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Optimization of the Compression Ratio of a Diesel Engine Running on Croton Bio-diesel

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Declaration

I, Shadrack Kimanzi Musyoka, hereby declare that this thesis represents my personal work, realized to the best of my knowledge. I also declare that all information, material and results from other works presented here, have been fully cited and referenced in accordance with the academic rules and ethics.

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Abstract

Over the past few decades, there has been growing concerns over the sustainability of petroleum derived fuels. Approximately 86.4 percent of the world depends on fossil fuel as their primary energy source. Numerous researches have found that fossil fuels are being depleted very rapidly and will be completely depleted in the near future. With no alternative energy source to the major fuel in the world, there will be the largest energy crisis ever experienced. Due to this, researchers all over the world are in constant search for alternative fuels that are renewable, sustainable, readily available and which can easily replace petroleum fuels. In Eastern Africa, one such potential renewable fuel source is the croton megalocarpus plant, which widely grows in the highlands of Eastern and Southern Africa.

A number of previous studies have identified croton seed biodiesel as a very promising fuel that can substitute diesel fuel in an internal combustion without any major modifications on the engine. However, most of these studies have focused more on the properties of the biodiesel rather than on its performance in an engine. Sivaramakrishnan studied Karanja biodiesel and found that a blend of B20 and compression ratio of 18 had even better performance than diesel fuel. Muralidharan and Vasudevan studied waste vegetable oil and found out that a blend of B40 had almost similar performance and lower emissions when compared to diesel fuel at higher compression ratios. Similar studies have also been done on other renewable fuel sources such as Jatropha and palm oil.

The aim of this study was to analyze the performance, combustion and emission characteristics of a variable compression ratio CI engine running on croton bio-diesel. Extensive research has shown that croton is relatively unexplored as a bio-diesel. Tests were done on blends of the biodiesel (B0, B20, B40, B60, B80 and B100) at compression ratios of 12, 14, 16 and 18. The performance characteristics to be considered included brake thermal efficiency, specific fuel consumption (SFC), brake thermal efficiency and brake mean effective pressure (BMEP). Combustion parameters to be considered included the mass fraction burnt, net heat release and mean gas temperature.

From the results of the study, croton biodiesel can be used in a diesel engine without any major modifications. The greatest challenge identified was the viscosity of the biodiesel, which affects the rate of the combustion process. As the compression ratio was increased from 12 to 18, the performance of the engine also increased. A blend of B20 has the best performance, with almost the same brake power as that of diesel fuel and a higher rate of heat release.

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List of Abbreviations

ASTM	American Society for Testing Materials
BMEP	Brake Mean Effective Pressure
BP	Brake Power
BSFC	Brake Specific Fuel Consumption
bTDC	Before Top Dead Centre
BTHE	Brake Thermal Efficiency
CI	Compression Ignition
CM	Croton Megalocarpus
CO	Carbon Monoxide
CR	Compression Ratio
DI	Direct Injection
FAAE	Fatty Acid Alkyl Ester
FMEP	Friction Mean Effective Pressure
GHG	Greenhouse Gases
HC	Hydrocarbon
IC	Internal Combustion
IMEP	Indicated Mean Effective Pressure
IP	Indicated Power
KSO	Kaner Seed Oil
MEP	Mean Effective Pressure
MFB	Mass Fraction Burnt
MGT	Mean Gas Temperature
MP	Millettia Pinnata
NASA	National Aeronautics and Space Administration
PM	Particulate Matter

PPME	Pongamia Pinnata Methyl Ester
SFC	SPecific Fuel COnsumption
SVO	Straight Vegetable Oil

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Chapter 1

Purpose and Significance of the Study

1.1 Overview

Fossil fuels, primarily petroleum and coal, are hydrocarbons formed from remains of ancient plants and animals. These plant and animal remains are exposed to heat and pressure in the absence of oxygen over a period of around a hundred million years. Since 1980s, the energy consumption globally has duplicated. The global energy mix is dominated by fossil fuels, representing over 80% of the total energy supplied in the world today. Due to the rapid economic development coupled with the modern life demands, the utilization of energy is expected to increase further. An increase in energy consumption will definitely cause unwanted emissions as well as depletion of the available fossil fuel reservoirs. Fossil fuels are only available in specific regions of the world and are nearing their maximum production. In the next few years, the world will reach its peak oil production. In 2010, the World Trade Organization recorded that fuels accounted for 15.8% of the total trade in primary products and merchandising, most of it being diesel fuel used in heavy duty engines and transportation. In addition, the world is faced with issues of environmental pollution and serious global warming. The major greenhouse gas, carbon dioxide, is mainly emitted by fossil fuels. Clearly, a serious crisis is approaching if the current trend in environmental degradation and fossil fuel depletion continues. To avert this looming crisis, there is need for extensive research on alternative fuel that is renewable, reliable, clean and economically viable. Any alternative to fossil fuel must be economically competitive, technically feasible, easily available and environmentally acceptable. Bio-fuels provide the most promising alternative to the fossil-derived fuels. Several bio-fuels such as corn oil, coconut oil and peanut oil have already been tested and found suitable. However, some ethical issues have been raised over the use of these bio-fuels since they are derived from foods. The study intends to use croton bio-diesel, which is not derived from foods. Croton oil is also available in abundance in most regions of Eastern and Southern Africa.

1.2 Objectives

General Objective

The main objective of the project is to investigate the effects of varying compression ratios on the performance, combustion characteristics and emissions of a compression ignition engine running on croton biodiesel-diesel blends.

Specific Objectives

1. To select the optimum compression ratio that will deliver optimum efficiency without knock for the various trans-esterified croton and diesel blends.

2. To recommend the best blend of the diesel and trans-esterified croton oil to be used based on the performance, combustion and emission characteristics.

3. To compare the performance and emissions of the CI engine when running on diesel fuel and when running on the trans-esterified croton oil and diesel blends.

1.3 Problem Statement

According to a projection by the US department of energy, the world's energy supply will reach its midpoint of depletion and maximum production around the year 2020. In addition, burning of fossil fuels for electricity production, transport and heating contributes the highest percentage of green house gas emissions. Greenhouse gases lead to problems such as global warming, melting of ice caps and rise of sea levels. The world is getting more concerned about the depletion of oil reserves, the environmental issues as well as the ozone layer depletion caused by combustion of fossil fuels. These issues have necessitated the search for alternative energy sources, with particular emphasis being laid on fuels that are renewable and more sustainable. A feasible solution to the twin crisis of fossil fuel depletion and environmental degradation is providing fuels of bio-origin.

1.4 Justification

In the recent past, a considerable effort has been directed towards obtaining fuels that can be used directly in a diesel engine without any major modifications to the engine. Compared to diesel fuel, bio-diesels have a number of superior combustion characteristics. The fuel characteristics of croton bio-diesel is almost similar to those of diesel fuel, which indicates that it can be directly used in a diesel engine. In addition, the bio-diesel can be mixed with diesel at any ratios, is free from sulphur and is bio-degradable.

Chapter 2

Literature Review

2.1 Introduction

Biodiesel is a renewable energy source that provides numerous benefits such as reducing the emission of greenhouse gases and pollutants, economic security and increasing energy diversity. Worldwide, biodiesel is considered one of the most promising alternative fuels to diesel fuel in the transportation sector since it has similar fuel properties to the conventional diesel. In addition, it can be used in power generation plants without major modifications. Previous analysis of carbon dioxide emission from pure biodiesel fuel indicates a 60% CO emission reduction compared to diesel [2].

The concern of finding an appropriate substitute for petroleum-based diesel is growing, generally because of the continuous diminishing of fossil fuel reserves, the price of petroleum-based diesel, dependency on fossil fuel imports and destruction of the environment. According to a study done in 2005, fossil fuel reserves of fuels used in IC engines were estimated to be depleted in 40 years if the rate of consumption increases at 3% per annum [3].

Globally, concerns on the price of petroleum and pollution from car emissions are on a continuous increase. The only feasible solution to these problems is using alternative fuels. Extensive research has been done to date on production of bio-diesels from edible vegetable oils rather than the non-edible ones due to high yields and easy processing of the former. Using edible vegetable oils poses a threat to food production and security [4]. For the purpose of continuous production of bio-diesels without affecting the food industry negatively, research on non-edible vegetable oils is being taken into careful consideration. Wastelands and non-cropped marginal lands can be used to grow appreciable quantities of crops that produce non-edible oil [5].

In the efforts to find a suitable alternative to diesel fuel, different countries have set targets and mandate to make use of biodiesel for transportation. The European Union has targets of using 10%

biodiesel by 2020. The Australian government has set a target of 20% biodiesel use by the year 2020. Other countries such as China have also set their targets. These commitments have greatly boosted the global biodiesel production. According to 2015 statistics, there was a 10.3% increase in biodiesel production from 2004 to 2014. Some countries, especially the developing countries, have not yet set targets for biodiesel use [2].

The most outstanding credit of biodiesels compared to other fuels of bio origin is the wide range of feedstocks available all over the world. Every region in the world has at least one readily available biodiesel feedstock. Most countries make use of the feedstock that is readily available in their region. For example, the United States widely uses soybean oil, Europe uses rapeseed oil extensively, while Malaysia produces a lot of biodiesel from palm oil. In connection to this, East African countries such as Kenya, Uganda, Tanzania and Rwanda have the opportunity to use biodiesel from croton megalocarpus due to its availability in the region [2].

Production of economical oils suitable for making bio-diesels will require new low cost non-edible oil crops. A possible alternative in providing bio-diesel in East Africa is Croton megalocarpus oil [3].

The properties of trans-esterified croton bio-diesel are within the range of the recommended ASTM values and are also comparable to those of petro-diesel. The properties of the bio-diesel are shown in table 2.1 [6].

Table 2.1: Physical Properties of Croton Bio-diesel

Property	**Biodiesel**	**Petro-diesel**	**ASTM recommended value**
Density (40^oC, g/cm^3)	0.8858	0.8231	N/A
Kinematic Viscosity (40^oC, cs)	4.51	2.87	1.9-6.0
Calorific Value (J/g)	39179	44648	
Cloud Point (^oC)	-1.5	4	
Pour Point (^oC)	-6.5	-2	
Flash Point (^oC)	>200	65	130 (Minimum)
Acid Number (mg KOH/g)	0.336	ND	0.8 (Maximum)

2.2 Diesel Engine

Since the appearance of the first diesel-powered engine in 1898, diesel engines have been widely used as a power source for transportation, power generation and other different industrial systems. The major engine types include the spark ignition engine and the diesel engine. The major difference between the two engine types is in the fuel ignition process. In the gasoline engine, ignition is triggered by a spark plug, while a diesel engine ignites by spontaneous compression ignition (CI).

The rise in temperature and pressure during the compression allows the fuel to be injected, atomized, vaporized and burnt.

The diesel engine typically operates between a compression ratio of 12 and 24. A higher compression ratio promotes efficienct combustion of the fuel due to the increased expansion stroke. Typically, diesel engines run lean where the air-fuel ratios are as high as 65:1, except when at full power. The control of the engine speed and power is done by varying the amount of fuel in the injection, not by throttling the intake air. This control mechanism gives diesel engines higher thermal efficiency. The low fuel consumption of diesel engine makes it to be preferred in applications such as railroad engines, trucks, power production, ships and other industrial applications.

The wide application of diesel engine in the transportation sector is due to its higher reliability, simple arrangement and design, higher thermal efficiency and more power at lower fuel consumption. Application of diesel engine depends on the engine speed, where engines with speeds above 1200rpm find their applications in the transportation sector. Engines with medium speeds of between 300 and 1200rpm are mainly used in ships, large pumps, electrical generators and compressors [7].

2.3 Engine Performance and Combustion Parameters

2.3.1 Brake specific fuel consumption

Brake specific fuel consumption (BSFC) is a parameter that clearly reflects the efficiency of any combustion engine which produces rotational power at the crankshaft or shaft by burning fuel. BSFC is used in automotive applications to determine the efficiency of an internal combustion engine. To produce energy, the internal combustion engine requires air and fuel. A dynamometer is used to measure the amount of fuel used as a mass flow rate(kg/s). The mass flow rate cannot be used to determine the efficiency of an internal combustion engine because it does not indicate the amount of power that can be extracted from the fuel. When the fuel mass flow rate (kg/s) is divided by the engine output power (W), the result is the brake specific fuel consumption (kg/J).

$$BSFC = m_f/P_e$$

Where:

m_f (kg/s) — fuel flow rate (measured on the engine dynamometer)

P_e(W) – effective (brake) engine power

BSFC(kg/J) – brake specific fuel consumption

Since the engine power is measured in kW and the fuel mass flow rate in g/s, the specific fuel consumption is obtained in g/kWh.

$$BSFC = m_f/P_e * 3600$$

Since the engine power is a product of the torque and speed, it is possible to express BSFC as a function of engine torque and speed.

$$BSFC = m_f/(w_e * T_e)$$

Where:

T_e(Nm) – effective engine torque

w_e(rad/s) – engine speed.

The engine torque (Nm) can also be expressed as a function of the mean effective pressure (BMEP).

$$T_e = (n_c V_d p_{me})/(2\pi n_r)$$

Where:

n_c – number of cylinders

$V_d(m^3)$ – cylinder displacement (volume)

p_{me}(Pa) – mean effective pressure n_r – number of crankshaft rotations for a complete engine cycle (n_r=2 for a 4-stroke engine)

The brake specific fuel consumption can, therefore, be expressed as a function of the mean effective pressure of the engine.

$$BSFC = (2\pi n_r m_f)/(n_c V_d p_{me} w_e)$$

A low brake specific fuel consumption is an indication of a more efficient engine. For a compression ignition engine, the brake specific fuel consumption is around 200g/kWh.

2.3.2 Brake mean effective pressure

The Mean Effective Pressure (MEP) is simply described as a theoretical parameter used for the measurement of internal combustion engine performance. Mean effective pressure is not an actual pressure measured within the engine cylinder although it contains the word pressure. The actual engine pressure changes continuously during the combustion cycle and is best illustrated by the pressure-volume (pV) diagram. The mean effective pressure is the average pressure in the cylinder in a complete cycle of the engine. It is a ratio of the work and the engine displacement:

$$p_{me} = W/V_d$$

The engine work can be expressed as:

$$W = p_{me} V_d$$

The power of the engine has a direct relationship with the work produced:

$$W = (n_r P)/n_e$$

By equating the two expressions for engine work, we can express the mean effective pressure as:

$$p_{me} = (n_r P)/(n_e V_d)$$

Also, power (P) is a product of torque (T) and speed (w):

$$P = wT = 2\pi n_e T$$

Therefore, the mean effective pressure can be expressed as a function of the engine torque:

$$p_{me} = (2\pi n_r T)/V_d$$

This expression indicates that the mean effective pressure is not affected by the engine speed (ne). Also, it is possible to compare internal combustion engines with different displacements since the torque has been divided by the engine capacity. For engines with multiple cylinders, it is necessary to take into account the total volume of the engine. The expression, therefore, becomes:

$$p_{me} = (2\pi n_r T)/(n_c V_d)$$

Where:

p_{me}(Pa) – mean effective pressure

$V_d(m^3)$ – engine (cylinder) displacement

n_r – number of crankshaft rotations in an engine cycle (n_r=2 for a 4-stroke engine)

n_c – number of cylinders

There are three different expressions of the mean effective pressure:

- Indicated mean effective pressure (IMEP) which is calculated with the indicted power.
- Brake mean effective pressure (BMEP) calculated from the dynamometer power (torque). BMEP indicates the actual output of the IC engine.
- Friction mean effective pressure (FMEP) which indicates the MEP lost by friction.

The following are the important points to note about brake mean effective pressure (BMEP):

- For a particular engine speed, the maximum brake mean effective pressure is obtained at full load.
- Throttling the engine leads to a decrease in the BMEP resulting from higher pumping losses.
- When a fixed engine displacement is considered, more effective torque is produced at the crankshaft if we increase the BMEP.
- A 2-stroke IC engine has almost double torque when compared to a 4-stroke engine for the same value of BMEP.
- A higher BMEP translates to higher thermal and mechanical stress on the engine components.

2.4 Biofuels

2.4.1 Croton Megalocarpus Hutch

Croton megalocarpus is found in East Africa and parts of Central and Southern Africa. It is distributed from Democratic Republic of Congo to Malawi, Zambia, Mozambique, and all countries in East Africa. Croton Megalocarpus is a multi-purpose plant, mainly being used for timber, medicine, firewood and as an auxillary plant [8].

The croton plant grows in high altitude areas, ranging from 4,000 to 6,700 feet, of East Africa and is commonly used as shade in coffee plantations and in homes.Croton megalocarpus Hutch belongs to the family Euphorbiaceae. The height of a fully grown plant is approximately 120 feet, with a trunk diameter of 2 to 4 feet and a clear cylindrical pole of 40 to 60 feet. It starts bearing nuts at 3 years and attains maturity at 11 years. The croton oil is obtained by pressing the dry nuts to obtain the straight vegetable oil (SVO), which is then processed to obtain the bio-diesel. The plant has been growing naturally in many parts of Kenya in homes, farms and forests [6].

