power, exhaust losses, cooling losses, and friction losses almost increase as engine speed increased. The percentage of brake power increases about 0.6%. In exhaust system, there is increase in 0.8% while in cooling system; there is increase about 0.6%. The percentage of friction losses is also increase for 1.6%.

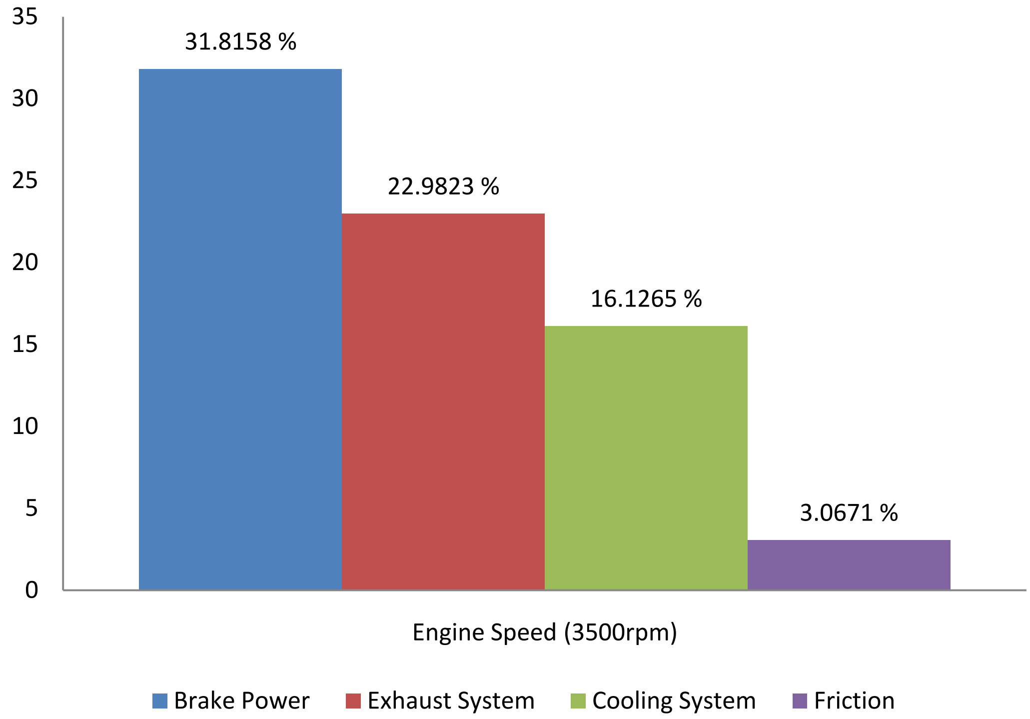

Figure 4.5: Heat Balance Bar Chart at 3500 rpm

Figure 4.5 present the heat balance bar chart at 3500 rpm. The percentage of fuel combustion that converts to useful work as brake power is about 31.8%. The rest was loss through the exhaust system (23.0%), cooling system (16.1%) and friction (3.1%). From the bar chart, the trend presents the percentage of energy balance for brake power, exhaust losses, cooling losses, and friction losses almost increase engine speed increased. The percentage of brake power increases is about 0.4%. In exhaust system, there is decrease in 1.5% while in cooling system; there is increase about 3.9%. The percentage of friction losses is also increase for 0.5%.

