Engine Performance and Exhaust Emission Characteristics of Fuel Oil-Diesel Fuel Mixture in a Single Cylinder Engine

by

Mohd Shah Rizal Bin Mohd Yusof

Dissertation submitted in partial fulfillment of
the requirements for the
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(Mechanical Engineering)

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CERTIFICATION OF APPROVAL

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Approved by,

Ch. rangkuti: 10/12/08.

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UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

July 2008

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

MOHD SHAHRIZAL BIN MOHD YUSOF

ABSTRACT

Cost for Diesel fuel is higher in recent years have cause problem to the consumer. It is necessary to find alternative fuel that is less expensive to be used for generating power. Fuel oil is a type of fuel that has been identified as a possible alternative source of fuel to be used for diesel engine. The engine performance and exhaust emission characteristics based on Diesel fuel and medium fuel oil (MFO) mixture is the main study in this project. Beside 100% Diesel fuel, a blend of 5%MFO, 10% MFO, 15% MFO and 20% MFO are used. The suitable viscosity of fuel mixture is studied as viscosity plays important criteria for the fuel to be used in diesel engine. Engine performance and exhaust emission of all fuel blends were investigated and compared with ordinary Diesel fuel in a single cylinder diesel engine without any modification. The experimental results show that the engine power and torque are lower than the value obtained from Diesel fuel. As for exhaust emission, adding MFO do increase CO emission and particulate matter, but reduce the NOx emission. It is proven that medium fuel oil can be used as an alternative fuel for diesel engine, with some lose of the engine performance and emission characteristics.

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CHAPTER 1
INTRODUCTION

1.1 Background Of Study

Diesel engine is one of the engine types that are currently being used world wide. Invented in 1892 by German engineer Rudolf Diesel, it used diesel cycle to operate. It is an internal combustion engine that powered by Diesel fuel. Until several years ago, the average price of Diesel fuel was usually lower than the average price of gasoline. From September 2004 onwards, the price of Diesel fuel has increase due to several reasons. This trend will remain in a long time especially due to product demand, imbalance supply, operating cost and tight refining capacity.

1.2 Problem Statement

Cost for Diesel fuel is higher in recent years have cause problem to the consumer. It is necessary to find alternative fuel that is less expensive to be used for generating power. Fuel oil is a type of fuel that has been identified as a possible alternative source of fuel to be used for diesel engine. Deep understanding of Diesel fuel, fuel oil and diesel engine principles are vital in this study. The engine performance and exhaust emission characteristics based on Diesel fuel and fuel oil mixture is the main study in this project.

A number of laboratory experiments are expected to be done to test various percentage of the fuels mixture. The test will be conducted using diesel engine and gas analyzer in order to study the engine performance and exhaust emission characteristics. All of these requires knowledge on how to operate the equipments, how to conduct the experiments, how to handle the mixture and others.

On top of that, basic knowledge of related theories, including combustion reactions, combustion by-products and emission curves are important for analysis of the data obtained through experiment. One important aspect that needs to be taken into account at first is the fuel oil viscosity. The mixture of both fuel oil and Diesel fuel must have certain value of viscosity, so that it can be used in diesel engine.

1.3 Objective Of Study

The objective of this project is to investigate the diesel engine performance and exhaust emission characteristics using a mixture of MFO and Diesel fuel.

1.4 Scope Of Study

There are two directions from which this study is focused on, that is the engine performance and exhaust emission characteristics. For engine performance, the brake horsepower (BHP), torque, exhaust temperature, fuel mass flow rate and brake specific fuel consumption (BSFC) for various percentages of fuel mixture and 100% diesel will be studied. The emission characteristics will involve on the amount of combustion by-product, namely carbon monoxide (CO), unburned hydrocarbon (HC), nitrogen oxides (NOx) and particulate matter (PM). This will be done by conducting a laboratory experiment for various percentages of fuel oil-Diesel fuel mixture, using single cylinder diesel engine and exhaust gas analyzer. Economic analysis for the fuel mixture will be conducted at the end of this study. Recommendation of the best percentage of fuel mixture will be concluded with regards to the engine power output and exhaust emission characteristics.

Literature studies via books, references, journals, internet and others are done to gather information regarding on the theories and to enhance the understanding of the study. Laboratory experiments are performed to study the fuel viscosity, engine performance and exhaust emission characteristics. The data obtained were analyzed theoretically and graphically using software. The combinations of result of the performance and emission characteristics are used as the base to choose the best mixture.

1.5 Significant Of Study

This study is basically help to determine the possibility of using fuel oil as an alternative for Diesel fuel in diesel engine. This study has the potential to increase the usage of medium fuel oil (MFO) and increase the feed stock of Diesel fuel. On top of that, this study will help to determine the right percentage of fuel mixture that can give optimum performance without sacrificing the exhaust emission. To do this, the performance and emission characteristics will be compared with the standard data. With all the data obtain, the right condition for optimum performance can be determined. Last but not least, it is hope that in the end of this study, the cost of running diesel engine can be reduced.

CHAPTER 2
LITERATURE REVIEW

2.1 Fuel Oil

Fuel oil is a product of heavy distillation from the oil refining process either as a distillate or residue. Fuel oil consist of a long hydrocarbon chains, mainly alkanes, cycloalkanes, and aromatic. This type of fuel is usually used as fuel for power stations, furnace and boiler. Fuel oil can be divided into six classes, based on its boiling temperature, chemical composition and usage. Table 2.1 below shows the summary of classification for fuel oil. The classification is based on the United States nomenclature and it is widely used in most of the world.

Table 2.1: Classification of fuel oil [3]

Name	Type	Heating Value (Btu/gal)	Chain Length	Remarks
No 1 fuel oil	Distillate	132 900-137 000	9-16	Distillate fuel oil
No 2 fuel oil	Distillate	137 000-141 800	10-20	Diesel fuel oil
No 3 fuel oil	Distillate	143 100-148 100		Light fuel oil
No 4 fuel oil	Distillate/Residual	146 800-150 000	12-70	Mixture of distillate and residual fuel oil
No 5 fuel oil	Residual	149 400-152 000	12-70	Residual fuel oil
No 6 fuel oil	Residual	151 300-155 900	20-70	Heavy fuel oil

The viscosity of every fuel oil increases with fuel oil number. For example No 1 fuel oil has the lowest viscosity while No 6 fuel oil is the most viscous among all the fuel oils. Due to the high viscosity, the heavy fuel oil sometimes needs to be heated before usage to make it flow. The price for each fuel oil decrease as the fuel number increase. The fuel oil is also being used in the maritime field, but using different classification. Table 2.2 below shows the classification used in maritime field.

Table 2.2: Marine classification of fuel oil [3]

Name	Alias	Remarks
Marine gasoil	MGO	Equivalent to No 2 fuel oil, made from distillate only.
Marine diesel oil	MDO	A blend of gasoil and heavy fuel oil.
Intermediate fuel oil	IFO	A blend of gasoil and heavy fuel oil, with less gasoil than marine diesel oil.
Medium fuel oil	MFO	A blend of gasoil and heavy fuel oil, with less gasoil than intermediate fuel oil.
Heavy fuel oil	HFO	Pure or nearly pure residual, equivalent to No 6 fuel oil.

2.2 Residual Fuel Oil

Residual fuel oil also known as bunker oil, bunk oil or number 6 fuel oil is oil that is thick, syrupy, black and tar-like liquid. It consists of complex hydrocarbons with high boiling points and viscosity, since it is residual product from the crude oil. This type of oil is usually used as fuel for steam boilers and power generators. Number 6 fuel oil is very stable, as a proof it cannot be ignited using burning matches and sparks in normal, cool condition. It can only be ignite if it is heated until its flashpoint, approximately about 150 degrees Fahrenheit. Table 1.3 below shows the properties of the residual fuel oil.

Table 2.3: Properties of the residual fuel oil [7]

Flash Point	150°F (min)
Boiling Point	>500°F
Auto-Ignition Temperature	765°F
Evaporation Rate	Negligible
Vapor Pressure	Less than 0.0001 mmHg
Vapor Density (air=1)	>1 (vapors are heavier than air)
Specific Gravity	0.97 (varies)
Solubility in water	Negligible
Percent volatiles	Negligible
Stability	Negligible
OSHA and NFPA classification	Class II combustible liquid

Its combustion produces darker than other fuel and it needs specific temperature for storage and pumping. Thus, due to these disadvantages, it is the cheapest oil in the market.

2.3 Performance Characteristics

The performance characteristics that are studied in this project are the brake horsepower (BHP), torque, exhaust temperature, fuel mass flow rate, and specific fuel consumption. Basically, engine performance is more precisely defined by the maximum power (maximum torque) available at each speed within the useful engine operating range. On top of that, it is define as the range of speed and power over which engine operation is satisfactory [19].

2.3.1 Torque and brake power

Engine torque is normally measured by dynamometer. Figure 2.1 shows operating principle of a dynamometer. The rotor is coupled electromagnetically, hydraulically or by mechanical friction to a stator, which is supported in low friction bearings. The stator is balanced with the rotor stationary. The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, springs or pneumatic means.

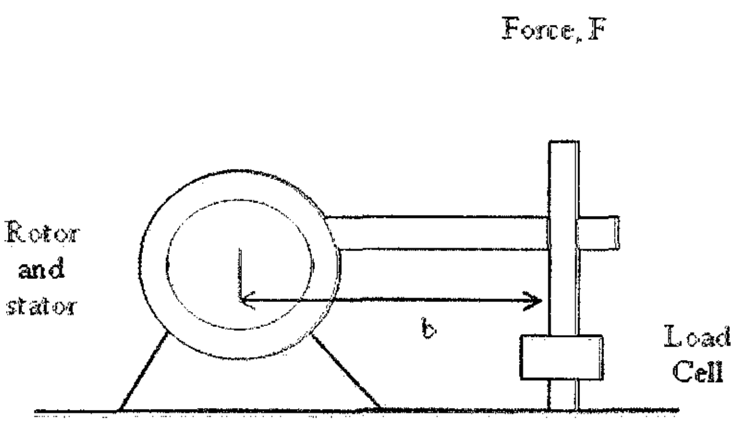


Figure 2.1: Schematic of principle of operation of dynamometer [19]

Using notation from the above Figure, if the torque exerted by the engine is T:

$$T=Fb \qquad [2.1]$$

The power P delivered by the engine and absorbed by the dynamometer is the product of torque and angular speed:

$$P=2\pi NT \qquad [2.2]$$

Where N is the crankshaft rotational speed. In SI units:

$$P(kW)= 2\pi N(^{rev}/_{s})T(N.m)x10^{-3} \qquad [2.3]$$

Or in U.S units: $$P(hp)=[N(^{rev}/_{min})T(lbf.ft)]/5252 \qquad [2.4]$$

Torque is a measure of an engine's ability to do work, while power is the rate at which work is done. The value of engine power measured as described above is called brake power P_b. This power basically is the usable power delivered by the engine to the load, in this case a 'brake'.

2.3.2 Brake Horsepower Correction Factor

The humidity, pressure and temperature of the ambient air induced into an engine, at a given speed, affect the air mass flow rate and the power output. Correction factors are used to adjust measured wide open-throttle power output value to standard atmospheric conditions to provide more accurate basis for comparisons between engines. The Corrected Brake Horse Power is given by,

$$P_{bc}=C_F P_{bm} \qquad [2.5]$$

Where P_{bm} is the measured brake horsepower and C_F is the correction factor, which can be determined by using this formula,

$$C_F=[P_{sd}/(P_m-P_v)][T_m/T_s]^{1/2} \qquad [2.6]$$

Where P_{sd} is a standard dry absolute pressure (736.6 mmHg), P_m is a measure of barometric pressure, P_v is a measure of ambient water vapor pressure, T_m is a measure of ambient temperature and T_s is a standard ambient temperature (29.4°C). The measured ambient water vapor pressure, P_v, can be calculated by using below equation,

$$P_v = P_{v,\,sat} - \text{Ø} \qquad [2.7]$$

Where Ø is relative humidity and $P_{v,\,sat}$ is the saturation water vapor pressure, which can be determine from this equation,

$$\text{Log}_{10} P_{v,\,sat} = 8.10765 - (1750.286/T_m + 235.15) \qquad [2.8]$$

Where T_m is a measure of ambient temperature.