A fully grown croton megalocarpus produces approximately 50kgs of seeds while one hectare can produce 5-10 tonnes of seeds in a year. Mature seed pods drop within a few weeks after they are ripe. Because of the poisonous alkaloids and the carcinogenic fatty acid esters of phorbol contained in the croton seed, it is not healthy to use the oil or cake in animal feed diet. However, the high nitrogen content of kernel cake is an indication that it can be used as an organic fertilizer [9].

2.4.2 Development of Biodiesel as an Alternative Fuel

Natural resources that our nation relies heavily upon such as oil, petroleum and natural gas are fossil fuels. These are non-renewable sources. They will eventually cease to exist. The thought of supply ending also causes a search for renewable sources that would never cease to exist. It is often reported that Rudolph Diesel designed his engine to run on peanut oil, but this is not the case. Diesel stated in his published papers that Otto Company showed that a small Diesel Engine was run on arachide (earth-nut or peanut) oil. It worked so smoothly that that only a few people were aware of it. The engine was constructed for using mineral oil, and was then worked on vegetable oil without any alterations being made. During a speech delivered in 1912, Rudolph Diesel said, the use of vegetable oils for engine fuels may seem insignificant today but such oils may become, in the course of time, as important as petroleum and the coal-tar products of the present time. [10]

Despite the widespread use of fossil petroleum-derived diesel fuels, interest in vegetable oils as fuels for internal combustion engines was reported in several countries during the 1920s and 1930s and later during World War 2 and 1970s oil crisis. Belgium, France, Italy, the United Kingdom, Portugal, Germany, Brazil, Argentina, Japan and China were reported to have tested and used vegetable oils as diesel fuels during this time. Some operational problems were reported due to the high viscosity of vegetable oils compared to petroleum diesel fuel, which results in poor atomization of the fuel in the fuel spray and often leads to deposits and choking of the injectors, combustion chamber and valves. Attempts to overcome these problems included heating of the vegetable oil, blending it with petroleum-derived diesel fuel or ethanol, pyrolysis and cracking of the oils [11].

On 31^{st} August 1937, G. Chavanne of the University of Brussels (Belgium) was granted a patent for a Procedure for the transformation of vegetable oils from their uses as fuels. This patent described the alcoholysis (often referred to as trans esterification) of vegetable oils using ethanol in order to separate the fatty acids from the glycerol by replacing the glycerol with short linear alcohols. The transesterification reaction is the basis for the production of modern biodiesel [12].

More recently in 1977, Brazilian scientist Expedito Parente invented and submitted for patent, the first industrial process for the production of biodiesel. This process is classified as biodiesel by international norms, conferring a standardized identity and quality. No other proposed biodiesel has been validated by the motor industry. Currently, Parentes company Tecbio is working with Boeing and NASA to certify bioquerosene (bio-kerosene), another product produced and patented by the Brazilian scientist.

Research into the use of trans esterified sunflower oil, and refining it to diesel fuel standards, was initiated in South Africa in 1979. By 1983, the process for producing fuel-quality, engine tested biodiesel was completed and published internationally. An Austrian company, Gaskoks, obtained the technology from the South African Agricultural Engineers and erected the first biodiesel pilot plant in November 1987, and the first industrial-scale plant in April 1989(with a capacity of 30,000 tons of rapeseed per annum)

In 1991, the European community proposed a 90% tax reduction for the use of bio fuels. Today 21 countries worldwide produce biodiesel. Throughout the 1990s, plants were opened in many European countries, including Czech Republic, Germany and Sweden. France launched local production of biodiesel fuel (referred to as diester) from rapeseed oil, which is mixed into regular diesel fuel at a level of 5% and into the diesel fuel used by some captive fleets (e.g. public transportation) at a level of 30%. Renault, Peugeot and other manufacturers have certified truck engines for use with up to that level of partial biodiesel; experiments with 50% biodiesel are underway [13].

In 1997, Kyoto protocol prompted resurgence in the use of biodiesel throughout the world. Under the protocol, 37 countries commit themselves under the reduction of four greenhouse gases (GHG) i.e. carbon dioxide, methane, nitrous oxide, sulphur hexafluoride and two groups of gases (hydrofluorocarbons and perfluorocarbons) produced by them. During the same period, nations in other parts of the world also saw production of local biodiesel starting up [14].

In September 2005, Minnesota became the first US state to mandate that all diesel fuel sold in the state contain biodiesel, requiring a content of at least 2% biodiesel. In 2008, ASTM published new Biodiesel Blend Specifications Standards.

2.4.3 Current Biodiesel Status in the World

For years, biofuels have been utilized to strengthen agricultural development domestically, increase energy self-sufficiency and reduce import costs [15]. In regions that have targets of minimizing emissions and increasing sustainability, the strategic focus has been on biomass-based transport fuels. Along with electric vehicles, the fuels are seen as a great leap towards low-carbon fuels that would increase the sustainability of the transport sector.The transport sector alone accounts for a third of the world's energy utilization, nearly a quarter of emissions of CO_2 from fossil fuels and half of oil consumption [16].

The supply of biofuels has increased globally by 8%, equalling 4% of the fuels used for transport in 2015. Policies such as the blending mandate have boosted this significant rise. Such policies are vital in insulating biofuels in instances where there is oil price flux and also foster greater utilization.

The only practical solution in providing low-carbon alternative in heavy machinery industries such as heavy freight, marine transport and aviation is the use of biofuels. Although there exists a wide opportunity and interest in biofuels, there has been a slow down in the near-term plant construction. The evident double-digit supply of 2010 is not there any more, a reflection of the existence of structural challenges and policy uncertainty. If the existing technical challenges and sustainability issues are resolved, biofuels have the ability to shape the future of the agricultural and transport sector [16].

In Europe, there has been commercial production of biodiesel since 1992, and 80% of biodiesel is consumed in European countries. In 2013, 10,367,000 tonnes of biodiesel were produced by EU countries.

Other countries also such as Asia (China and India in particular) and United States have also experienced double digits growth in biodiesel markets. In these countries, the respective governments have set targets of replasing 15% of petrodiesel with biodiesel by 2020. However, due to the high feedstock prices, the biodiesel industry has been under pressure since 2007. The global leaders in biodiesel production in 2017 are shown in figure 2.1 [17]

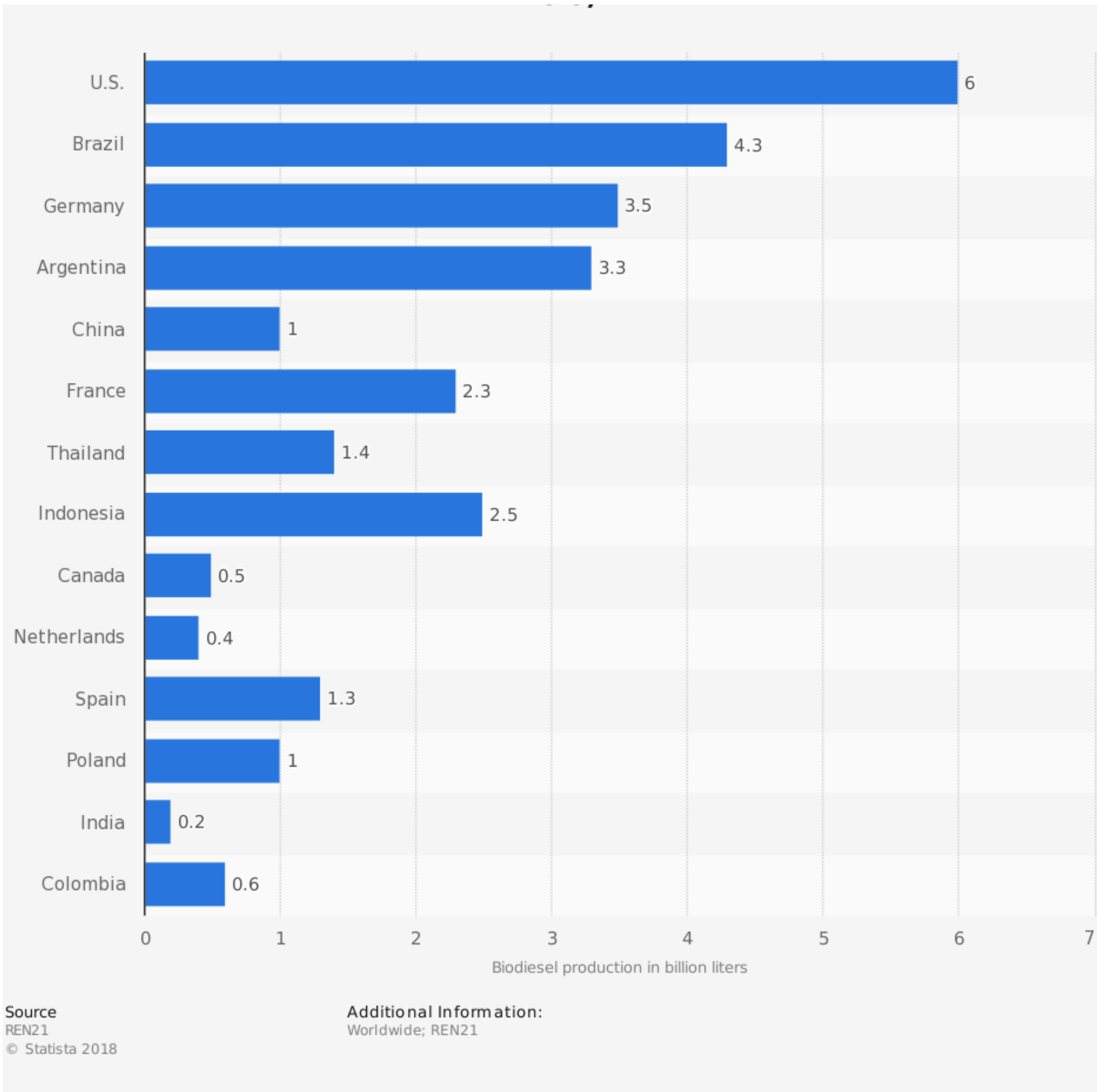

Figure 2.1: Leading biodiesel producers in 2017 (in billion liters)

2.4.4 Key Issues with Biodiesel Sustainability

A number of sustainability concerns have been raised on the use of biofuels. These issues affect the economy, society and ecological systems. The following are some of the concerns that have been raised on biofuels.

The Food-Fuel Debate

The periods 2006-2008 and 2010-2011 experienced a lot of agriculture prices volatility globally. The main agricultural foods affected during the periods were cereals, oilseed and sugar [1]. All these three products are used as feedstocks for the production of biofuels. This led to serious concerns about the competition for farmland between food and biofuels. Although price fluctuations of food products are common, the number of affected countries and the degree of volatility for these two cases was quite high [18]. Figure 2.2 shows the variation of agriculture prices from 1990 to 2015.

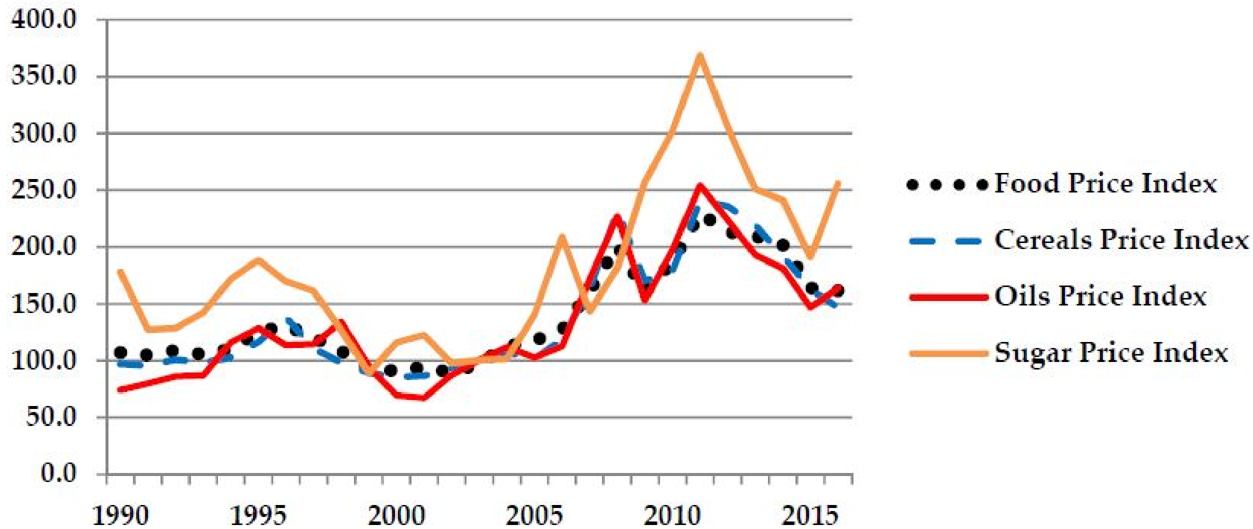

Figure 2.2: Food price index from 1990 to 2015 [1]

During this same period where there was an increase in food prices, there was also rise in the amount of biofuel produced, leading to questions on how the two were linked. In the G20 summit of 2008 and 2011, the major concerns were how the biofuels were impacting food prices and how land meant for food production was being used to produce feedstock for biofuels [19]. However, further analysis indicated that a number of other factors such as weather conditions, high oil prices, biofuel production and investor speculation may have contributed to the change in food prices. The debate on food versus biofuels still remains as there is no any conclusive argument on the issue [19].

Land

Land has been a major issue of concern for a long time in the development process of biofuels. It is expected that land use for societal expansion, food and biofuels will continue to grow proportionally to the population. According to an estimate by the United Nations Food Agriculture Organization, cultivated land grew by 159 million hectares (Mha) from 1961, with agricultural production growing by a factor of 2.5-3 and irrigated land doubling. During this period, genetics, pesticides, fertilizers and use of digital information played a great role in reducing food shortages. Although a recent estimates shows that the land dedicated to biofuel crops is less than 3%, there are still concerns over indirect land use change and land grabbing for biofuels. There, however, exists opportunities to expand cultivatable land in some regions, especially in Africa, Central America and South America [20].

Water

Just as is the case with land, there are issues over the limits of water that can be used for biofuels. Approximately 80% of fresh water is currently being utilized for agriculture, and water scarcity is already being experienced in a number of regions in the world. The number of countries experiencing water shortages is expected to rise to 55 by 2050 from the current 30.

Development of biofuels coupled with rising food production could affect the supply and quality of water. If more fertilizers are used to produce more energy and food crops, this could accelerate the water problems already being experienced. However, there are some programs being developed such as strategically planting switchgrass to minimize groundwater contamination by nitrates. The compounding problems of limited water resource and rising energy demand leave the debate still open for discussion and monitoring [21].

Biodiversity

Converting land for production of biofuels may have a greater impact on the biodiversity. The process involves clearing forests, which leads to disruption of natural habitats for numerous species. However, research is still ongoing on the impact of cultivating energy crops on the biodiversity [16].

2.4.5 Biodiesel Production Procedure

Generally, there are several well-established and accepted technologies used for biodiesel fuel production. It is suitable to reduce the viscosities of animal fats and vegetable oils to obtain products that have suitable properties that can be used in a diesel engine. There are a number of modification procedures for production of better quality bio-diesels which include microemulsifications, direct use and blending, trasnesterification and pyrolysis of vegetable oil [22].

Direct Use and Blending of Oils

Since 1900 when Dr. Rudolph Diesel, the inventor of diesel engine, first used peanut oil in the compression ignition engine, the use of alternative fuels from vegetable oils has been in existence. Direct use of vegetable oil in a diesel has been found to have a lot of problems to the engine and may cause engine failure. Although vegetable oil has been in use in the engine for over 100 years, extensive research has only been done in the past few decades. Mixing vegetable oils directly with diesel helps to reduce the viscosity of the fuel. High viscosity of vegetable oils is the main problem associated with the use of pure vegetable oils in a CI engine. Direct use of vegetable oils in the diesel engine has been found to be impractical and unsatisfactory. The obvious problems include acid composition, high viscosity, gum formation due to oxidation, carbon deposits, polymerization, free fatty acid content and lubricating oil thickening. Since blending and heating of vegetable oils does not change their molecular structure, the polyunsaturated character remains [23].

Microemulsion of Oils

A potential solution to the problem of high viscosity in vegetable oil is the microemulsification, which is the formation of microemulsions (co-solvency). A microemulsion can be defined as a colloidal equilibrium dispersion of fluid microstructures that are optically isotropic with dimensions in the range 1-150 nm formed spontaneously from two liquids that are immiscible and one or more nonionic or ionic amphiphiles [24].