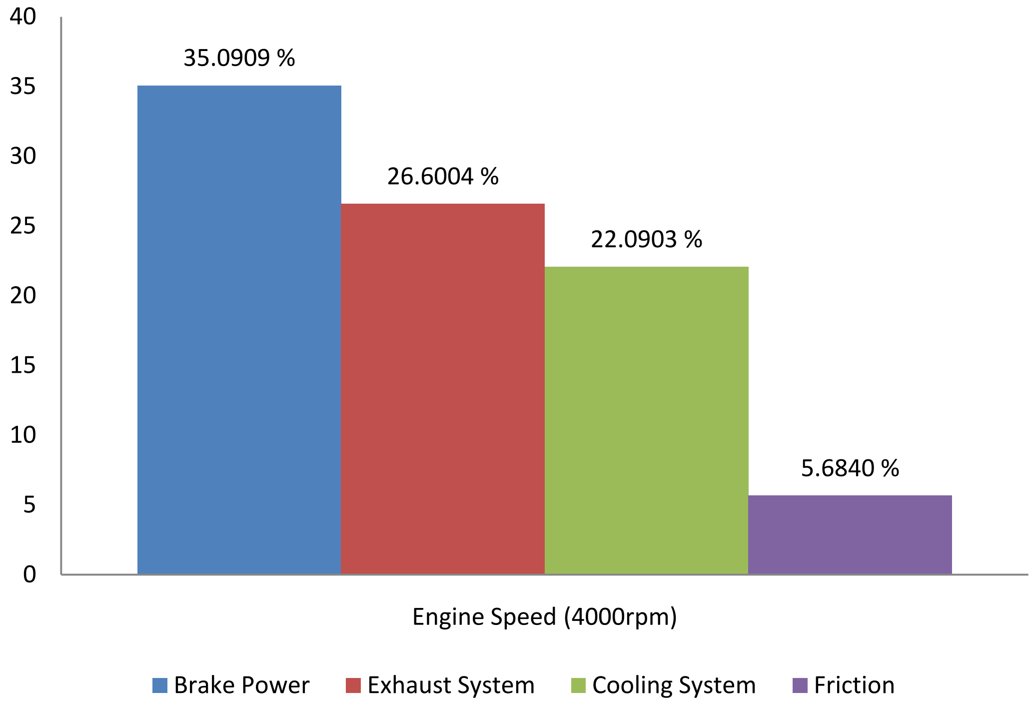

Figure 4.6: Heat Balance Bar Chart at 4000 rpm

Figure 4.6 present the heat balance bar chart at 4000 rpm. The percentage of fuel combustion that converts to useful work as brake power is about 35.1%. The rest was loss through the exhaust system (26.6%), cooling system (22.1%) and friction (5.7%). From the bar chart, the trend presents the percentage of energy balance for brake power, exhaust losses, cooling losses, and friction losses almost increase as engine speed increased. The percentage of brake power increase is about 3.3%. In exhaust system, there is increase in 3.6% while in cooling system; there is increase about 6.0%. The percentage of friction losses is also increase for 2.6%.

4.3 FACTOR OF THE HEAT LOSSES

From the heat balance analysis, the graph of heat losses through the exhaust system, cooling system and friction at different engine speed as the dependent variable versus the factor that influence the heat loss are plot.

4.3.1 Cooling Losses

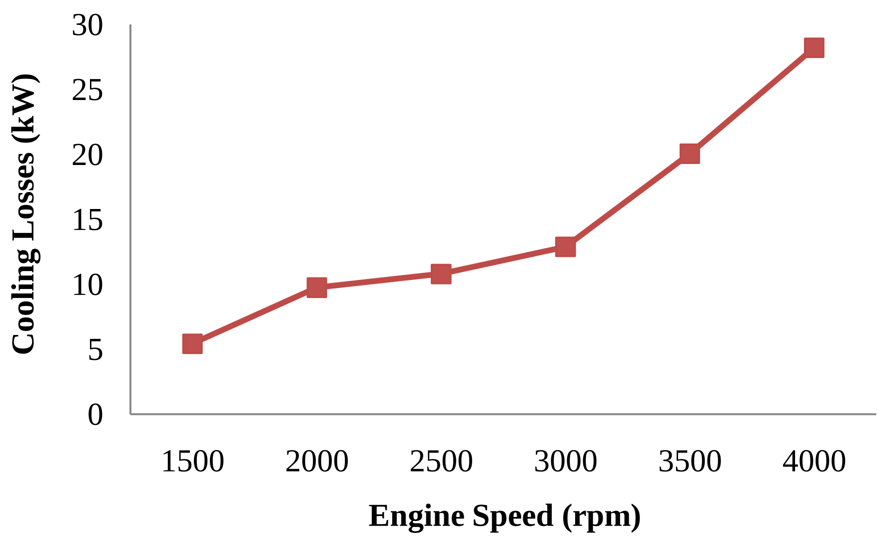

Figure 4.7: Effect of Engine Speed to Cooling Losses

Figure 4.7 present the effect of engine speed to cooling system. The trends of the graph show the cooling losses are proportional to engine speed. Energy losses through the cooling system slightly increase between 1500 rpm to 3000 rpm. At 3000 rpm, coolant losses start increase rapidly until 4000 rpm. The higher energy losses through coolant system occur at engine speed of 4000 rpm and there are about 28.2 kW. The coolant losses extremely low at engine speed of 1500 rpm and there are about 5.4 kW.