2.3.3 Fuel Mass Flow Rate and Brake Specific Fuel Consumption

The fuel consumption is a measure of fuel flow rate; mass flow rate per unit time, m_f. A more useful parameter that usually being used is the Brake Specific Fuel Consumption (BSFC), which define as the fuel flow rate per unit power output. It measures how efficiently an engine is using the fuel supplied to produce work.

$$BSFC = m_f/P \qquad [2.9]$$

And the unit in SI is BSFC (mg/J) = m_f (g/s)/P (kW) [2.10]

Where m_f is the fuel mass flow rate per unit time and P is the power produced for the combustion of the fuel. The lowest BSFC value is more desirable compared to the higher value.

The best value of BSFC for diesel engine is different. The best value of BSFC for diesel engine are lower than the gasoline engine and is about 55 μg/J = 200 g/kWh [19]. The measurement of the engine's efficiency is actually the ratio of the work produced per cycle to the amount of the fuel energy supplied per cycle that can be released in the combustion process. The fuel energy supplied, which can be released by combustion, is given by the mass of fuel supplied to the engine per cycle times the heating value of the fuel.

$$\text{Fuel energy supplied} = m_f Q_{HV} \qquad [2.11]$$

The heating value of fuel, Q_{HV} defines its energy content. Typical heating values for the commercial hydrocarbons fuels used in engine are in the range between 42 to 44 MJ/kg[19].

The measurement of an engine efficiency, that are also called Fuel Conversions Efficiency, η_f, is given by,

$$\eta_f = W_c / m_f Q_{HV} = P / m_f Q_{HV} = 1 / bsfcQ_{HV} \qquad [2.12]$$

2.4 Emission Characteristics

The combustion that occurred on each type of engine is very different. As for compress ignition engine or diesel engine, it has separate fuel and air streams that combust as they are mixed together. The chemical reaction, which produces a diffusion flame, takes place at the interface between the fuel and the air. The heat release begins at a relatively high value and then decreases as the available oxygen is depleted.

The main combustion products from internal combustion engine emissions are namely nitrogen oxide (NOx), carbon monoxide (CO), hydrocarbons (HC) and particulates (PM). These products are a significant source of air pollution. Internal combustion engines are the source of roughly half of the NOx, CO and HC pollutants in air. As for example, NOx reacts with water vapor to form nitric acid, and reacts with solar radiation to form ground level ozone, both of which cause respiratory system problems. And hydrocarbons can cause cellular mutations and also contribute to the formation of ground level ozone too.

2.4.1 Carbon Monoxide

Carbon monoxide appears in the exhaust of rich-running engines since there is insufficient oxygen to convert all the carbon in the fuel to carbon dioxide. The most important engine parameter influencing carbon monoxide emissions is the fuel air equivalence ratio. The chemical reaction for combustion when air is sufficient,

$$C_nH_n + O_2 \rightarrow CO_2 + H_2O + CO$$

From the reaction above, it can be clearly seen that carbon monoxide will be produced when insufficient air or oxygen is drawn into the combustion chamber. Lacks of oxygen will only permits the carbons atoms to combine with only one oxygen atom to produce CO instead of two which will produce CO_2. For fuel rich mixture, CO concentrations in the exhaust increase steadily with increasing of fuel-to-air ration since the amount of excess fuel increases. And for fuel lean mixture, CO concentrations in the exhaust vary little with fuel-to-air ratio. Since diesel engines run lean overall, their emission of carbon monoxide are low and generally not considered a problem. It does appear the direct injection diesel engines emit relatively more CO than indirect injection diesel engine.

2.4.2 Nitrogen Oxides

Nitrogen oxides (NOx) are formed throughout the combustion chamber during the combustion process due to the reaction of atomic oxygen and nitrogen. The reactions forming NOx are very dependent on temperature, so the NOx emission from the engine scale proportionally to the engine load and NOx emission are relatively low during engine start and warm-up.

The reaction mechanism that produced NO is called "Zeldovich mechanism" [19], in which NO is formed in the high temperature burned gases left behind the flame front. The prompt mechanism occurs within the flame front, and relatively small if the volume of the high temperature burned gases is much greater the instantaneous volume of the flame front, as in usual, the case in internal combustion engine. The chemical reactions are [19],

$O + N_2 \text{====} NO + N$, a nitrogen dissociation reaction triggered by an oxygen atom. This reaction is endothermic.

$N + O \text{====} NO + O$, a nitrogen atom reacts exothermically with an oxygen molecule to form nitric oxide and an oxygen atom.

$N + OH \text{====} NO + H$, an exothermic reaction between a nitrogen atom and a hydroxide radical which forms nitric oxide and a hydrogen atom.

2.4.3 Unburned Hydrocarbon Emission (HC)

Hydrocarbons describe the large family of emissions composed of hydrogen and carbon in a variety of chemical bonds. These range from simple non reactive methane molecules (CH_4) to more complex and active chemical chains like benzene (C_6H_6) and butane (C_4H_8). Hydrocarbons (HC) are formed when fuel is not adequately oxidized, or burned. In diesels, incomplete combustion of the fuel results in soot formation, visible as large clouds of black smoke, containing up to 0.5% of the fuel mass. During startup, and subsequent misfire, unburned fuel may condense and produce clouds of white smoke [16]. Overall, the level of HC emitted as a pollutant is strongly dependent upon the fuel distribution and resulting combustion inside the cylinder.

Hydrocarbon emissions can be split into two major groups: non-reactive and reactive. This grouping stems from the chemical reactivity of the molecules with respect to the formation of smog. Hydrocarbons play a secondary role in ozone formation by accelerating the formation of NO_2, which reacts with O_2 to produce ozone, the basic component of smog. The reactive components include all hydrocarbon chains except methane, which is highly stable and also gives rise to the term "non-methane organic gases" which include all non-methane hydrocarbons and oxygenates. In addition to participating in smog formation, many oxygenates are also irritants to the eyes and lungs. Further many of these molecular chains are not found in the fuel prior to combustion, demonstrating the complex chemical kinetics that occurs inside a combustion chamber.

One of the factors in the production of hydrocarbon emissions is the quenching of the flame front as it approaches the relatively colder surfaces of the cylinder walls and piston. These surfaces absorb heat energy to such an extent that combustion cannot sustain itself within the fuel-air mixture. Crevices and gaps such as those seen between the cylinder walls and piston dominate this mechanism as hydrocarbons quenched at the walls are readily oxidized later in the cycle [19]. Cold starting of an engine demonstrates this problem drastically as the relatively cold surfaces of the combustion chamber cause excessive amounts of black smoke. One source unique to direct injection diesels comes from the fuel injector tips. Fuel leftover in the nozzle tips after injection has ceased slowly evaporates and seeps slowly into the combustion chamber where it may or may not be oxidized.

The major source, however, contributing to HC emissions are the localized rich or lean condition found within the combustion zone. As the spray is injected, the air mixes with the outer edges of the fuel producing very lean zones that oxidize in a non self-sustaining manner and seldom to completion. As the spray continues to mix with the air, these lean zones expand outward leaving more combustible mixtures behind in the center of the chamber. The amount of HC left unburned is then a function of the mixing rate (or turbulent swirl) of the engine, the cylinder conditions and because of its association of the prior two, the ignition delay. According to Heywood, there is a non-linear relationship between the ignition delay and the amount of HC produced. Leanness, however, is not the sole condition aiding hydrocarbon emissions. Overly rich mixtures will also result in incomplete combustion, a condition that can be caused by insufficient mixing of the oxygen in the air with the fuel spray. This is especially the case just after the injector nozzles have ceased spraying as the pressure forcing the fuel out has dropped and the remaining fuel enters the combustion chamber at low speed. The low velocity of the fuel causes under mixing of the fuel-air to occur, which of course generates an overly rich region. Desorption of HC from the layer of oil that coats the cylinder walls adds to the overall level found in exhaust gas and is controlled by the characteristics of the fuel being used and its ability to be absorbed by the oil layer.

Engine operating conditions play a role in HC emissions mainly as a function of the load on the engine. Idle and light load conditions generate overall fuel to air ratios of around 100:1 and this causes an excess of over lean regions in the injected fuel spray. Consequently, light load and idle produce substantially more HC emissions than full load [19]. On the other end of the spectrum over fueling of the engine at high loads will produce excessive HC through insufficient oxygen supplies.

The timing of the injection produces an effect on HC as well. If the timing is advanced away from top dead center and away from the optimum timing, the ignition delay lengthens, allowing a higher percentage of the total fuel injected to mix with the air and impinge on the cylinder walls. This also produces more areas of lean mixtures, hindering efficient combustion and raising the amount of unburned HC [16]. On the other hand, retarding the advance produces overly rich regions with insufficient time to combust with the end result being visible smoke. In a similar vein, lengthening the physical time that the injectors are open and spraying fuel into the cylinders reduces HC at low load, but at high load leads to an increase in smoke and particulates [16].

2.4.4 Particulate Matter

Diesel particulates consist of principally of combustion generated carbonaceous material (soot), on which some organic compounds have become absorbed. Most particulate material results from incomplete combustion of fuel hydrocarbons. The composition of the particulate material depends on the condition in the engine exhaust and particulate collection system. At temperature above 500°C, the individual particles are principally cluster of many small spheres of carbon. As temperature decreases, the particles become coated with absorbed and condensed high molecular weight compound that include unburned hydrocarbons, oxygenated hydrocarbons and polynuclear aromatic hydrocarbons. Smoke also forms in diesel engine because combustion in diesel engine is heterogeneous. The rate amount of particulate matter produced can be determined by,

$$PM\ (g/bhp - hr) = (m_{d,\,PM}) / (P_{bhp}\, t_s) \qquad [2.13]$$

Where $m_{d,\,PM}$ is dry mass of particulate matter, P_{bhp} is brake horse power and t_s are sampling time.

2.5 Expected Mixture Results

The result of engine performance obtained using 100% Diesel fuel will be used as benchmark for the mixture fuel. Based on theory and findings, there would be a difference in the trend of the performances curve and the emission as compared with the 100% Diesel fuel. This is maybe due to the chemical composition of the medium fuel oil used.

For the torque and brake horse power, the expected curve will be in the same shape, but the value will be less than performance using 100% Diesel fuel. This expected trend will also followed by brake mean effective pressure since the brake horse power has a direct proportional relationship with the brake mean effective pressure. As for the brake specific fuel consumption, the curve will be similar to 100% Diesel fuel but the value will be higher.

In this study, the method of testing the mixture is by increasing the percentage of medium fuel oil. For each blend, it is expected that more percentage will give less brake power, less torque, higher fuel consumption and higher temperature. As for emission characteristic, it is expected that the amount of exhaust gases, namely CO, HC and NOx will increases. Since NOx formation is likely influence by temperature, as the temperature increases and the energy of the fuel increases, the NOx production will also increases. Another exhaust by-product that needs to be considered in this project is particulate matter (PM). Mixing Diesel fuel with medium fuel oil will make the fuel less oxygenates. Thus less complete combustion will happen and this will produced less CO_2. As less CO_2 being produced, the particulate matter will increase. This is shown by the smoke or soot produced.

CHAPTER 3
METHODOLOGY

3.1 Overall Procedure and Steps

Step by step procedures have been made as to meet the objective of this project and to ensure that this project is organized and structured.

3.1.1 Literature Review

Literature review has been done during the initial stage of this project. All kind of sources that are available and reliable such as books, journals, articles, web pages and other researches has been reviewed, referred and studied. Literature review has been done to cover theory, performance and emission aspect, fuels, diesel engine and other element that have relation with this project. This step has aid to understand more about the fuel used, both Diesel fuel and MFO. Other similar studies and researches have been referred as a guide to ensure that the methodology is correct and suitable.

3.1.2 Laboratory Experiments and Testing

Series of experiments and testing have been conducted in order to test the outcome of using various mixtures of Diesel fuel and medium fuel oil. The interest of this experiment is to find the engine performance and emission characteristic, on top of satisfying other objective of this project.

3.1.3 Analysis of Data and Results

Analysis of the data and results are performed in order to complete the objective and to come out with a conclusion about the fuel mixture. Data will be put in graphical form as to help in analysis and comparison.

3.2 Tools/Equipment Used

A number of equipment has been used for the experiments and testing. Among the important equipment is viscometer, single cylinder diesel engine and exhaust gas analyzer.

3.2.1 Single Cylinder Diesel Engine Test Bed

Single cylinder diesel engine (Figure 3.1) is used to study the performance and emission of using the fuel mixture. The specifications of this equipment are shown in Table 3.1.