Pyrolysis of Oils

Pyrolysis is the process of converting one organic substance into another organic substance by heating or heating with the aid of a catalyst. The pyrolyzed material can be animal fat, methyl esters of fatty acids, natural fatty acids or vegetable oil. A promising technology of producing bio-diesel is converting animal fats and vegetable oils that are mainly composed of triglycerides by thermal cracking reactions. The technology is more promising in areas with well-established hydro-processing industry since the technology has huge similarities to the conventional petroleum refining. The liquid product of thermal decomposition of vegetable oil has fuel properties that approach those of diesel fuel. Pyrolysis of triglycerides can either be a catalytic or non-catalytic process. Because of the structures and the multiplicity of the reactions in the thermal decomposition of triglycerides, the mechanisim is very complex, as shown in figure 2.3 [25].

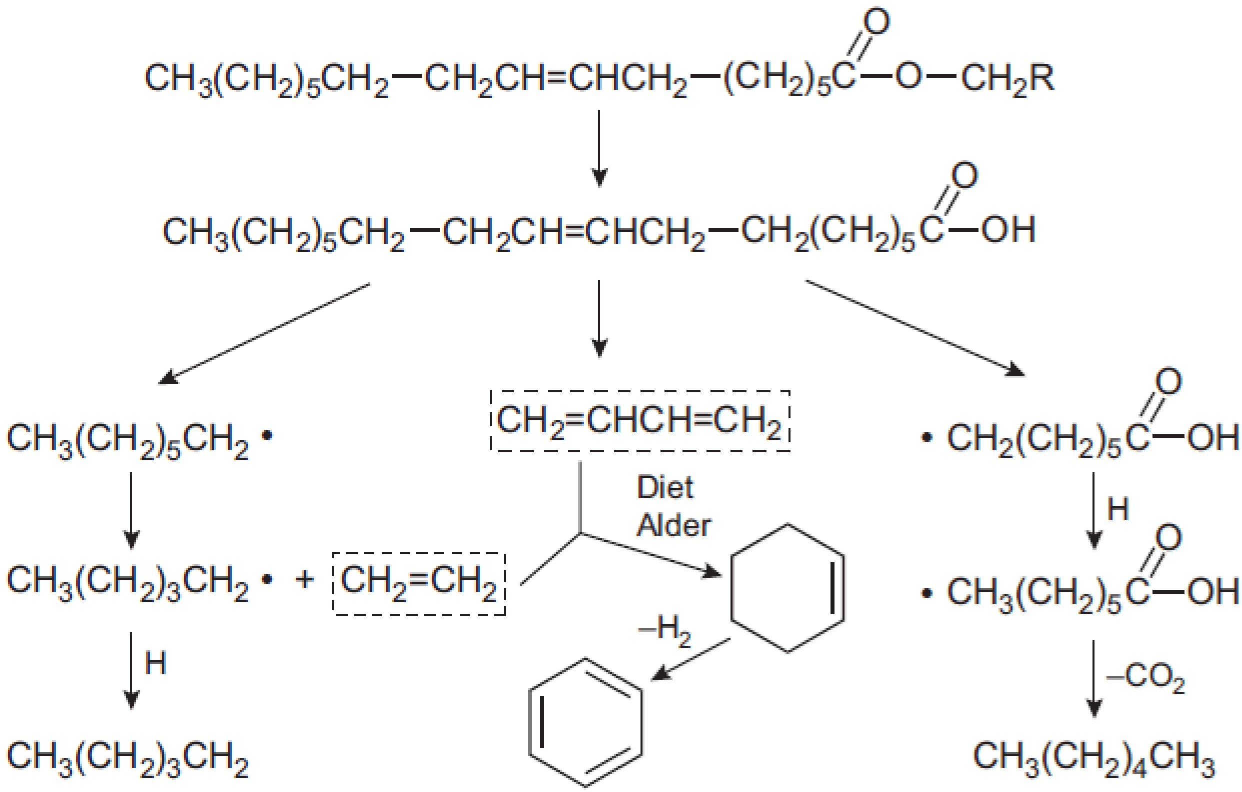

Figure 2.3: The mechanism of thermal decomposition of triglycerides

The process of thermal cracking and pyrolysis has a number of disadvantages that makes researchers not to prefer it. For modest throughputs, the equipment required for pyrolysis and thermal cracking is very expensive. In addition, although the products from the process have similar chemical composition to that of gasoline and diesel fuel derived from petroleum, any environmental benefits of using oxygenated fuel are removed during the process of thermal processing since oxygen is removed [23].

Transesterification of Oils

Transesterification is the most common technology used to produce bio-diesel. Transesterification of oils (triglycerides) using alcohol gives the main product as biodiesel (fatty acid alkyl esters, FAAE) and the by-product as glycerine. The basic process of transesterification is shown in figure 2.4. The process involves a series of reactions, the first being the conversion of triglycerides to diglycerides. The diglycerides are then converted to monoglycerides while the monoglycerides are converted into glycerol, with each glyceride yielding one methyl ester molecule at each step [23].

Also called alcoholysis, the process of transesterification involves the exchange of alcohol from an ester by another alcohol, just like the hydrolysis process, except that instead of water, alcohol is used. The most critical variables in the process include the kind of feedstock, concentration and type of catalyst, reaction time, mixing intensity (rpm), alcohol to oil ratio and reaction temperature [26].

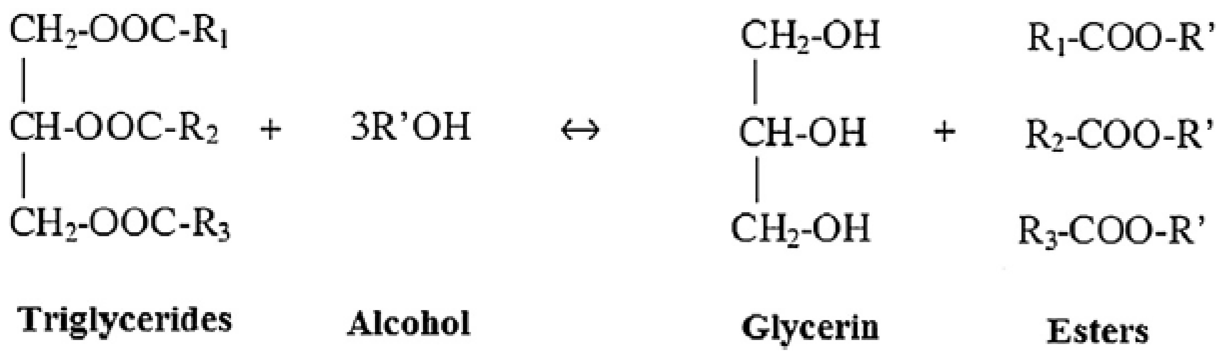

Figure 2.4: Transesterification reaction of triglycerides with alcohol

2.4.6 Biodiesel Fuel Properties

According to ASTM, biodiesel is defined as "a fuel comprised of monoalkyl esters of long-chain fatty acids derived from vegetable oils or animal fats, designated B100" [27]. The properties and composition of any biodiesel depend on the unsaturation degree of the fatty acid alkyl chains. The combustion properties when compared to diesel fuel are affected by lack of aromatic compounds, higher number of double bonds and molecular weight, and oxygen presence in the esters. Every fuel has its specific macroscopic properties that are interdependent. It is possible to generalise the influence of a particular fuel property on the engine performance and emissions, although fuel properties are difficult to isolate, independent and vary independently. Heating value, flow characteristics, cetane number and viscosity are the fuel properties that mostly affect the performance of an engine [27].

Biodiesel is an oxygenated fuel, containing between 10-15% oxygen by weight, and lower hydrogen and carbon contents when compared to diesel fuel. This difference means that biodiesel has approximately 10% lower mass energy content when compared to diesel fuel. However, the volumetric energy content is just 5-6% lower than diesel due to the higher fuel density of the biodiesel. The properties of biodiesel can greatly vary from one feedstock to the other.

Kinematic Viscosity

Viscosity is defined as the measure of a fluid's resistance to flow caused by the internal friction of the fluid, and mainly affected by temperature and the molecular structure. In fuels, the kinematic viscosity is a very critical property since it affects the injection behaviour. Generally, higher viscosity results into poor fuel atomization. It also causes narrower injection spray angle, larger droplet sizes, greater in-cylinder penetration and poorer vaporization. All these lead to higher emissions, poorer combustion and increased oil dilution [28]. Typically, biodiesel has higher viscosity than diesel fuel, often by a factor of two. High viscosity means that the pump is unable to sufficiently supply fuel to the pumping chamber, resulting in engine power loss. On the other hand, too low viscosity leads to fuel leakage through the injection system seals. Most of the problems associated with high viscosity are more visible during cold-start and low ambient temperature engine conditions [29].

Density

The density of the fuel is a key parameter that affects the performance of the engine. Since fuel metering by the injection pump is by volume, not by mass, the mass of fuel injected varies depending on the density. Variation in the mass of fuel injected affects the air-fuel ratio and the energy content in the combustion chamber, which in turn affects the combustion. To inject the same mass of fuel, lower density fuels require more injection duration. Generally, biodiesel fuels have a slightly higher density than mineral diesel (less than 6%) [30].

Energy Content

Energy content refers to the amount of energy per unit volume or mass that is given out during the combustion process. Although a high-density fuel has a higher energy content per unit volume, a low-density fuel will have higher energy content per unit mass. Unless the injection of the fuel is optimized for each individual fuel, fuels that have different energy contents will produce different power outputs.Biodiesel contains lower mass energy values when compared to mineral diesel. Generally, biodiesel's mass energy content (MJ/kg) is approximately 10% lower than that of mineral diesel. To produce the same amount of engine power as that of mineral diesel when using biodiesel, one would need to increase the fuel flow rate [31].

Cetane Number

The cetane number of a fuel is based on the ignition quality. A higher cetane number means shorter ignition delay, which leads to less amount of fuel being injected during the premix burn and more fuel injection in the diffusion burn portion [32]. This has the impact of reducing the cylinder pressure rise and may lower the cylinder temperatures. On the other hand, a fuel with lower cetane number advances the ignition timing due to the shorter ignition delay, hence increasing the combustion pressure and temperatures. Typically, biodiesels have higher cetane number than mineral diesel due to the long-chain HC groups(with vitually no aromatic structures or branching) [31].

Cloud and Pour Points

The cloud point of a fuel is defined as the temperature at which small crystals or wax begin to form in the fuel. The fuel loses its ability to flow and begins to fully or partially solidify at this temperature. On the other hand, the lowest temperature at which the fuel flows is called the pour point. Compared to diesel fuel, biodiesels have significantly higher pour points and cloud points. This characteristic of biodiesel may cause major problems in the engine by plugging the flow line. This has caused a lot of difficulties in using biodiesels in cold climates [33].

2.5 Experimental Studies on Biofuels

Muralidharan and Vasudevan investigated the performance, combustion and emission characteristics of single cylinder multi-fuel engine fueled with waste cooking oil methyl ester and its 20, 40, 60 and 80% blends with diesel. The experiment was conducted at a fixed engine speed of 1500 rpm and 50% load. The results indicated that using waste cooking methyl oil leads to longer ignition delay, lower heat release rate, maximum rate of pressure rise and higher mass fraction burnt at higher compression ratio compared to diesel. The blend B40 was found to give the maximum thermal efficiency. Use of the blends also resulted to lower levels of carbon monoxide and hydrocarbons, although the nitrogen oxides emissions increased [34].

According to a study by Rodrigo C. Costa and Jos R. Sodr on the influence of compression ratio on the performance of a spark ignition engine, higher compression ratios were found to improve the engine performance. The engine was a 1.0L, 8-valve and 4-cylinder fueled by a blend consisting of 78% gasoline and 22% ethanol [35].

Another study was conducted on the effect of compression ratio and injection timing on a DI engine running on Jatropha methyl ester. The performance parameters under investigation were brake thermal efficiency(BTHE), fuel consumption (BSFC) and emissions. It was found that increasing both the compression ratio and injection timing increases the BTHE but lowers the BSFC and emissions. The optimum combination was a CR of 18 and an IP of 250 bar for the small engine (3.5kW) used [36].

Researchers have also done an investigation on the effect of compression ratio on the performance, emission and combustion characteristics of a diesel engine running on raw biogas. The test was done on a 3.5kW single cylinder, water cooled, direct injection, variable compression ratio engine. Conversion of the engine to a biogas run dual fuel was done by connecting a venturi gas mixer at the inlet manifold. Experiments were conducted at varying compression ratios and different loading conditions, while the injection timing was fixed at 23 degrees bTDC. At 100% load, the maximum replacement of the diesel fuel was found to be 79.46%, 76.1%, 74% and 72% at CR of 18, 17.5, 17 and 16 respectively. There was also an average reduction in hydrocarbon and carbon monoxide emissions by 41.97% and 26.22% in dual fuel mode when the CR was increased from 16 to 18. Due to a reduction in the volumetric efficiency, there was higher emission of carbon monoxide and hydrocarbon in dual fuel mode compared to the diesel mode [37].

Experimental studies have been conducted on the combined effects of compression ratio,injection timing and nozzle opening pressure on the performance and emissions of a CI engine running on an emulsion fuel obtained from carbon black. The fuel consisted of 10% carbon black, 2% water, 85% diesel and 3% surfactant. The experiment was done on a single cylinder, 4 stroke, air cooled DI engine with a power of 4.4kW at a constant speed of 1500 rpm. The compression ratio was varied in steps of 1 from 16.5 to 18.5, the injection timing was advanced to a maximum of 3 degrees crank angle, while the nozzle opening pressure was set at 200, 220 and 240. The brake thermal efficiency and emissions of the emulsion fuel were found to be lower compared to diesel fuel at higher compression ratios [38].

Sivaramakrishnan studied the performance and emission characteristics of a multi-fuel engine with variable compression ratio using Karanja biodiesel-diesel blend. In the study, the impact of compression ratio on the brake thermal efficiency, fuel consumption and exhaust gas emissions was investigated. The optimum conditions were found to be a compression ratio of 18 and 25 % biodiesel-diesel blend [39].

Investigation was also done on the performance and emission characteristics of a variable compression ratio DI engine fuelled with pre-heated palm oil. The aim was to determine the suitability of raw palm oil as a fuel when pre-heated in the temperature range of 90 degrees. The experiment was conducted at compression ratios of 16, 17, 18, 19 and 20 and a constant speed of 1500 rpm. The parameters under investigation were indicated mean effective pressure, mechanical efficiency, brake power and emission characteristics. A blend of B20 was found to give a 14.6% higher mechanical efficiency when compared to diesel at higher compression ratios. The blend also produced a 6% higher brake power when compared to diesel. However, the indicated mean effective pressure was lower than that of diesel fuel. An increase in the blending ratio and compression ratio led to a decrease in the emission of CO and HC, but an increase in carbon dioxide emission. Optimum engine performance was obtained at a compression ratio of 20:1 and using B20 as fuel during full load [40].

Alireza Shirneshana conducted a study on the NO_x, CO_2, CO and HC emissions of a diesel engine running on blends of waste frying oil methyl ester. The experiments for the emission characteristics were done at a constant engine speed of 1800rpm and at four engine loads. Four fuel blends were tested (B20, B40, B60 and B80). The study concluded that the use of biodiesel reduced the concentration of CO and HC in the exhaust gases but increased the concentration of CO_2 and NO_x. Generally, a higher biodiesel concentration in the fuel reduced the emissions from the engine [41].

Another study was done by C.W. Mohd Noor, M.M. Noor and R. Mamat on the feasibility of biodiesel as an alternative to the diesel fuel in marine engine applications. The researchers noted that although biodiesels provide a promising alternative to diesel fuel in the marine engines, there are numerous challenges facing the adoption of biodiesel in the sector. Most engine manufacturers give conditional warranties to discourage the use of biodiesels. They, however, concluded that biodiesels have a very bright future in the marine industry if these issues are solved since they have shown to reduce emissions and give almost the same performance to that of diesel fuel [42].

A study was also carried out on the effects of trout-oil methyl ester on the performance and emissions of a diesel engine. The preparation of trout oil methyl ester was by transesterification. Using a single-cylinder, indirect injection, naturally aspirated diesel engine, tests were done on the trout oil and the various blends (B10, B20, B40 and B50). The results indicated significant changes in the engine power as well as improvements in the emissions for some blends (B40 and B50), except NO_x emissions which increased. Hydrocarbon (HC) and carbon monoxide (CO) emissions were shown to decrease by 45% and 13% respectively when trout oil methyl ester was used [43].

A group of researchers did a comparative evaluation of the edible and non-edible biodiesel feedstocks that are promising alternatives to diesel fuel. They deduced that using biodiesels lowered the brake thermal efficiency and brake power slightly but increased the brake specific fuel consumption (BSFC). The biodiesels also reduced the CO, CO_2, HC and PM emissions but increased NO_x emissions. The final conclusion form the research was that biodiesel can be used in the diesel engine without the need for any engine modifications. This can help a lot in reducing the dependency on fossil fuels, fulfilling the energy demand and reducing emissions [44].

