

**COMPARATIVE STUDY ON COMBUSTION AND EMISSION
CHARACTERISTICS ON JATROPHA OIL BIODIESEL WITH
DIESEL**

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**RESEARCH PROJECT SUBMITTED IN PARTIAL
FULFILLMENT OF THE
REQUIREMENTS FOR THE DEGREE OF MASTER
OF ENGINEERING**

**FACULTY OF ENGINEERING
UNIVERSITY OF MALAYA
KUALA LUMPUR**

2012

Abstract

This study is about comparison of combustion and emission characteristics between biodiesel and diesel in internal combustion engine. Study on the performance, combustion and emissions were carried out using Jatropha Oil Biodiesel blends together with conventional diesel (JB10 and JB20) and pure fossil diesel (JB0). The results show that diesel engine operated by Jatropha blends has comparable output with diesel operated engine with different speeds, where the blends give slightly lower output for peak temperature, heat release rate, torque and brake power which is mainly caused by lower heating value of biodiesel compared to diesel. Exhaust emissions such as CO, CO₂ and HC seems to be lower for higher blends, credits to the excessive amount of oxygen in the biodiesel to promises cleaner combustion. The heat release rate has an effect on the exhaust emission as it determines the characteristics of flame. Due to analogous combustion and emissions characteristics of Jatropha biodiesel blends and diesel on the same engine type, Jatropha biodiesel blends are feasible to be used in internal combustion engine.

Abstrak

Kajian ini adalah tentang perbandingan ciri-ciri pembakaran dan pelepasan ekzos antara biodiesel dan diesel dalam enjin pembakaran dalaman. Uji kaji dan pengiraan ke atas pembakaran dan pelepasan ekzos telah dijalankan dengan menggunakan Jatropha Methyl Ester digabungkan bersama-sama dengan diesel konvensional (JB10 dan JB20) dan diesel tulen (JB0). Keputusan menunjukkan bahawa enjin diesel yang dikendalikan menggunakan campuran Jatropha dan diesel menghasilkan keputusan yang setanding dengan enjin diesel yang beroperasi pada throttle penuh pada kelajuan yang berbeza, di mana, campuran biodiesel dan diesel tersebut menghasilkan prestasi yang lebih rendah untuk suhu puncak, kadar pembebasan haba, tork dan kuasa brek atas sebab utamanya adalah nilai pemanasan biodiesel yang rendah bagi campuran biodiesel yang lebih tinggi berbanding diesel. Pelepasan ekzos seperti CO, CO₂, dan HC kelihatan lebih rendah bagi campuran yang lebih tinggi, kredit kepada kandungan oksigen yang tinggi dalam biodiesel yang membekalkan oksigen yang mencukupi untuk menghasilkan pembakaran yang lebih bersih dan sempurna. Kadar pembebasan haba juga dipercayai mempunyai kesan ke atas pelepasan ekzos kerana ia menentukan ciri-ciri nyalaan api. Disebabkan oleh ciri-ciri pembakaran dan pelepasan ekzos yang agak tidak jauh berbeza bagi campuran Jatropha biodiesel dan diesel pada jenis enjin yang sama, campuran Jatropha biodiesel adalah layak untuk digunakan di dalam enjin pembakaran dalaman sebagai bahan bakar alternatif.

Acknowledgement

I would like to express my deepest gratitude to Dr Md Abul Kalam, my dissertation supervisor for his guidance, encouragement, insightful opinions, and expertise throughout the process of my master dissertation. A special thanks to my examiner for their willingness to evaluate my report with patience and justice.

I would like to extend my greatest appreciation to colleagues for their valuable suggestions and critics for this study. Their enthusiastic assistance and positive encouragement are really appreciated. Not to forget, PhD student, Liaquat for his co-operative effort throughout the dissertation.

Last but not least, a special thanks to my family especially my wife for the supports and advices. This has been my greatest experience so far to catch the glimpse of the future in Mechanical Engineering field. I hope that I will be able to apply the knowledge that I have gained.

Table of Contents

Abstract	i
Abstrak	ii
Acknowledgement	iii
Table of Contents	iv
List of Figures	vii
List of Tables	ix
List of Symbols and Abbreviations.....	x
1.0 Introduction.....	1
1.1 Objectives.....	4
1.2 Scope of the study	5
2.0 Literature Review.....	6
2.1 Introduction	6
2.1.1 Vegetable oil.....	9
2.1.2 Production of biodiesel.....	10
2.2 Properties of biodiesel.....	11
2.2.1 Density of biodiesel	13
2.2.2 Viscosity of biodiesel	14
2.2.3 Flash point	16
2.2.4 Heating value	16

2.3 Diesel engine combustion analysis	17
2.3.1 Diesel Engine Operation.....	17
2.3.2 Heat release rate.....	19
2.4 Performance analysis.....	20
2.5 Diesel engine emission.....	20
2.5.1 Carbon dioxide (CO ₂).....	21
2.5.2 Carbon monoxide (CO)	21
2.5.3 Unburned Hydrocarbons (HC)	22
3.0 Methodology	23
3.1 Introduction	23
3.2 Biodiesel properties measurement	23
3.2.1 Density and Viscosity.....	24
3.2.2 Flash point measurement.....	26
3.2.3 Heating value measurement.....	27
3.3 Heat release rate and combustion analysis.....	28
3.4 Performance measurement	32
3.5 Emission performance.....	34
4.0 Result and Discussion	35
4.1 Introduction	35
4.2 Effect of biodiesel fraction on properties of fuels.....	35
4.2.1 Effect of biodiesel fraction on density.....	35

4.2.2 Effect of biodiesel fraction on viscosity	36
4.2.3 Effect of biodiesel fraction on heating value	38
4.2.4 Effect of biodiesel fraction on flash point	39
4.3 Effect of biodiesel fraction on combustion characteristics	41
4.4 Effect of biodiesel fraction on engine performance	47
4.5 Effect of biodiesel fraction on emission.....	49
5.0 Conclusion	52
6.0 References.....	54

List of Figures

Figure 1.1: GDP per energy use (USD per kg of oil)	1
Figure 1.2: Energy Use (kg of oil per capita)	2
Figure 1.3: Total Energy use (ktoe)	2
Figure 2.1: Jatropha L. Curcas	6
Figure 2.2: Seeds of Jatropha L Curcas	8
Figure 2.3: Stoichiometric Transesterification Reaction	10
Figure 2.4: Diesel engine cycle.....	18
Figure 2.5: Four stroke diesel engine mode operation.....	18
Figure 3.1: Homogenizer	23
Figure 3.2: Anton Paar (SVM 3000) Viscometer	24
Figure 3.3: Pensky-Martens Closed Cup Tester	26
Figure 3.4: IKA C 2000 Calorimeter	27
Figure 3.5: Piston cylinder diagram.....	29
Figure 3.6: Engine test bed setup.....	32
Figure 3.7: TF120M Diesel Engine	33
Figure 3.8: Data Acquisition Layout	33
Figure 3.9: Bosch petrol and diesel emission analyser	34
Figure 4.1: Density of biodiesel blends at 40°C	36
Figure 4.2: Kinematic viscosity of biodiesel blends (mm^2/s) at 40°C	37
Figure 4.3: Heating value of biodiesel blends (J/kg)	39
Figure 4.4: Volume vs crank angle	41

Figure 4.5: Piston speed vs crank angle.....	42
Figure 4.6: Adiabatic cylinder pressure vs crank angle	42
Figure 4.7: Adiabatic cylinder temperature	43
Figure 4.8: Mass burn fraction.....	44
Figure 4.9: Combustion pressure vs crank angle for biodiesel blends	45
Figure 4.10: Heat release rate crank angle for biodiesel blends	46
Figure 4.11: Cumulative heat release for biodiesel blends	47
Figure 4.12: Engine torque vs engine speed for biodiesel blends.....	48
Figure 4.13: Brake power vs engine speed for biodiesel blends.....	48
Figure 4.14: Unburned hydrocarbon (HC) emission vs engine speed for biodiesel blends	49
Figure 4.15: CO ₂ emission vs engine speed for biodiesel blends	50
Figure 4.16: CO emission vs engine speed for biodiesel blends	51

List of Tables

Table 2.1: Production of oil	7
Table 3.1: Volume fraction	24
Table 3.2: Engine specification for combustio analysis	28
Table 3.3: Air property values	29
Table 3.4: Engine specification for performance analysis.....	34
Table 4.1: Density of biodiesel blends at 40°C.....	35
Table 4.2: Kinematic viscosity of biodiesel blends (mm ² /s) at 40°C	37
Table 4.3: Heating value of biodiesel blends (J/kg)	38
Table 4.4: Flash point of biodiesel blends (°C).....	40

List of Symbols and Abbreviations

M_i	Mass fraction
Q_{HV}	Energy content
SG_{blend}	Specific Gravity of blend
SG_i	Specific Gravity
c_v	Specific heat content of air at constant volume
m_f	Mass of fuel in the combustion chamber
m_m	Total mass of fuel and air mixture
x_b	Mass burn fraction
θ_o	Crank angle where the combustion started
$\Delta\theta$	Combustion duration
B10	10% biodiesel + 90% Diesel
B100	100% biodiesel
B20	20% biodiesel + 80% Diesel
BDC	Bottom dead center
BMEP	Brake Mean Effective Pressure
BSFC	Brake specific fuel consumption
CDM	Clean Development Mechanism
CI	Compression-ignition engine
CO	Carbon Monoxide
CO ₂	Carbon Dioxide
CV	Calorific value
GDP	Gross Domestic Product
GHG	Green House Gases
HC	Unburned Hydrocarbon

HRR	Heat release rate
HV	Heating value
ICE	Internal Combustion Engine
JB0	0% Jatropha biodiesel + 100% Diesel
JB10	10% Jatropha biodiesel + 90% Diesel
JB20	20% Jatropha biodiesel + 80% Diesel
m	mass of fuel mixture
P	in-cylinder pressure
R	gas constant of air
SI	Spark-ignition engine
T	in-cylinder temperature
TDC	Top dead center
V	in-cylinder volume
V _c	Clearance volume
V _d	Displacement volume
x	Biodiesel fraction
γ	ratio of specific heats
η	Kinematic viscosity (mm ² /s)
θ	Crank angle
μ	dynamic viscosity (mPa-s)
ρ	density (kg/m ³)

1.0 Introduction

The development of transportation, industries and other needs depends greatly on energy especially from fuel. In 2010, the world energy use has been climbed more than 5%, with China and India show no signs of slowing down in energy consumption and keep continued their extreme demand for all forms of energy (Lapillonne, 2010)

Nowadays, an economic property of a nation can be determined by its per capita energy consumption. As can be seen in Figure 1.1, most of the developing countries' gross domestic product per kg of oil used to produce energy increases. This indicates that at the present time, energy plays a very important role in economy.

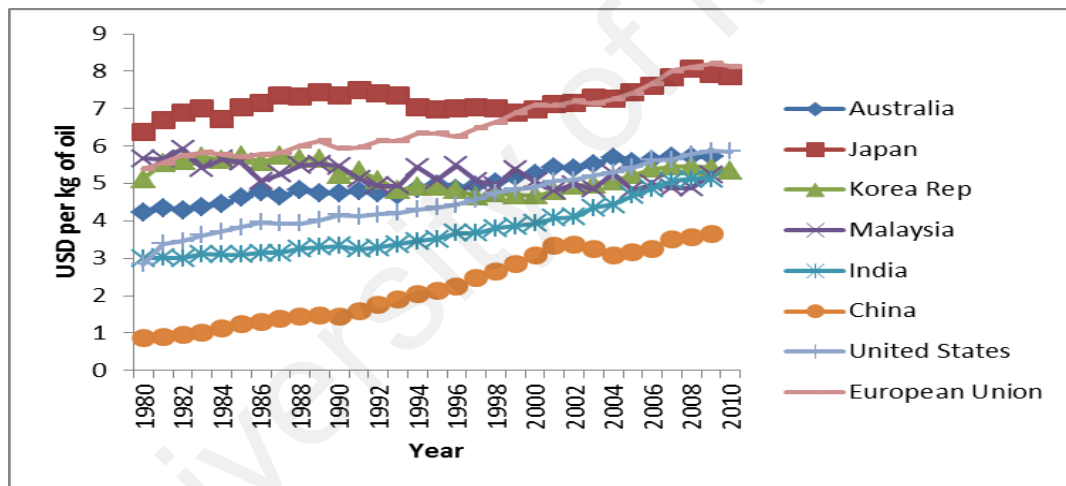


Figure 1.1: GDP per energy use (USD per kg of oil)

With respect to the outcome that gives more turnovers for more energy used, all countries in the world are now chasing higher profit by utilizing more fuel every year as can be seen in Figure 1.2 and Figure 1.3. Figure 1.2 represents the energy consumption per capita for several developing countries, as can be comprehended; most of the countries' energy consumption per capita has been increased persistently within 30 years. The total energy consumption of each of the countries can be obtained in Figure 1.3; the trend obviously

shows that the energy consumption keeps increasing every year because it contributes to the generation of income to those nations. Malaysia also seems to practise the same principle, as can be seen that the total energy consumption by Malaysia in Figure 1.3 reached highest new record at 72,589 ktoe in 2007.