Table 3.1: Diesel Engine Test Bed Specification [13]

Manufacturer	'Robin' - Fuji DY23D
Model	TD111
Engine Type	Diesel, Single cylinder, 4 stroke
Net Weight	45 kg
Maximum Speed	3750 rpm

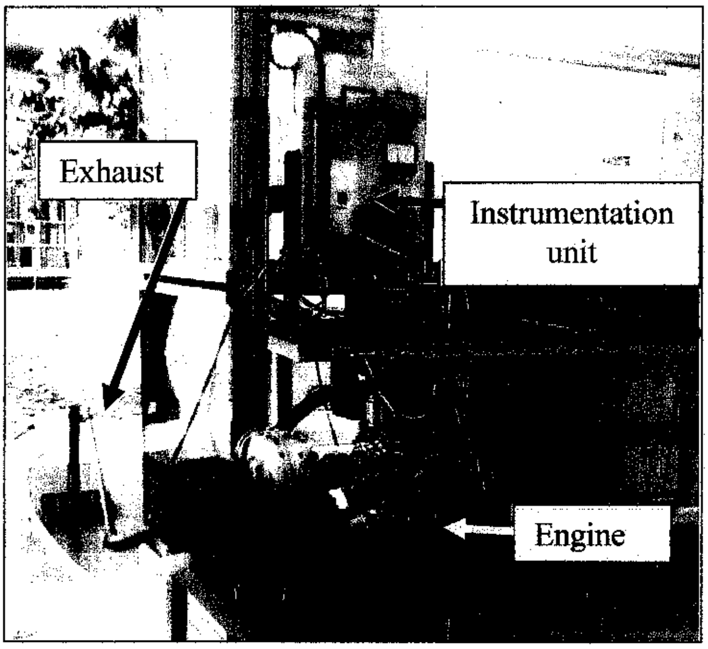


Figure 3.1: Engine test bed and instrumentation

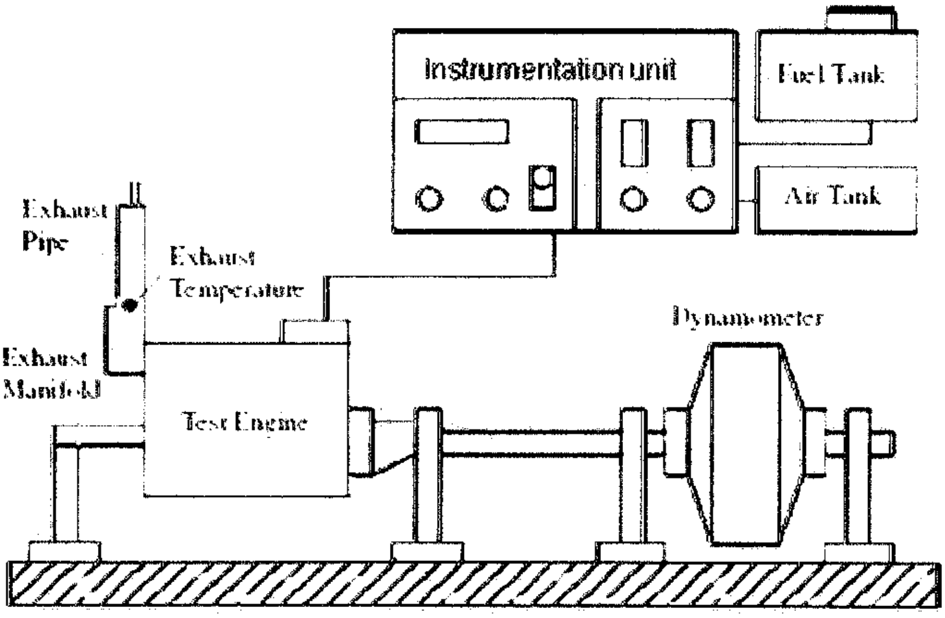


Figure 3.2: Schematic diagram of experimental setup

3.2.2 Exhaust Gas Analyzer

This equipment is used to study the emission characteristic. It detects the type and amount of emission from the exhaust of diesel engine.

3.2.3 Viscometer

This equipment is used to determine the viscosity of the fuel mixture with different percentage of medium fuel oil.

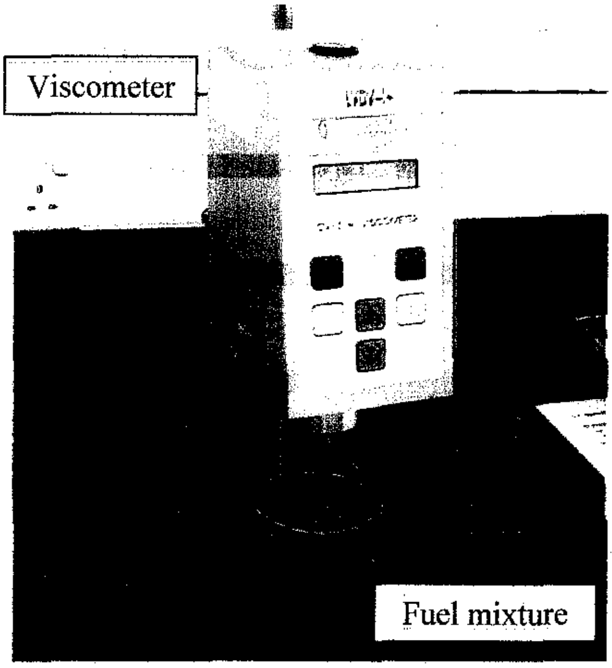


Figure 3.3: Viscometer used

3.2.4 Fuel Used

The fuels used in this study are Diesel fuel and medium fuel oil (MFO). The medium fuel oil were mix with Diesel fuel in different percentage; 5% MFO, 10% MFO, 15% MFO, 20% MFO and 100% diesel. The specifications of this fuel and Diesel fuel are shown in Table 3.2. MFO fuel is the fuel oil chosen mainly because of its low price and potential to be an alternative source of fuel.

Table 3.2: Specification of the medium fuel oil and Diesel fuel used [8], [9]

Specification	Value (MFO)	Value (Diesel Fuel)
Kinematic Viscosity	15.0-20.0 mm^2/s	1.6-5.8 mm^2/s
Flash Point	68°C (max)	60°C
Sulfur Content (mass)	1.00 %m (max)	0.3%m
Water Content (volume)	0.75%vol (max)	0.05%vol
Ash Content (mass)	0.10 %m (max)	0.01%m
Density (at 15 °C)	990.0 kg/m^3 (max)	-
Pour Point	24°C (max)	15°C
Hydrogen Sulphide	2 mg/kg (max)	-
Asphaltenes	0.5%mm (min)	-
Sodium (as a metal)	150 mg/kg (max)	-
Carbon Residue	180.0%mm (max)	0.1% (Micro Carbon) 0.1% (Conradson Carbon)

3.2.5 Computer Software

The ordinary computer software used is Microsoft Office. It is used for documentation work. GT Suite and GT Power software were used to make a simulation regarding the engine performance before the laboratory testing, function to obtain the highest theoretical value for each performance aspect in ideal environment.

3.3 Experiment/ Testing Methodology

In this study, there are a number of experiments and testing conducted. The experiments were conducted with scientific and engineering approach, to get the reliable, repeatable and accurate data.

3.3.1 Viscosity Determination: Fuel characterization

Viscosity plays important role in this study, as the fuel that can be used in diesel engine have a specific range of viscosity. Basically, both of this fuel can be mixed together, and both of it are miscible in each other.

Eight samples are being prepared. The sample is 100% diesel, 1% MFO, 5% MFO, 10% MFO, 15% MFO, 20%MFO, 50% MFO and 100% MFO. All of the samples are mixed in volumetric based. For example, 20% MFO means that in 100ml of the mixture, there are 20ml MFO and 80ml Diesel fuel. The fuel mixture is being put in a beaker. Viscometer is used to determine the mixture viscosity. All the data are recorded three times and the average is taken, then graph is produced. Using the graph, the correct amount of percentage that suitable for testing are selected, based on the specific range obtained in literature review.

3.3.2 Engine Performance and emission characteristics using 100% Diesel fuel: Diesel fuel familiarization

In this experiment, the fuel used is 100% diesel. This experiment is done to obtain the data of engine performance and emission characteristic for 100% diesel. The desired data from this experiment is; torque, brake power, temperature, specific fuel consumption and emission characteristics. Before the experiment being conduct, the instrumentation unit is being calibrated. Refer Appendix 3.1 for the calibration procedure.

3.3.3 Engine Performance and emission characteristics using different percentage of Diesel fuel-Medium fuel oil mixture

In this experiment, four sample of oil mixture with different percentage are being prepared and tested. The sample is 5% MFO, 10% MFO, 15% MFO and 20% MFO. The procedure for this testing is the same with the experiment using 100% Diesel fuel. Data are recorded three times and the average is taken.

3.3.4 Emission characteristics using exhaust gas analyzer

The desired data for this experiment is the emission characteristics for every sample of fuels. In this experiment, gas analyzer is used. The probe put in the end of the exhaust will send the smoke sample for further analysis by the gas analyzer. Three readings are taken and the average is calculated. Refer Appendix 3.2 for the procedure of this experiment.

3.3.5 Simulation Using GT-Power

The objective of this simulation is to get the ideal value of engine performance in ideal condition and environment. This data will be used to verify the data obtained by laboratory testing. The software used is from GT-Suite software, GT-Power. All the specification and other important item such as pipe dimension are taken. This data is needed to produce the simulation.

CHAPTER 4
RESULTS AND DISCUSSION

4.1 Viscosity Determination: Fuel Characterization

Viscosity is one of the key aspects in this study. As two different fuel oils are mixed, the viscosity will change. In this study, the viscosity is determined using viscometer. The range of kinematic viscosity that suitable for diesel engine based on ASTM D975 is from 1.3 mm^2/s until 4.1 mm^2/s (at 40°C), or in cP unit 1.066 cP to 3.362 cP. Therefore, in this study, the viscosity of the mixture must be in this range.

Based on the experiment, eight samples are being prepared; 100% diesel, 1% MFO, 5% MFO, 10% MFO, 15% MFO, 20%MFO, 50% MFO and 100% MFO. The kinematic viscosity of each sample is tabulated in Table 4.1 below. The example of calculation to convert unit from mm^2/s to cP are attached in Appendix 4.1.

Table 4.1: Viscosity of MFO-Diesel fuel mixture

Diesel fuel (%)	MFO (%)	Viscosity at room temperature (cP)
100%	0%	1.13
99%	1%	1.15
95%	5%	1.28
90%	10%	1.40
85%	15%	1.73
80%	20%	1.88
50%	50%	5.91

Based on the result obtained, it is clearly showed that mixture of 50% of both fuels will produce viscosity that exceeds the specified range. Graph of viscosity versus percentage of oil was constructed, as Figure 4.1 below.

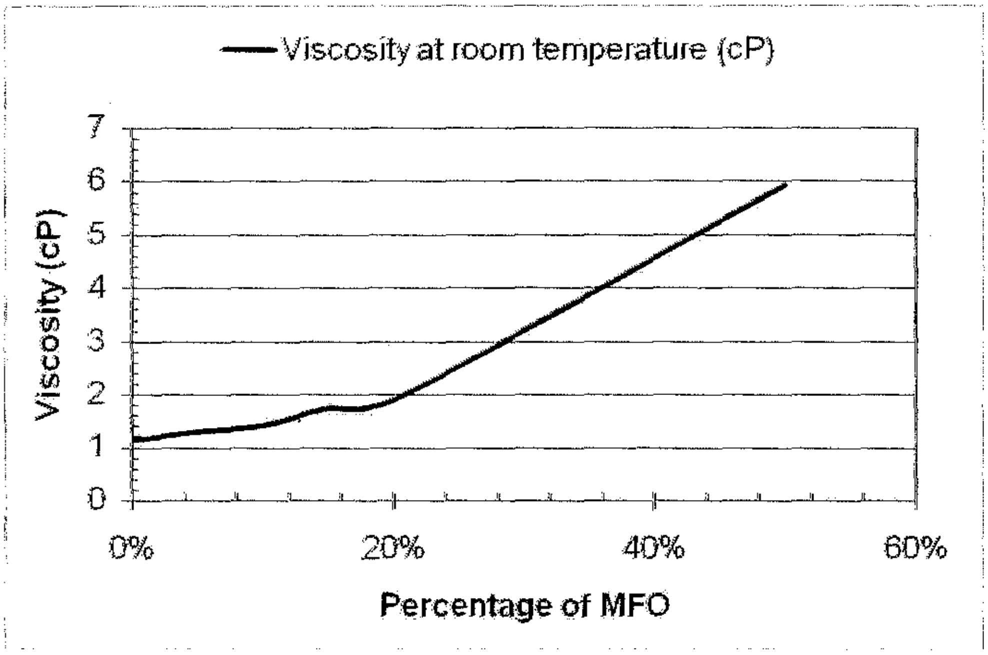


Figure 4.1: Viscosity versus fuel percentage

From Figure 4.1 above, as the fuel oil percentage increase, the viscosity will increase as well. It can be seen clearly that 50% of both oil will result viscosity of 5.91 cP, which exceed the range. Based on the viscosity range obtained, the maximum percentage of fuel oil that can be mixed is 31%. Refer Appendix 4.1.