An experimental study was done on biodiesel from binary mixture of poppy and waste cooking oil. The performance and emission characteristics of this biodiesel and its blends (B5, B10, and

B20) were determined at different loads (25%, 50%, 75% and 100%) and engine speeds (1220-2400 rpm). The performance parameters considered include torque, brake thermal efficiency (BTE), brake specific fuel consumption (BSFC) and brake power (BP). The results indicated an increase in the brake power with speed, while BSFC initially decreased up to a speed of 1400 rpm then and then increased. The results also indicated that for all the blends, the torque, BTE, BSFC and BP were lower compared to diesel fuel. Using biodiesel also reduced the emissions of carbon monoxide(CO), particulate matter (PM) and hydrocarbon (HC) but increased the NO_x emissions [45].

Abhishek Paul, Rajsekhar Panua and Durbadal Debroy also did a study pn the performance, combustion and emission characteristics of diesel-ethanol-biodiesel blends in CI engine. In their work, the percentage of Pongamia pinata methyl ester (PPME) was fixed at 50% while the concentration of ethanol was varied from 5% to 20%. It was found out that a blend D35E15B50 provided the best engine performance, with an decrease in the brake specific fuel consumption and an increase in brake thermal efficiency at full load. A combustion analysis showed an increase in heat release rate and cylinder pressure, indicating general improvement in the combustion characteristic of the engine. There was also a substantial improvement in the emissions of carbon monoxide and unburned hydrocarbon [46].

An evaluation was also done on the performance, combustion and emission characteristics of the blends of croton megalocarpus (CM) and Millettia pinnata (MP), commonly known as Karanja, in a diesel engine. It was noted that both CM20 (20% croton megalocarpus) and MP20 (20% Karanja) reduced the brake thermal efficiency by 1.41% and 3.36% and brake power by 0.53% and 3.70% respectively. A mixture of 5% Karanja, 15% CM and 80% diesel (MP5CM15) produced a better heat release rate and higher cylinder peak pressure and shorter ignition delay when compared to diesel fuel. The blend MP5CM15 was established as a suitable alternative to diesel fuel, except for the increased NO_x emissions [47].

A research has also been done on the exhaust emissions, combustion and performance characteristics of a diesel engine fuelled with croton biodiesel with antioxidant. Experiments were done on pure croton biodiesel (B100), B20 (20% croton, 80% diesel), and pure diesel fuel. Addition of antioxidants in the biodiesel lowered the brake specific fuel consumption, but had no efffect on the combustion characteristics. The final conclusion from the study was that biodiesel from croton magalocarpus oil can partially substitute mineral diesel in a CI engine [48].

Biplab, Niranjan and Ujjwall conducted a thermodynamic analysis of a diesel engine with variable compression ratio running on palm oil methyl ester. The experiments were carried out in a direct injection, water cooled, single cylinder variable compression ratio engine. The engine was at full load brak mean effective pressure of 4.24 bar and a constant speed of 1500 rpm. It was observed that when the compression ratio was increased, the heat release rate and peak pressure also increased [49].

Another study was also done on waste cooking oil to determine its operational characteristics when operating in a CI engine with variable compression ratio. The test engine was run at a constant speed of 1500 rpm, a 200 bar injection pressure, varying engine load and compression ratios 15:1 and 17.5:1. The performance parameters considered are brake specific energy consumption and brake thermal efficiency, while emissions included hydrocarbons (HC), carbon monoxide (CO) and oxides of nitrogen (NO_x). The results demonstrated that biodiesel-diesel blends had fewer emissions when compared to diesel and, therefore, the blends are suitable in a diesel engine [50].

Pankaj Dubey and Rajesh Gupta studied the impact of dual fuel from turpentine oil and Jatropha biodiesel on a variable compression ratio, single cylinder diesel engine. The experiments were conducted on blends of 50%, 70%, 90% and 100% Jatropha biodiesel and turpentine oil. Since Jatropha biodiesel is highly viscous, it was blended with turpentine oil, which has a relatively lower viscosity, to achieve a viscosity that is comparable to that of diesel fuel. The target was also to develop an alternative fuel that is also cost effective and which will help in eliminating the dependency on petroleum derived fuels. In both aspects of performance and emissions, the dual fuel was found to be a suitable substitute to diesel fuel. Experiments were done at compression ratios of 15.5, 17, 18.5 and 20. Using the dual fuel blend (JBT 50) at CR 20, the brake thermal efficiency increased by 2.17% and NO_x, HC and CO emissions reduced by 4.21%, 17.5% and 13.04% respectively [51].

Another study was one on the development of a hybrid reactor to be used for the production of biodiesel from Kaner Seed Oil (KSO). The hybrid reactor combines both mechanical stirring and hydrodynamic cavitation processes. It was found out that the biodiesel production from the hybrid reactor was significantly higher than using one process. The hybrid reactor improved the efficiency (time saving), cost (high yield) and environmental friendliness (lower catalyst %). A performance study on the biodiesel produced from the hybrid reactor indicated that as the compression ratios were increased (from 16 to 18), the engine performance improved when using blends of the biodiesel [52].

Studies have also been done on the impact of compression ratio and fuel injection pressure on a CI engine fuelled with blends of palm kernel oil-eucalyptus oil. The biodiesel was prepared from palm kernel oil by the process of transesterification then blended with eucalyptus oil. The blends used in the experiments were B5 (5% eucalyptus oil, 95% palm kernel biodiesel), B10 (10% eucalyptus oil, 90% palm kernel biodiesel) and B15 (15% eucalyptus oil, 85% palm kernel biodiesel). The injection pressures under study were 220, 210 and 200 bar. In addition, the compression ratios tested were 14, 16.5 and 19. The parameters under consideration included brake thermal efficiency, mechanical efficiency and exhaust gas temperature. It was noted that as the compression ratio, injection pressure and blending percentage of eucalyptus oil were increased, there was an improvement in the combustion and performance characteristics and emissions were lowered [53].

Senthil Ramalingam, Silambarasan Rajendran and Ravichandiran Nattan conducted a study on the influence of compression ratio and injection timing on the combustion, performance and emission characteristics of a diesel engine operated on Annona methyl ester. Brake thermal efficiency, specific fuel consumption and emissions were measured for a fuel blend of the Annona methyl ester (A20). It was discovered that the blend A20 can be sufficiently used in a diesel engine without any modifications. Further, an injection timing of 30^obTDC and a compression ratio of 19.5 produced the optimum operating conditions, close to diesel fuel. For the same blend, a combined increase in injection timing and compression ratio reduces specific fuel consumption, increases brake thermal efficiency and reduces emissions [54].

Sohan Lal and S.K. Mohapatra also investigated the emission and performance characteristics of diesel engine with variable compression ratio using biomass from producer gas. The maximum diesel saving attained at compression ratios 18, 16, 14 and 12 were 64.3%, 57.14%, 31.82% and 8.7%. When the compression ratio was increased from 12 to 18, the hydrocarbon emissions reduced by 63.62%. NO_x and SO_x emissions also reduced with the dual fuel mode [55].

Hariram and Vagesh Shangar studied the behaviour of a CI engine when the compression ratio is varied. The compression ratios tested were 16, 17 and 18 at varying loads. As the compression ratio was reduced from 18 to 16, the exhaust gas temperatures increased while the brake thermal efficiency reduced. In addition, the brake specific fuel consumption increased as the compression ratio reduced. Also, there was an increase in the ignition delay period and a reduction in peak cylinder peak pressure as the compression ratio was reduced. For higher compression ratios and loads, there was higher heat energy. In general, the compression ratio of 18 provided the best combustion and performance parameters [56].

2.6 Summary

Due to the diminishing volumes of fossil fuels and their impact on the environment, it is vital to find alternative fuels that can directly substitute them. So far, bio-diesels are the most feasible alternative source of energy to fossil fuels. Bio-diesels can be used directly in a diesel engine with or without modifications to the engine. Modifying the engine is too costly to both the manufacturer and the user. The chemical and physical characteristics of bio-diesels are slightly different from those of petroleum diesel, which poses a challenge in direct utilisation of the fuels. Using the bio-diesels directly in a diesel engine, therefore, affects its performance and emissions. To investigate clearly the extend of these effects, extensive research is required on bio-diesels to determine their suitability in the diesel engine. To date, considerable effort has been put into research on blended fuels. A huge number of studies on this topic only focus on engine operation at a constant compression ratio. Studies on variable compression ratio are limited.

2.7 Gaps

The available literature clearly indicates that most studies on bio-diesels have focused on operation of the engine at a constant compression ratio. Very few researchers have conducted experiments on variable compression ratio. In addition, there are no documented engine experiments done on croton bio-diesel, which is a very attractive non-edible biofuel, especially in East Africa where it is abundant.

The aim of the study is to optimize the compression ratio of a diesel engine running on croton bio-diesel. This study will provide more insights on the neglected aspect of compression ratio in an engine and also assess the viability of croton as a bio-diesel.

Chapter 3

Methodology

3.1 Bio-diesel fuel Production

The method adopted for producing the bio-diesel was by trans-esterification using an alkali-catalyst. There is already a worldwide accepted procedure of trans-esterification of bio-diesels using an alkali catalyst.

3.1.1 Trans-esterification of bio-diesel

Trans-esterification of bio-diesel is done in two main steps. The steps include:

1. *Acid value determination*

    - Use a filter bag to strain oil well into a large stockpot. The filter will catch even the tiniest particles of waste.
    - Mix 5 grams of sodium hydroxide with 500 milliliters of distilled water.
    - Mix in 4ml of oil and 40 ml of propan-2-ol in a beaker. Warm the contents in hot water until the mixture is clear.
    - Slowly add the water and sodium hydroxide to the beaker noting how many milliliters you add until the Ph is 8.5. Divide the number of milliliters of the water and sodium hydroxide mix you added to the beaker by 4. This is how much sodium hydroxide you need.

2. *Transesterification.*

- Place the sodium hydroxide until you have the amount determined by the acid value test. Seal the bag so that the sodium hydroxide does not absorb moisture. Wear plastic gloves as you do this.
- Pour quickly 200ml of methanol into container 2 using a funnel and close the container.
- Pour sodium hydroxide into the same container using a different funnel and screw the cap on the container.
- Rock the plastic container from side to side until the methanol has completely dissolved the sodium hydroxide and the liquid is clear. The solution inside the container will become hot and turn into sodium methoxide.
- Heat the oil to 1300F and pour it into a blender.
- Add the sodium methoxide into the blender.
- Close the blender tightly and mix on the lowest setting for 30 minutes.
- Pour the contents into another container and cap it.
- Allow the mixture to cool for 24 hrs. Two layers will then emerge. The one on top is the biodiesel and the one at the bottom is glycerol.
- Separate the biodiesel from the glycerol.
- Test the biodiesel. Pour equal parts of biodiesel and water into a clean container and shake for 10 minutes. If the biodiesel separates from the water you have made a good fuel.
- Place the biodiesel in a clean separating funnel with two cups of water. Shake the bottle for 10 minutes. After the biodiesel and water separate, poke a hole at the bottom and let the water drain. Cover the hole with duct tape. Repeat the process until the biodiesel looks clear.

3.2 Blends Preparation

The blends were prepared by mixing the right proportions of the different fuels using calibrated containers. Depending on the desired ratios, the fuels were accurately measured using the calibrated containers and mixed to form the blends. The blends prepared include B0, B20, B40, B60, B80 and B100, where B0 is pure diesel and B100 is 100% biodiesel.

3.3 Preparing the Engine for the Tests

The set up consists of a single cylinder, four stroke, multi-fuel, research engine connected to an eddy current dynamo-meter for loading. The test engine is designed to operate on both diesel and petrol. It can also be coupled with an emissions analyser for measurement of the emissions. The engine also allows for varying the compression ratio without stopping it. The test engine used for the tests is shown in figure 3.1. Care has to be, therefore, taken to ensure that the diesel head is mounted before the start the experiment. The procedure of mounting the diesel head is available in the test engine manual. The engine specifications are provided in table 3.1.

Table 3.1: Engine Specifications

EngineManufacturer	Kirloskar Oil Engines, India
EngineType	1 Cylinder, 4 Stroke
BoreandStroke	87.5mm by 110mm
Capacity	661cc
Fuel	Multi-fuel. Diesel mode: power 3.5kW, speed 1500rpm CR range 12:1 to 18:1 Injection variation 0-25 deg BTDC
Dynamometer	Eddy Current Type, water cooled with loading unit
PropellerShaft	With universal joints
FuelTank	Capacity 15lit, Type: Dual compartment, with fuel metering pipe of glass
CrankAngleSensor	Resolution 1 deg, Speed 5500rpm with TDC pulse
PiezoPoweringUnit	Make: Apex, Model: AX-409
EngineControlHardware	Fuel injector, Fuel pump, Ignition coil, idle air
DigitalVoltmeter	Range 0-20V, panel mounted
TemperatureSensor	Type RTD, PT100 and thermocouple, type K
TemperatureTransmitter	Type two wire, Input RTD PT100, Range 0-100 deg C, Output 4-20 Ma and type two wire, Input thermocouple , Range 0.1200 deg C, Output 4-20 Ma
Loadindicator	Digital range 0-50kg, Supply 230VAC
LoadSensor	Load cell, type strain gauge, range 0-50kg
FuelFlowTransmitter	Pressure transmitter, Range 0-500mm WC
AirFlowTransmitter	Pressure transmitter, Range (-)250mm WC
Software	"Enginesoft" Engine performance analysis software

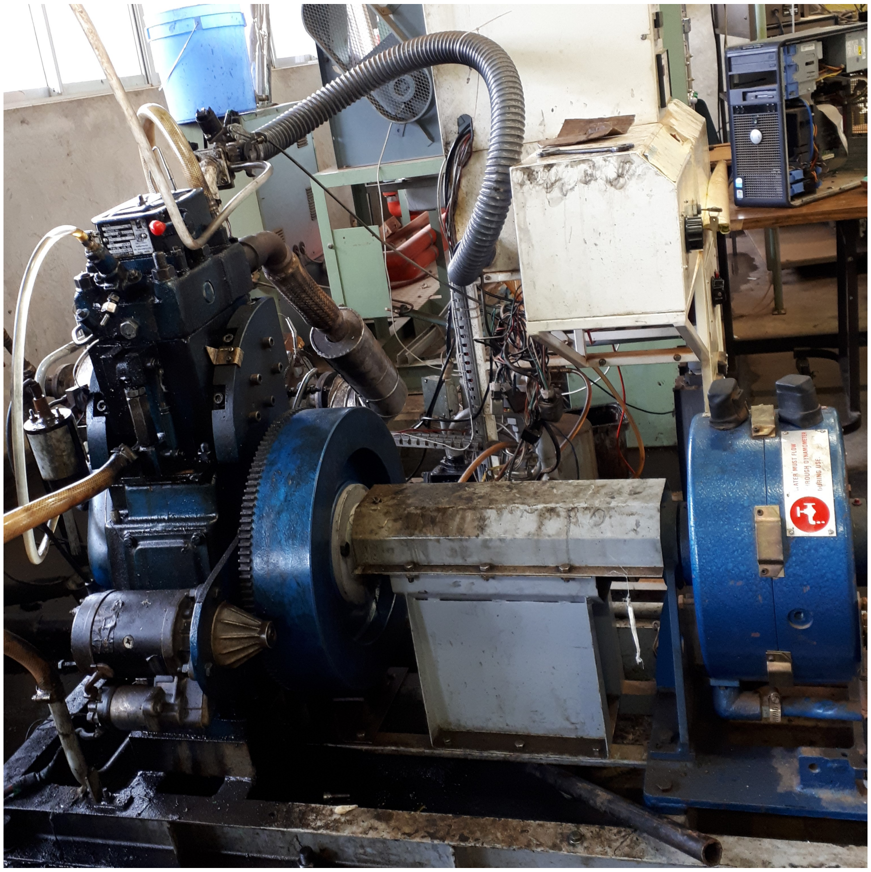

Figure 3.1: Test engine

3.4 Conducting the Tests

The test engine was connected to the "Enginesoft" software that is used to record the readings. The software interface is shown in figure 3.2. The engine was operated on pure diesel fuel at first to provide the baseline data for the other tests. For each of the compression ratios from 12 to 18, the measurements were taken from the software as well as from the emissions analyser by varying the load from 0kg to 12kg. The procedure was then repeated several times, using a different fuel blend each time. The performance parameters considered include brake power, brake thermal efficiency, specific fuel consumption and brake mean effective pressure. The emissions considered include carbon dioxide, oxides of sulphur and oxides of nitrogen. Figure 3.3 shows the emissions analyzer on the test bench. From these measurements, it was possible to obtain the engine performance and emissions for each compression ratio and fuel blend.