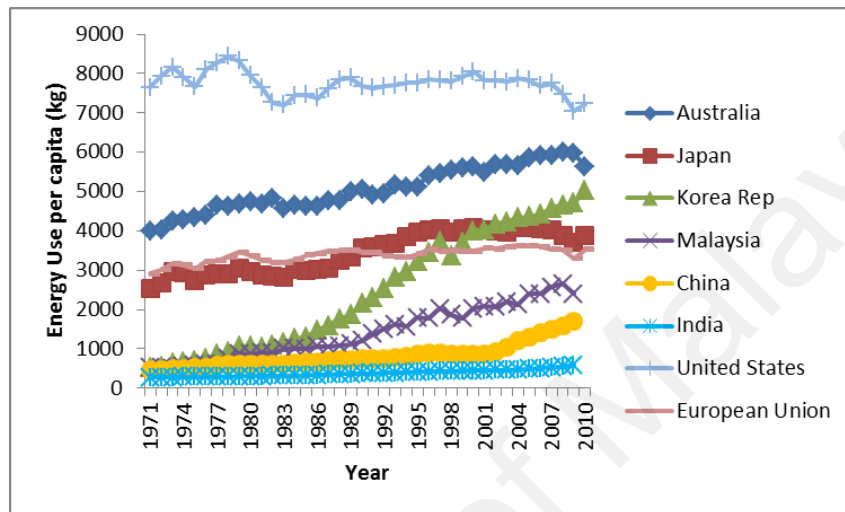


Figure 1.2: Energy Use (kg of oil per capita)

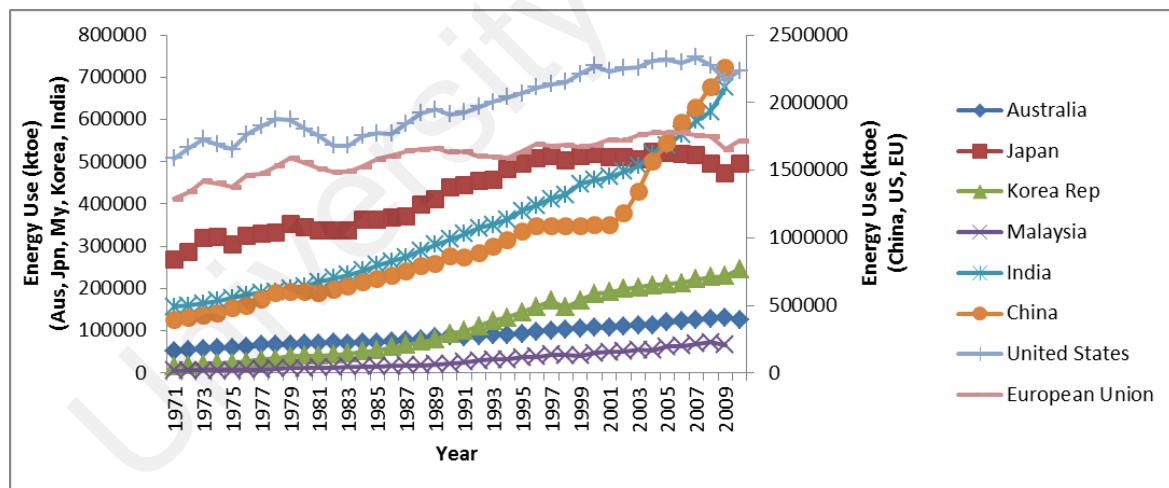


Figure 1.3: Total Energy use (ktoe)

The demand for fuels rises quickly all over the world from time to time. Diesel is one of the most important energy resources. Diesel engines are becoming more popular nowadays due to advantage in fuel efficiency (Kurani & Sperling, 1988) and low emission of CO₂, CO, HC (Hedegaard et al., 2008; Walsh, 1999). In order to protect the environment for long term

energy precautions, it is necessary to develop alternative fuels which have comparable combustion, performance and emission characteristics to the conventional diesel as substitutes. Before a severe calamitous stage arrives, the effort to search for alternative fuel based on renewable energy becomes highly crucial. Hopefully the adaptation of alternative fuels can help to solve these issues.

An alternative energy such as oil from fruits or vegetables, which is also known as biodiesel, is very encouraged to replace the scarce fossil oil resources. Now, bio-fuels are getting a new attention because of global stress on reduction of greenhouse gases (GHGs) and clean development mechanism (CDM). CDM is one of the policies that create opportunities to use biodiesels originating from developing countries to contribute to CO₂ emission reduction goals and solve energy supply matter (Rao, 2006). Biodiesel is becoming more significant and can be produced from vegetable oil resources. Some of them can be used directly while some of them still need to be formulated to have the similar properties to the conventional diesels. The important requirement for biodiesels to be a more maintainable as substitute for fossil diesel is, they must be from renewable raw material and the production of the biodiesel must has a lower negative impact on environment than that of fossil fuel (Janulis, 2004).

For the diesel engines, an important modification also need to be done to make it compatible with the use of vegetable oils and their derivatives as alternative fuels. On the other hand, the alternative fuels should be available to be obtained at low cost, environmental friendly and capable to fulfil energy security needs without sacrificing engine's operational performance.

Promptly increasing prices as can be seen in Malaysia recently and fears regarding petroleum availability have attracted scientists and researchers to put effort on alternative sources for fuel. This alternative fuel topic is very crucial in researches as it will not only help

to replace petroleum fuel as renewable energy, but also might help to recover the economy of certain countries to its better state.

1.1 Objectives

The objectives of this study are:

1. To compare the physico-chemical properties of Jatropha biodiesel blends and diesel.
2. To simulate the combustion characteristics of Jatropha blends and diesel in combustion chamber.
3. To analyse the engine performance of the blends and diesel in diesel engine.
4. To study the exhaust emission gas released during engine operation using the blends and diesel.

This is because, until now, there is no biodiesel yet has perform as good as diesel, thus expectantly the blends will give much more similar characteristics with diesel than the Jatropha biodiesel alone. The blends may help to reduce the consumption of diesel although it is not yet capable to provide 100% replacement of the conventional diesel.

1.2 Scope of the study

This study is using several equipment to measure the properties (Anton Paar SVM 3000 Viscometer), IKA C 2000 Calorimeter), performance (Yanmar TF120M Diesel Engine), and emission (Bosch emission analyser).

This study focuses on the comparison study of the performance, emission and combustion between diesel and alternative diesel which in this study we use Jatropha biodiesel and its blends. The type of the fuels used in this study are JB0 (0% Jatropha biodiesel + 100% Diesel), JB10 (10% Jatropha biodiesel + 90% Diesel) and JB20 (20% Jatropha biodiesel + 80% Diesel).

This study will analyse properties such as density, viscosity, flash point, and high heating value, emissions such as carbon dioxide (CO₂), carbon monoxide (CO), and unburned hydrocarbon (HC), combustion characteristics and performance such as heat release rate break mean effective pressure (BMEP), brake power output and torque.

The results of the study will show the comparison between conventional diesel and Jatropha biodiesel blends. From that, the compatibility and the capability of Jatropha biodiesel as an alternative fuel will be recognized.

2.0 Literature Review

2.1 Introduction

Jatropha L. Curcas (Figure 2.1) is a small tropical tree belongs to the Euphorbiaceae family that has multipurpose uses of significant economic importance (Aregheore et al., 2003). It is producing fruits that hold seeds containing oil that is toxic and not suitable for human and animal consumption (Makkar & Becker, 1997). However, the oil is extractable to become biodiesel that meets European and American standards (Azam et al., 2005).



Figure 2.1: *Jatropha L. Curcas*

Jatropha was once cited by investment bank Goldman Sachs as one of the best candidates for future biodiesel production (Barta, 2007). It is capable of producing seeds containing 27-40% oil, averaging 34.4%. (Achten et al., 2007; Achten et al., 2008). Table 2.1 shows the potential of *Jatropha L. Curcas* in producing oil among other crops.

Table 2.1: Production of oil (Fairless, 2007)

Biodiesel crop	Litres of oil per hectare
Oil palm	2,400
Jatropha	1,300
Rapeseed (canola)	1,100
Sunflower	690
Soya bean	400

Jatropha can be grown in all countries falling under tropical climate which means that Malaysia is very suitable to be a producer of the plant. Jatropha L. Curcas cultivates fine with about more than 600mm rainfall per year and it capable to withstand a long period of drought. It prefers a climate with temperatures 20°C to 28°C and can endure a very light frost. One tonne of Jatropha L. Curcas seeds will yield up to 600litres of biodiesel. Jatropha L. Curcas seeds can produce up to 60% oil content depending on (Palm Plantations Of Australia Pty Ltd, 2010):

1. Production capacity of the plants
2. Application of advanced pruning techniques
3. Improved pollination results from establishment of bee colonies
4. The moisture level of the soil
5. The nutrient level of the soil
6. Application of foliar fertilizer 30 days before harvest
7. Stage of ripening at harvest
8. Use of high quality processing equipment
9. Processing completed within 24 hours of harvest

There are several costs and benefits in cultivating *Jatropha* (Macintyre, 2007):

1. *Jatropha* needs at least 600mm of rain a year to flourish. But, it still can survive for three consecutive years of drought by dropping its leaves.
2. It is capable to prevent soil erosion, and soil around it will be enriched by the dropped leaves.
3. Alkaline soils are more preferred in growing the plant
4. The cost of 1,000 *Jatropha* saplings in Pakistan is about £50 (RM245)
5. The cost of 1kg of *Jatropha* seeds in India is about £70 (RM343). An area of two metres square needs to be provided for each *Jatropha* seeding.
6. 20% of seedlings planted will not survive
7. *Jatropha* seedlings yield seeds in the first year after plantation.

The seeds of *Jatropha* (Figure 2.2) are resulting *Jatropha* oil after it being crushed, and the oil can be burnt in a standard diesel car, whereas the residue of the crushed seeds can also be processed into biomass to power electricity plants (Macintyre, 2007). The oil content in the seed with cover is 25% to 30%, but without cover, it has 50% to 60% oil content in the kernel (Lele, 2011; Winkler et al., 1997).



Figure 2.2: Seeds of *Jatropha* L. Curcas

Many experiments about Jatropha biodiesel are on-going in several African countries, Malaysia, India, Brazil, China, and the Philippines because of the capability of these countries to produce the plant themselves. Recently, scientists in India have successfully used biodiesel from Jatropha on a car without any alterations to the engine. The diesel has been tested in a C Class Mercedes Benz and has completed about 6,500 kilometres across India with no deterioration happened to the engine with its usage (NTD Television, 2011).

2.1.1 Vegetable oil

The idea of using vegetable oil as an alternative to diesel has been started for a long time and not new. At the 1911 Paris Exposition, Dr Rudolph Diesel operated his diesel engine using peanut oil and declared that 'the diesel engine can be fed with vegetable oils and will help considerably in the development of the agriculture of the countries which utilize it'. Heavy duty vehicles in South Africa before World War II was one of the first vehicles functioned by using transesterified vegetable oil or also called as biodiesel (A. Demirbas, 2003).

Vegetable oils have become more important because of its advantages, although it got several disadvantages but yet still can be improved. Some of the advantages of vegetable oils are their renewability, biodegradability, lower aromatic content, lower sulphur content, high heat content although still lower than diesel, ready ability and liquid nature portability (A. Demirbas, 2003). Meanwhile, the disadvantages of vegetable oils are their higher viscosity, lower volatility and the reactivity of unsaturated hydrocarbon chains.

Vegetable oils are lipid material derived from plants such as palm oil, corn and Jatropha. Vegetable oils have almost equivalent energy density, heat of vaporization, cetane number and stoichiometric air/fuel ratio with conventional diesel. Moreover, they are also biodegradable, non-toxic and capable to extensively reduce the pollution from the emission. Vegetable oils may lead to significant reductions in emissions of carbon monoxide, sulphur oxides, smoke, poly aromatic hydrocarbons, and particulate matter (Bona et al., 1999;

Murayama, 1994). And more importantly, the role of bio-fuels to greenhouse effect is insignificant because the CO₂ produced from combustion can be directly used in the photosynthesis process in the plants (Agarwal, 1998; Narayan, 1992; Sharma & Jain, 2009).

The use of straight vegetable oil has met problem due to its high viscosity. Many engine inconveniences found while using raw vegetable oils as fuel such as carbon deposits on piston and head of engine, coking of injectors on piston and head of engine and excessive engine wear (Agarwal, 2007; Harwood, 1984; Ma & Hanna, 1999). To overcome this problem, transesterification of vegetable oils has been suggested by most researchers (De et al., 1999; Selmi & Thomas, 1998) to reduce viscosity. This transesterified vegetable oil is termed as biodiesel.

2.1.2 Production of biodiesel

Transesterification is a chemical reaction to convert vegetable oil to biodiesel. In this transesterification process, vegetable oil reacts with alcohol such as ethanol or methanol in presence of catalyst and results various components of vegetable oil break down to form new compounds of glycerol and esters as shown in Figure 2.3.

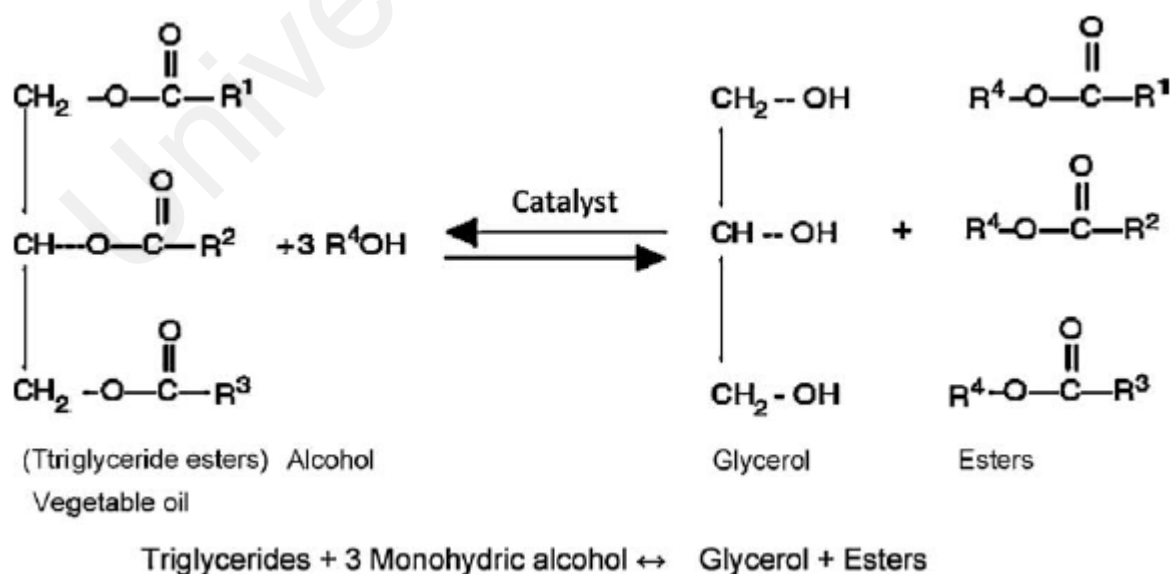


Figure 2.3: Stoichiometric Transesterification Reaction

The triglycerides in the vegetable oil are converted into alkyl esters which also well-known as biodiesel. The product of it depends on the alcohol used, methyl esters will be formed as if methanol is used, and ethyl esters will be formed as if ethanol is used. The reaction with alcohol is to separate the fuel from glycerine as alcohol will replace glycerine in the reaction. The separated glycerine during the process then will either float to the surface or sink to the bottom of vessel depending on its phase. The glycerine can be separated from the biodiesel using centrifuge.