4.2 Engine Performance Characteristics

Five samples have been successfully tested for engine performance characteristics using single cylinder diesel engine. This basically covers brake hose power (BHP), torque, brake specific fuel consumption (BSFC) and exhaust temperature. The data obtained for all mixture; 5% MFO, 10% MFO, 15% MFO, 20% MFO and 100% Diesel are tabulated in Appendix 4.4.

4.2.1 Torque

Appendix 4.4 shows the tabulated data of torque for every mixture. Based on these data, graph of torque versus engine speed are plotted, as shown by Figure 4.2.

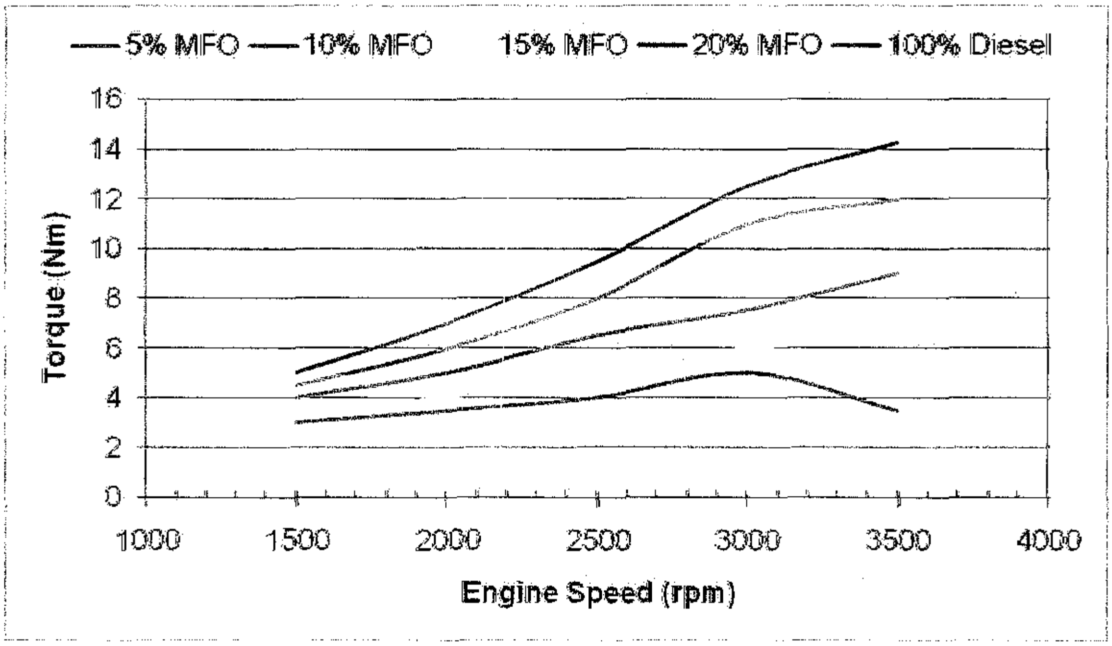


Figure 4.2: Torque versus engine speed

Based on the torque versus engine speed above, the torque increased as the engine speed increased. Theoretically, the torque will increase until the engine speed reaches a maximum value and will decrease gradually after that. This is due to aerodynamic friction (inlet and outlet air flow of cylinder) and mechanical friction (between cylinder wall and piston). At low speed, the pressure developed on the piston head due to combustion process can easily overcome both frictions to produce torque. Even though the torque increased with engine speed, the amount of increment is decreasing since the friction increase with engine speed. At high speed, the friction are much greater compared to the pressure developed. Aerodynamic friction cause less air to enter the combustion chamber thus less complete combustion occurred. Thus, torque seems to decrease slowly with the engine speed.

The trend for 100% Diesel fuel and other mixture in torque curve is similar. It is also noticed that there is a decreased of torque when more MFO is added. The percentage of torque reduction for all mixtures is described in a Table 4.2.

Table 4.2: Torque percentage reduction value for every mixture

Engine Speed (rpm)	**Percentage reduction of Torque (Nm)**			
	Percentage of MFO in mixture			
	5 %	**10%**	**15%**	**20%**
1500	10.00	20.00	30.00	40.00
2000	14.28	28.57	42.85	50.00
2500	15.78	31.57	47.36	57.89
3000	12.00	40.00	52.00	60.00
3500	15.78	36.84	54.38	75.43
% Avg	13.57	31.40	45.32	56.67

Based on this table, the average of torque reduction increased as the percentage of MFO increased, from 13.57% for 5% MFO to 56.67% for 20% MFO. This is comparable with BHP reduction trends. Similar to that, this is caused by incomplete combustion, releasing less energy, which converted into the torque, will decrease.

4.2.2 Brake Horse Power (BHP)

In this section, the uncorrected BHP is discussed. The data of BHP values for all mixtures are in Appendix 4.4. Based on the data, BHP versus engine speed curve is plotted, as shown in Figure 4.3.

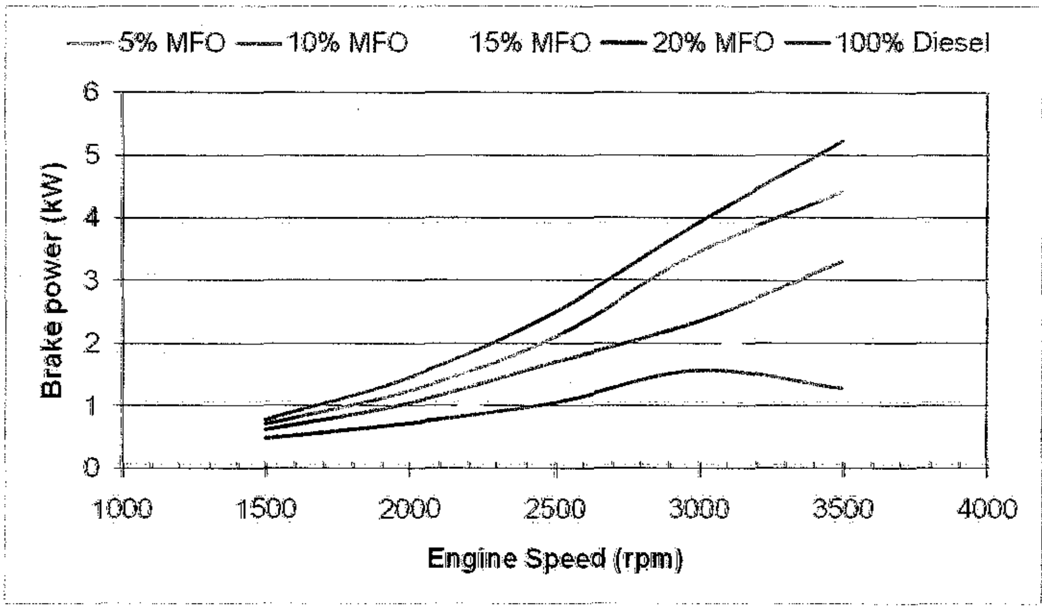


Figure 4.3: BHP versus engine speed

From Figure 4.3, it can be seen that the power output from diesel engine combustion process increase with speed. This is because; more complete cycles have been done by the engine within specific period of time. Thus, the power cycles performed by the engine will increase resulting in more power output.

BHP curve for 100% Diesel fuel is plotted for the basis of comparison between mixtures of fuel. As shown in Figure 4.3, the BHP curves for all mixtures have similar trend, although the power output decreased with increasing percentage of MFO in the mixture. Even though the brake horse power increased with engine speed, the degree of increment does decline slowly towards maximum engine rpm. This situation can be related to the power output formula, where power output is a function of torque developed and engine speed. The percentage of reduction in power output for MFO mixture compared to the 100% Diesel fuel is shown in Table 4.3.

Table 4.3: BHP percentage reduction for every mixture

Engine Speed (rpm)	Percentage reduction of BHP			
	Percentage of MFO in mixture			
	5 %	10%	15%	20%
1500	9.93	20.00	30.06	40.00
2000	14.25	28.58	42.83	50.00
2500	15.80	31.56	47.40	57.90
3000	12.02	40.00	52.02	60.02
3500	15.79	36.85	54.39	75.45
% Avg	13.55	31.39	45.34	56.67

Based on Table 4.3 above, the average percentage reduction of BHP increased as more MFO is added into the mixture. Even though there is a lot hydrocarbon as percentage of MFO increased, the diesel engine cannot do complete combustion, thus the total output gain from the mixture will be less and decreased.

4.2.3 Corrected Brake Horse Power

The brake power obtained is corrected to standard atmospheric condition. This provides more accurate result, as it includes the environment effect on the engine performance. Appendix 4.3 shows the example on how to calculate corrected brake horse power. Figure 4.4 show the corrected BHP.

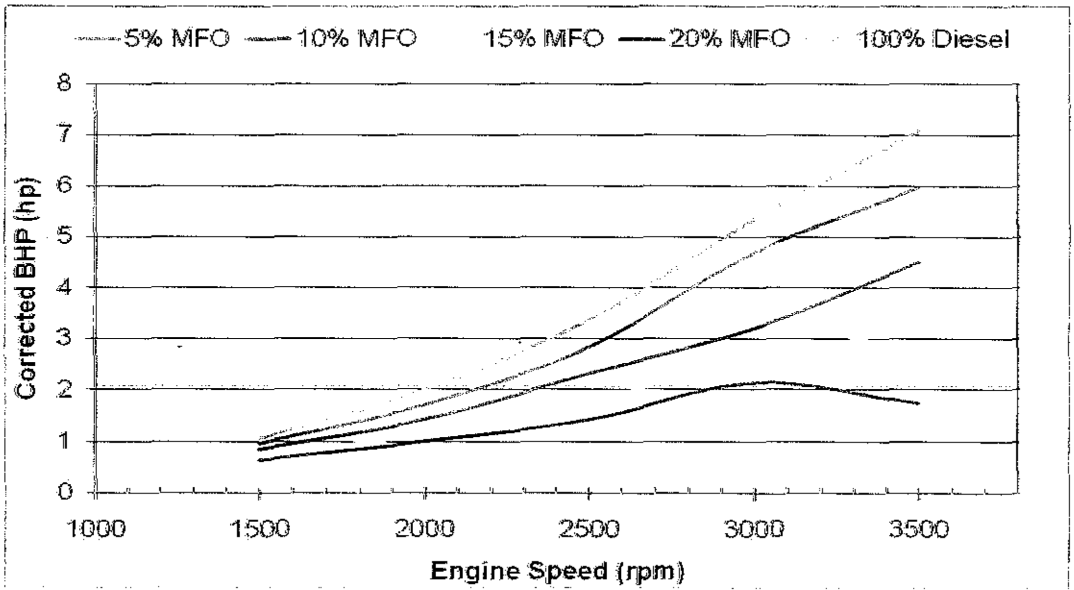


Figure 4.4: Corrected BHP versus engine speed

The trend is similar to the brake power curve, mainly because this is the corrected version of that data. Table 4.4 shows the percent reduction of corrected BHP.

Table 4.4: Percentage reduction of corrected BHP value for every mixture

Engine Speed (rpm)	Percentage reduction of Corrected BHP			
	Percentage of MFO in mixture			
	5 %	10%	15%	20%
1500	10.04	20	30.14	40
2000	14.23	28.57	42.80	50.20
2500	15.79	31.56	47.39	58.03
3000	12.01	40.01	52.03	60.02
3500	15.80	36.24	54.66	75.46
% Avg	13.57	31.27	45.40	56.74

4.2.4 Brake Specific Fuel Consumption

Appendix 4.4 summarized the value of BSFC obtained from the experiments for each mixtures of fuel. Similar to other data, the BSFC value versus engine speed is plotted and shown in Figure 4.5.