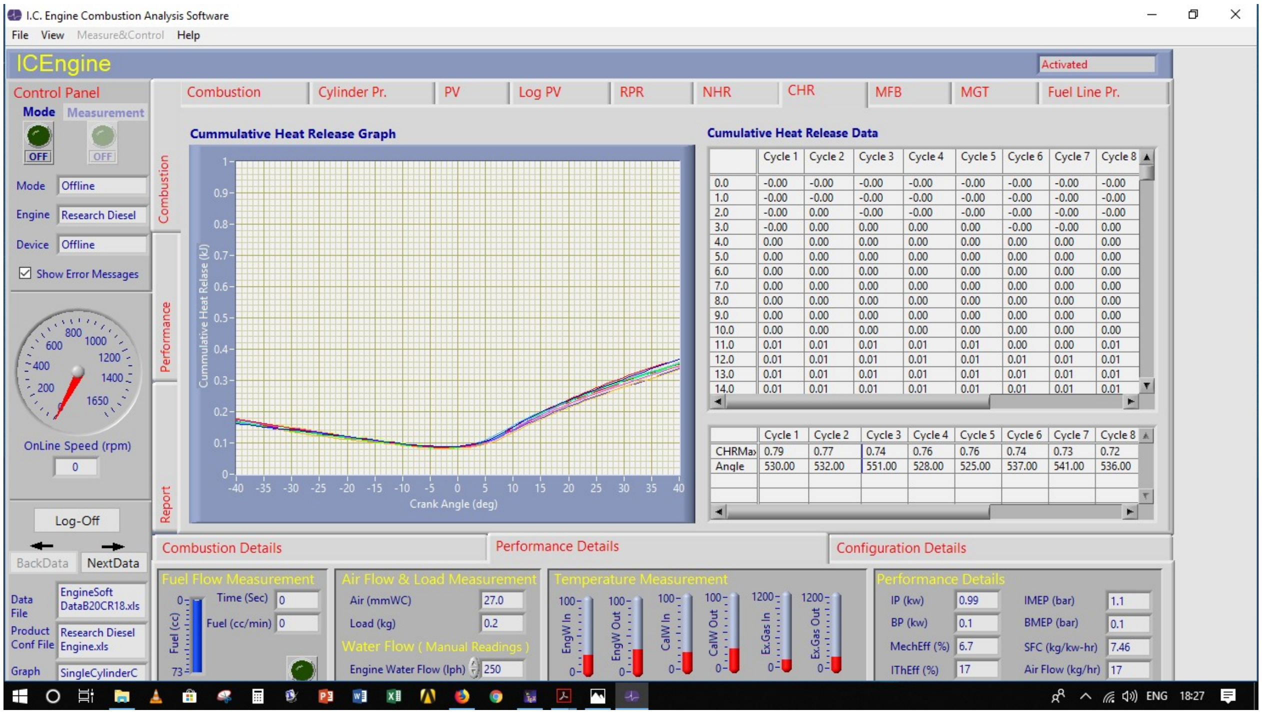

Figure 3.2: "Enginesoft" software interface for taking engine measurements

Figure 3.3: Emissions analyzer on the test bench

Chapter 4

Results and Discussions

4.1 Performance Characteristics

The performance parameters considered include brake power, brake thermal efficiency, brake mean effective pressure and specific fuel consumption. These parameters are adequate to determine the performance of the engine under the different loading conditions and fuel types. The performance analysis was done for each fuel at different compression ratios then the different fuels were compared at given compression ratios.

4.1.1 Blend B0(100% diesel)

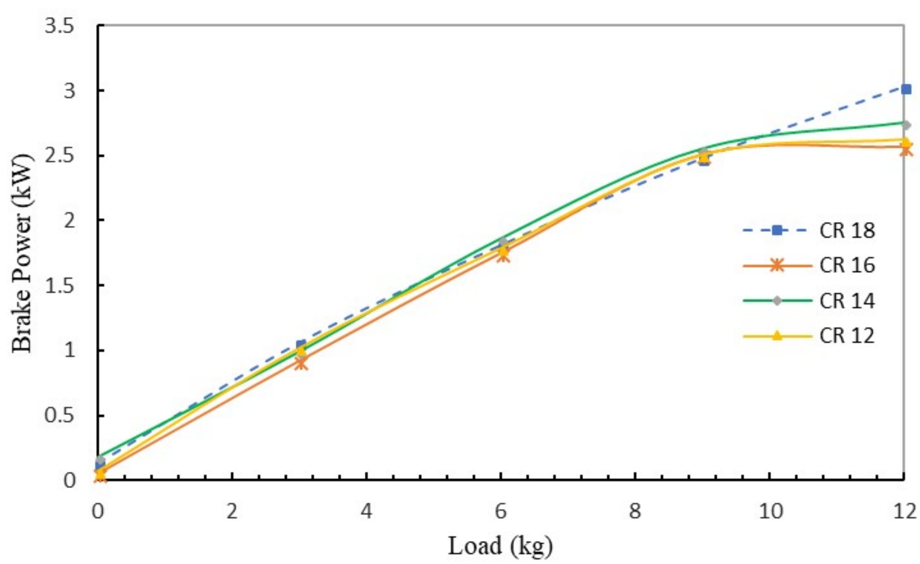

Figure 4.1: Variation of brake power with load at different compression ratios for diesel fuel

Diesel fuel is the base fuel for the research since the test engine is designed to run purely on diesel. According to Figure 4.1, the brake power of the engine increases gradually with loading for each compression ratio. For any particular load, the brake power is higher for higher compression ratio. It is evident from the graph that CR 12 has the lowest brake power while CR 18 and 16 have the highest brake power. This means that as the compression ratio is increased, the brake power also increases for all loads.

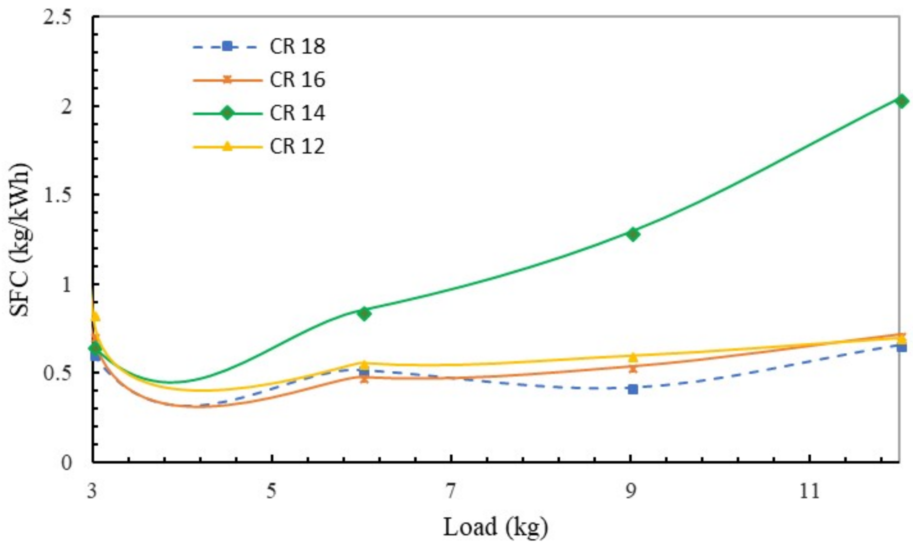

Figure 4.2: Variation of specific fuel consumption (SFC) with load at different compression ratios for diesel fuel

Figure 4.2 shows the variation of the specific fuel consumption with the engine load for pure diesel. The specific fuel consumption slightly decreases with an increase in the load for all compression ratios. For all the loads, the specific fuel consumption is lower for higher compression ratios of 18 and 16 than those of CR 14 and CR 12. A lower specific fuel consumption is desired in an engine since it means a higher fuel efficiency. Higher compression ratios ensures that almost all the fuel burns in the combustion chamber, hence the higher fuel efficiency leading to a low specific fuel consumption.

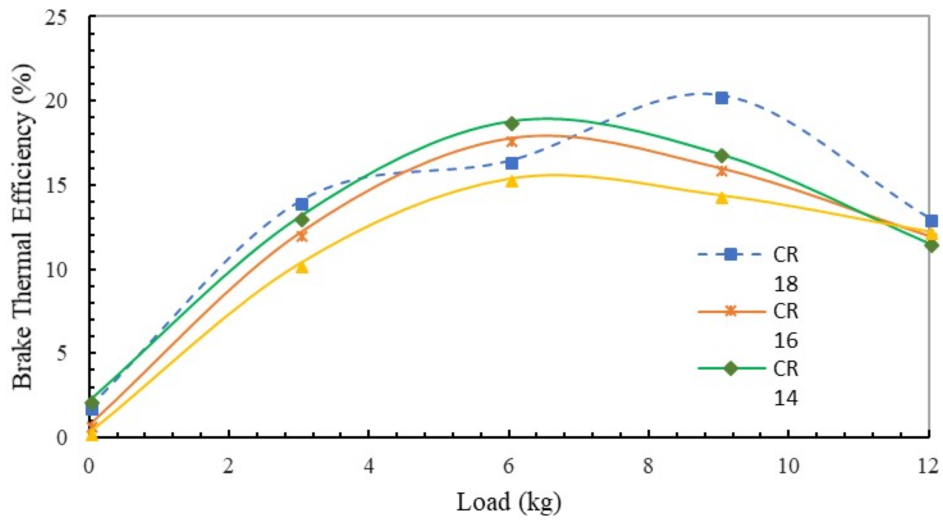

Figure 4.3: Variation of brake thermal efficiency with load at different compression ratios for diesel fuel

Figure 4.3 shows how the brake thermal efficiency varies with load for diesel fuel. The brake thermal efficiency gradually increases with loading for all compression ratios up to a load of about 7kg then it starts to decrease. For the different compression ratios, CR 18 has the highest brake thermal efficiency, while CR 12 has the lowest for all the loads. This is an indication that higher compression ratios translate to higher brake thermal efficiency. A higher compression ratio leads to more pressure in the cylinder, hence the reason for the improved thermal properties of the fuel blend.

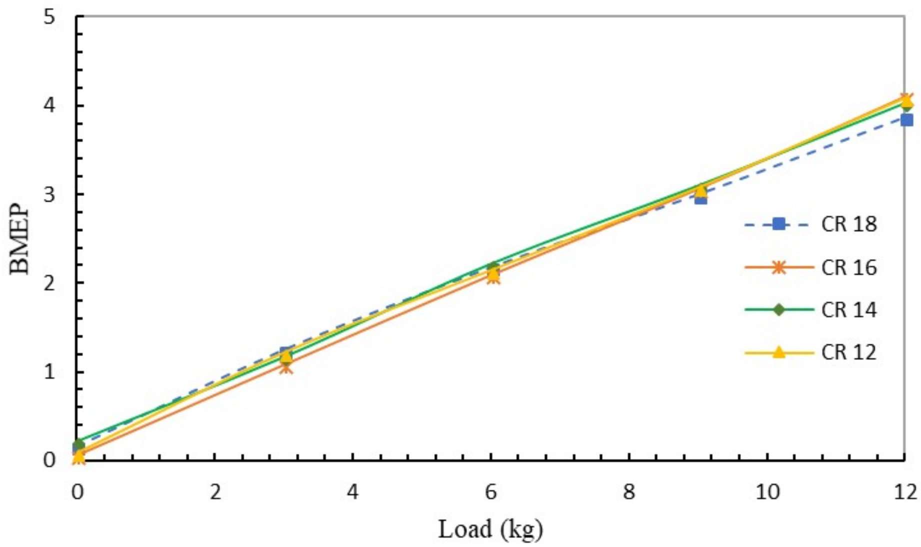

Figure 4.4: Variation of brake mean effective pressure with load at different compression ratios for diesel fuel

Figure 4.4 indicates the variation of brake mean effective pressure with load for diesel fuel. It is worth noting that the brake mean effective pressure does not indicate any considerable difference for all the compression ratio. All the compression ratios indicate a linear increase in the brake mean effective pressure as the load is increased. Since there is no much distinction, this is an indication that varying the compression ratio does not have a major impact on the average cylinder pressure of the engine.

4.1.2 B20 (20% biodiesel, 80% diesel)

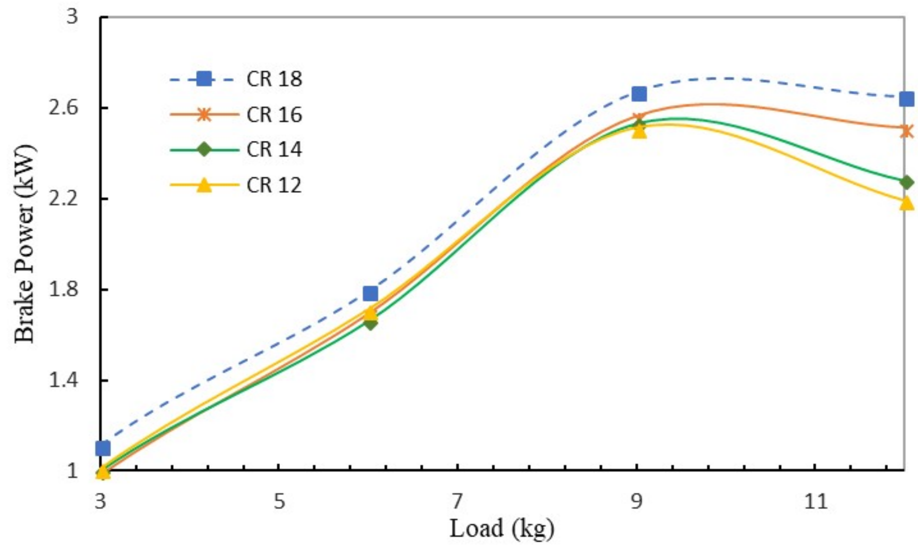

Figure 4.5: Variation of brake power with engine load for blend B20

Figure 4.5 shows the variation in brake power with loading for blend B20. The trend in brake power shows a continuous increase with loading up to the 9kg load where it begins to stagnate for all compression ratios. For all compression ratios studied, higher compression ratios of 18 and 16 have higher brake power than the lower compression ratios of 14 and 12. Higher compression ratio results into more brake power for the engine.

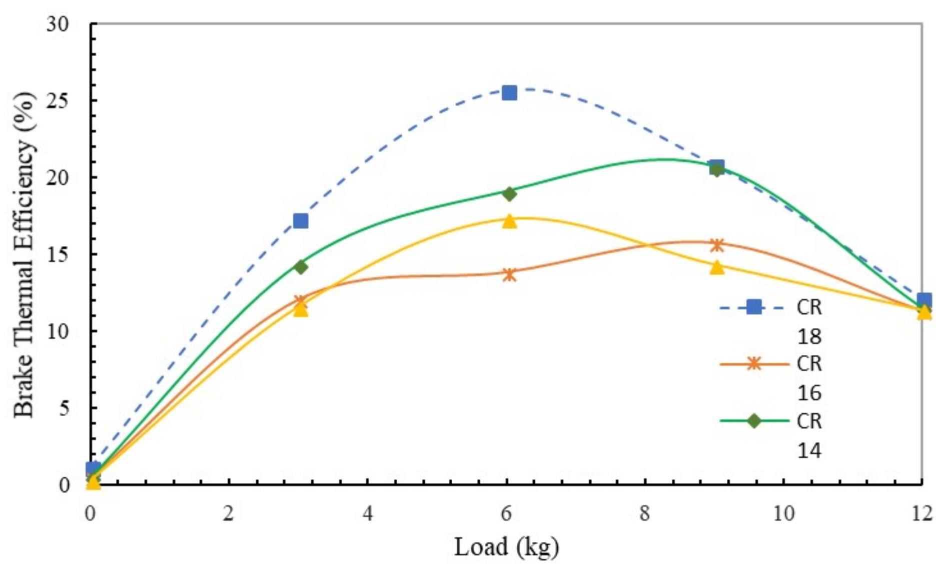

Figure 4.6: Variation of brake thermal efficiency with engine load for blend B20

Figure 4.6 shows how the brake thermal efficiency varies when the engine load is increased. the brake thermal efficiency shows an increase up to a load of 6kg then it begins to decrease gradually for all compression ratios. The graphical representation clearly indicates that CR 18 has higher brake thermal efficiency for almost all loads while CR 12 has the lowest brake thermal efficiency. The higher compression ratio improves the combustion characteristics of the fuel which in turn leads to higher brake thermal efficiency by converting more fuel energy into mechanical energy.