The advantage of the transesterification process is; it produces lesser viscosity biodiesel which makes it capable to substitute diesel used to operate diesel engines (Basha et al., 2009; Korus et al., 1985; Ramadhas et al., 2005). This process is removing the obstruction of high viscosity of vegetable oil which was the major hindrance for it to be used in diesel engine. Transesterification may reduce viscosity to a certain level but the transesterification of waste cooking oil has need of post-processing and an extra stage from raw material to fuel (Hribernik & Kegl, 2009).

However, the process is costly and followed by a time consuming process separating the fuel from glycerine causes the final product price beats the price of fossil diesel fuel (Kalam et al., 2010).

2.2 Properties of biodiesel

Biodiesel is considered as future diesel fuel. It is non-flammable and also non-explosive as it has higher flash point compared to diesel. Biodiesel is also biodegradable, non-toxic and could reduce emissions when combusted. Biodiesel also requires no engine modification but accomplishes substantial performance and emissions. As biodiesel is still new compared to other established fuel, the production of biodiesel is still under improvement and very costly which make it not widely used as vehicles fuel. Furthermore, the current worldwide vegetable oil production is still not enough to substitute the petroleum based diesel use around the

world. Biodiesel is currently also not capable to fully replace the diesel because of its drawback such as high viscosity and low energy content which means higher amount of biodiesel compared to diesel is needed to give the same amount of heat energy as per unit diesel which may cause very high brake specific fuel consumption.

Biodiesel blends are frequently represented as; “BXX” with “XX” indicates the percentage amount of biodiesel contained in the mixture. As an example, B10 is 10% of biodiesel and 90% of petroleum diesel. Other than that, pure biodiesel or B100 has strong solvent properties that release rust and other engine contaminants, which may plug oil filters and fuel injectors, and degrade natural rubber gaskets and hoses in vehicles.

B20 blend has been the most demanded fuel. India aims to use 20% Jatropha biodiesel blend with fossil diesel by 2017 (Agoramoorthy et al., 2009; Fairless, 2007). It is because of the small amounts of biodiesel performances as a lubricant in the ultra-low sulphur diesel fuel to aid with lubrication to protect from engine wear, as the sulphur compounds in petroleum diesel provide much of the lubrication. After numerous researches done, it is determined that by using B20 will improve lubrication of the engine, reduce unwanted odours from exhaust gas and assist in cleaning injectors, fuel lines, pumps and tanks. (Boat U.S. Foundation, 2011)

B100 or higher blends levels require equipment and exceptional handling. The attempt to use the blends can be managed with heaters and changing the materials of engine seal and gasket. Nonetheless, a very special maintenance is needed to sustain the engine, consequently the U.S Department of Energy and National Renewable Energy Laboratory do not recommend the use of high blends level biodiesels, apart from where the exposure of diesel particulate matter elevated exposed to human and health anxieties worth the attention to equipment and fuel handling (NREL, 2009).

As for now, biodiesel is the best alternative fuel that has low emission and good performance. The use of biodiesel in a conventional compression ignition engine shows a

significant reduction of carbon monoxide, unburned hydrocarbons and particulate matter compared to the emissions released from the combustion of diesel fuel. While the production of nitrogen oxides is either slightly increase or slightly decrease depending on the method of testing used and the duty cycle of the engine.

2.2.1 Density of biodiesel

Density is the ratio of the mass to the volume of a particular entity. The density of biodiesel usually varies in the range of 800 and 900 kg/m³. Most of the studies done previously observed that the density of biodiesel does not change significantly because the densities of oil are close to the density of the pure 100% biodiesel (Graboski & McCormik, 1998).

Specific gravity is defined as a ratio of density of a substance which is either liquid or solid to the density of water at a specific pressure and temperature. Density is an important property in fuel; it may affect the performance characteristics such as cetane number and heating value (Mustafa & Jon, 1999). The variations in the fuel density will influence engine output power due to a different mass of fuel as the diesel fuel injection is measured using volumetric parameter (Bahadur et al., 1995).

Since the biodiesel is typically denser than diesel, the density of biodiesel blends increasing with the increase of biodiesel in volume fraction. An equation has been suggested by Clements to determine the specific gravity of different blends of biodiesel at a particular temperature (Mustafa & Jon, 1999). The specific gravity of the blend has been reflected to be proportional to mass fractions of the constituents in the equation 2.1.

$$SG_{blend} = \sum SG_i M_i \quad (2.1)$$

Where SG_{blend} is the specific gravity of the blend, SG_i is the specific gravity of the component i, and M_i is the mass fraction of the component i.

On the other hand, 1st degree empirical equation has been proposed to relate the density of a biodiesel with a percentage of a biodiesel present in a mixture as shown in equation 2.2 (Alptekin & Canakci, 2008).

$$D = Ax + B \quad (2.2)$$

Where D is density (g/cm^3), A and B are constants which differ with the type of the biodiesel, and x is the biodiesel fraction.

Even though the density of biodiesel is greater, the heating value which indicates the amount of energy content in fuel of biodiesel is lower on both mass and volume basis compared to diesel. Due to this drawback, more biodiesel need to be injected into the combustion chamber in order to give the engine the same amount of power as diesel which may cause biodiesel to have higher brake specific fuel consumption.

2.2.2 Viscosity of biodiesel

Viscosity is a measure of the fluid's resistance to flow. The viscosity of biodiesel is higher than the viscosity of diesel and previous research stated that the biodiesel viscosity can be up to 1.6 times that of diesel at 40°C (Bhale et al., 2009).

There are two types of viscosity; dynamic and kinematic. Dynamic viscosity is the tangential force per unit area required to move one horizontal plane with respect to the other at unit velocity when maintained a unit distance apart by the fluid, while kinematic viscosity is the ratio between dynamic viscosity and density.

Viscosity is one of the most significant elements that majorly affect the combustion quality of fuels. In diesel engine, the fuel is sprayed into compressed air and atomized into

small droplets close to the nozzle exit. The viscosity of fuels contributes different effects on the atomization quality, size of fuel drop and penetration (Heywood, 1988; Lichty, 1967). A high viscosity liquid fuel leads to improper atomisation which in turn tips to incomplete combustion. In long term of high viscosity biodiesel usage, it will commonly cause the development of gumming, ring sticking, the formation of injector deposits and incompatibility with conventional lubricating oils (Ryan et al., 1984). Furthermore, the specific fuel consumption and the exhaust gas temperature also expected to be reduced as the viscosity decrease (Pramanik, 2003).

The correct atomisation which takes place in the fuel injectors will be hard to achieve due to the fuel's high viscosity. The healthy mixing of fuel and air will only occur by experiencing very efficient atomisation. Hence, it is substantial to set the viscosity value in the right range. The high viscosity issue of vegetable oils has been approached in some ways, such as oils preheating, blending with other fuels with lower viscosity like conventional diesel, transesterification and thermal pyrolysis (Pramanik, 2003).

Alptekin and Canakci (2008) had studied about the correlation between viscosity and different percentages of blends of biodiesel. The test was done on several biodiesels at 40°C such as palm oil biodiesel, soybean oil biodiesel, corn oil biodiesel, sunflower oil biodiesel and conventional diesel. A second degree empirical equation has been developed to estimate the viscosity of blends by taking the fraction of biodiesel in the mixture as the main variable.

$$\eta = Ax^2 + Bx + C \quad (2.3)$$

Where η is the kinematic viscosity (mm^2/s), A, B, and C are coefficients depend on the type of biodiesel used, and x is the biodiesel fraction. This empirical formula can only be used for constant temperature as for the viscosity equation for varying temperature may use much more complicated equation.

2.2.3 Flash point

Flash point is the lowest temperature which is corrected to the standard atmospheric pressure of 101.3 kPa (760mmHg) where the vapour of a specimen will ignite under specified test conditions when a test flame applied to it. The temperature of the liquid must be adequately high to create vapour concentration above the liquid surface at or above the lower flammability limit of the vapour to form a flammable vapour-air mixture above the surface of a flammable liquid.

Flash point is important in defining flammable and combustible liquids in transportation and safety protocols. Typically, flash point is measured in air. Flash points could be affected by the type of oxidant gases employed in specific process. Many chemical processes might relate to oxidant gases such as oxygen, chlorine and oxides of nitrogen, which could affect the flash point of the sample. Flash point is also used to determine the potential of vapour explosion in a particular process. Previous study on flash point shows that biodiesel has higher flash point compared to diesel (NREL, 2009).

The flash point of diesel fuel is the lowest temperature at which the fuel will be ignited under identified conditions. Flash point temperature is a very important aspect to be known in dealing with diesel fuel and safety to avoid any undesirable circumstances. Due to high flash point temperature, diesel is essentially safer than gasoline.

2.2.4 Heating value

The other most important properties that could distinguish a fuel are its energy content. Energy content also can be known as heating value (HV), calorific value (CV) or heat of combustion. The heating value is determined by getting all the products of combustion back to the previous pre-combustion temperature.

The heating value is acquired by the complete combustion of a unit quantity, usually mass (J/kg), of fuel in a combustion chamber under carefully defined conditions. There are many methods for approximating the HV of biofuel, such as proximate analysis, ultimate analysis or chemical analysis (Parikh et al., 2005; Parikh J, 2005; Sheng & Azevedo, 2005). Some studies have been done in estimating the heating value of vegetable oils by using physical properties such as density and viscosity (A. Demirbas, 2000).

2.3 Diesel engine combustion analysis

2.3.1 Diesel Engine Operation

Diesel engine or also known as compression-ignition (CI) engine is an internal combustion engine (ICE) that initiate ignition to burn the fuel that injected into the combustion chamber by using the heat of compression. The operation is very different with spark-ignition (SI) engine as SI engine needs spark plug to ignite the fuel in the combustion chamber. Because of the different operation of these two engines, both of them could not use the same type of fuel, where SI engine uses gasoline as fuel, while CI engine uses diesel, biodiesel or vegetable oils as fuel.

Engine diesel and its thermodynamic cycle was found and developed by Rudolph Diesel (1858-1913) by referring to Carnot cyclic process, a principle that describes the ideal process of fuel combustion in engine, which discovered by Sadi Carnot (1796-1832). The diesel engine has the highest thermal efficiency due to its very high compression ratio. This cycle can operate with higher compression ration than the Dual cycle and Otto cycle due to the only element that being compressed is air and there is no possibility for auto-ignition of fuel to occur. If all three cycles run at the same compression ratio, Diesel engine will give the lowest thermal efficiency. However, due to practicability, the Diesel engine cannot use the same compression ratio as Otto and Dual engine, therefore if optimal compression ratios are

used for every cycle, the Diesel engine will give the highest thermal efficiency among all three types of engines.

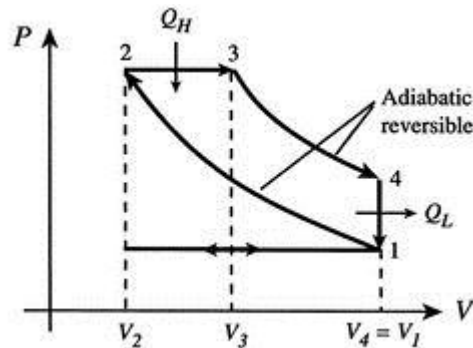


Figure 2.4: Diesel engine cycle

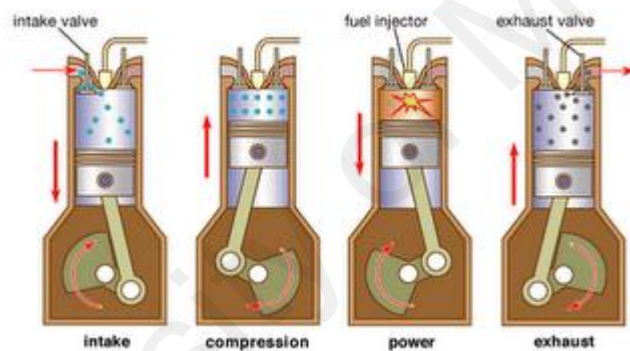


Figure 2.5: Four stroke diesel engine mode operation

Figure 2.4 and Figure 2.5 shows the diesel engine cycle used in compression-ignition engine operation. The four stroke diesel engine mode operation is separated into four modes, which are:

1. From 4-1 (Intake stroke): The piston moving down, pure air sucked into the combustion chamber.
2. From 1-2 (Compression stroke): The piston moving up and compresses the air sucked before.
3. From 2-3 (Power stroke): A moment before the piston reaches the top dead center, fuel is injected and ignited due to very high pressure forcing the piston moves down.

4. From 3-4 (Exhaust stroke): The piston travels back to top and pushing out the exhaust gas produced from the combustion out of the exhaust valve.

The main difference between combustion in compression-ignition engine (CI) and spark-ignition engine (SI) is the method of the combustion. The difference caused by the way of the ignition method is, the flame in the SI engines travels at a certain direction as it is initiated by the spark, whereas there is no flame propagation with a direction in CI engines.