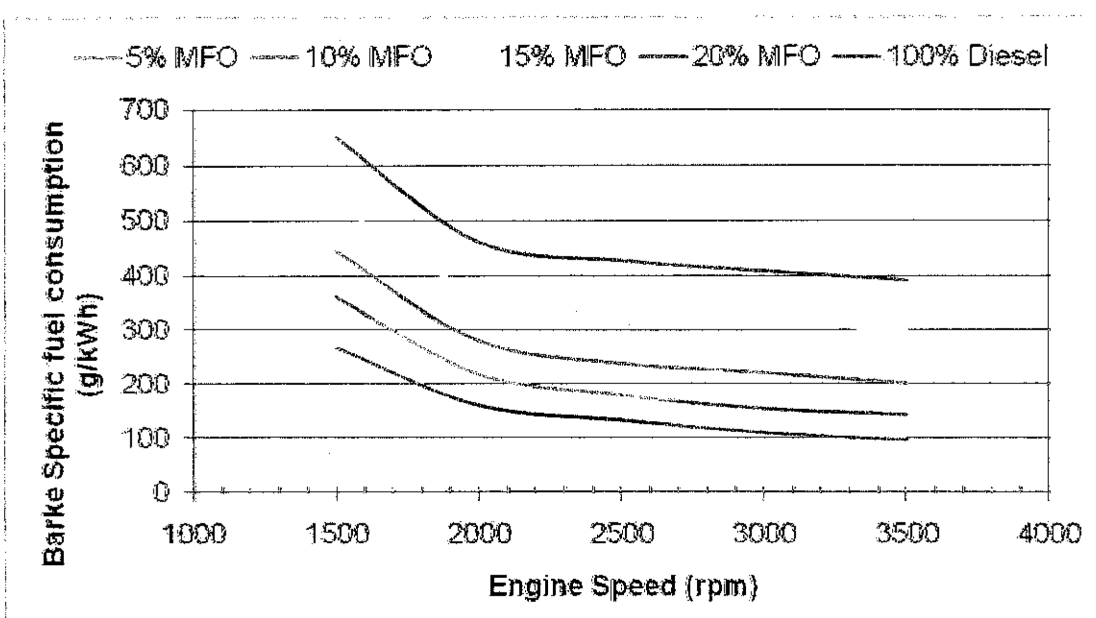


Figure 4.5: BSFC versus engine speed

Based on Figure 4.5, BSFC decrease as the engine speed increased. Theoretically, the BSFC curve should be a mirror image of torque curve. Referring to Figure 4.5 it can be concluded that as the engine speed increased, more fuel is consumed to produce more power and torque.

The curve pattern for BSFC is similar between all fuel mixtures. It is noticed that for mixture that have MFO, the BSFC value increased as the percentage of MFO increased. This might be due to the need of more fuel combustion to get the power output. The percentage of increase in BSFC for all mixture compared to 100% diesel is tabulated in Table 4.5.

Table 4.5: BSFC percentage increment value for every mixture

Engine Speed (rpm)	**Percentage increment of BSFC (g/kW-hr)**			
	Percentage of MFO in mixture			
	5 %	**10%**	**15%**	**20%**
1500	34.97	65.96	98.40	143.05
2000	36.10	73.81	128.03	186.06
2500	36.29	78.84	143.92	220.60
3000	40.34	99.23	183.25	268.23
3500	48.49	107.3	208.50	301.90
% Avg	39.24	85.02	152.42	223.86

Based on Table 4.5, the average percentage increment of BSFC increased as percentage of MFO increased. This is due to lower energy content of the mixture. Although add MFO will increase the hydrocarbon content and theoretically will also increased the energy content of the mixture, the engine cannot obtained the full power due to the incomplete combustion. Thus, in order to cope up or maintain power output, more fuel mixture has to be burn so that the energy produced is also maintained.

4.3 Engine Exhaust Emission Characteristics

All the mixture including 100% Diesel fuel has been successfully tested for the emission characteristics. In this aspect, there are four elements that will be discussed; nitrogen oxide, carbon dioxide, unburned hydrocarbon and particulate matter.

4.3.1 Nitrogen Oxide (NOx)

The amount of NOx produced is being measured by Exhaust Gas Analyzer. Appendix 4.4 and Figure 4.6 below showed the data obtained.

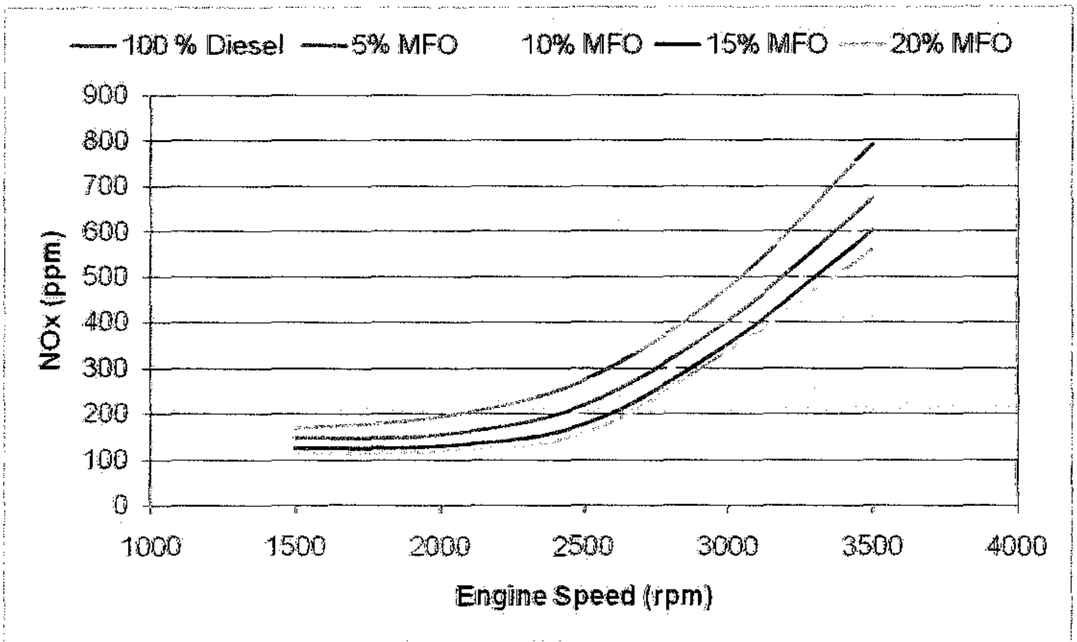


Figure 4.6: NOx emission versus engine speed

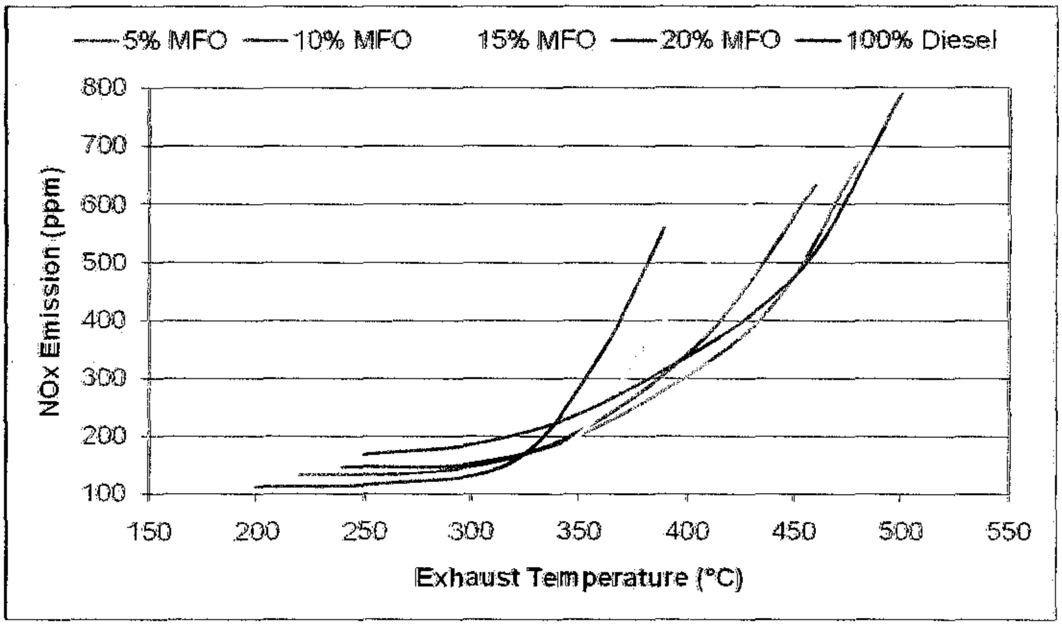


Figure 4.7: Exhaust temperature versus NOx emission

Based on the tabulated data and NOx curve, the NOx produced increase with engine speed. This condition is mainly because the production of NOx is depending on the engine temperature. Basically, higher engine temperature will result in higher production of NOx. A sufficient amount of energy is needed for nitrogen molecules to react with oxygen molecules to form NOx. Therefore, higher engine speed will increase the combustion process which produce heat (energy). As a result, the greater amount of energy in the form of heat is released, the greater amount of NOx is produced. At high engine speed, less time is available for the heat to dissipate thus the heat is trapped in the combustion chamber, which also contribute to the NOx formation.

From the graph plotted, it can be seen that the amount of NOx produced decreased with increasing MFO percentage. The percentage of reduction of NOx is shown in Table 4.6.

Table 4.6: Percentage reduction of NOx emission value for every mixture

Engine Speed (rpm)	**Percentage reduction of NOx (ppm)**			
	Percentage of MFO in mixture			
	5 %	**10%**	**15%**	**20%**
1500	13.02	21.30	26.63	33.14
2000	20.75	26.94	33.16	38.34
2500	20.15	24.17	35.89	42.49
3000	15.16	21.05	26.11	28.84
3500	14.92	19.97	24.02	29.20
% Avg	16.8	22.68	29.16	34.40

Referring to Table 4.6 above, the NOx reduced 16.8% when 5% MFO is being added to Diesel fuel. As more MFO is being added, the percent of NOx reduction increased. This trend happen because of the power output is reduced by the addition of MFO. This reduction in power indicates less heat (energy) is being produced, thus engine temperature will also decreased. As a result, less NOx is produced.

4.3.2 Carbon Monoxide (CO)

On top of NOx, Carbon Monoxide is also being measured by the exhaust gas analyzer. The data of CO produced are depicted in Appendix 4.4 and plotted in Figure 4.8.

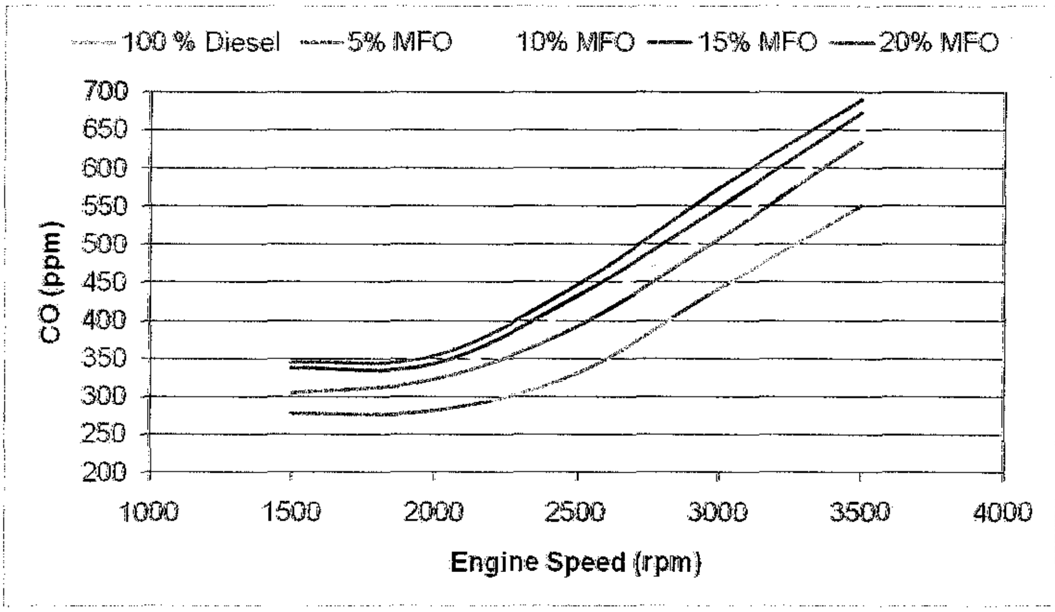


Figure 4.8: CO emission versus engine speed

Based on the graph above, the emission of CO increase as the engine speed increase. Once the engine speed increased, more fuel are injected into the combustion chamber, consequently increasing the likelihood of incomplete combustion to occur. This will result in higher CO production. Figure 4.8 also shows that as the MFO percentage increase, the CO emission will also increase.

The percentage of increment for CO emission is tabulated in Table 4.7 below.