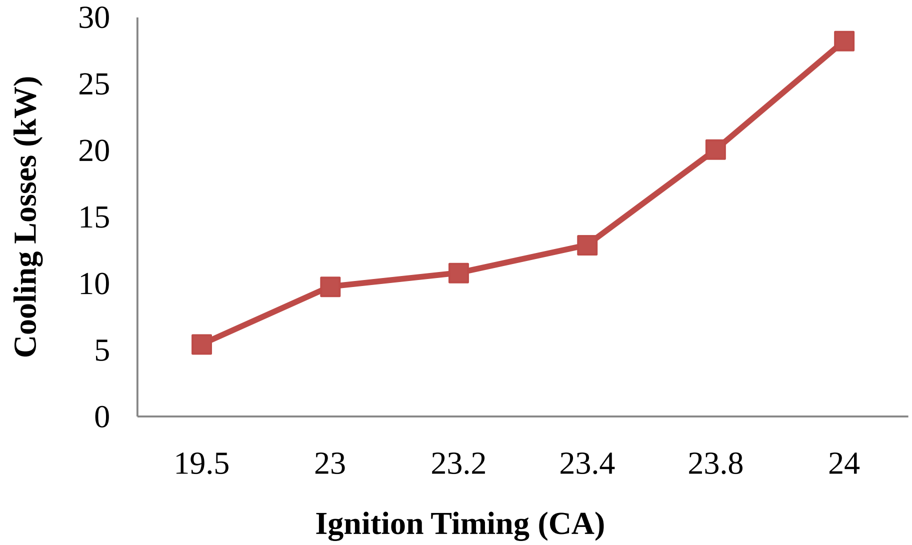

Figure 4.8: Effect of Ignition Timing to Cooling Losses

Figure 4.8 present the effect of ignition timing to cooling system. The trends of the graph show the cooling losses are proportional to ignition timing. Energy losses through the cooling system increase slightly between 19.5 ˚CA to 23.4 ˚CA. At 23.4 ˚CA, coolant losses start increase rapidly until 24.0 ˚CA. The higher energy losses through coolant system occur at ignition timing of 24.0 ˚CA and there are about 28.2 kW. The coolant losses extremely low at ignition timing of 19.5 ˚CA and there are about 5.4 kW.

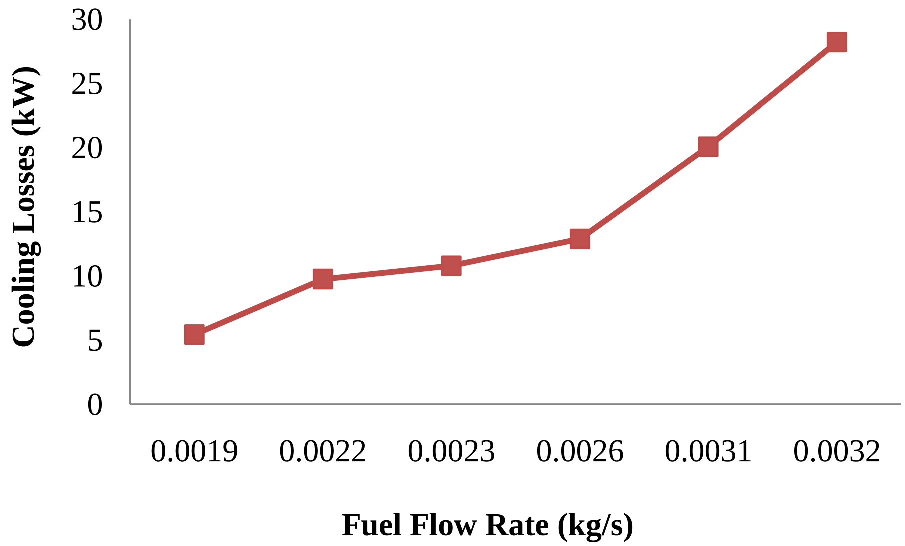

Figure 4.9: Effect of Fuel Flow Rate to Cooling Losses

Figure 4.9 present the effect of fuel flow rate to cooling system. The trends of the graph show the cooling losses are proportional to fuel flow rate. Energy losses through the cooling system increase slightly between 0.0019 kg/s to 0.0026 kg/s. At 0.0026 kg/s, coolant losses start increase rapidly until 0.0032 kg/s. The higher energy losses through coolant system occur at fuel flow rate of 0.0032 kg/s and there are about 28.2 kW. The coolant losses extremely low at fuel flow rate of 0.0019 kg/s and there are about 5.4 kW.

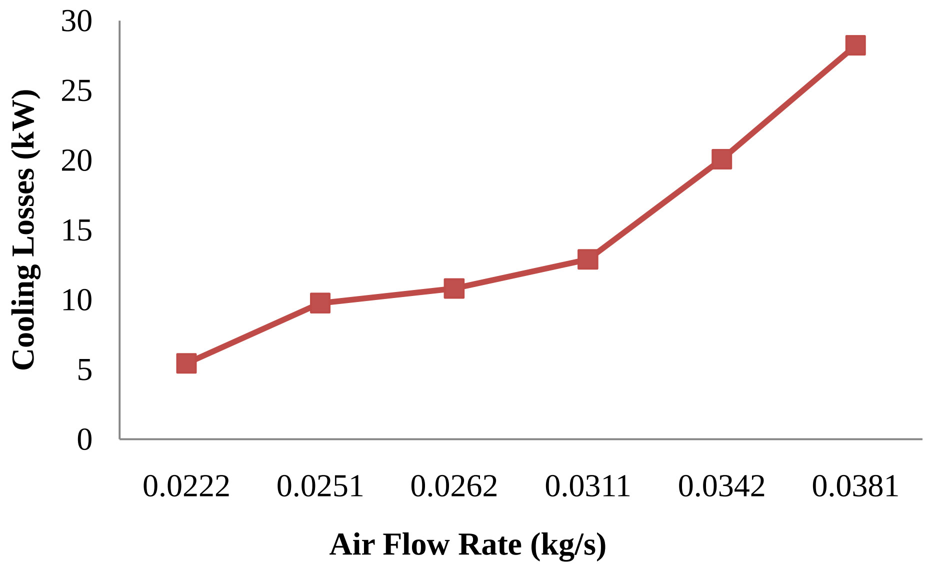

Figure 4.10: Effect of Air Flow Rate to Cooling Losses

Figure 4.10 present the effect of air flow rate to cooling system. The trends of the graph show the cooling losses are proportional to air flow rate. Energy losses through the cooling system increase quietly between 0.0222 kg/s to 0.0311 kg/s. At 0.0311 kg/s, coolant losses start increase rapidly until 0.0381 kg/s. The higher energy losses through coolant system occur at air flow rate of 0.0381 kg/s and there are about 28.2 kW. The coolant losses extremely low at fuel flow rate of 0.0222 kg/s and there are about 5.4 kW.