The air-fuel mixture inside the combustion chamber in CI engines is non-homogeneous as air is only substance being compressed until at the end of the compression stroke. Injection of the fuel only occurs near the cylinder reaches top dead center (TDC). After the injection, the fuel must undergo the following stages to have a proper combustion;

1. Atomization: The fuel break into very small droplets. The smaller the size of the droplets injected the quicker and more efficient will the atomization process be.
2. Vaporization: The droplets of fuel evaporated in the combustion chamber due to the hot air and high pressure created by the high compression.
3. Mixing: The fuel vapour mix with the high compressed air to form a combustible air-fuel mixture. The mixing occurs because of high fuel injection velocity added to swirl and turbulence in the cylinder.
4. Self-ignition: The air-fuel mixtures usually start to ignite after the injection. Reaction caused by high temperature leads to final constant combustion process.
5. Combustion: Combustion finally takes place after the self-ignition. At the time of combustion, around 70-95% of the fuel in the combustion chamber is already in vapour state.

2.3.2 Heat release rate

Heat release rate (HRR) is a mechanism that stimulates the characteristics of fire. As HRR increases, the rate of pressure and temperature increase and resulting fire development

accelerated. Consequently, the increase in HRR will reduce concentration of oxygen and also increase the production of gases and particulate matters as the products of incomplete combustion. The analysis of heat release rate is very significant as it can help to identify some remarkable features that can help to describe the combustion mechanism of the fuels in the engine.

The heat release rate of biodiesel usually lower compared to phase of premixed combustion. The heat release rate may cause different in-cylinder gas temperature and pressure. Typically, the lower the heat release rate, the lower the peak temperature and peak pressure will be. This occurrence will affect the production of exhaust gases especially oxides of nitrogen as it is greatly depends on peak temperature. It has major influence on power output, exhaust emissions and fuel consumption (Banapurmath et al., 2008). Ordinarily, the lower blends have higher peak heat release rate due to lower viscosity and better formation of spray compared to higher blends.

2.4 Performance analysis

Engine performance parameters such as torque and brake power can be measured by using dynamometer. Torque can be used to indicate the capability of engines to do work. It is defined as force acting at a moment distance with units of Nm. Power indicates the rate at which work is done. Engine torque and power are likely lower when using biodiesel due to the biodiesel's lower energy per unit volume compared to diesel. Engine torque and power are likely lower when using biodiesel due to the biodiesel's lower energy per unit volume compared to diesel.

2.5 Diesel engine emission

For several decades, air pollution was observed to be ominously instigated by road transport, and it is worrying especially for the places which is dense with population such as megacities with risk to effect on human health (Liaquat et al., 2010). There are several aspects that may

affect the emissions of combustion such as engine load, engine speed, and inlet, outlet and in-cylinder temperature. Properties such as viscosity, heating value and ignition timing also do affect the level of emissions. The most substantial gas emissions released from the combustion of biodiesel are carbon dioxide, carbon monoxide, unburned hydrocarbon, particulate matter and oxides of nitrogen.

2.5.1 Carbon dioxide (CO₂)

Carbon dioxide (CO₂) is one of most common gases emitted during combustion or burning. The complete combustion in engine or also known as stoichiometric combustion only produces CO₂ and H₂O. This means in any combustion must at least contain CO₂ in the exhaust gas. CO₂ is much preferable than CO, even catalytic oxidation of CO and HC has been introduced to change CO to CO₂. The CO₂ emitted from the biodiesel burning can be used by plants in photosynthesis system. This will make the reduction of CO₂ expected to be very significant (Jayed et al., 2009).

2.5.2 Carbon monoxide (CO)

Carbon monoxide (CO) is a toxic gas formed from incomplete combustion. As diesel run with a stoichiometric or lean mixture, the production of CO is normally insignificant. As if the diesel runs in rich mixture, there will be improper burning of hydrocarbon which will lead to excessive amount of carbon and will produce CO instead of CO₂. An oxidation catalyst in exhaust tailpipe can help to decrease CO and unburned emission, but as a drawback, it will increase the amount of CO₂. Carbon monoxide can cause health issue such as damages sensitivity and thinking, slackens reflexes, and causes tiredness. It has a synergistic action with other pollutants to encourage illness with respiratory complications (Faiz et al., 1990).

2.5.3 Unburned Hydrocarbons (HC)

Unburned hydrocarbon (HC) is formed due to rich mixture which makes the fuel not combusted completely in the combustion chamber. The unburned hydrocarbon is released together with other exhaust gas during the exhaust stroke. HC is an organic compound in the gaseous state, while the hydrocarbon which is still in solid state after the combustion is called as particulate matter. In diesel engine, HC emissions usually caused by fuel and air mixing such as over-mixed, over lean regions formed before ignition and under-mixed, fuel injected at low velocity near the end of combustion. Low molecular weight compounds of HC may cause eye irritation, coughing and sneezing, exhaustion and signs similar to drunkenness while heavy molecular weight compounds of HC may contribute to lung disease (Faiz, et al., 1990).

3.0 Methodology

3.1 Introduction

This research utilized qualitative methods including properties measurement, combustion analysis, engine testing, and emission measurement. The review of literature studies showed that there are relationships between properties, combustion characteristics, engine performance and exhaust gas emission. This study will identify the characteristics of diesel and Jatropha biodiesel blends, and correlates them later in results and discussion section.

3.2 Biodiesel properties measurement

For this study, the properties of Jatropha biodiesel blends with conventional diesel were compared. A homogenizer (Figure 3.1) used to mix the Jatropha biodiesel and conventional diesel so that they are not going to be separated due to different density.



Figure 3.1: Homogenizer

There are two types of blends used in this study, which are JB10 and JB20. The volume fraction of those two types of blends can be comprehended from Table 3.1.

Table 3.1: Volume fraction

Blends	Volume fraction (%)	
	Jatropha	Diesel
JB0	0	100
JB10	10	90
JB20	20	80

3.2.1 Density and Viscosity

The density measurement of Jatropha Biodiesel in this study was uses ASTM D7042 test method. This test method proposes a procedure for the coexisting measurement of dynamic viscosity, and the density of liquid petroleum. The kinematic viscosity obtained by dividing the dynamic viscosity and the density at the same test temperature.

The measurements were done at temperature 40°C by using Anton Paar (SVM 3000) as shown in Figure 3.2. The measurements were automatically run for three times for each sample and averaged by the viscometer.



Figure 3.2: Anton Paar (SVM 3000) Viscometer

The proposed idea by Alptekin and Canakci (2008) was used to correlate the density with the percentage of the Jatropha biodiesel in the blends in this study. The measured data were correlated as a function of volume fraction by using empirical polynomial equation

which is obtained from regression. By using the equation, the density of the Jatropha biodiesel blends for the rest of the volume fraction can be estimated by only using density measurements at 0%, 10%, 15%, 20% and 100% Jatropha biodiesel blends. Second degree polynomial equation was used as it gives better correlation result compared to the linear equation. The general form of the equation as a function of biodiesel fraction is given by

$$\rho = Ax^2 + Bx + C \quad (3.1)$$

Where ρ is density (kg/m³), A, B, and C are coefficients and x is the biodiesel fraction.

There were two types of viscosity measured, which are dynamic and kinematic. Same as density, both of the viscosities also measured using ASTM D7042 test method. Anton Paar (SVM 3000) viscometer as shown in Figure 3.2 was used to measure the viscosities. The viscometer automatically calculates the kinematic viscosity and delivers measurement results which are equivalent to ISO 3104. The viscometer is capable to measures dynamic and kinematic viscosity from less than 1 mm²/s up to 20000 mm²/s in a temperature range from 56°C to 105°C in one measuring cell.

The measured data were correlated as a function of volume fraction by using empirical second degree polynomial equation which is obtained from regression. By using the equation, the dynamic and kinematic viscosity of the Jatropha biodiesel blends for the all volume fraction can be estimated by only using viscosity at 0%, 10%, 15%, 20% and 100% Jatropha biodiesel blends. Second degree polynomial equation was used as it gives better correlation result compared to linear equation. The general form of the equation as a function of biodiesel fraction for kinematic and dynamic viscosity is given by equation 3.2 and 3.3.

$$\eta = Ax^2 + Bx + C \quad (3.2)$$

Where η is the kinematic viscosity (mm²/s), A, B, and C are coefficients and x is the biodiesel fraction.

$$\mu = Ax^2 + Bx + C \quad (3.3)$$

Where μ is the dynamic viscosity (mPa-s), A, B, and C are coefficients and x is the biodiesel fraction.

3.2.2 Flash point measurement

The Pensky-Martens Closed Cup Tester (Figure 3.3) was used to measure the flash point of all three samples, diesel, JB10 and JB20. The flash point tester is operated according to the standard test method ASTM D93 (ASTM, 2000). Due to the measurement is prone to error such as parallax and random errors which might cause by environmental factors like temperature and air circulation.

Fuel was filled into the cup, closed and heated continuously. To measure the flash point of the fuel, a hot rod is used to ignite the vapour of the fuel. The cup was open for every increment of 2-3°C of fuel temperature, and the hot rod directed to the mouth of the cup. Once an ignition noticed, that instant temperature of the fuel is its flash point. The measurement was taken three times and averaged to get the best approximation value.

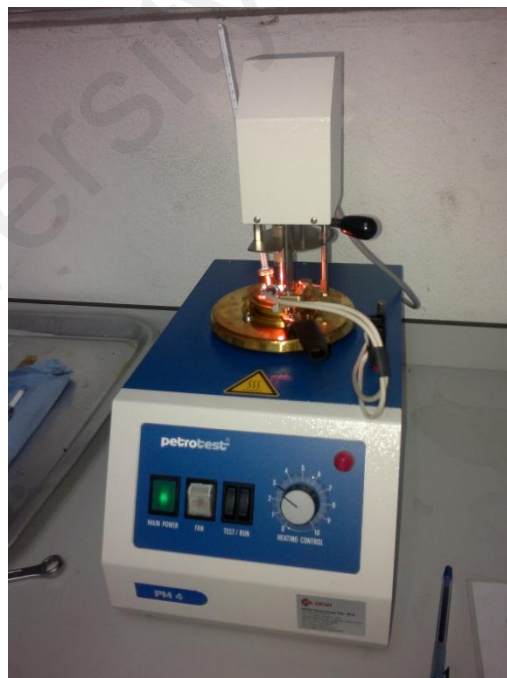


Figure 3.3: Pensky-Martens Closed Cup Tester

If the pressure of the place where the test taken is not equal to 101.3 kPa (760mm Hg), the flash point needs to be corrected as below equations:

$$\text{Corrected flash point} = C + 0.25 (101.3 - K) \quad (3.4)$$

$$\text{Corrected flash point} = F + 0.06 (760 - P) \quad (3.5)$$

$$\text{Corrected flash point} = C + 0.033 (760 - P) \quad (3.6)$$

Where C is the observed flash point in °C, F is the observed flash point in °F, P is the ambient barometric pressure in mm Hg, and K is the ambient barometric pressure in kPa.

3.2.3 Heating value measurement

In determining the heating value of the diesel and biodiesel, IKA C 2000 calorimeter (Figure 3.4) was used. IKA C 2000 calorimeter system used to determine the gross calorific value of solid and liquid materials in accordance to DIN 51900, BS 1016 T5, ISP 1928, ASTM 5468 and ASTM 4809.



Figure 3.4: IKA C 2000 Calorimeter

The combustion calorimeter measures heat that rises from burning. The sample is weighed into a digestion vessel and filled with oxygen (with pressure of oxygen operating pressure is 30 bars). The burning process is started by means of an ignition spark. The experiment ends when the sample is fully burned. By measuring the temperature increase, the

heating value of the sample can be calculated. Some technical data of the calorimeter can be found from Table 3.2.

3.3 Heat release rate and combustion analysis

Combustion analysis has been done by calculating the heat release rate based on the heating value of each blends. The first important data needed to calculate heat release rate is in-cylinder pressure versus crank angle in the operating engine.

To simulate the heat release from the three fuels combustion, engine specifications as shown in Table 3.2 was used. The in-cylinder pressure can be assessed by using the ideal gas law.

$$PV = mRT \quad (3.7)$$

Where P indicates the in-cylinder pressure, V is volume, m is the mass of gas in the cylinder, R is gas constant for air with value of 0.287 kJ/kg K and T is the temperature (Pulkrabek, 2004). When analysing what occurs within engines during the operating cycle and exhaust flow, this paper uses the air property values in Table 3.3.

Table 3.2: Engine specification for combustion analysis

Engine Specification	Value	Unit
Compression ratio	16	
Displacement Volume, Vd	1000	cc
Bore, B	10	cm
Rod Length, l	16.8	cm
Starting compression pressure	100	Bar
Starting compression temperature	333	Kelvin
Starting combustion (crank angle)	-20	Degree
Stop combustion (crank angle)	45	Degree
Duration of combustion	65	Degree
Air/Fuel Ratio	18	
Speed, N	2000	rpm

Table 3.3: Air property values (Pulkrabek, 2004)

Constant	Value
cp	1.108 kJ/kg.K
cv	0.821 kJ/kg.K
k	cp/cv=1.35
R	cp-cv=0.287 kJ/kg.K

The instantaneous volume can be found by using numerical methods.