Table 4.7: Percentage increment of CO emission value for every mixture

Engine Speed (rpm)	**Percentage increment of CO (ppm)**			
	Percentage of MFO in mixture			
	5 %	**10%**	**15%**	**20%**
1500	10.11	18.05	22.02	24.91
2000	14.95	18.15	22.06	25.97
2500	18.48	26.36	30.60	35.15
3000	14.97	18.14	24.26	29.93
3500	14.98	17.96	22.14	25.41
% Avg	14.69	19.73	24.22	28.27

As shown in Table 4.7, the emission of CO increased by 14.69% when 5% MFO is being added. Greater amount of MFO will result in higher CO emission. With the addition of MFO, carbon content will increase, thus higher chances of incomplete combustion to occur. Instead of producing CO_2, carbon molecules produced CO in the combustion chamber. Temperature also is a factor contributing to the formation of CO emission. Lower gas temperature in cylinder engine will prevent the CO component from effectively converting to CO_2 [10].

4.3.3 Unburned Hydrocarbon Emission (HC)

Beside NOx and CO, Hydrocarbon emission (HC) is also being measured using exhaust gas analyzer. The data of HC emission is tabulated in Appendix 4.4 and shown in Figure 4.9.

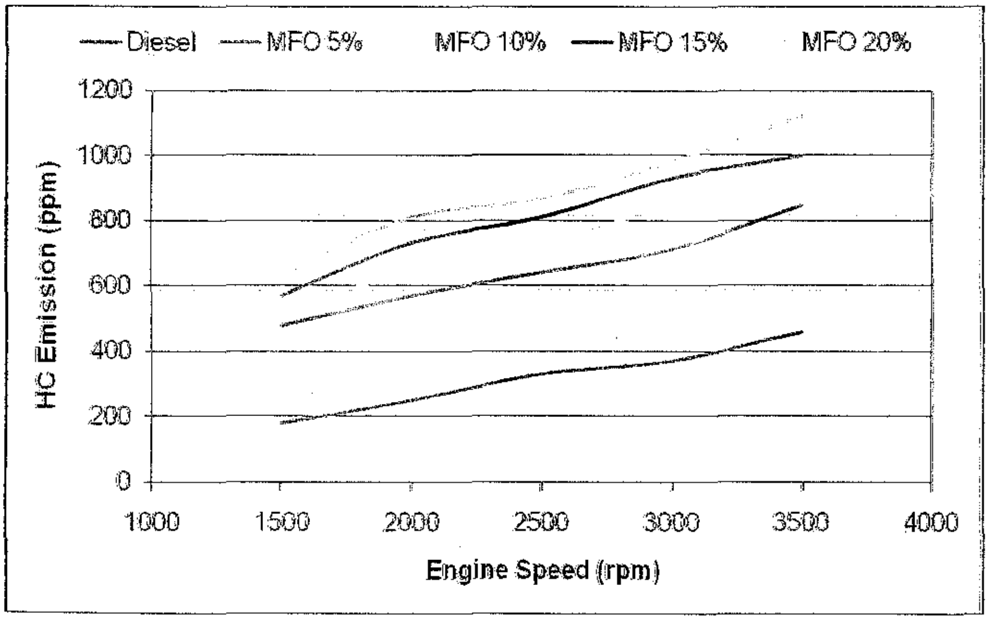


Figure 4.9: Unburned HC emission versus engine speed

Based on Figure 4.9, the amount of unburned HC produced increase as the engine speed increase, from 1500 rpm to 3500 rpm. As the engine speed increased, more fuel are injected and burned in the combustion chamber. Thus, more chance of incomplete combustion which results on the production of unburned HC.

On top of that, it is noticed that as the percentage of MFO increase, the HC production will also increase. Beside the incomplete combustion, this could also be due to the low volatility, which effected the spray formation in combustion chamber and thus lead to slow combustion [22]. Referring G. Nagarajan, A.N Rao and S. Renganarayan, the fuel spray does not propagate deeper into the combustion chamber and gaseous HC remain along the cylinder wall and the crevice volume left unburned.

4.3.4 Particulate Matter (PM)

During the laboratory testing of fuel mixture, it is noticed that the engine produced smoke and soot. Smoke emission were visually observed and appeared notably higher than 100% Diesel fuel. It is observed that as the MFO percentage increased, the more soot and smoke exist. On top of that, the smoke becomes darker with the increasing of MFO in the fuel.

4.5 Economic Evaluation

Each of the fuel mixture tested in the experiment is evaluated. The basic task in this economic evaluation is to predict the fuel mixture price per liter. The price of both fuels; Diesel fuel and MFO are known, that is RM 2.07 per liter and RM 1.03 per liter respectively. Note that both of these prices are non subsidized price per liter. By using the percentage of each fuel in the mixture, the price per liter is predicted. Refer Appendix 4.6 for a sample of calculation. The price per liter for each fuel mixture is shown in Figure 4.10 while the percentage reduction in price per liter is tabulated in Table 4.9.

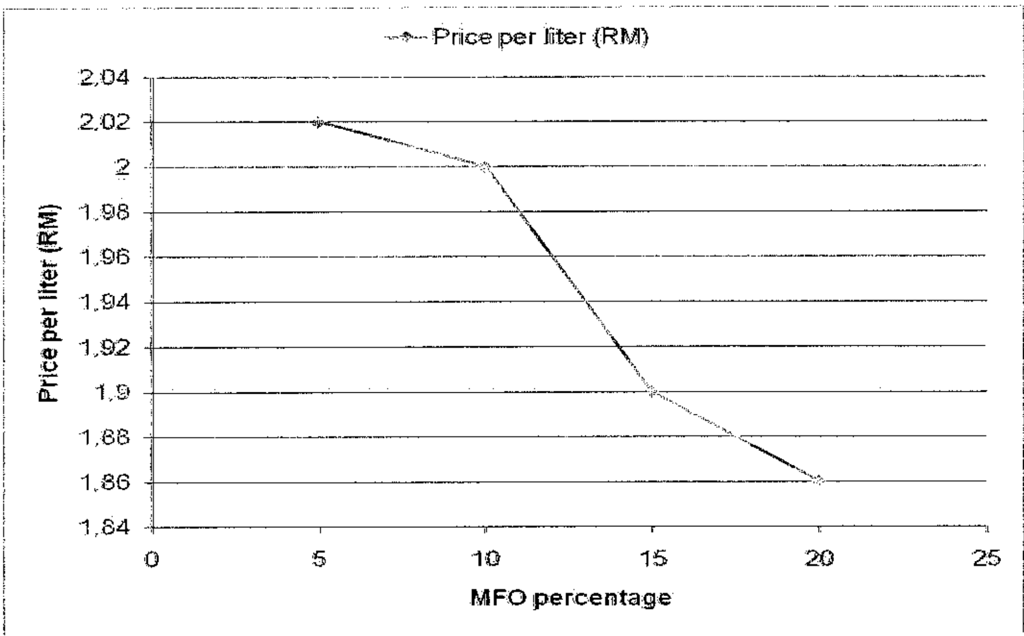


Figure 4.10: Price per liter of each MFO percentage

Table 4.9: Price per liter for each fuel mixture

Fuel Mixture	Price per liter	Percentage reduction of price per liter
5% MFO	RM 2.02	2.41%
10% MFO	RM 2.00	3.38%
15% MFO	RM 1.90	8.21%
20% MFO	RM 1.86	10.14%

As shown in Table 4.9 the price of the mixture per liter decreases as the amount of MFO increases.

4.6 Selection of Best Diesel fuel-MFO Fuel Mixture

Based on the result obtain and presented in this section, it is proven that mixture of Diesel fuel-MFO fuel can be used as alternative fuel for diesel engine. One of the objectives of this project is to recommend the suitable amount of MFO to be mix with Diesel fuel with regards to the performance and emission characteristics.

Based on the experimental results, the relationship between every result obtained and the MFO percentage are illustrated in Figure 4.11 and 4.12.

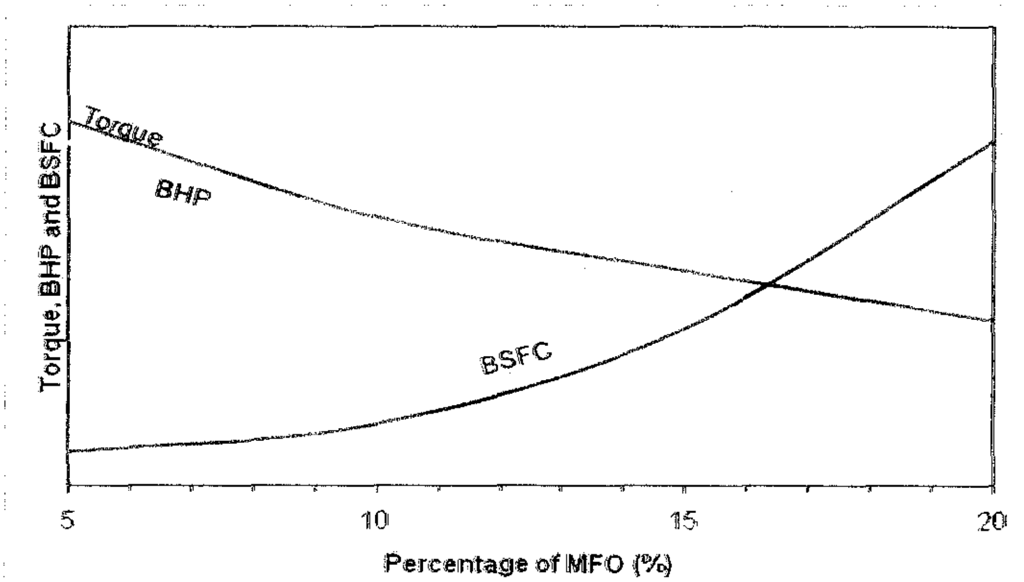


Figure 4.11: Relationship between engine performance, price and MFO percentage

Figure 4.11 shows the relationship between engine performance and the amount of MFO added. As more of MFO fuel added, torque and BHP will reduce while BSFC will increase generally.

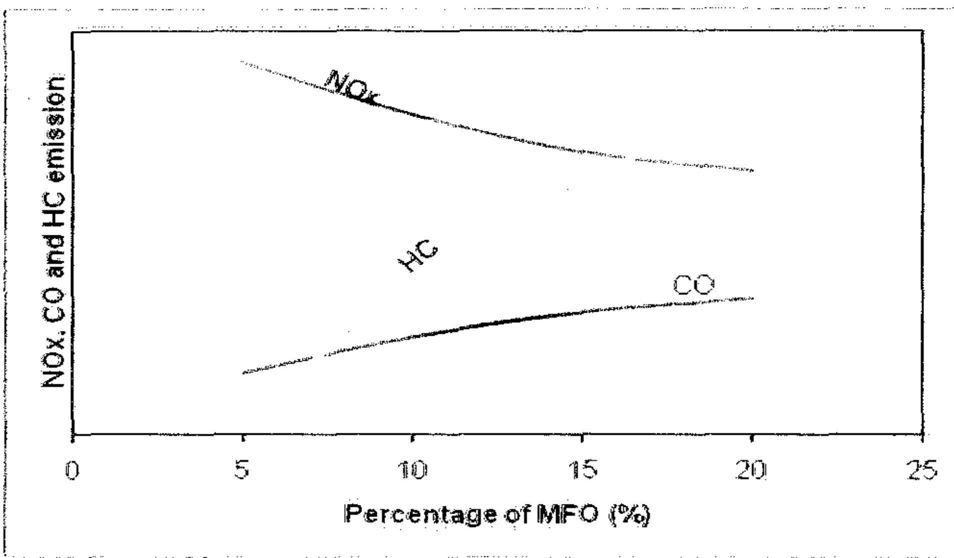


Figure 4.12: Relationship between exhaust emission and MFO percentage

Figure 4.12 shows the relationship between exhaust emission and MFO percentage. It has been found that as more percentage MFO in the mixture, the emission of HC and CO increases while the emission of NOx decreases.

Based on this result, the selection of the best Diesel fuel-MFO mixture depends on certain criteria.

- The reduction of torque and BHP must be in an acceptable value. The reduction should be not more than 15% [25].
- Increase in BSFC must be in acceptable value.
- Maximum cost reduction without sacrificing performance and emission aspect.
- Low exhaust emission characteristic, namely NOx, CO and HC emission.

Based on the criteria above, the best Diesel fuel-MFO mixture is Diesel fuel with 5% MFO. This mixture only reduces the torque and BHP up to 13.57% and 13.55% respectively. Among all the mixture tested, it also has the lowest BSFC value. In exhaust emission characteristics, 5% MFO produce less CO and unburned HC emission compared to the other MFO mixture. On top of that, it also reduced the NOx emission (up to 16.8%) compared when using 100% Diesel. The price per liter for 5% MFO is still lower compared than RM 2.07 per liter for diesel fuel.