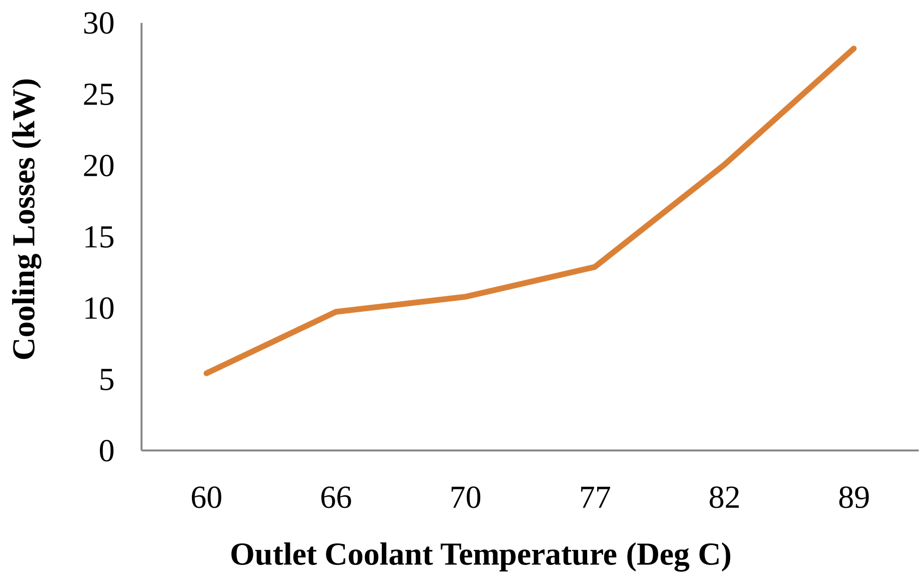

Figure 4.11: Effect of Outlet Coolant Temperature to Cooling Losses

Figure 4.11 present the effect of outlet coolant temperature to cooling system. The trends of the graph show the cooling losses are proportional to outlet coolant temperature. Energy losses through the cooling system increase quietly between 60 ˚C to 77 ˚C. At 77 ˚C, coolant losses start increase rapidly until 89 ˚C. The higher energy losses through coolant system occur at outlet coolant temperature of 89 ˚C and there are about 28.2 kW. The coolant losses extremely low at fuel flow rate of 60 ˚C and there are about 5.4 kW.

The most factors that influence the coolant losses are outlet coolant temperature. When the engine temperature is same or cooler than ambient temperature, the valve in thermostat is closed. The increase engine temperature cause wax in the cylinder to expand to open the valve in thermostat. This allows coolant to flow from water pump through the cylinder block and back to radiator. When the engine temperature increase, heat absorb by engine coolant become increase. As the result, the outlet coolant temperature increase and cause the increasing of heat losses through the cooling system.

4.3.2 Exhaust Losses

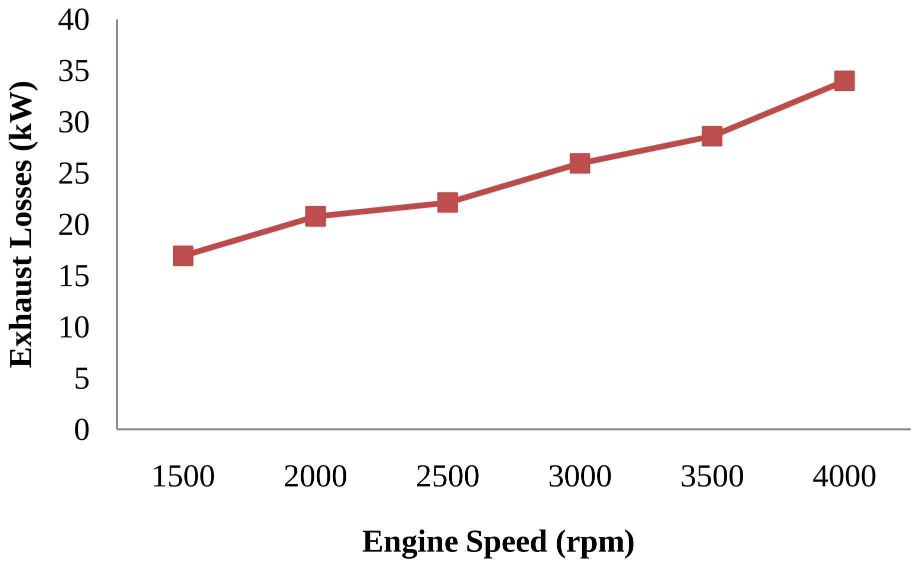

Figure 4.12: Effect of Engine Speed to Exhaust Losses

Figure 4.12 present the effect of engine speed to exhaust system. The trends of the graph show the exhaust losses are proportional to engine speed. Energy losses through the exhaust system increase slightly between 2000 rpm to 3500 rpm. At 3500 rpm, exhaust losses start increase rapidly until 4000 rpm. The higher energy losses through exhaust system occur at engine speed of 4000 rpm and there are about 34.0 kW. The exhaust losses extremely low at engine speed of 1500 rpm and there are about 16.9 kW.