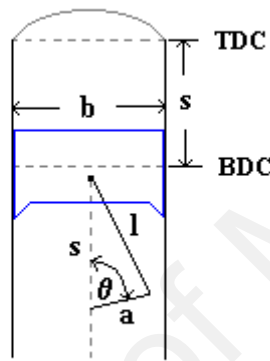


Figure 3.5: Piston cylinder diagram

Top dead center (TDC) is where the location of the crank when the crank angle is at 0° , while the bottom dead center (BDC) is where the volume of the combustion chamber at maximum when the crank angle is at 180° . The displacement volume of the chamber can be represented by equation 3.8.

$$V_d = \frac{\pi}{4} b^2 s \quad (3.8)$$

Where V_d is displacement volume, b is bore and s is stroke. Meanwhile the volume of cylinder at any crank angle can be calculated using equation 3.9 below.

$$V = V_c + \left(\frac{\pi b^2}{4} \right) (l + a - s) \quad (3.9)$$

Where V_c is clearance volume, l is the connecting rod length, a is the crank offset and s is the piston position as shown in Figure 3.5. The piston position, s and piston speed can be determined by using equation 3.10 and 3.11 in a function of crank angle, Θ .

$$s = a\cos(\theta) + (l^2 - a^2\sin^2(\theta))^{1/2} \quad (3.10)$$

$$\frac{ds}{d\theta} = -a\sin(\theta) - \frac{l^2\sin\theta\cos\theta}{\sqrt{l^2 - a^2\sin^2\theta}} \quad (3.11)$$

By substituting equation 3.10 into equation 3.9, the instantaneous volume of cylinder at any crank can be defined in term of crank angle as shown in equation 3.12.

$$V(\theta) = V_c + \frac{\pi b^2}{4} \left(l + a(1 - \cos(\theta)) - (l^2 - a^2\sin^2(\theta))^{1/2} \right) \quad (3.12)$$

Adiabatic in cylinder pressure and temperature can be calculated by using equation 3.13 and 3.14 respectively.

$$P(\theta_i) = P(\theta_{i-1}) \left(\frac{V(\theta_{i-1})}{V(\theta_i)} \right)^k \quad (3.13)$$

$$T(\theta_i) = \frac{P(\theta_i)V(\theta_i)T(\theta_{i-1})}{P(\theta_{i-1})V(\theta_{i-1})} \quad (3.14)$$

Where θ_{i-1} indicates the instantaneous condition at the previous 1° crank angle from the current θ_i . The initial cylinder temperature and pressure were assumed to be 333K and 100bar respectively.

The mass burn fraction of fuel for every crank angle during combustion period can be estimated by using Wiebe function as expressed in equation 3.15 below (Yeliana et al., 2011).

$$x_b(\theta) = 1 - \exp \left[-n \left(\frac{\theta - \theta_o}{\Delta\theta} \right)^{m+1} \right] \quad (3.15)$$

Where n is the efficiency parameter, m is the shape factor, x_b is the mass burn fraction, θ is the instantaneous crank angle, θ_o is the crank angle where the combustion started and $\Delta\theta$ is the combustion duration. Although the θ_o indicates the start of combustion, due to the delay in flame development, the spark timing does not necessarily match with the first location of

combustion detected. Thus, θ_o can be accepted as the spark discharge. The n and m are constants that used to define the burn rate characteristic, where usually equal to 5 and 2 respectively. Nevertheless, these constants may result in a large estimation error especially in combustion knock applications. The start of combustion, combustion duration and end of combustion are assumed to be same for all three blends as shown in Table 3.2.

The in cylinder temperature during the combustion period can be estimated by using equation 3.16 (Pulkrabek, 2004).

$$m_f Q_{HV} = x_b(\theta_i) m_m c_v (T(\theta_i) - T(\theta_{i-1})) \quad (3.16)$$

Where m_f is the mass of fuel in the combustion chamber, Q_{HV} is the energy content or heating value of the fuel, m_m is the total mass of fuel and air in the combustion chamber, c_v is the specific heat content of air at constant volume. The variable x_b took into account that not all of the fuel was burned and meaning that only a portion of the chemical energy was converted into thermal energy. The mass of the fuel was calculated using the air-fuel ratio and remarking that the sum of mass of air, m_a , and mass of fuel, m_f , equal to the total mass, m_m . The heating value, Q_{HV} depends on the type of fuel.

Heat release rate study is commonly practiced on compression ignition engines. It analyses the amount of heat need to be added to the combustion chamber in order to produce the pressure deviations. By using the first law of thermodynamics, net heat release rate can be expressed as in equation 3.17 (Abd Alla, 2002).

$$\frac{dQ_{net}}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \quad (3.17)$$

Where γ is the ratio of specific heats (c_p/c_v), Q_{net} is the net heat release rate, P is the instantaneous in-cylinder pressure and V is the instantaneous in-cylinder volume.

3.4 Performance measurement

The performance test were conducted on a four-stroke, single cylinder DI diesel engine Yanmar TF 120 (Figure 3.7) which was coupled to an eddy current motoring dynamometer as shown in Figure 3.6. No modifications were done on the engine while running Jatropha biodiesel blends. Figure 3.8 shows the schematic diagram of the experimental data acquisition layout used by the Yanmar TF 120 and dynamometer. Dyno-Max 2000 is used as data acquisition and analysis software to control the engine running conditions and analyse the performance parameters. The performance parameters include torque and brake mean effective pressure. The engine speed and load can be controlled either manually using the controller box or automatically using the Dyno-Max 2000. All the data received by controller box from the dynamometer can be transferred to Dyno-Max 2000 for better analysis.



Figure 3.6: Engine test bed setup



Figure 3.7: TF120M Diesel Engine

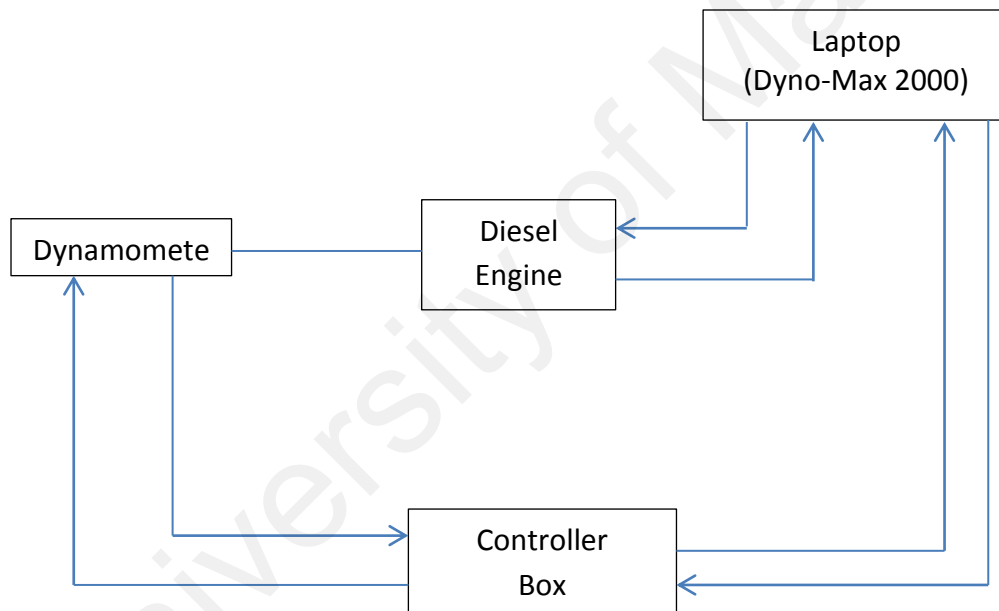


Figure 3.8: Data Acquisition Layout

The engine specifications are given in Table 3.4. This setup allows the study of engine performance for torque and brake power. The performance test has been done at constant full throttle with variable engine speeds from 1200rpm to 2500rpm.

Table 3.4: Engine specification for performance analysis

Engine type	Four-stroke DI diesel engine
Number of cylinders	One
Cylinder bore x stroke (mm)	92 x 96
Displacement (L)	0.638
Compression ratio	17.7
Maximum engine speed (RPM)	2400
Maximum power (kW)	7.7
Cooling system	Radiator cooling
Power take – off position	Flywheel side

3.5 Emission performance

Bosch petrol and diesel emission analyser Figure 3.9 was used to measure the emission of the Jatropa biodiesel blends. Emission gases measured are carbon monoxide (CO), carbon dioxide (CO₂), and unburned hydrocarbon (HC).



Figure 3.9: Bosch petrol and diesel emission analyser

The same engine used for performance analysis was used for emissions characteristics study. The engine was started using diesel fuel and warmed up before the emission readings are taken using the analyser's probe. The emissions were measured using the exhaust gas

analyser after the engine reached its steadied working condition. The engine exhaust emissions were investigated at full throttle with range of speed from 1200rpm to 2500rpm.

4.0 Result and Discussion

4.1 Introduction

This chapter will elaborate on the findings that have been obtained from the implementation of methodology of this project. This chapter will perform the correlation between results to determine the cause and effect of each result.

4.2 Effect of biodiesel fraction on properties of fuels

This subtopic will discuss about the effect of biodiesel fraction on fuel blends properties such as density, viscosity, heating value, and flash point.

4.2.1 Effect of biodiesel fraction on density

The density of Jatropha Biodiesel was measured in 40°C temperature with various blends. Table 4.1 shows the density of Jatropha Biodiesel from 0% to 100% volume fraction.

Table 4.1: Density of Biodiesel Blend at 40°C

Biodiesel	JB0%	JB10%	JB15%	JB20%	JB100%
Density (g/cm ³)	0.8272	0.8306	0.8322	0.8338	0.8625

The density is range is not large as the minimum value is 0.8272g/cm³ for conventional diesel and the maximum value is 0.8625g/cm³ for the 100% biodiesel fraction. As can be seen from Table 4.1 and Figure 4.1, the density is increasing as the biodiesel volume fraction increases.

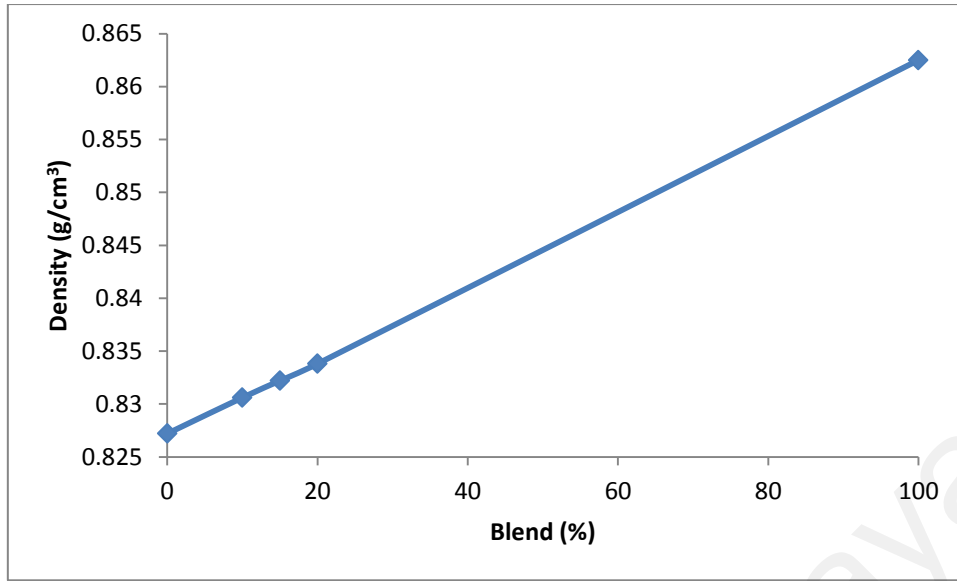


Figure 4.1: Density of biodiesel blends at 40°C

Figure 4.1 represents the measured density and calculated polynomial trend lines for regression analysis. The density of biodiesel increases with respect to the amount of biodiesel in the mixture. A second degree polynomial correlation was developed by the measured density at several points. The correlation is described by the equation 4.1.

$$\rho = (3 \times 10^{-7})x^2 + 0.0003x + 0.8272 \quad (4.1)$$

Where ρ is the density of each blend with x amount of volume fraction at temperature 40°C. The regression correlation and the experimental value have R^2 of 1.

4.2.2 Effect of biodiesel fraction on viscosity

The effect of the biodiesel volume fraction on the kinematic viscosity of biodiesel was probed and equivalent numerical analysis were built. Table 4.2 shows the kinematic viscosity with respect to the Jatropha biodiesel blends. It can be seen from the table that the viscosity of Jatropha biodiesel drops relatively with the fraction of diesel added into it. The viscosity of Jatropha biodiesel decreased significantly was observed for JB20 blends and JB10 with a reduction of 20.53% and 22.24% were respectively. And the blends also have higher kinematic viscosity compared to conventional diesel (JB0), 4.29% (JB20) and 2.05% (JB10).

Table 4.2: Kinematic viscosity of biodiesel blends (mm²/s) at 40°C

Biodiesel	JB0%	JB10%	JB15%	JB20%	JB100%
Kinematic viscosity (mm ² /s)	3.3873	3.4569	3.4978	3.5329	4.4455

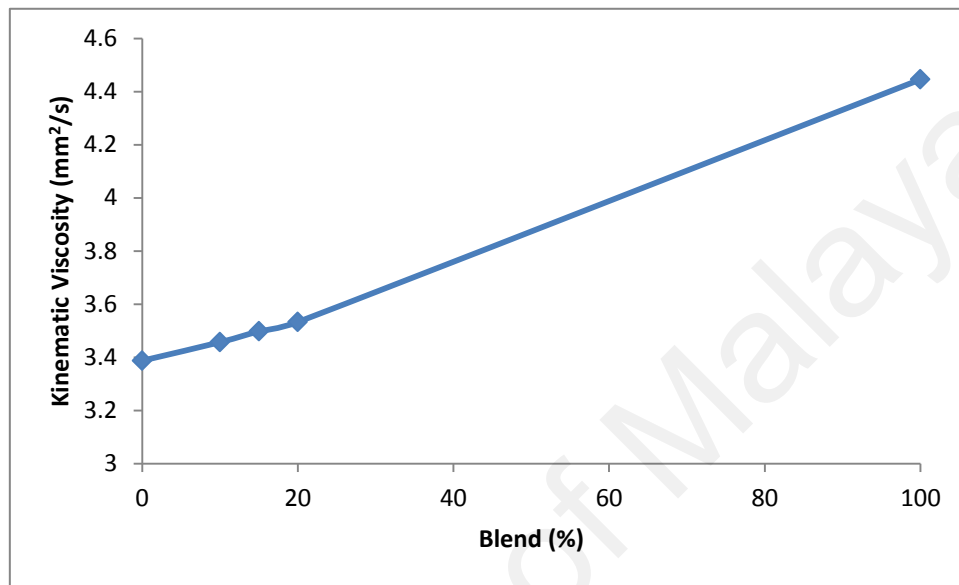


Figure 4.2: Kinematic viscosity (mm²/s) of biodiesel blends at 40°C

Figure 4.2 shows the trend of the kinematic viscosity versus the volume fraction of biodiesel blends. The trend indicates that the kinematic viscosity is directly proportional to the volume fraction, as the volume fraction increase, the kinematic viscosity will also increase.