4.7 Summary

The values of 100% Diesel fuel are used as baseline in this analysis of performance and emission characteristics. From the study conducted, the suitable viscosity for diesel engine is 1.06cP to 3.36cP. From the results obtained, it shows that the addition of MFO have reduced the performance aspect. BHP and torque experienced a drop in performance up to 56.74% and 56.67% respectively, compared when using 100% Diesel fuel. Whereas for BSFC, the value increased up to 223.86% compared with the normal value. This means that the fuel consumption is doubled when adding more MFO into the fuel blend.

For emission characteristics, a reduction on NOx quantity down to 34.40% has been observed. Carbon Monoxide has increased up to28.27% while the HC emission does increase with increasing percentage of MFO up to 1120 ppm. Both CO and HC emission can also be observed naked eye by referring to the particulate matter produced (smoke and soot), since smoke and soot can act as an indicator of amount of CO and unburned HC produced. It is found that the price per liter of the fuel can be decrease with the addition of MFO, from RM 2.07 per liter initially down to RM 1.86 per liter.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

This project is being carried out with the goal of studying the engine performance and exhaust emission characteristics of Diesel fuel-Fuel oil (MFO) mixture in a single cylinder engine. It is found that the suitable viscosity for diesel engine is between 1.06cP to 3.36cP [14]. The maximum amount of MFO that can be added is 31%.

For performance characteristics, conclusions obtained are;

- BHP and torque experienced a drop in performance up to 56.74% and 56.67% from normal performance using 100% Diesel.
- BSFC experienced an increment more than double, up to 223.56% from its normal value.

For exhaust emission characteristics, conclusions that can be made are;

- NOx emission decreased as more MFO being added into the fuel mixture, up to 34.40%.
- The emission of CO and HC increased as the amount of MFO increased.

From this project, it can be concluded that Fuel oil (MFO) can be used as an alternative fuel source for diesel engine, but with certain lose in performance and emission characteristics. Economically, using MFO only give minor price reduction, down to 10.14%.

5.2 Recommendation

For future work, it is recommended that durability and reliability testing should be done to identify if there is any negative effect towards engine parts when using this fuel mixture. It is proposed that the engine should be running in long hours, as example 8 hours continuously. The performance and emission must be monitored. The engine can be disassembled later to check the effect of fuel mixture towards parts such as piston.

Also for continuation of this project, it is recommended that more narrow range of Diesel fuel-MFO fuel mixture is studied. In this project, the amount of MFO tested by percentage is 5%, 10%, 15% and 20%, with 5% increment. For future work, the study should test MFO percentage with 1% difference, as this can give more accurate MFO mixture for recommendation.

Last but not least, in this project, no modification is done to the engine. For future work, some modification can be done to the engine. For example, heater can be used to heat the fuel before entering the combustion chamber. The rise of temperature can reduce the mixture viscosity. This mean that more percentage of MFO can be add. Thus it can help it determining the suitable percentage of MFO. Besides that, modification can also been done so that the performance reduction and emission characteristics can be improved.

REFERENCES

[1] Moran Michael J. and Howard N. Shapiro, 2004, *Fundamentals of Engineering Thermodynamics,* John Wiley and Sons, 406-408

[2] Cengel Yunus A. and Michael A. Boles, 2006, *Thermodynamics, An Engineering Approach,* Mc Graw Hill, 500-502

[3] Wikipedia, The Free Encyclopedia< http://en.wikipedia.org/wiki/Fuel_oil> assessed on 7th February 2008

[4] Energy Concepts, Characteristics of No 6 (Residual) Fuel Oil <http://energyconcepts.tripod.com/energyconcepts/heavy_oil.htm> assessed on 7th February 2008

[5] ABC Impact, Marine Fuels, assessed on 7th February 2008 <http://dev.ulb.ac.be/ceese/ABC_Impacts/glossary/marinefuels.php>

[6] Altun S, et al. 2008, "The comparison of engine performance and exhaust emission characteristics of sesame oil-Diesel fuel mixture in a direct injection diesel engine" in *Renewable Energy (2008)* Science Direct

[7] Energy Boss, Marketing LLC, Bunker Fuel Specification Sheet

[8] Topaz Energy Ltd, Production Supply Specification, *Medium Fuel Oil-1.00% Sulfur* 1 Dec 2005.

[9] http://www.mymesra.com.my, assessed on 21st February 2008

[10] Lin CY, Lin HA. "Diesel engine performance and emission characteristic of biodiesel produced by the peroxidation process." Fuel 2006; 85:298-305

[11] Doraldo MP,Ballesteros EA, Arnal JM, Gomez J, Lopez FJ. "Exhaust emission from a diesel engine fuel with transesterified waste olive oil." Fuel 2003;82(11):1311-5

[12] Agarwal D, Sinha S,Argawal AK. "Experimental investigation of control of NO_x emissions in biodiesel-fueled compression ignition engine." Renew Energy 2006;31:2356-69

[13] Operating Manual-TD110-TD115 Test Bed and Instrumentation for Small Engines, TQ Education and Training Ltd.

[14] Biodiesel Association of Canada<http://www.greenfuels.org/biodiesel/tech/ASTM-D975.pdf>,assessed on 20th February 2008.

[15] http://www.gtisoft.com/broch_gtsuite.html, assessed on 13th August 2008.

[16] Degobert, P, Automobiles and Pollution, Society of Automotive Engineers, Warrendale, Penn., 1995.

[17] Holman J.P. Experimental methods for engineers (6th ed.), McGraw-Hill, New York (1994).

[18] Edwards, J., Combustion : Formation and Emission of Trace Species, Ann Arbor Science Publishers, Inc., Ann Arbor, Mich., 1974.

[19] Heywood, J., Internal Combustion Engine Fundamentals, McGraw Hill, Inc., New York, 1988.

[20] Schafer, F. and van Basshuysen, R., Reduced Emissions and Fuel Consumption in Automobile Engines, Springer-Verlag, Vienna, 1993.

[21] Nayarajan G. A.N Rao & S Renganarayan, "Emission Performance Characteristics of Neat Ethanol Fuelled DI Diesel Engine." International Journal of Ambient Energy 23 (3) pp 149-158, 2002.

[22] Hebbal O.D, K. Vijayakumar Reddy, K. Rajagopal, Science Direct, vol 85, issues 1445 page 2187-2194, Oct 2006.

[23] Ganesh D, G Nagarajan and M. Mohamed Ibrahim, "Study of Performance, Combustion and Emission Characteristics of Diesel Homogenous Charge Compression Ignition (HCCI) Combustion with External Mixture Formation." Science Direct, Volume 87, Issues 17-18, December 2008, Pages 3497-3503.

[24] Argawal AK. "Biofuel (alcohol and biodiesel) application as fuels for internal combustion engines." Prog Energy Combust Sci 2007;33(3):233-310

[25] James P, 2003, "An Intro to E-Diesel Commercialization & Standard." O_2 Diesel Inc, California

APPENDICES

APPENDIX 3.1

Procedure for calibration of instrumentation unit

Calibration Procedures:

1. Set the Span control to its maximum clockwise position.
2. Shake the engine vigorously to overcome the stiction of bearing seals.
3. Adjust the Zero control until the torque meter reads zero.
4. Check that the Zero is accurate by shaking the engine again.
5. Hang a load of 3.5 kg on the calibration arm of the TD115.
6. Shake the engine until the torque meter reading settles down to a constant value.
7. Adjust the Span control to give a torque reading of 8.6 Nm.
8. Remove the calibration load and repeat steps (2) to (8) until satisfied Zero and Span settings are correct.

Test Procedures:

1. Start the engine.
2. Advance the throttle or rack control to its maximum position.
3. Note the maximum speed of the engine. (The dynamometer water flow should still be the trickle used for starting).
4. Keep the throttle or rack control open and slowly adjust the needle valve to increase the flow of water through the dynamometer until the needle valve is fully open. Note the engine speed.
5. Choose at least five speeds between the two extremes at which to take readings of engine performance.
6. Keep the throttle open and reduce the water flow to a trickle, so that the engine returns to its maximum speed.
7. When the engine has settled down to a steady output, record the readings of speed, torque, exhaust temperature and air consumption. Operate the fuel tap beneath the pipette so that the engine takes its fuel from the pipette. Time the consumption of 8 ml of fuel. Turn the tap so that the pipette again fills. Enter the data in the Table.

8. Check that the temperature of the water flowing out of the dynamometer is less than 80°C. If the temperature is higher than this, increase the water flow to cool the dynamometer bearing seals.
9. Increase the flow of water into the dynamometer until the engine speed drops to the next highest selected value. Because the time response of the dynamometer is fairly slow, the needle valve has to be operated slowly.
10. Allow time for the engine speed to stabilize before taking another set of results. If the dynamometer is too sensitive to obtain the desired speed, it will help if the drain tap is partially closed. Do not close fully.
11. Repeat step (9) until the dynamometer needle valve is fully open.

APPENDIX 3.2

Procedure for emission characteristics experiment

Procedures:

1. The content and equipment must be checked to make sure it is in good condition.
2. Assemble every parts of gas analyzer properly.
3. Switch on the analyzer.
4. The analyzer will display 30 seconds stabilizing sequence. Let the sequence completed.
5. Ensure that the pump is running and operating.
6. Checked the battery level. It is advisable to be above 40% or else it should be charged.
7. Place the probe inside the exhaust pipe.
8. Press "Data" button to capture the emission data. Make sure that the data is for desired engine speed.
9. Wait until the reading become stable. Press "Record" button to record the data.
10. Repeat step (7) to (9) for every desired engine speed.
11. Tabulate all readings obtained and analyze the result.

APPENDIX 4.1

Sample Calculation of Converting cSt to cP

The range of Diesel fuel kinematic viscosity (based on ASTM D975) is about 1.3-4.1 mm^2/s (at 40°C) [14]. Using the conversion and equation below, the range of viscosity in cP unit is 1.066cP – 3.362 cP. (given that SG for diesel is 0.82)

1 cSt = 1 mm^2/s

cP = (cSt)(SG)

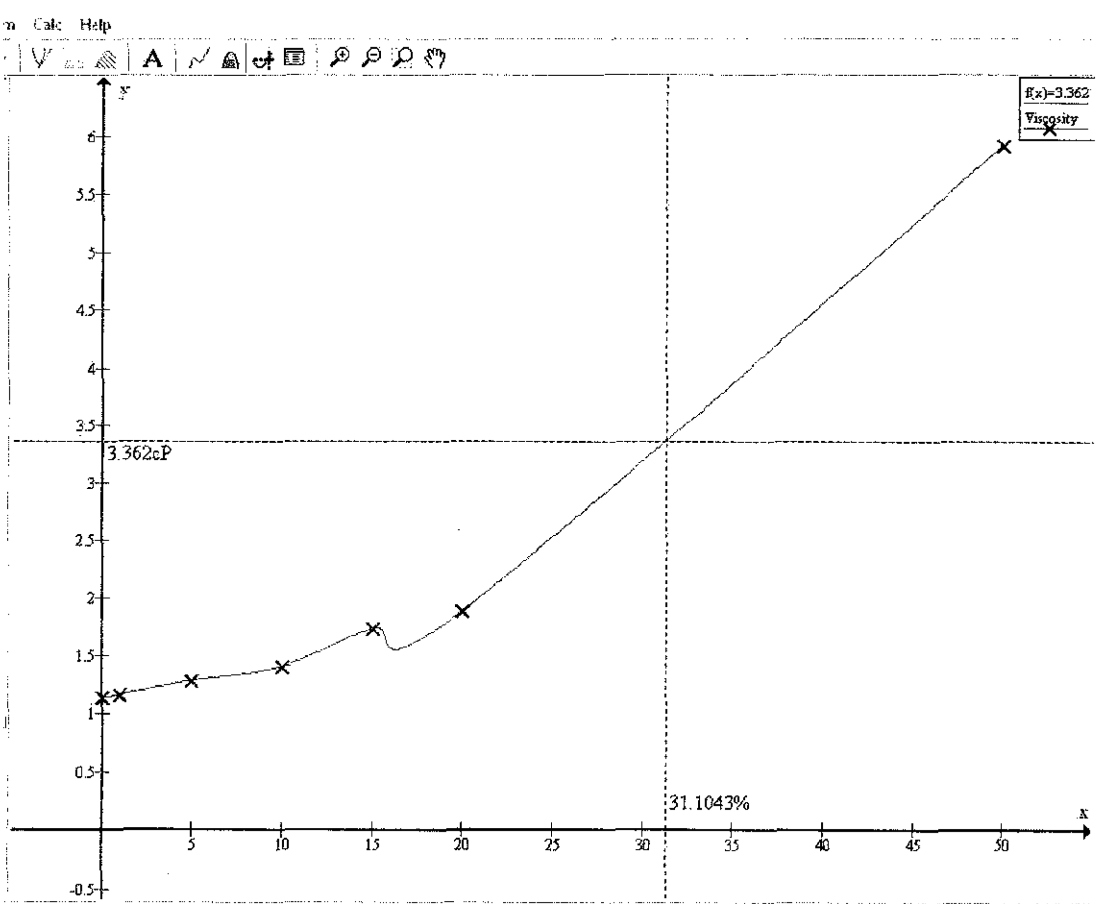
for 1.3 mm^2/s = 1.3 cSt and 4.1 mm^2/s = 4.1 cSt,

thus,

cP = (1.3)(0.82)=1.066 cP

cP = (4.1)(0.82)=3.362 cP

Thus, the minimum value of viscosity is 1.066 cp and the maximum value of viscosity is 3.362 cP. Graph of viscosity and MFO percentage is constructed. The maximum percentage of MFO is determined when the viscosity is 3.362cP, as below.