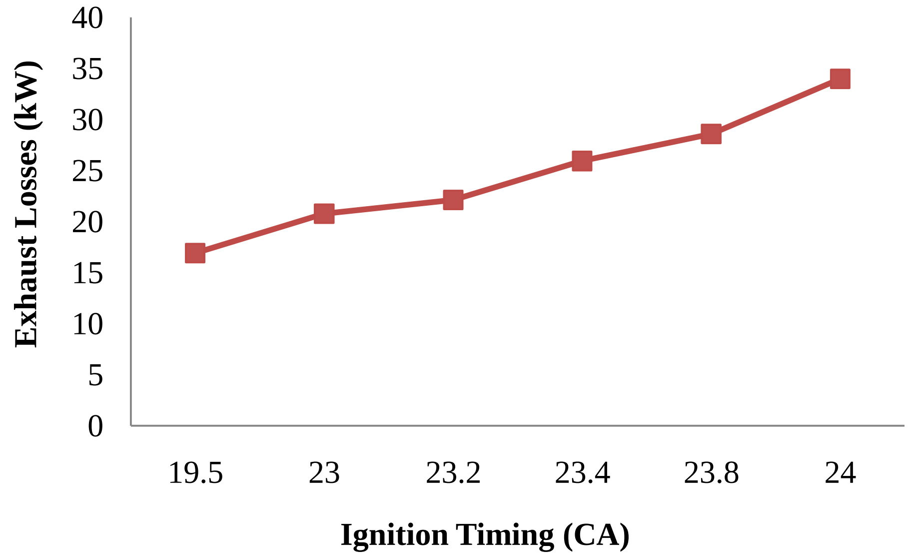

Figure 4.13: Effect of Ignition Timing to Exhaust Losses

Figure 4.13 present the effect of ignition timing to exhaust system. The trends of the graph show the exhaust losses are proportional to ignition timing. Energy losses through the exhaust system increase quietly between 23 ˚CA to 23.8 ˚CA. At 23.8 ˚CA, exhaust losses start increase rapidly until 24 ˚CA. The higher energy losses through exhaust system occur at ignition timing of 24 ˚CA and there are about 34.0 kW. The exhaust losses extremely low at ignition timing of 19.5 ˚CA and there are about 16.9 kW.

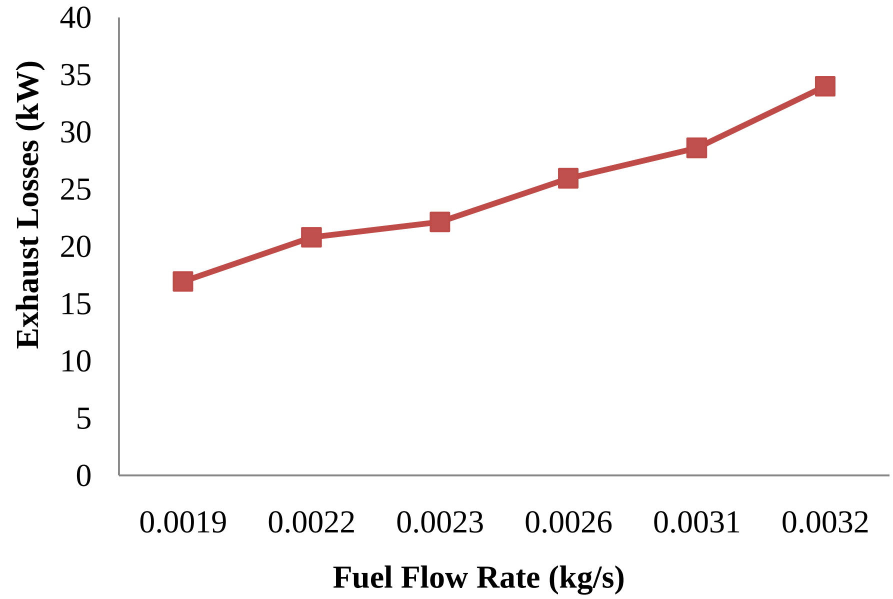

Figure 4.14: Effect of Fuel Flow Rate to Exhaust Losses

Figure 4.14 present the effect of fuel flow rate to exhaust system. The trends of the graph show the exhaust losses are proportional to fuel flow rate. Energy losses through the exhaust system increase slightly between 0.0022 kg/s to 0.0031 kg/s. At 0.0031 kg/s, exhaust losses start increase rapidly until 0.0032 kg/s. The higher energy losses through exhaust system occur at fuel flow rate of 0.0032 kg/s and there are about 34.0 kW. The exhaust losses extremely low at fuel flow rate of 0.0019 kg/s and there are about 16.9 kW.