The correlation equation can be obtained in second degree polynomial equation as described in Equation 4.2, with R^2 is equal to 1.

$$\eta = (4 \times 10^{-5})x^2 + 0.0066x + 3.3873 \quad (4.2)$$

Where η is the kinematic viscosity of each blend with x amount of volume fraction at temperature 40°C.

4.2.3 Effect of biodiesel fraction on heating value

The effect of the biodiesel volume fraction on the heating value of biodiesel was probed and equivalent numerical analysis were built. Table 4.3 shows the heating value with respect to the Jatropha Biodiesel blends. As we already distinguish, the conventional diesel has the highest heating value compared to the pure biodiesel and the blends with value of 45547.5 J/kg, while the pure biodiesel has the lowest value 39965 J/kg (less 12.26% from the conventional diesel) which may cause power loss. The lower heating value of biodiesel requires larger amount of fuel to be injected into the combustion chamber to produce the same power as conventional diesel (Canakci, 2007).

As we learnt that although biodiesel has very high density compared to conventional diesel, it still has lower heating value compared to conventional diesel, thus by blending both type of fuels may increase the heating value of biodiesel as shown in Table 4.3 below.

Table 4.3: Heating Value of biodiesel blends (J/kg)

Biodiesel	JB0%	JB10%	JB15%	JB20%	JB100%
Heating Value (J/kg)	45547.5	45114	44837.5	44672	39965

For JB20, the heating value is 11.778% higher than pure biodiesel (44672 J/kg), JB15 diesel has 12.19% higher and JB10 has 12.88% higher. This means to produce the same amount of energy as diesel, JB10 needs 1.0096% more than diesel in term of mass basis. While JB15 needs 1.0159%, JB20 needs 1.02% and JB100 needs 1.14% more mass than diesel to get comparable energy with diesel during combustion.

Figure 4.3 shows the trend of the heating value with respect to the volume fraction of biodiesel blends. The trend indicates that the heating value is inversely proportional to the volume fraction, as the volume fraction increase, the heating value will decrease.

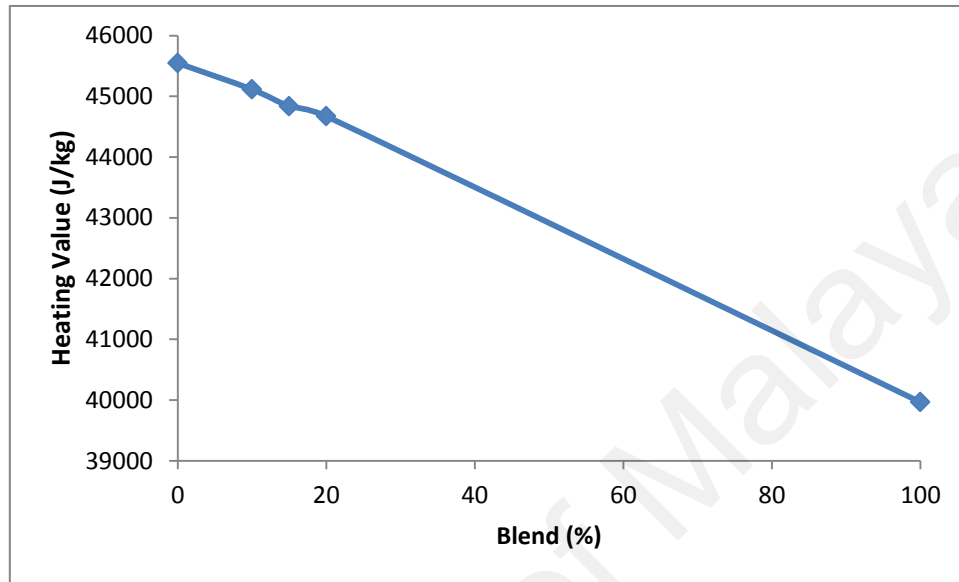


Figure 4.3: Heating Value of biodiesel blends (J/kg)

The correlation equation can be obtained in second degree polynomial equation as described in Equation 4.2, with R^2 is equal to 0.9999.

HV =

$$-0.1355x^2 - 42.218x + 45547.5 \quad (4.2)$$

Whereas **HV** is the heating value of each blend with x amount of volume.

4.2.4 Effect of biodiesel fraction on flash point

Table 4.4 shows the reading obtained from the thermometer at the Pensky-Martens Closed Cup Tester. The atmospheric pressure in the lab in University of Malaya is assumed to be 101.3 kPa (760mm Hg), therefore no correction need to be done on the readings. The readings are inconsistent due to several factors, thus average reading need to be calculated from at least three readings to get the best approximation for the flash point.

Table 4.4: Flash Point of biodiesel blends (°C)

Biodiesel	Flash point (°C)			
	1st Reading	2nd Reading	3rd Reading	Average
JB0% (Diesel)	70	69	71	70
JB10%	75	72	74	74
JB20%	85	90	88	88

Diesel fuel usually has flash points in between 60°C and 100°C as it is designed for high compression engine. As can be seen from Table 4.5, the flash points increase as the amount of Jatropha biodiesel increasing in the mixture. Diesel flash point is the lowest among the three samples tested with average measurement, 70°C. For JB10, the average flash point measured increased 5.7% to 74°C, while JB20 blend gives average flash point 88°C, which means 20% increment. This means that the Jatropha biodiesel is safer to be handled as its vapour is harder to be combusted by spark as its needs higher temperature compared to diesel.

4.3 Effect of biodiesel fraction on combustion characteristics

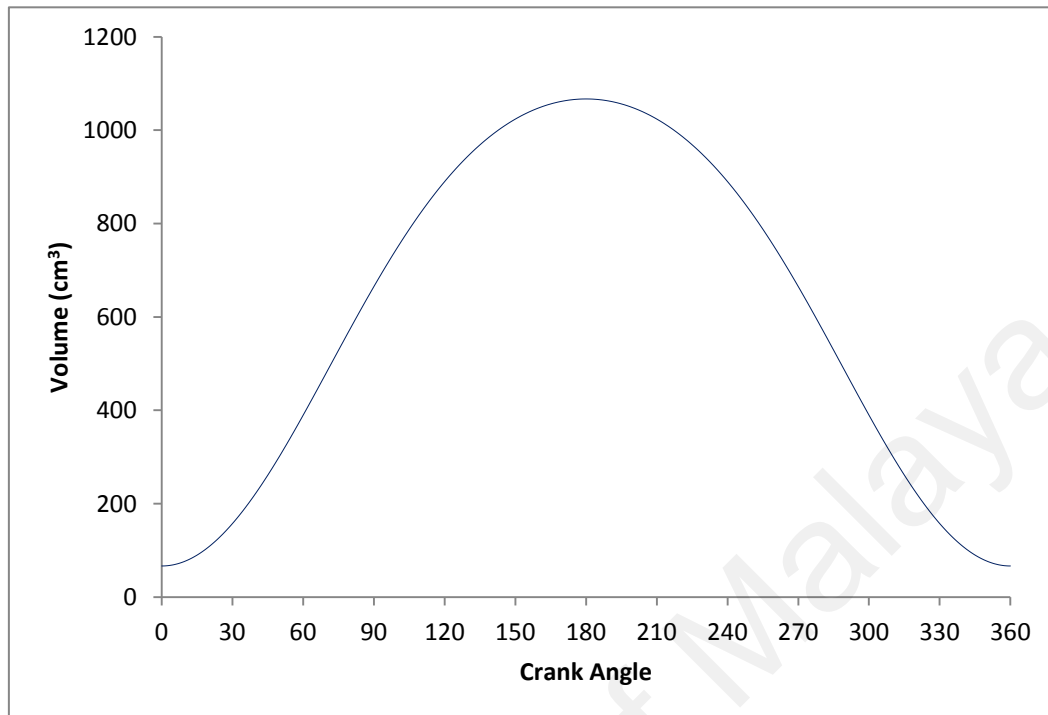


Figure 4.4: Volume vs Crank Angle

Figure 4.4 indicates the volume versus crank angle characteristic for the chosen engine specification. This information is very important to calculate the theoretical in-cylinder pressure and temperature for every crank angle. As shown in the figure, the maximum volume will be at crank angle 180° CA or at bottom dead center (BDC), which is equal to the sum of displacement volume and clearance volume, 1066cc (1.066L). Meanwhile, the minimum volume will be at 0° CA and 360° CA, as the piston surface reaches top dead center (TDC), with the volume equal to only clearance volume, 66cc (0.066L).

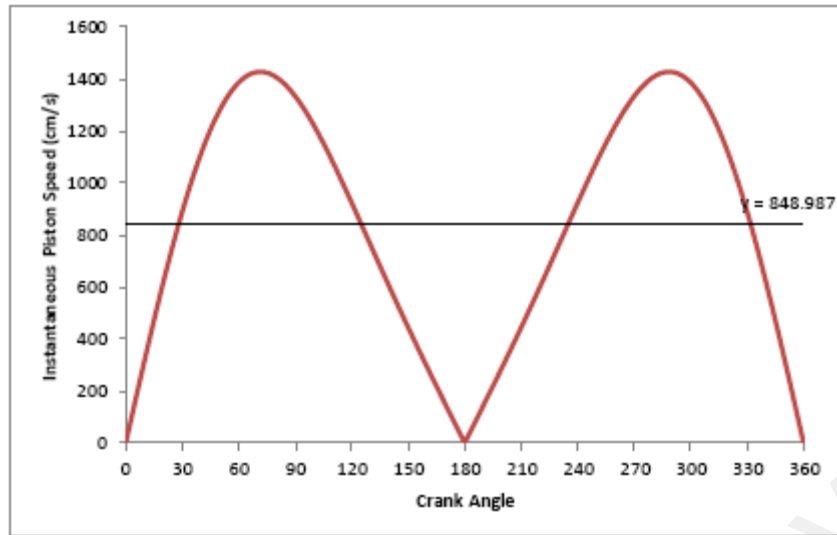


Figure 4.5: Piston Speed vs Crank Angle

Figure 4.5 shows the instantaneous piston speed at every crank angle. This characteristic is useful to calculate the heat coefficient before calculating the heat release rate. The average piston velocity is 848.987cm/s.

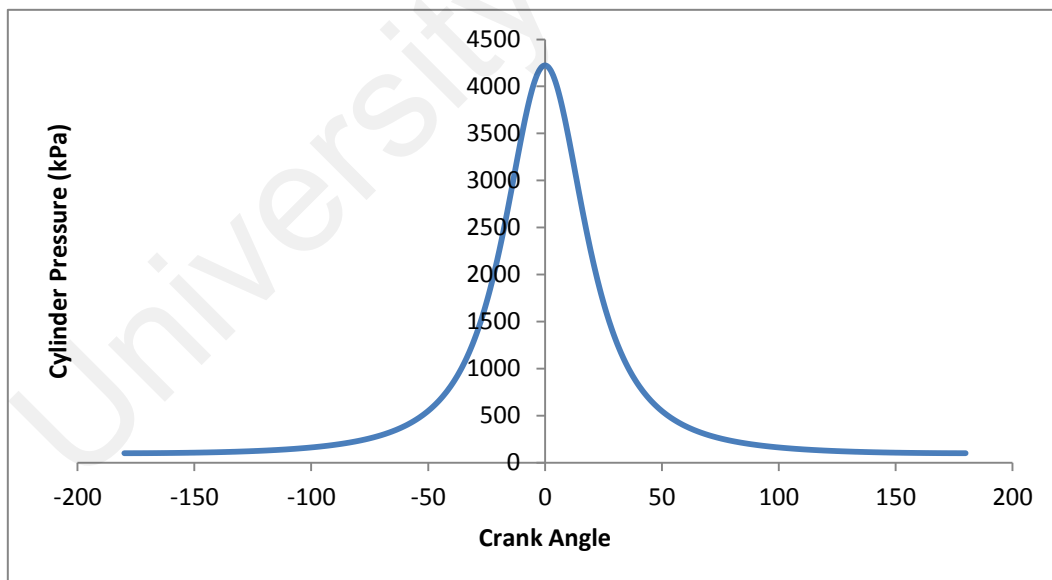


Figure 4.6: Adiabatic Cylinder Pressure vs Crank Angle

Figure 4.6 represents the instantaneous adiabatic in-cylinder pressure in the combustion chamber. The adiabatic cylinder pressure occurred without the presence of fuel.

The peak pressure for adiabatic process is 4222kPa at TDC as it is the place with smallest volume.

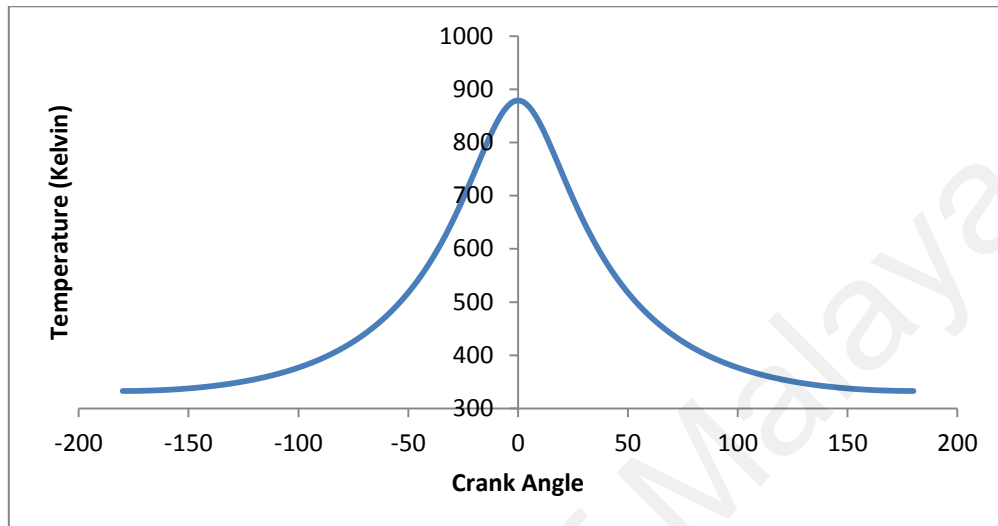


Figure 4.7: Adiabatic Cylinder Temperature

Figure 4.7 shows the instantaneous in-cylinder temperature for adiabatic process which there is no heat loss or gain in the process. Same as peak pressure, the peak temperature also seen at TDC with value of 878 Kelvin.