APPENDIX 4.2

Typical spreadsheet layout for result calculations

	A	B
1	Engine Type	SI/Diesel/4 Stroke
2	Date	
3	Fuel	Diesel/Diesel-MFO
4	. Specific gravity	0.84 (Diesel)
5	Barometric pressure (kPa)	Read Barometer
6	Ambient Temperature (°C)	Read Thermometer
7	Air Density (kg/m^3)	= B5x1000/(287x(B6+273))
8	Speed (rev/min)	Read Tachometer
9	Torque (Nm)	Read Gauge
10	Brake power (kW)	= 2πxB9xB10/6000
11	Fuel; time for 8 ml (s)	Read Stopwatch
12	Fuel mass flow rate (kg/hr)	= B4x8x3.6/B12
13	Specific fuel consumption (g/kWh)	= B13x1000/B11

APPENDIX 4.3

Sample calculation for Corrected BHP **(100% Diesel fuel at 1500 rpm)**

According to Heywood, Corrected Brake Power, $P_{bc} = C_F\ P_{bm}$ [19]

While $C_F = [P_{sd}/P_m\text{-}P_r]/[T_m/T_s]^2$

Saturated water vapor pressure $P_v = P_{v,\,sat} - Ø$ at ambient temperature

$Log_{10}\ P_{v,\,sat} = 8.10765 - (1750.286/T_m + 235.15)$

Take $T_m = 35.9\ °C$

$Log_{10}\ P_{v,\,sat} = 8.10765 - (1750.286/35.9 + 235.15)$

$= 1.6466$

$P_{v,\,sat} = 10^{1.6466}$

$= 44.32$ mmHg or 5.909 kPa

Ambient-water vapor pressure, Pv (assume relative humidity $Ø = 0.65$)

$P_v = P_{v,\,sat}\ x\ Ø$

$= 5.909\ x\ 06.5$

$= 3.8439$ kPa

Given that $P_{sd} = 98.274$ kPa, $T_s = 29.4°C$ and $P_{bm} = 0.785$ (obtain from experiment), the power correction factor C_F,

$C_F = [P_{sd}/P_m\text{-}P_r]/[T_m/T_s]^2$

$= [98.274/\ (102\text{-}3.8439)]/[271.05/302.55]^{0.5}$

$= 1.0119$

$P_{bc} = C_F\ P_{bm}$

$= 1.0119\ x\ 0.785 =$ <u>0.794 kW or 1.065 hp</u>

APPENDIX 4.4

Table for each engine performance and emission characteristic

Torque value for every mixture

Engine Speed (rpm)	Torque (Nm)				
	100% Diesel fuel	Percentage of MFO in mixture			
		5 %	10%	15%	20%
1500	5.00	4.50	4.00	3.50	3.00
2000	7.00	6.00	5.00	4.00	3.50
2500	9.50	8.00	6.50	5.00	4.00
3000	12.50	11.00	7.50	6.00	5.00
3500	14.25	12.00	9.00	6.50	3.50

BHP value for every mixture

Engine Speed (rpm)	BHP value (kW)				
	100% Diesel fuel	Percentage of MFO in mixture			
		5 %	10%	15%	20%
1500	0.785	0.707	0.628	0.549	0.471
2000	1.466	1.257	1.047	0.838	0.733
2500	2.487	2.094	1.702	1.308	1.047
3000	3.927	3.455	2.356	1.884	1.570
3500	5.223	4.398	3.298	2.382	1.282

Corrected BHP value for every mixture

Engine Speed (rpm)	Corrected BHP value (hp)				
	100% Diesel fuel	Percentage of MFO in mixture			
		5 %	10%	15%	20%
1500	1.065	0.958	0.852	0.744	0.639
2000	1.988	1.705	1.42	1.137	0.995
2500	3.374	2.841	2.309	1.775	1.416
3000	5.328	4.688	3.196	2.556	2.13
3500	7.087	5.967	4.519	3.231	1.739

BSFC value for every mixture

Engine Speed (rpm)	BSFC (g/kW-hr)				
	100% Diesel fuel	Percentage of MFO in mixture			
		5 %	10%	15%	20%
1500	267.85	361.54	444.53	531.42	651.01
2000	161.78	220.19	281.20	368.92	462.80
2500	133.25	181.61	238.31	325.03	427.23
3000	111.17	156.02	221.49	314.89	409.37
3500	97.45	144.70	202.07	300.64	391.66

Exhaust temperature value for every mixture

Engine Speed (rpm)	Exhaust Temperature (°C)				
	100% Diesel fuel	Percentage of MFO in mixture			
		5 %	10%	15%	20%
1500	200	210	220	240	250
2000	260	270	290	300	310
2500	320	340	350	360	370
3000	360	380	410	435	450
3500	390	430	460	480	500

Percentage increment of exhaust temperature value for every mixture

Engine Speed (rpm)	Percentage increment of Exhaust Temperature (°C)			
	Percentage of MFO in mixture			
	5 %	10%	15%	20%
1500	5.00	10.00	20.00	25.00
2000	3.48	11.54	15.38	19.23
2500	6.25	9.38	12.50	15.63
3000	5.55	13.89	20.83	25.00
3500	10.26	17.95	23.08	28.21
% Avg	6.11	12.55	18.36	22.61

NOx emission value for every mixture

Engine Speed (rpm)	NOx (ppm)				
	100% Diesel fuel	Percentage of MFO in mixture			
		5 %	10%	15%	20%
1500	169	147	133	124	113
2000	193	153	141	129	119
2500	273	218	207	175	157
3000	475	403	375	351	338
3500	791	673	633	601	560

CO emission value for every mixture

Engine Speed (rpm)	CO (ppm)				
	100% Diesel fuel	Percentage of MFO in mixture			
		5 %	10%	15%	20%
1500	277	305	327	338	346
2000	281	323	332	343	354
2500	330	391	417	431	446
3000	441	507	521	548	573
3500	551	634	650	673	691

Unburned HC emission value for every mixture

Engine Speed (rpm)	Unburned HC (ppm)				
	100% Diesel fuel	Percentage of MFO in mixture			
		5 %	10%	15%	20%
1500	180	480	510	570	630
2000	250	570	650	730	810
2500	330	640	720	810	870
3000	370	710	850	930	980
3500	460	850	910	1000	1120

APPENDIX 4.5

Engine performance obtained by GT Power software and experiment

Performance characteristics using 100% Diesel fuel

Engine Speed (rpm)	Torque (Nm)			Brake Horsepower (hp)			BSFC (g/kW-hr)		
	GT Power	Experiment	% difference	GT Power	Experiment	% difference	GT Power	Experiment	% difference
1500	4.81	5.00	3.80	1.01	1.05	2.00	285.81	267.85	6.70
2000	7.24	7.00	3.42	2.02	1.96	3.06	166.97	161.78	3.20
2500	9.89	9.50	4.11	3.55	3.33	6.60	121.39	133.25	8.90
3000	12.91	12.50	3.28	5.51	5.26	4.75	102.28	111.17	7.99
3500	13.81	14.20	2.74	6.73	7.00	3.86	108.26	97.45	11.09

Performance characteristics using 5% MFO mixture

Engine Speed (rpm)	Torque (Nm)			Brake Horsepower (hp)			BSFC (g/kW-hr)		
	GT Power	Experiment	% difference	GT Power	Experiment	% difference	GT Power	Experiment	% difference
1500	4.31	4.50	4.22	0.91	0.95	4.21	386.39	361.54	6.87
2000	6.11	6.00	1.83	1.73	1.68	5.00	228.24	220.19	3.66
2500	8.41	8.00	5.12	2.97	2.80	6.07	169.80	181.61	6.50
3000	11.43	11.00	3.90	4.79	4.63	3.46	151.02	165.02	8.48
3500	11.61	12.00	3.25	5.71	5.89	3.06	162.12	144.70	12.04

Performance characteristics using 10% MFO mixture

Engine Speed (rpm)	Torque (Nm)			Brake Horsepower (hp)			BSFC (g/kW-hr)		
	GT Power	Experiment	% difference	GT Power	Experiment	% difference	GT Power	Experiment	% difference
1500	3.83	4.00	4.25	0.87	0.84	3.57	468.93	444.53	5.48
2000	4.81	5.00	3.80	1.34	1.40	4.28	302.61	281.20	7.61
2500	6.67	6.50	2.61	2.41	2.28	5.70	233.15	238.31	2.17
3000	8.15	7.50	8.66	3.38	3.15	7.30	210.94	221.49	4.76
3500	8.53	9.00	5.22	4.11	4.39	6.37	228.36	202.07	13.01

Performance characteristics using 15% MFO mixture

Engine Speed (rpm)	Torque (Nm)			Brake Horsepower (hp)			BSFC (g/kW-hr)		
	GT Power	Experiment	% difference	GT Power	Experiment	% difference	GT Power	Experiment	% difference
1500	3.58	3.50	2.28	0.76	0.73	4.11	556.44	531.42	4.70
2000	4.15	4.00	3.75	1.16	1.12	3.57	381.62	368.92	3.44
2500	5.24	5.00	4.80	1.85	1.75	5.71	335.51	325.03	3.22
3000	6.44	6.00	7.33	2.66	2.52	5.55	307.98	314.89	2.19
3500	6.27	6.50	3.53	3.01	3.19	5.64	320.91	300.64	6.74

Performance characteristics using 20% MFO mixture

Engine Speed (rpm)	**Torque (Nm)**			**Brake Horsepower (hp)**			**BSFC (g/kW-hr)**		
	GT Power	**Experiment**	**% difference**	**GT Power**	**Experiment**	**% difference**	**GT Power**	**Experiment**	**% difference**
1500	3.15	3.00	5.00	0.66	0.63	4.76	675.18	651.01	3.71
2000	3.58	3.50	2.28	1.01	0.98	3.06	485.67	462.80	4.94
2500	4.30	4.00	7.50	1.51	1.40	7.85	417.73	427.23	2.23
3000	5.28	5.00	5.60	2.19	2.10	4.28	398.79	409.37	2.58
3500	4.03	3.50	15.14	2.07	1.71	3.66	406.39	391.66	3.76

APPENDIX 4.6

Sample calculation for price per liter (5% MFO)

Price per liter of Diesel fuel (unsubsidized) = RM 2.07

Price per liter of MFO fuel = RM 1.03

In 1 liter of 5% MFO mixture consist of 950 ml Diesel fuel and 50 ml MFO.

Thus, for 950 ml Diesel fuel,

= (950/1000) x RM 2.07

= RM 1.96

For 50 ml MFO fuel,

= (50/1000) x RM 1.03

= RM 0.05

Total cost per liter of 5% MFO

= RM 2.07 + RM 0.05

= RM 2.01

Percentage of reduction = (RM 2.07-RM 2.01)/RM 2.07 x 100%

= 2.41%

APPENDIX 4.7

TD111 Diesel engine specification

Engine Manufacturer	'Robin' - Fuji DY23D
Engine Provider	TQ Education and Training Ltd.
Model	TD111
Engine Type	Diesel, Single cylinder, 4 stroke
Piston Displacement	230 cm^3
Stroke	60 mm
Bore	70 mm
Compression ratio	2:1
Maximum Speed	3750 rpm
Net Weight	45 kg
Dimension	Nett: 460 x 420 x 440mm