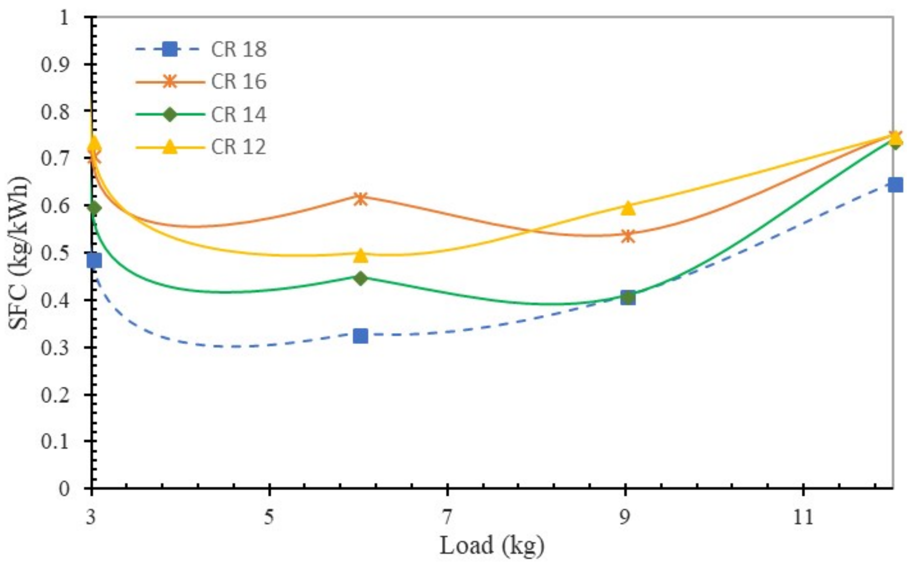

Figure 4.7: Variation of specific fuel consumption with engine load for blend B20

Figure 4.7 show the change in the specific fuel consumption of the engine when its load is varied. It is evident that the specific fuel consumption fluctuates throughout the various loads for all loads and all compression ratios. There is no particular trend for the change in specific fuel consumption when the compression ratio is varied, although it is clear that CR 18 provides the lowest SFC for all loads. A lower specific fuel consumption is an indication of higher fuel efficiency, meaning the engine is able to use less fuel to produce more work. Higher compression ratios, therefore, increase the fuel efficiency.

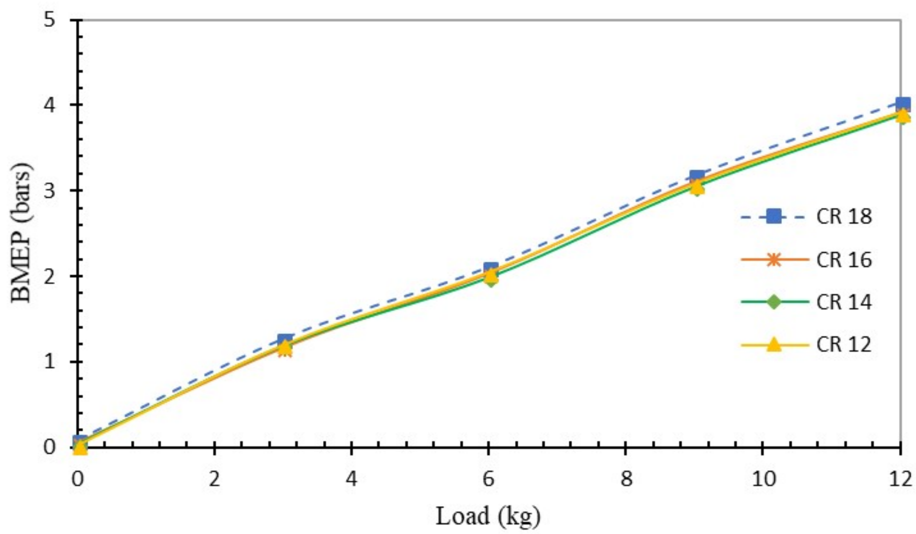

Figure 4.8: Variation of brake mean effective pressure with engine load for blend B20

Brake mean effective pressure increases gradually with increased loading for all compression ratios with fuel blend B20. Figure 4.8 shows the trend in the BMEP with change in load. There is a very slight difference between the values of the different compression ratios, although CR 18 stands out as having the higher brake mean effective pressure. This is an indication that higher compression ratio slightly increases the brake mean effective pressure for the B20 blend.

4.1.3 B80 (80% biodiesel, 20% diesel)

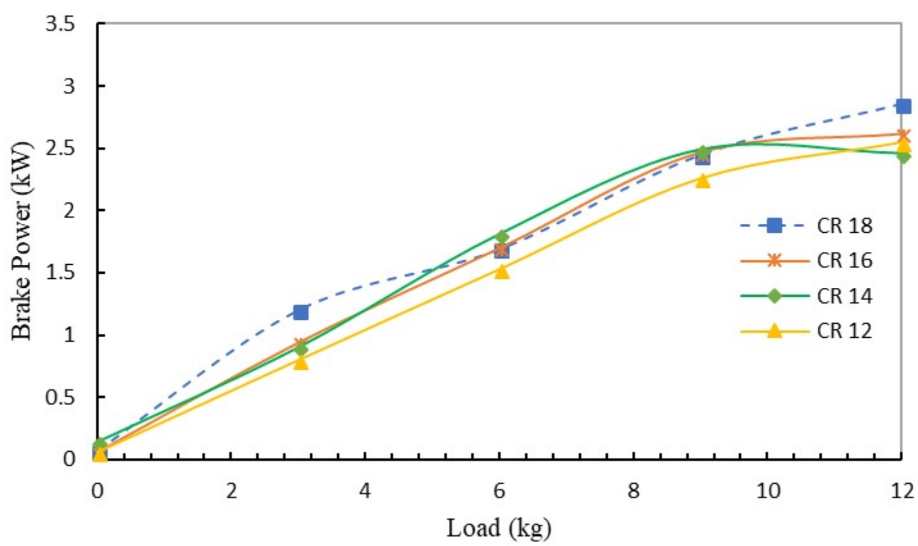

Figure 4.9: Variation of brake power with engine load for blend B80

Figure 4.9 shows the brake power against load for the fuel blend of B80. It is noted that the brake power increases gradually as the load is increased for all the compression ratios. There is minimal difference in the values of the brake power for the different compression ratios, although it is clear that CR 12 has the lowest brake power. The low brake power for CR 12 is an indicator that decreasing the compression ratio tends to lower the brake power also. Engines should be operated in conditions that provide the highest possible brake power.

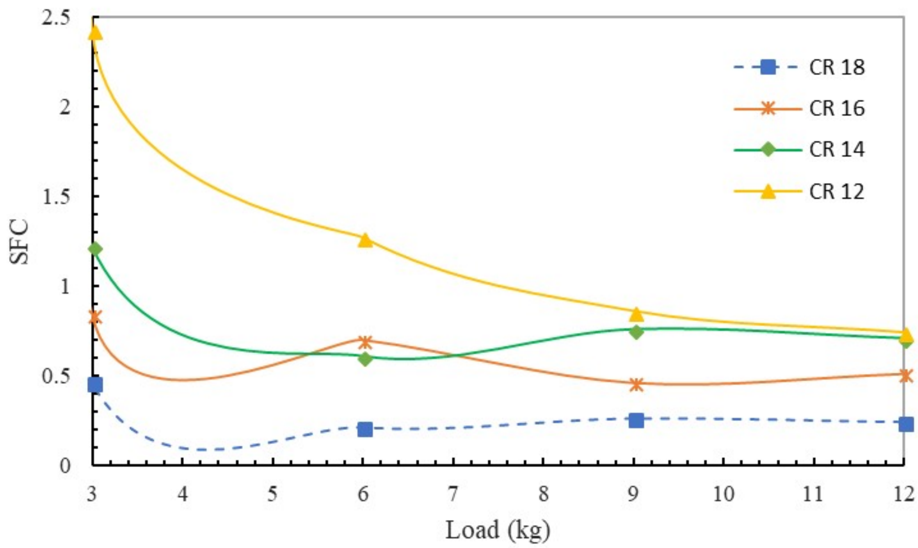

Figure 4.10: Variation of specific fuel consumption (SFC) with engine load for blend B80

Figure 4.10 shows how the specific fuel consumption changes with the load for the fuel blend B80. The specific fuel consumption has a very clear trend for this fuel blend. The values decrease up to a load of about 6kg and then remain almost constant for the rest of the loads. This is the case for all the compression ratios. In addition, it is evident that as the compression ratio is increased, the specific fuel consumption decreases. Higher compression ratio of 18 has the lowest specific fuel consumption while CR 12 has the highest. Since low specific fuel consumption values are an indication of higher engine efficiency, it means that increasing the compression ratio also increases the engine efficiency. It is, therefore, desirable to have higher compression ratios in order to attain lower values of specific fuel consumption.

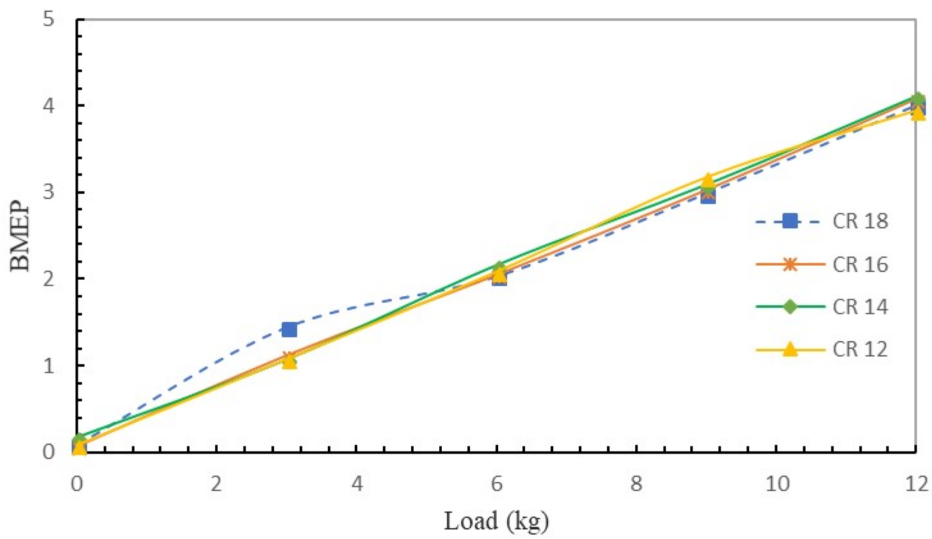

Figure 4.11: Variation of brake mean effective pressure with engine load for blend B80

Figure 4.11 shows the brake mean effective pressure versus load for fuel blend B80. The brake mean effective pressure increases in an almost linear manner as the load is increased for all the compression ratios. There is a very slight or no difference between the graphs of the different compression ratios. The trend is a good indicator that varying the compression ratio has limited impact on the brake mean effective pressure for the fuel blend B80.

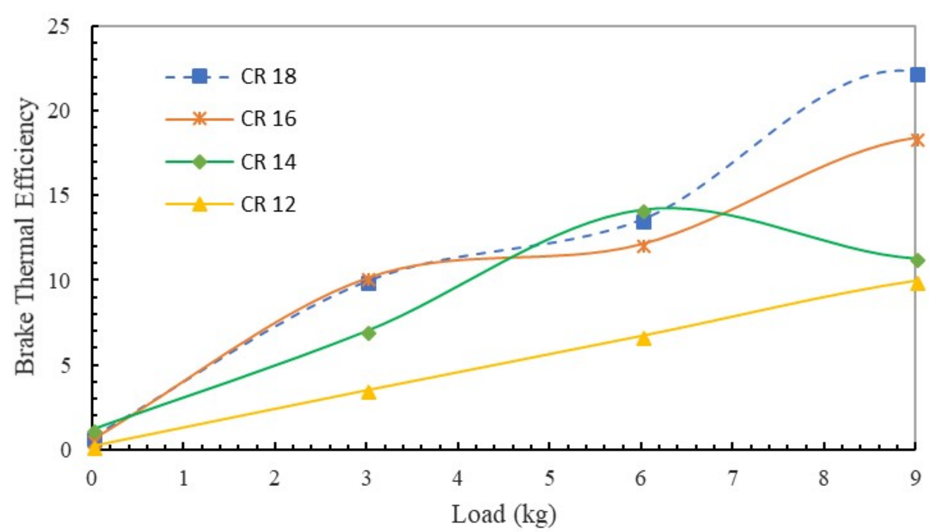

Figure 4.12: Variation of brake thermal efficiency with engine load for blend B80

The brake thermal efficiency increases with an increase in loading then begins to stagnate at higher loads for all the compression ratios. Figure 4.12 illustrates the trend of brake thermal efficiency with loading for different compression ratios using fuel blend B80. There is a clear trend on the impact of varying the compression ratio. CR 18 has the highest values of brake thermal efficiency for all the loads while CR 12 has the lowest. This indicates that as the compression ratio is increased, the brake thermal efficiency also increases. The brake thermal efficiency shows how well the engine converts heat from the fuel into useful mechanical energy. A higher brake thermal efficiency means that the engine conversion rate of the fuel heat to mechanical energy is high, which is the desirable characteristic of an efficient engine. With higher brake thermal efficiency, more heat from the fuel is converted to mechanical energy.

4.1.4 B100 (100% biodiesel)

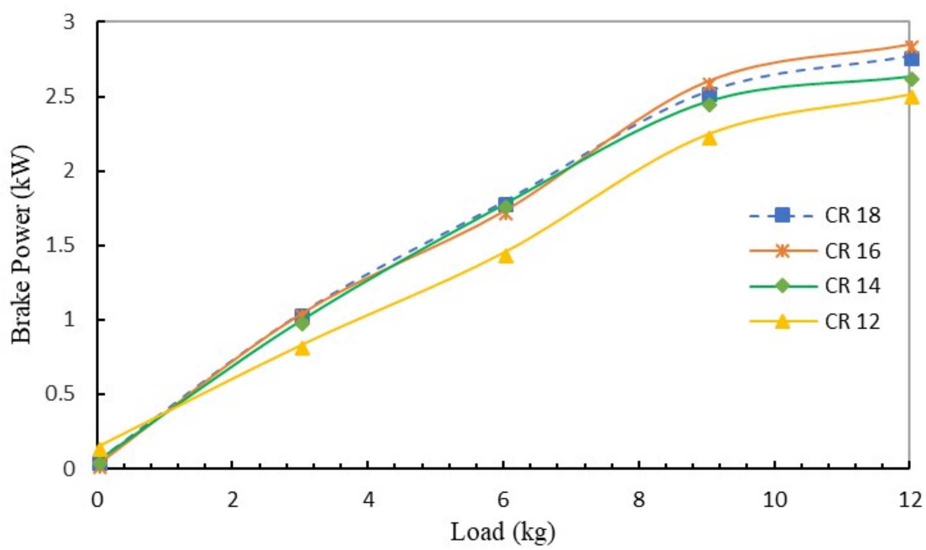

Figure 4.13: Variation of brake power with engine load for blend B100

Figure 4.13 shows the variation of brake power with load at various compression ratios for pure croton biodiesel. The figure shows that as the loading is increased, the brake power also gradually increases for all the compression ratios. Although there is minimal difference in the values of compression ratios for CR 14, 16 and 18, it can be observed that higher compression ratios slightly increase the brake power. CR 12 produces a lower brake power than the other compression ratios, indicating that lower compression ratios lead to lower brake power.

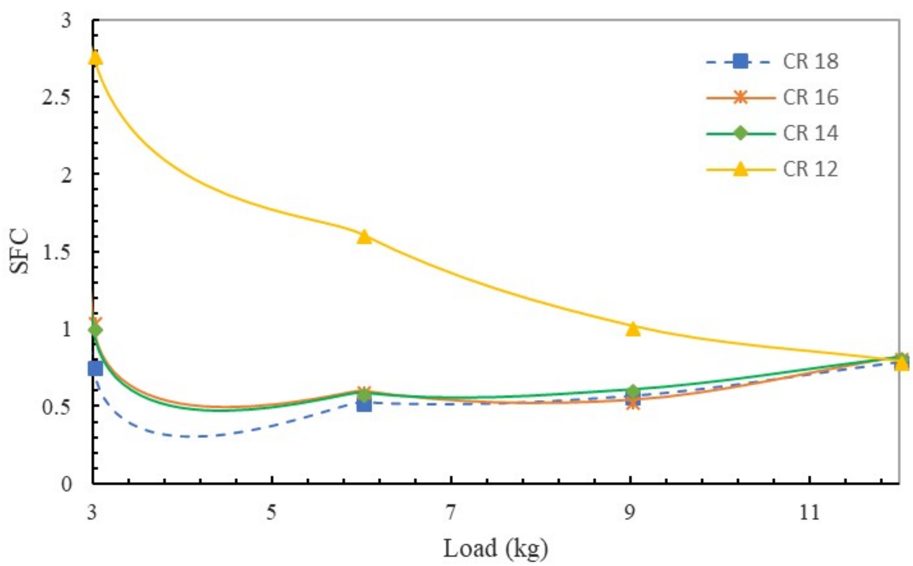

Figure 4.14: Variation of specific fuel consumption with engine load for blend B100

Figure 4.14 shows the specific fuel consumption against engine load for pure biodiesel at different compression ratios. The specific fuel consumption decreases with loading up to a load of about 4kg then slightly increases as the load is increased for all the compression ratios. The graphical representation indicates that CR 12 has the highest specific fuel consumption while CR 18 has the lowest. This means that as the compression ratio is increased, there is a decrease in the specific fuel consumption. A lower specific fuel consumption is an indication of higher fuel efficiency in the engine since the engine consumes less fuel per kWh of power produced.