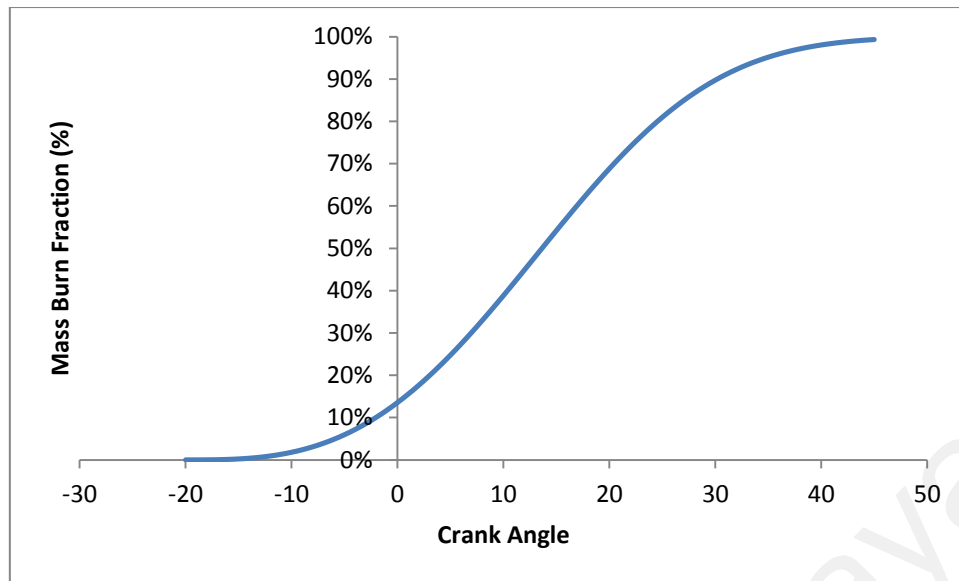


Figure 4.8: Mass Burn Fraction

Figure 4.8 expresses the mass burn fraction of fuel in the combustion chamber starting from 20° bTDC until 45° aTDC with crank angle duration of 65° . The characteristic was estimated using Wiebe function. The instantaneous mass burn fraction of fuel is needed to estimate the rate of fuel has been burned with respect to crank angle which helps to approximate the instantaneous in-cylinder pressure and temperature in the combustion chamber.

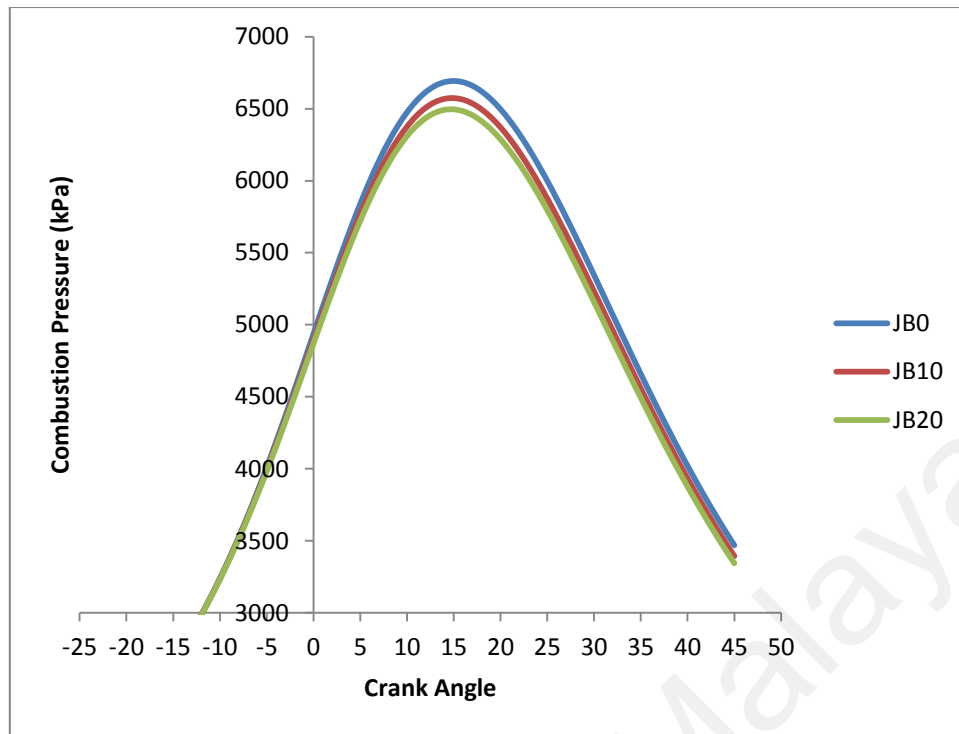


Figure 4.9: Combustion Pressure vs Crank Angle for biodiesel blends

Figure 4.9 indicates the calculated in-cylinder pressure for all three types of blends. The fuel diesel (JB0) gives the highest pressure peak with the value of 6692 kPa at 14°CA. The JB10 and JB20 blends produce peak pressure less 1.78% and 2.95% compared to JB0 respectively.

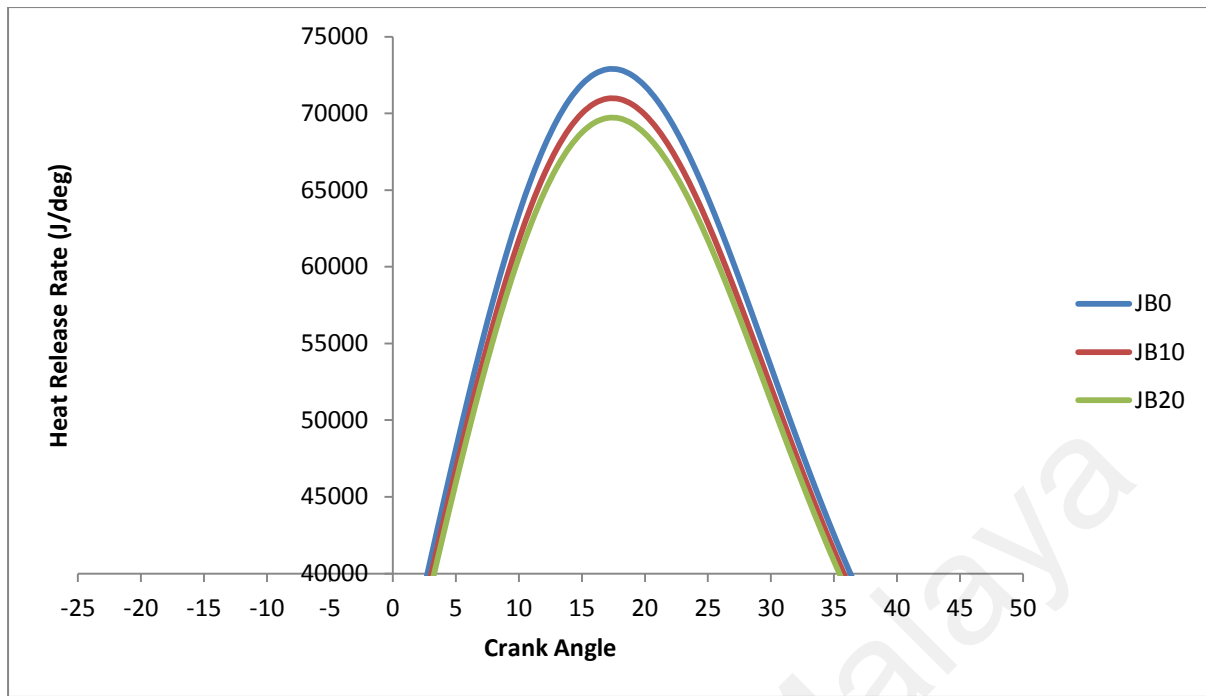


Figure 4.10: Heat Release Rate vs Crank Angle for biodiesel blends

By using the in-cylinder pressure data from Figure 4.9, rate of heat release of the combustion can be estimated and expressed into Figure 4.10. The fuel diesel gives the highest heat release rate with peak of heat release rate value of 73kJ/°CA. The JB10 and JB20 blends peak of heat release rate are lower than JB0, with the values of 71kJ/°CA and 69kJ/°CA respectively.

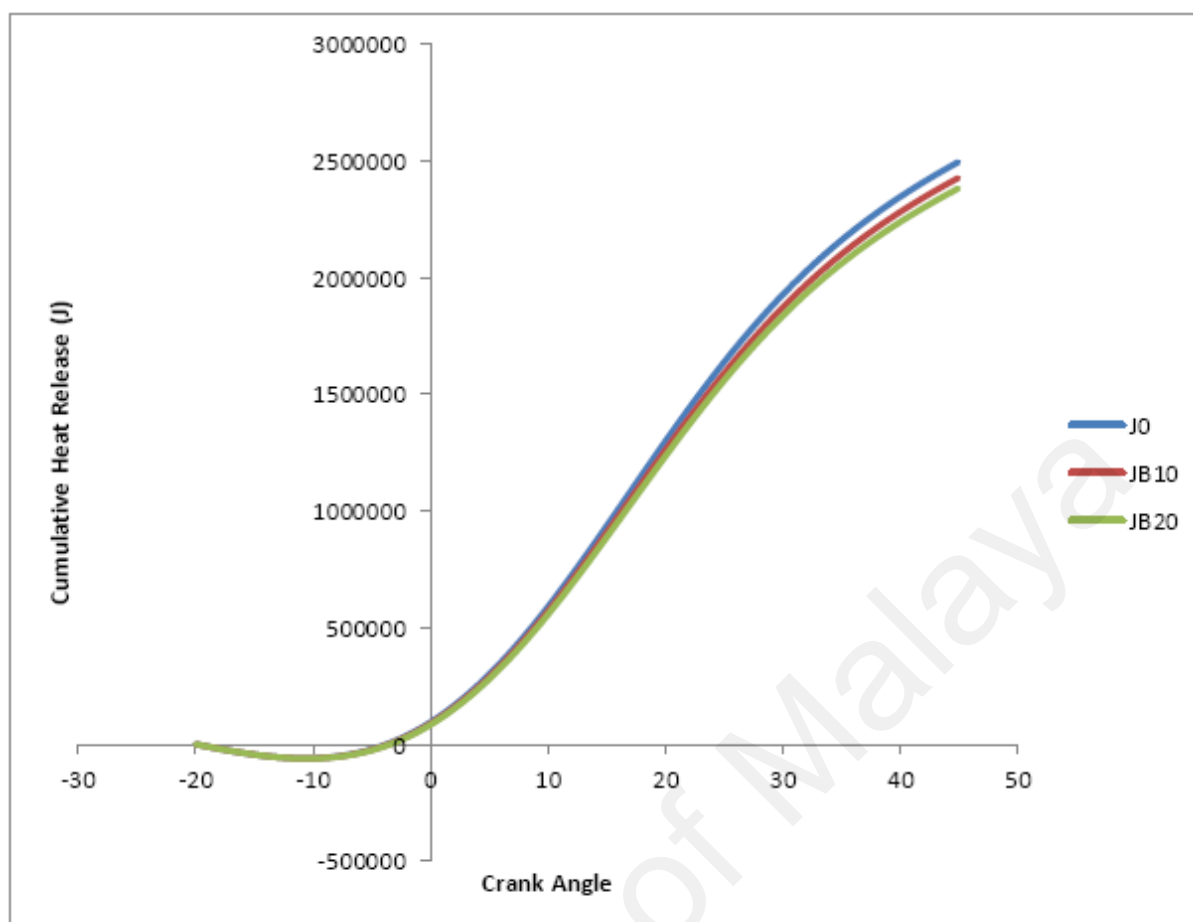


Figure 4.11: Cumulative Heat Release for biodiesel blends

Figure 4.11 shows the cumulative heat released from the fuel combustion. At the end of the combustion, the total heat released for JB0, JB10 and JB20 are 2.49MJ, 2.42MJ and 2.38MJ respectively. There is a small negative heat release just after the injection started due to the evaporation of fuel uses energy from the surrounding gas (Ashgriz, 2011).

4.4 Effect of biodiesel fraction on engine performance

Engine torque versus engine speed at constant full throttle is shown in Figure 4.12. On average, JB0 gives the highest torque following with JB10 and JB20. This shows that the higher the percentage of Jatropha biodiesel, the lower the torque will be produced.

The value of engine torque can be used to derive the value of brake power. Figure 4.13 represents the engine brake power produced in engine speed in the range of 1200 rpm

and 2500 rpm. It was found that at constant fuel injection, the brake power increases with increasing engine speed until 2300 rpm where the brake power drops due to increasing friction in piston and cylinder. The maximum brake power for JB0, JB10 and JB20 are at near 2300rpm with 4.04kW, 3.94kW and 3.89kW respectively. The reason both JB20 has the lowest brake powers while JB0 has highest brake powers is mainly because of their heating values where B0 has highest heating value whilst JB20 has lowest heating value among those three.