The most factors that influence the exhaust losses are fuel flow rate. The air fuel ratio for combustion will decrease when the fuel flow rate increase. The equivalence ratio become rich by the lower of air fuel ratio. At a given compression ratio the temperature after combustion reaches a maximum when the mixture slightly rich. Increase of maximum temperature at high peak pressure, the higher exhaust gases temperature is produced. Higher exhaust gases temperature contribute the increasing of losses through the exhaust system.

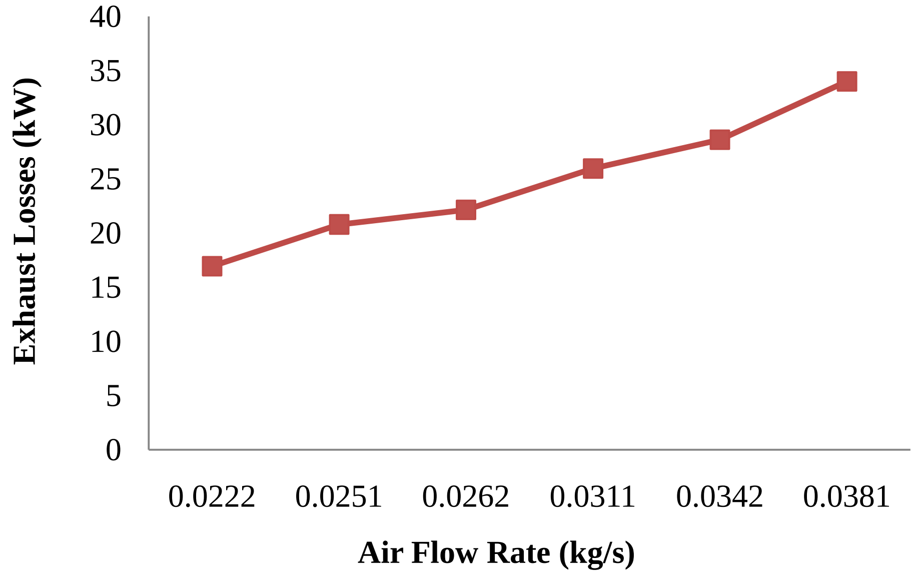

Figure 4.15: Effect of Air Flow Rate to Exhaust Losses

Figure 4.15 present the effect of air flow rate to exhaust system. The trends of the graph show the exhaust losses are proportional to air flow rate. Energy losses through the exhaust system increase slightly between 0.0251 kg/s to 0.0342 kg/s. At 0.0342 kg/s, exhaust losses start increase rapidly until 0.0381 kg/s. The higher energy losses through exhaust system occur at air flow rate of 0.0381 kg/s and there are about 34.0 kW. The exhaust losses extremely low at fuel flow rate of 0.0222 kg/s and there are about 16.9 kW.

4.3.3 Friction Losses

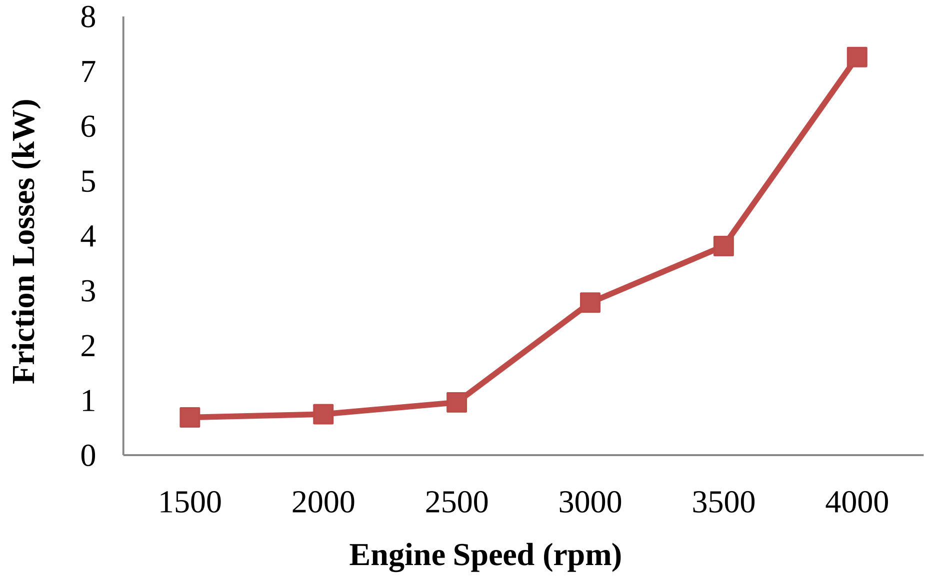

Figure 4.16: Effect of Engine Speed to Friction Losses

Figure 4.16 present the effect of engine speed to friction power. The trends of the graph show the friction losses are proportional to engine speed. Energy losses through the friction power increase quietly between 1500 rpm to 2500 rpm. At 2500 rpm, friction losses start increase rapidly until 4000 rpm. The higher energy losses through friction power occur at engine speed of 4000 rpm and there are about 7.3 kW. The friction losses extremely low at engine speed of 1500 rpm and there are about 0.7 kW.