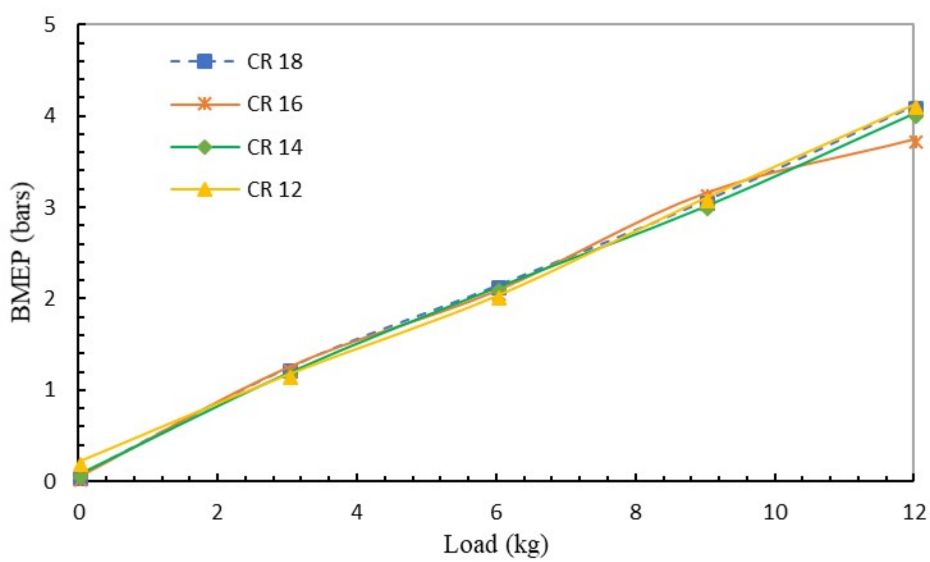

Figure 4.15: Variation of brake mean effective pressure with engine load for blend B100

Figure 4.15 illustrates how the brake mean effective pressure varies with load for pure biodiesel. the brake mean effective pressure increases linearly as the load is increased. There is no any notable difference in the values of the brake mean effective pressure when the compression ratio is varied. All compression ratios seem to almost fall on the same line.

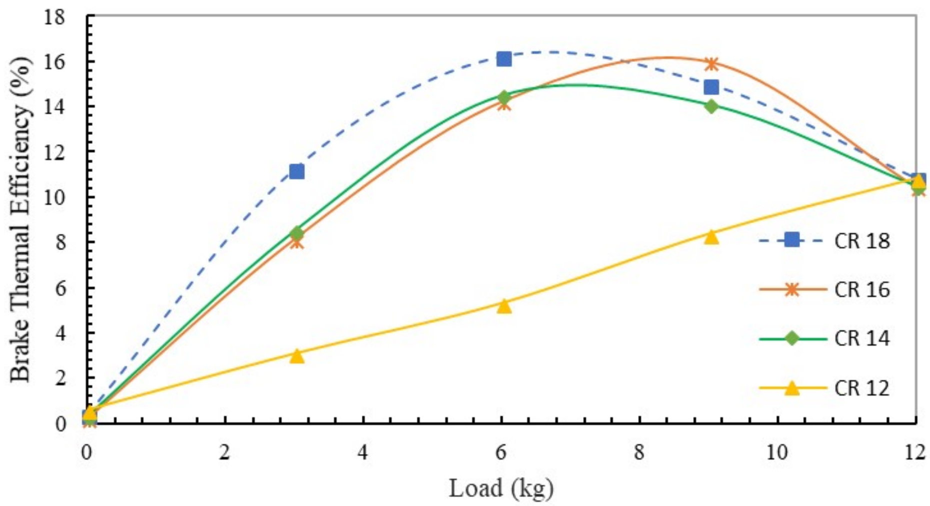

Figure 4.16: Variation of brake thermal efficiency with engine load for blend B100

Figure 4.16 is an illustration of the variation of brake thermal efficiency with load for pure croton biodiesel at various compression ratios. The brake thermal efficiency gradually increases with loading up to a load of 6kg then it begins to decrease, except for CR 12 where there is an almost linear increase with loading. From the graph, it is clear that higher compression ratios result in higher brake thermal efficiency for all loads. To attain higher brake thermal efficiencies, the highest compression ratio should be adopted (CR 18 for this case).

The analysis of pure diesel fuel, pure croton biodiesel and blends B20 and B80 all lead to similar conclusions. It is evident that higher compression ratios increase the performance of the engine. This trend is also true for blends B40 and B60. Although increasing the compression ratio has minimal effect on the brake power and the brake mean effective pressure (BMEP), it has huge effects on the specific fuel consumption and brake thermal efficiency. The specific fuel consumption and brake thermal efficiency are measures of how efficiently the energy in the fuel is being converted to mechanical energy to do useful work. Therefore, CR 18 has the optimum engine performance. The results are in agreement with, among others, studies done by Sivaramakrishnan [39], Senthil [54] and Nagaraja [40]. Sivaramakrishnan noted that optimum conditions for the operation of the diesel engine powered by Karanja biodiesel-diesel blend were at a compression ratio of 18. R. Senthil, R. Silambarasan and N. Ravichandiran also concluded that better engine performance was obtained at the highest compression ratio.

4.1.5 Comparison of Different Fuel Blends

The previous sections on comparing performance at different compression ratios gave the conclusion that higher compression ratios translate to higher performance. Since it was established that CR 18 produces the optimum performance, this section focuses on comparing the various blends at the optimum compression ratio. This comparison will enable the determination of the best fuel blends that should be used in the engine to give the best performance without any modification to the engine. The same performance parameters are considered, but in this case, a comparison is made between the blends, not the compression ratios.

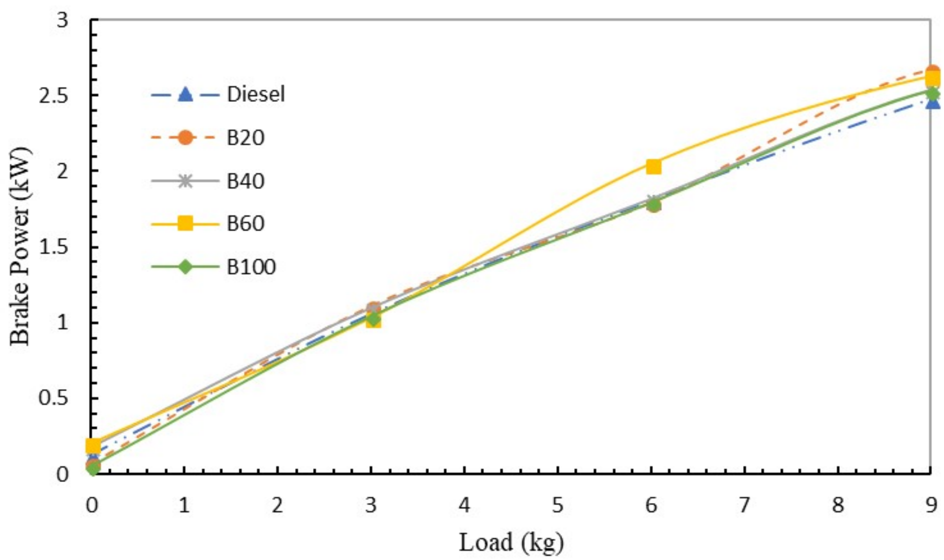

Figure 4.17: Variation of brake power with load for various fuel blends (B0 to B100) at CR 18

Figure 4.17 shows the brake power of different fuels and fuel blends at a compression ratio of 18. For all the fuels, there is an almost linear increase of brake power as the load is increased up to a load of 9kg where the brake power becomes almost constant. The values of brake power for the different fuels is almost indistinguishable since there are very slight differences. This is an indication that compared to diesel fuel, using the different fuel blends has very little or no impact on the brake power of the engine. It is possible to attain similar brake power with that of diesel when using any blend of diesel and croton seed bio-diesel.

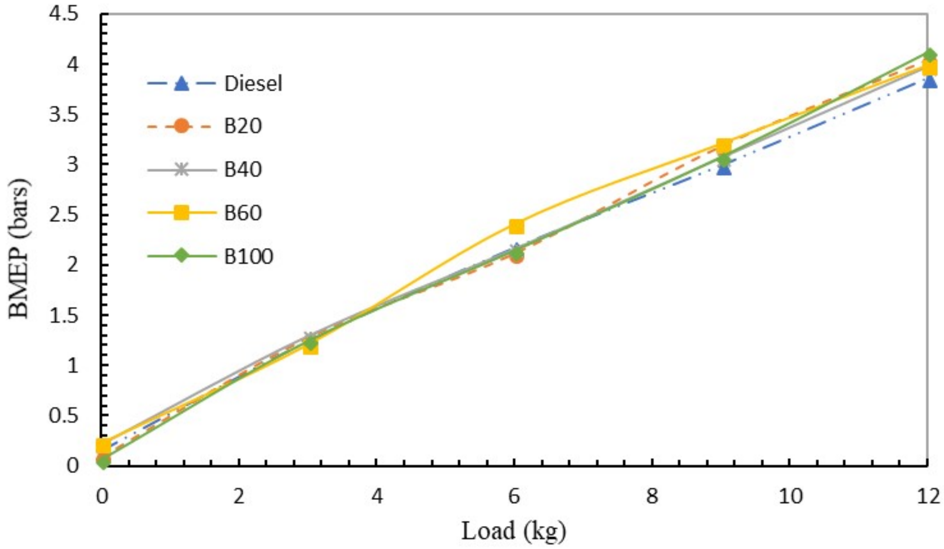

Figure 4.18: Variation of brake mean effective pressure for the various fuel blends at CR 18

Figure 4.18 shows a comparison of the brake mean effective pressure of the different fuel blends at a compression ratio of 18. The brake mean effective pressure for the different fuels also indicates an increasing trajectory with an increase in the load. As is the case with brake power, there is a very slight difference between the values of the brake mean effective pressure of the different fuels. It can, therefore, be concluded that compared with diesel fuel, using different blends of croton seed oil has very little or no impact on the change in the brake mean effective pressure.

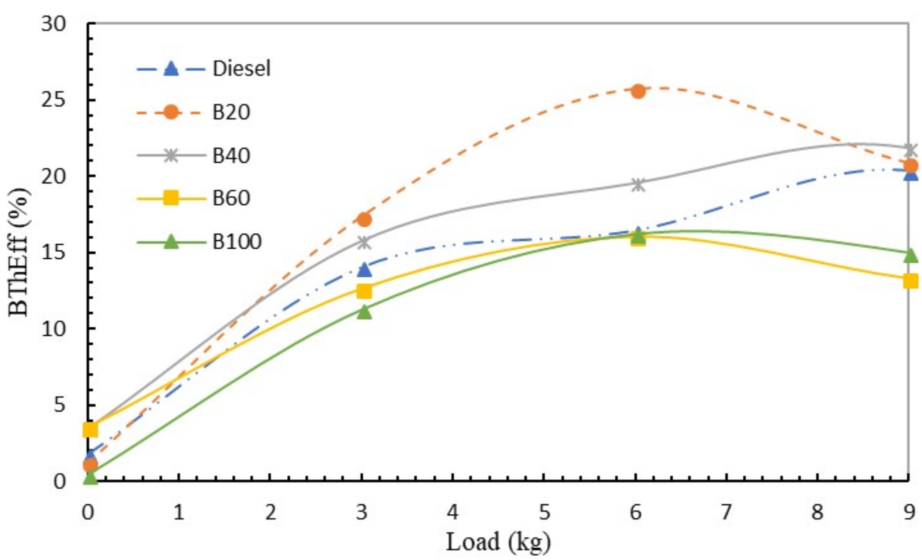

Figure 4.19: Variation of brake thermal efficiency with load for different blends at CR 18

Figure 4.19 shows the variation of brake thermal efficiency with load for the various fuel blends at the optimum compression ratio. The figure shows a gradual increase up to a load of 6kg then a slight decrease as the load is increased further. A comparison of the different fuel and fuel blends used indicates that some croton seed oil blends have better brake thermal efficiency than pure diesel fuel. Using croton seed oil alone as a fuel without blending leads to a decrease of the brake thermal efficiency. As the diesel is blended with some percentage of the croton bio-diesel, the brake thermal efficiency also increases up to the blend B20 where the brake thermal efficiency begins to decrease. As seen in the graph, B20 has the highest brake thermal efficiency for all the loads, with the highest being 25% at a load of 6kg. B40 also exhibits higher brake thermal efficiency than pure diesel fuel. As the amount of bio-diesel is increased to 60% (B60), the brake thermal efficiency becomes lower than that of diesel fuel. Pure croton bio-diesel has the lowest brake thermal efficiency for most of the loads. The lower brake thermal efficiency of croton bio-diesel can be attributed to its viscosity. The higher viscosity makes it difficult for the fuel to properly mix with air, therefore, the fuel is not completely combusted. In addition, the energy value of biodiesels is lower than that of petroleum based fuels. A lower brake thermal efficiency is an indication of lower energy conversion from chemical energy in the fuel to mechanical energy required to do work. The results are in agreement with a comparative evaluation done on both edible and non-edible feed stocks, which concluded that using biodiesels tends to decrease the brake thermal efficiency of the engine [44].

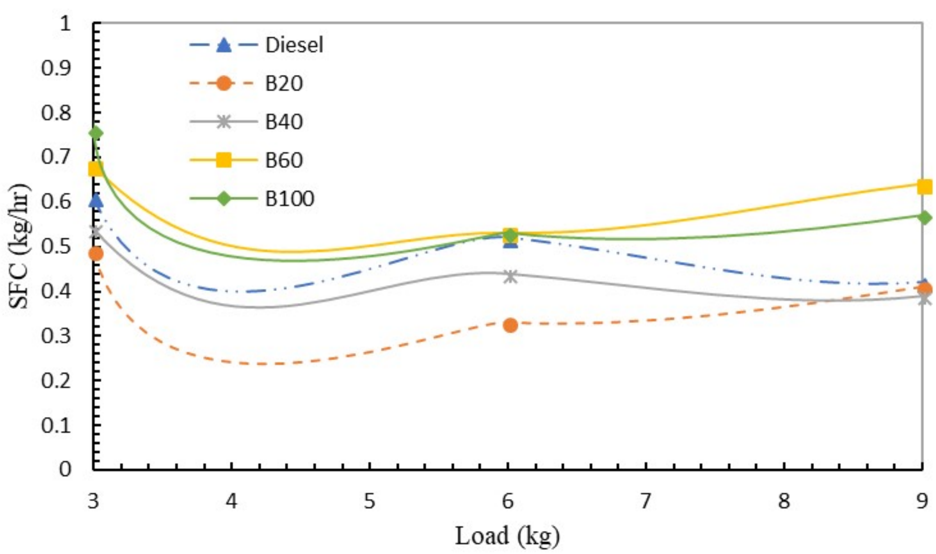

Figure 4.20: Variation of specific fuel consumption with load for different blends at CR 18

Figure 4.20 illustrates provides a comparison of the specific fuel consumption of the various fuel blends at CR 18. The specific fuel consumption appears to decrease as the load is increased up to a load of 6kg then it becomes almost constant as the load is increased further for all the fuels. Figure 4.20 clearly indicates that different fuels and fuel blends have different specific fuel consumption values. Croton seed oil fuel blend B20 gives the lowest specific fuel consumption values for all the loads. This is an indication that addition of a small percentage of croton bio-diesel to the diesel fuel lowers the specific fuel consumption. However, higher percentages of the biodiesel above 40% result into higher specific fuel consumption than that of pure diesel fuel. The increase in the specific fuel consumption with increased biodiesel percentage can be attributed to poor fuel combustion due to higher viscosity of the biodiesel, which in turn reduces the engine power. When the engine power reduces without a reduction in the amount of fuel consumed, the specific fuel consumption increases. The desired condition is to obtain the lowest values of specific fuel consumption without any change in the brake power. The lower specific fuel consumption of blend B20 means that the fuel produces the highest fuel efficiency. The results are in agreement with a research done on poppy and waste cooking oil, which indicated that blending the diesel with small percentages of biodiesel reduces the specific fuel consumption [45].

4.2 Combustion Analysis

Figure 4.21, Figure 4.22 and Figure 4.23 show the net heat release, mass fraction burnt and the mean gas temperature of the different fuel blends at half load. It is worth noting that all the tests were done at the engines default ignition timing (23 degrees bTDC) since most of these combustion parameters are affected by the ignition timing and the ignition delay.