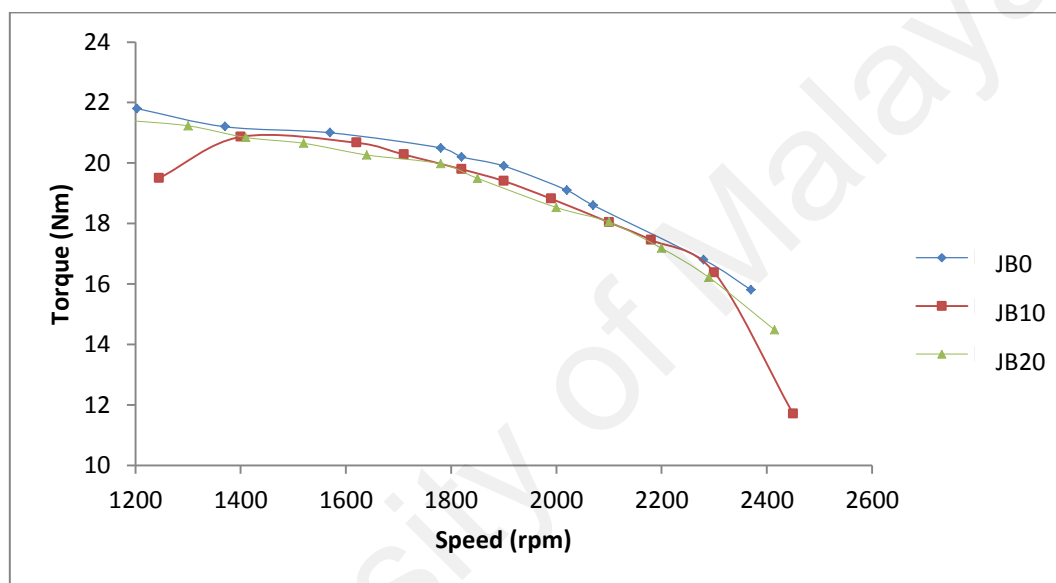


Figure 4.12: Engine torque vs engine speed for biodiesel blends

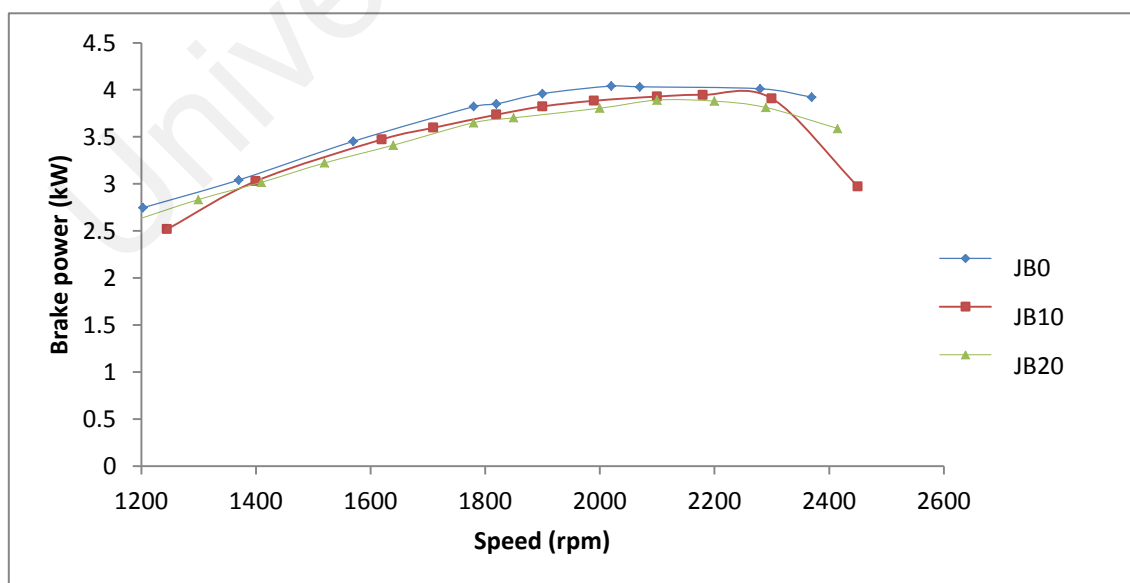


Figure 4.13: Brake power vs engine speed for biodiesel blends

4.5 Effect of biodiesel fraction on emission

The unburned hydrocarbon (HC) versus engine speed is shown in Figure 4.14. The unburned hydrocarbon is the outcome of incomplete combustion because of flame slaking in crevice regions in the cylinder wall. The unburned HC also may increase due to excessive rich fuel-air mixture which means that the content of oxygen in the mixture is insufficient to provide complete combustion for the hydrocarbon. Moreover, fuel misfire also may cause more hydrocarbons in exhaust. From the figure, it can be seen that JB20 produced lowest HC, followed by JB10 and JB0. This may be because of the oxygen content in vegetable oils and biodiesels is higher than diesel, thus make the higher blends have more thorough combustion compared to lower blends and diesel. On average, JB0, JB10 and JB20 produce 10 ppm, 8.9 ppm and 7.75 ppm respectively.

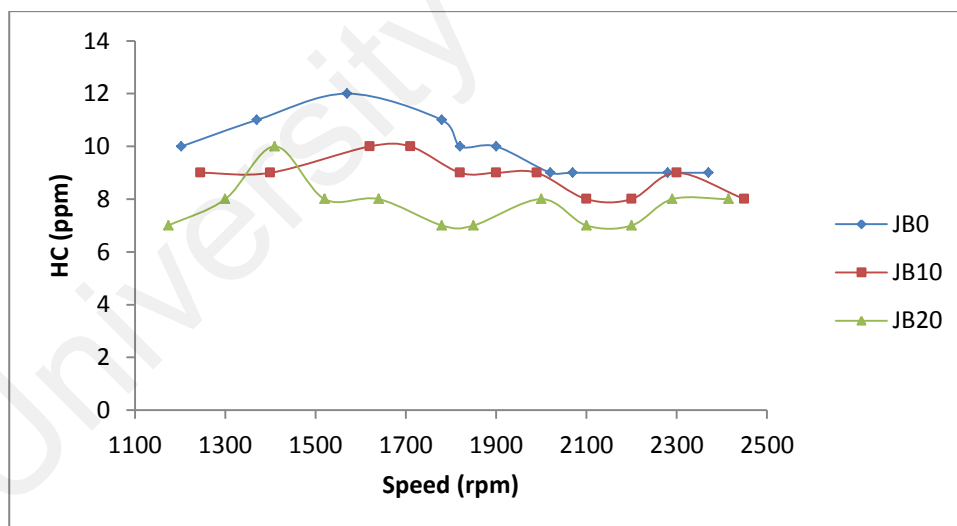


Figure 4.14: Unburned hydrocarbon (HC) emission vs engine speed for biodiesel blends

The emission of CO_2 is shown in Figure 4.16 with engine speed as dependant variable. The CO_2 exhaust emission can be used to determine the combustion performance. The higher the CO_2 content in exhaust gas can articulate that the fuel has better combustion.

The peak of CO₂ emission can be found in the range of 1500 rpm and 2000 rpm, probably means that the best combustion occurs in that range of engine speed for all three blends. On average, JB0, JB10 and JB20 produced 4.216%vol, 3.948%vol and 3.740%vol respectively.

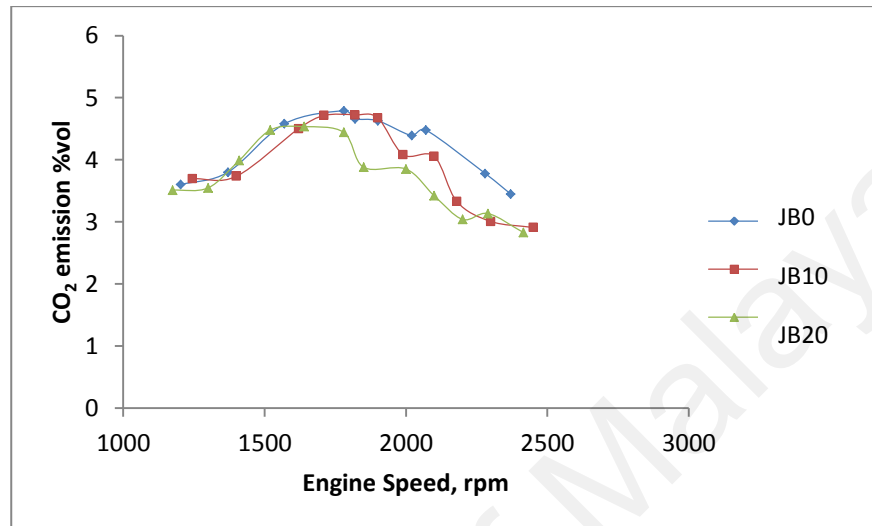


Figure 4.15: CO₂ emission vs engine speed for biodiesel blends

Figure 4.16 below shows CO emission versus engine speed. Carbon monoxide, CO is a toxic gas that produced from incomplete combustion probably due to incomplete combustion of fuel with low oxygen concentration. The production of CO in exhaust is significantly reliant on the air-fuel ratio relative to the stoichiometric proportion and fuel combustion performance in the cylinder engine (Kalam, et al., 2010). On average, JB0 recorded the highest CO emission, followed by JB10 and JB20 with measurement of 0.083%vol, 0.076%vol and 0.067%vol respectively. This is because the higher the blends, the higher the oxygen content in the fuel because the high oxygen content in vegetable oils.

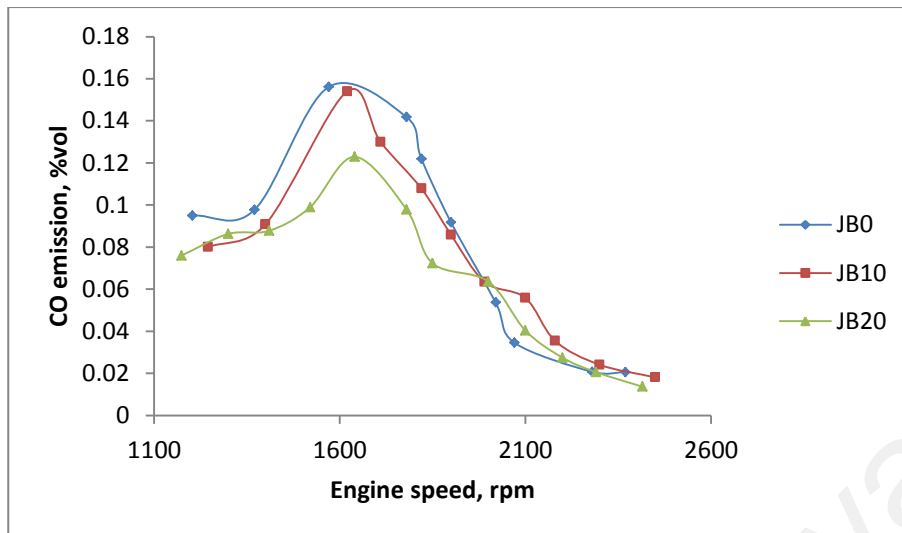


Figure 4.16 CO emission vs engine speed for biodiesel blends

The dependency of community on petroleum can be reduced with the existence of alternatives such as biodiesel and in the same time can help to overcome high price of fuel. To gain almost comparable quality of fuel, two types of blend have been tested, JB10 (10% Jatropha biodiesel + 90% diesel) and JB20 (20% Jatropha biodiesel + 80% diesel).

5.0 Conclusion

In this study, seeds of *Jatropha L. Curcas* were utilized into biodiesel and blended with conventional diesel fuel to give comparable emission, performance and combustion characteristics to diesel. It is abundant, eco-friendly and renewable to substitute or complement fossil fuels. With the resources of fossil energy keep decreasing, biodiesel has been anticipated as alternative of the non-renewable fuel.

From the study of the fuel properties for those three fuels, comparative study has been done on density, viscosity, heating value and flash point. Density, viscosity and flash point increases as the percentage of *Jatropha* biodiesel increased in the blends. The difference in density may influence engine output as the fuel injection uses volumetric parameter, therefore the mass of fuel injected will be vary for different type of blends. Biodiesel tends to have higher density because biodiesel contains oxygen, thus better combustion can be achieved with a higher fuel-air ratio. Higher viscosity may cause poor spray injection atomisation which may lead to development of gumming, formation of deposits and issue with conventional lubricating oils. Higher flash point makes the fuel much safer to be handled as the lowest temperature of its vapor to be ignited if exposed to the flame is higher. However, as the biodiesel portion in blends increased, the heating value decreased as biodiesel has lower heating value compared to diesel. Heating value is very important in affecting the performance and combustion characteristics of fuels, as higher heating value can lead to higher torque, brake power, fuel economy, peak pressure and peak temperature.

From combustion analysis, the peak pressure for JB0 is highest followed by higher blend JB10 and JB20 which are less 1.78% and 2.95% respectively compared to JB0. The main cause of this trend is the different heating value between the blends. The different in peak pressure leads to variation of heat release rate during combustion, where JB0 recorded the highest peak of heat release rate, 73kJ/°CA followed by JB10 (71 kJ/°CA) and JB20 (69

$\text{kJ/}^{\circ}\text{CA}$). Heat release rate describes the fire characteristics, where the higher the heat release rate, the rate of pressure and temperature will also become rapid and lead to acceleration of fire development. As a result, the concentration of oxygen reduced and the production of exhaust gases increased as the combustion of fuels were less proper.

For the engine performance, as expected from the value of energy content or heating value, conventional diesel, JB0 recorded the best performance followed by JB10 and JB20. JB0 has the highest maximum brake power with value of 4.04kW, JB10 has 2.47% lower with the value of 3.94kW and JB20 has the lowest among those three blends with 3.89kW (3.71% lower than JB0 and 1.27% lower than JB10).

The biodiesel has found to have lower emission of exhaust gas because the oxygen content in the fuel compound is higher than diesel. The presence of excessive oxygen in the fuel helps to produce better combustion and reduce the production of carbon monoxide, carbon dioxide and unburned hydrocarbon.

The analysis of combustion, performance and emission obtained comparable characteristics between diesel and Jatropha biodiesel blends. It shows that Jatropha biodiesel has potential to help to reduce the diesel usage by blending both of them as standalone biodiesel will result poor performance although has more environmental friendly emission.

6.0 References

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