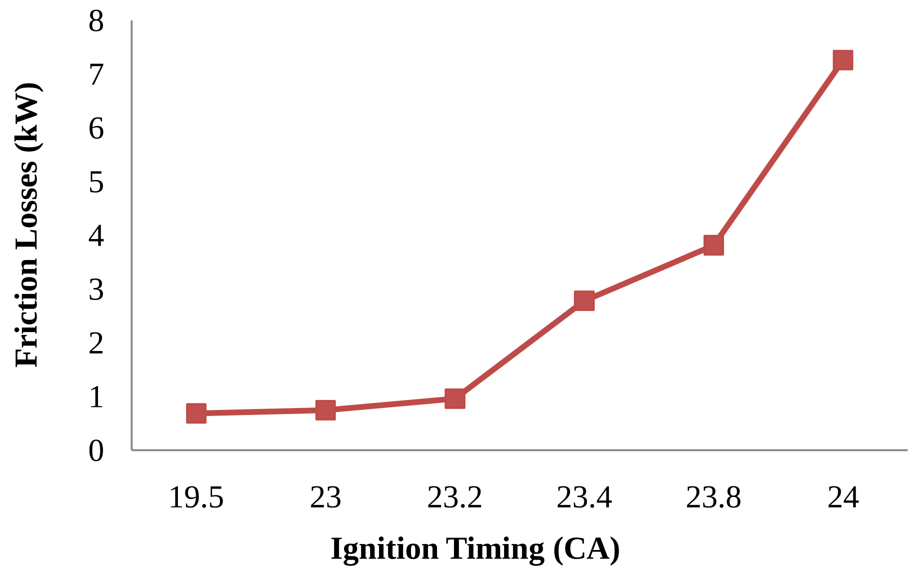

Figure 4.17: Effect of Ignition Timing to Friction Losses

Figure 4.17 present the effect of ignition timing to friction power. The trends of the graph show the friction losses are proportional to ignition timing. Energy losses through the friction power increase quietly between 19.5 ˚CA to 23.2 ˚CA. At 23.2 of crank angle, friction losses start increase rapidly until 24 ˚CA. The higher energy losses through friction power occur at ignition timing of 24 ˚CA and there are about 7.3 kW. The friction losses extremely low at ignition timing of 19.5 ˚CA and there are about 0.7 kW.

The most influential factors that influence the friction losses are ignition timing. Ignition timing parameter influenced the resultant peak pressure and the timing for peak pressure. Usually the spark should occur at about 15 degrees before TDC. When the spark occur too early the combustion take place before the compression stroke is completed and the pressure so developed would oppose the piston motion. As the result, large works transferred to the piston and increase the friction. If the spark occurs too late, the piston would have already completed a certain part of the expansion stroke before the pressure arise occurs and a corresponding amount of engine power is lost.

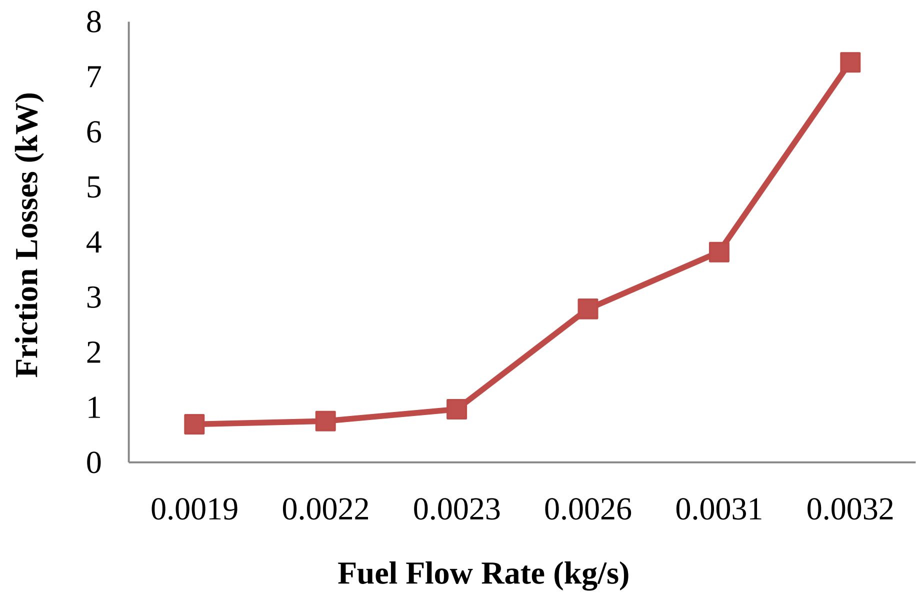

Figure 4.18: Effect of Fuel Flow Rate to Friction Losses

Figure 4.18 present the effect of fuel flow rate to friction power. The trends of the graph show the friction losses are proportional to fuel flow rate. Energy losses through the friction power increase quietly between 0.0019 kg/s to 0.0023 kg/s. At 0.0023 kg/s, friction losses start increase rapidly until 0.0032 kg/s. The higher energy losses through exhaust system occur at fuel flow rate of 0.0032 kg/s and there are about 7.3 kW. The exhaust losses extremely low at fuel flow rate of 0.0019 kg/s and there are about 0.7 kW.

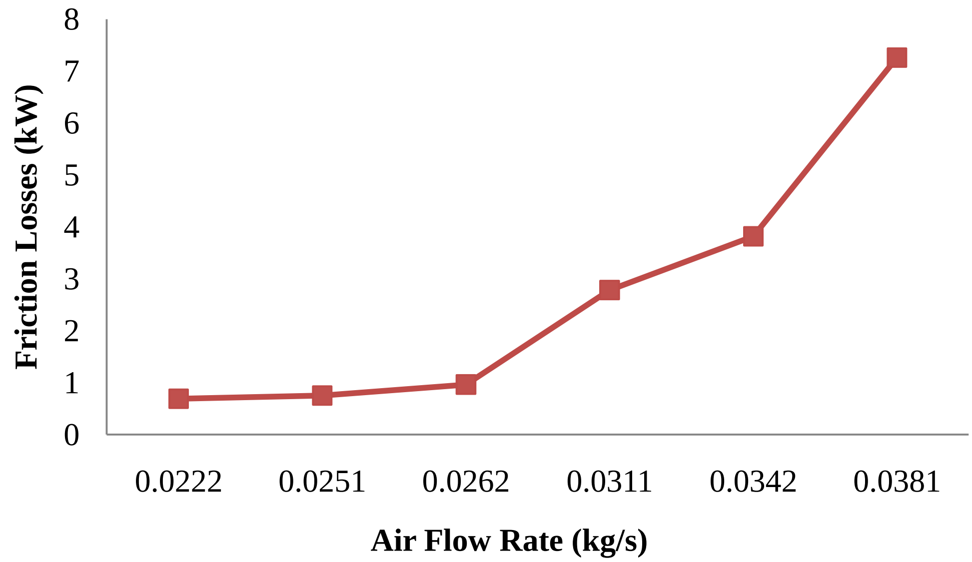

Figure 4.19: Effect of Air Flow Rate to Friction Losses

Figure 4.19 present the effect of air flow rate to friction power. The trends of the graph show the friction losses are proportional to air flow rate. Energy losses through the friction power increase quietly between 0.0222 kg/s to 0.0262 kg/s. At 0.0262 kg/s, exhaust losses start increase rapidly until 0.0381 kg/s. The higher energy losses through friction power occur at air flow rate of 0.0381 kg/s and there are about 7.3 kW. The friction losses extremely low at fuel flow rate of 0.0222 kg/s and there are about 0.7 kW.

4.4 SUMMARY

This chapter has presented the result from analysis of data. The result was including the percentage and amount of heat of fuel combustion convert as a useful work and loss through the coolant, exhaust, and friction. Bar chart diagram present the trend of energy distribution at different engine speed of wide open throttle condition. In this chapter also, the factor of heat losses through the coolant, exhaust, and friction was present as a graph.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

5.1 CONCLUSION

The overall study has conducted energy balance study on four stroke cylinder in-line Mitsubishi Magma (4G15) 1.5 liters spark ignition gasoline engine. The combustion of energy fuel transferred as useful work at brake power and heat losses to cooling system, exhaust system and friction. Figure 3.1 present the distribution of energy in a typical Spark Ignition engines as a function of engine speed. The energy balance in the engine operation is different at low, medium and high engine speed. Analysis of the experimental data is very important in order to see the different of the heat balance at different engine speed. The sample calculation at 4000 rpm of engine speed is show in chapter 3. Only 30% of the combustion fuels are converted to useful work as a brake power and about 70% are loss. Figure 4.1, 4.2, 4.3, 4.4, 4.5, and 4.6 present the comparison of the heat balance percentages at different engine speed. As example, in figure 4.1, about 20.3% are converted as brake power, 21.7% loss to exhaust, 7% loss to cooling system and 0.9% loss to friction power. As engine speed increases, the indicated power increases while brake power increases to maximum and then decrease and the friction power increases and become dominant at higher speeds. On the other hand, as engine speed increases, exhaust temperature increases and thus, exhaust losses increase. Moreover, mechanical losses increase with increasing engine speed. The graph trend is important in order to determine the factor of heat losses. Figure 4.7 until figure 4.19 show the trend of the graph is directly proportional. The parameter such as engine speed, ignition timing, fuel flow rate, and air flow rate and outlet coolant temperature are the most factor that influence higher losses to coolant, exhaust and friction. This factor is very important to future work in order to improve the power output, minimizing the energy losses in engine operation and reduced the emission from the exhaust gases system.

5.2 RECOMMENDATIONS FOR FUTURE WORKS

Vast knowledge and experiences gathered from the study have suggested the following recommendations for further improve the experimental work in term of the whole system to get the accurate data. In this study, the value of the exhaust gas temperature needs to estimate. In addition to this, some emission testing equipment is scheduled to be added to the diagnostic testing system. Monitoring exhaust gas constituents might aid in understanding both the exhaust temperature trends as well as energy losses through the exhaust. Finally, running an energy balance test in future at different engine operation could give a good indication of the data.

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