4.2.1 Net Heat Release

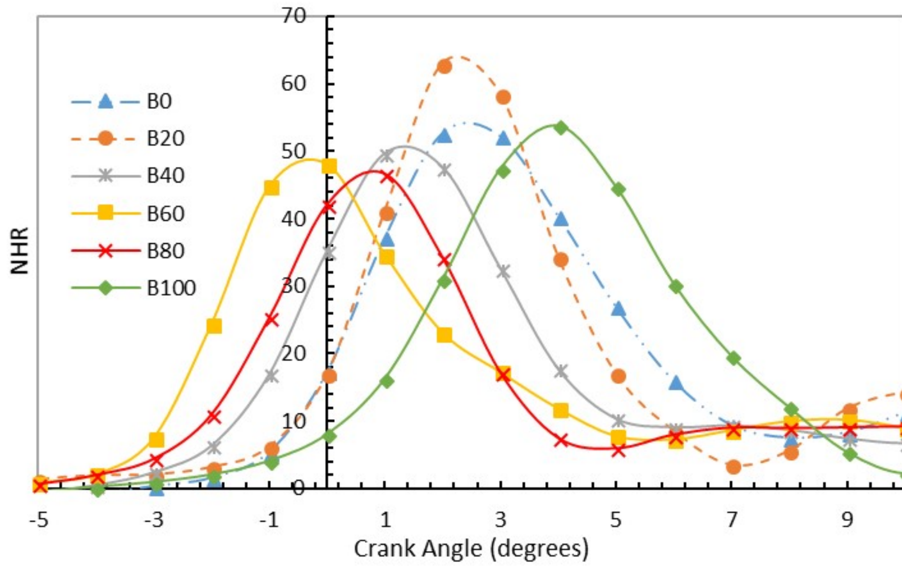

Figure 4.21: Variation of net heat release with crank angle for various blends at CR 18

Figure 4.21 represents the comparison of the net heat release for the different fuel blends. The difference in the net heat release curves for the various fuel blends is mainly caused by the difference in the ignition delay. The shorter the ignition delay, the earlier the curve starts to rise. It can be observed that B60 has the shortest ignition delay, while B100 has the longest ignition delay. The variation in the ignition delay can be attributed to the difference in the autoignition temperatures and the cetane number. Croton seed oil has a slightly lower autoignition than diesel fuel, therefore, it is expected to have a shorter ignition delay. However, due to its high viscosity, density and lower cetane number, pure croton biodiesel has a longer ignition delay than diesel fuel. Blending the diesel fuel with small volumes of the biodiesel leads to a reduction of the ignition delay up to the blend B60 beyond which the viscosity of the blend begins to rise, hence a longer ignition delay. However, a very short ignition delay is not desirable since it may cause engine knock. From the fuel blends analysed, the one with the most appropriate net heat release was B20. The blend has a considerably lower ignition delay which will not cause any engine knock. In addition, the fuel blend has the highest peak. The results are in agreement with a research done by Abhishek Paul, Rajsekhar Panua and Durbadal Debroy, where it was concluded that the biodiesels improved the heat release of the engine [46].

4.2.2 Mass Fraction Burnt (MFB)

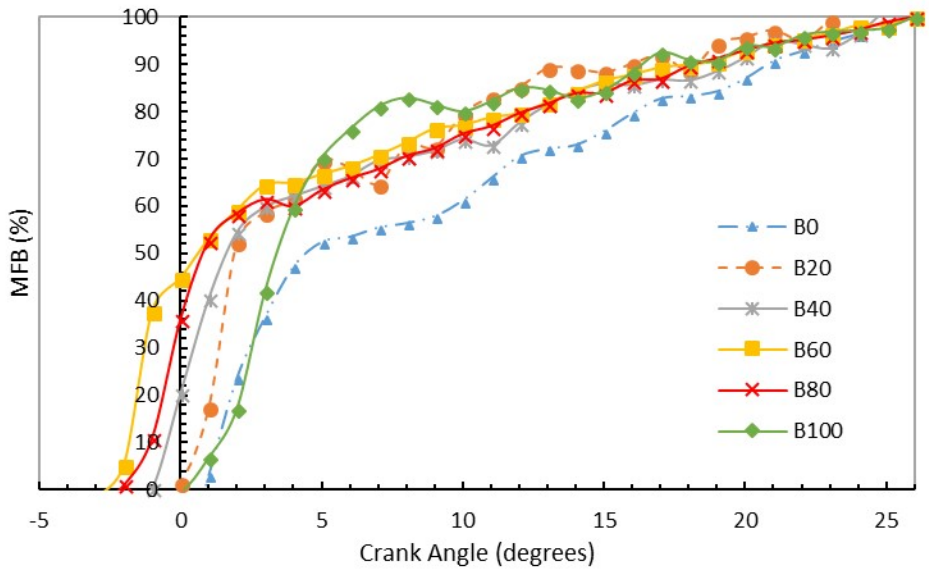

Figure 4.22: Variation of mass fraction burnt with crank angle for various blends at CR 18

Figure 4.22 shows the mass fraction burned for the various fuel blends. The mass fraction burned is mainly affected by the rate of the combustion process. The combustion process in an internal combustion engine should be fast enough to provide the required engine power. From the results, it is evident that diesel fuel burns rapidly up to about 50% of the fuel then it begins to burn more moderately as the crank angle increases. On the other hand, pure croton biodiesel burns rapidly up to about 80% of the fuel before the combustion rate decreases. Both cases are not very healthy for the engine. When 50% of the fuel does not burn in the fast combustion, there is a possibility of having unburned fuel at the end of the combustion process, which increases the emissions. In addition, when most of the fuel burns rapidly, it causes a lot of stress on the engine and it may even damage the engine components. Blending the diesel fuel with small volumes of the biodiesel increases the rate of combustion. Blend B20 gives the most appropriate curve for the mass fraction burned, with approximately 70% of the fuel burning rapidly before the slow combustion. The fuel blend also is found to burn completely by 22 degrees after TDC, leading to less residual fuel in the exhaust gases. This does not only reduce emissions but also provides more energy on the engine. The results are similar to those obtained in a research on Karanja and croton megalocarpus biodiesel-diesel blends, where the study concluded that the biodiesels increased the combustion process and reduced the ignition delay [47].

4.2.3 Mean Gas Temperature (MGT)

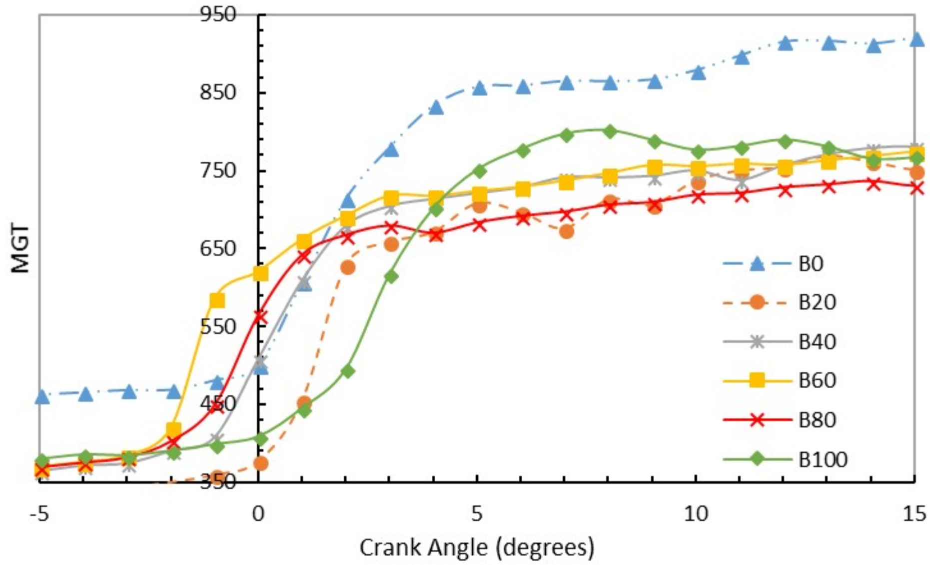

Figure 4.23: Variation of mean gas temperature with crank angle for various blends at CR 18

Figure 4.23 shows how the mean gas temperature varies with the crank angle for diesel fuel and the various fuel blends. It is clearly indicated that pure diesel has the highest gas temperature when compared to biodiesel and its blends. The mean gas temperature of a fuel is affected by the cetane number. The croton megalocarpus biodiesel has a higher cetane number than diesel due to its long long-chain hydrocarbon groups with no branching or aromatic structures. A higher cetane number has the effect of lowering the cylinder pressure rise and cylinder temperatures. Therefore, blending reduces the mean gas temperature of the engine. These results are in agreement with the explanations given by other researchers [31].

In conclusion, blending diesel with biodiesel improves most of the combustion parameters. From the results obtained, it is evident that a blend of B20 produces the best net heat release and mass fraction burnt curves at the optimum compression ratio of 18. This is a clear indication that blending the diesel with 20% biodiesel improves the combustion characteristics of the fuel. The croton biodiesel can, therefore, partially replace diesel fuel engine in a CI engine and produce better combustion.

4.3 Emission Analysis

Combustion in an internal combustion engine produces a mixture of gases, popularly known as the exhaust gases. Some of these gases are harmful to the environment and human beings. For the purpose of this study, the focus was mainly carbon monoxide (CO), which, although indirectly, has been identified to greatly contribute to global warming. CO also causes a lot of health hazards.

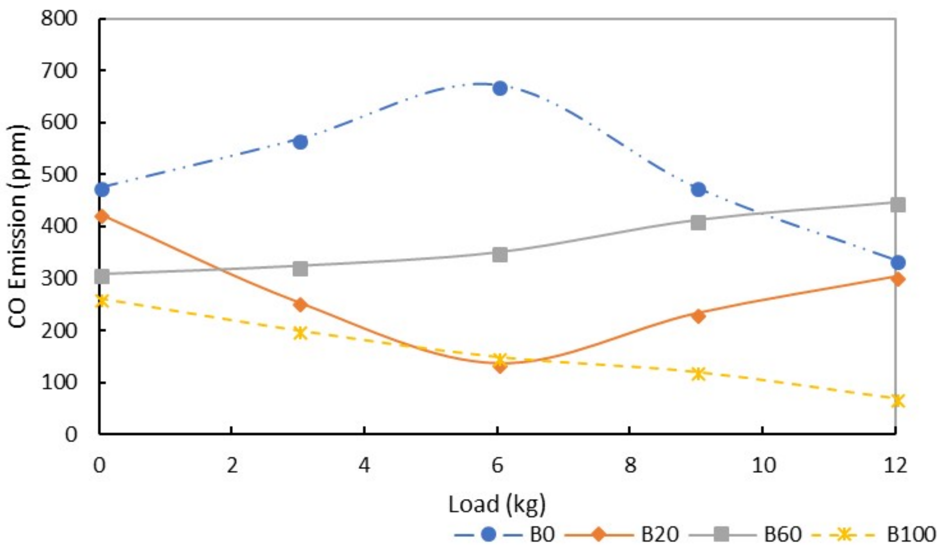

Figure 4.24: Variation of carbon monoxide emission with load for various blends at CR 18

Figure 4.24 shows the variation of CO emission with load for diesel fuels and the various diesel-biodiesel blends. Since the previous sections have already established that the optimum performance of the engine occurs at a compression ratio of 18 for all the fuel blends, the emission analysis was also done at CR 18. The general trend shows that as the load is increased, the CO emisssion also decreases. This is caused by the increase in the combustion temperature as the engine is loaded. It can be clearly deduced from the figure that the CO emission of pure diesel fuel is higher than that of the blends. Also, pure croton biodiesel produces the lowest CO emissions for all the loads. The trend in the CO emission can be attributed to the concentration of oxygen in the different fuels. Studies have shown that biodiesels have a higher concentration of oxygen than petroleum-derived diesel. The higher amounts of oxygen in the fuel minimizes the level of incomplete combustion of the fuel. Since CO is as a result of incomplete combustion, the pure biodiesel will have lower levels of CO than mineral diesel. The results are in tandem with a study by Alireza Shirneshana [41], which concluded that biodiesel reduces the CO emissions in the engine.

Chapter 5

Conclusions and Recommendations

The aim of the study was to optimize the compression ratio of a diesel engine running on croton biodiesel. Several croton biodiesel-diesel blends were prepared for testing (B0, B20, B40, B60, B80 and B100), All the blends were run in the diesel engine and the optimum compression ratio was obtained. In addition, the results from the blends were compared at the optimum compression ratio to obtain the blend with the best performance, combustion and emission characteristics. The blend with the best characteristics is the most suitable alternative to diesel fuel in the engine. The following are the conclusions from the study:

1. A higher compression ratio is desirable in a biodiesel-diesel blend fuelled engine. At higher compression ratio, the brake thermal efficiency is higher, the specific fuel consumption reduces and the emissions are lower. The study shows that CR 18 produces the optimum performance and combustion characteristics for all the fuels and fuel blends tested. A higher compression ratio improves the combustion of the fuel, hence also improving the performance characteristics.

2. Increasing the percentage of croton bio-diesel in the blends improves the engine performance. However, higher percentages of the bio-diesel have negative impacts on engine performance. As stated earlier, croton biodiesel has some undesirable characteristics which make it difficult for the fuel to be used in pure form in an engine without any modifications. However, blending some small percentages of the biodiesel with petroleum diesel was shown to produce excellent results. From the study, the fuel blend B20 (20% biodiesel and 80% diesel) was found to produce the optimum performance in the test engine at a compression ratio of 18. Since it was shown that B20 has no any negative impact on the engine and its performance, adopting the idea of blending all diesel fuels would go a long way in ensuring that the rate of depletion

of petroleum-based fuels is greatly reduced.

3. Croton megalocarpus biodiesel reduces the emissions of carbon monoxide (CO) in the exhaust gases. Pure croton biodiesel produces the lowest CO emissions in the engine. As the concentration of biodiesel in the fuel is increased, the emissions also reduce. Blending the diesel fuel with croton biodiesel can, therefore, help in reducing carbon monoxide emission to the atmosphere and minimize environmental degradation caused by exhaust gas emissions from engines.

4. Croton bio-diesel can be used in a diesel engine without any or with minimal modifications to the engine. Actually, it was observed that the performance of pure croton bio-diesel does not greatly deviate from that of pure diesel fuel. However, the bio-diesel has some undesirable characteristics that affect its combustion characteristics. Blending is, therefore, necessary in order to minimize some of these undesirable physical characteristics of the biodiesel. The croton biodiesel has a higher viscosity and density, which affects its performance in an engine. A high viscosity fuel will not easily mix with air in the combustion chamber and causes a longer ignition delay.

The following are the recommendations for future study:

1. The high viscosity of croton biodiesel is a major concern for its suitability in a diesel engine. More research should be done on ways of reducing its viscosity such as preheating the fuel or using other additives. The viscosity of the fuel affects its injection as well as combustion.

2. More efficient and economical ways of converting the croton oil into biodiesel need to be devised. Although the current method of transesterification produces a biodiesel with almost similar properties to those of diesel fuel, the chemical processes involved are complex and the process itself makes the biodiesel more expensive. If there was a way of producing the biodiesel more cheaply and efficiently, the idea of blending would gain more interest.

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Appendices

Appendix A: Performance Results

BO(Pure Diesel)

Load	CR 18	CR 16	CR 14	CR 12
Brake Power (BP)				
0	0.05	0.03	0.05	0.15
3	1.04	1.04	0.99	0.83
6	1.79	1.73	1.77	1.45
9	2.53	2.6	2.46	2.24
12	2.77	2.85	2.63	2.51
Brake Mean Effective Pressure (BMEP)				
0	0.06	0.04	0.07	0.22
3	1.24	1.24	1.18	1.17
6	2.14	2.08	2.12	2.03
9	3.07	3.15	3.01	3.1
12	4.11	3.74	4.03	4.12
Brake Thermal Efficiency (BThEff)				
0	0.4	0.21	0.34	0.61
3	11.23	8.14	8.52	3.09
6	16.2	14.21	14.5	5.31
9	14.99	15.98	14.11	8.37
12	10.83	10.45	10.51	10.82
Specific Fuel Consumption (SFC)				
0	21.34	41.76	25.36	14.04
3	0.76	1.05	1.01	2.77
6	0.53	0.6	0.59	1.61
9	0.57	0.54	0.61	1.02
12	0.79	0.82	0.82	0.79

B20 (20% Biodiesel)

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Brake Power (BP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.07	0.04	0.04	0.03
3	1.11	0.98	1	1.01
6	1.79	1.69	1.66	1.71
9	2.67	2.56	2.53	2.51
12	2.65	2.51	2.28	2.19

Brake Mean Effective Pressure (BMEP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.08	0.05	0.04	0.03
3	1.27	1.16	1.19	1.2
6	2.11	2.03	1.99	2.05
9	3.18	3.1	3.05	3.08
12	4.04	3.91	3.89	3.91

Brake Thermal Efficiency (BThEff)

Load	CR 18	CR 16	CR 14	CR 12
0	1.15	0.53	0.52	0.36
3	17.34	12.08	14.37	11.63
6	25.72	13.85	19.1	17.31
9	20.85	15.76	20.72	14.37
12	12.14	11.36	11.53	11.41

Specific Fuel Consumption (SFC)

Load	CR 18	CR 16	CR 14	CR 12
0	7.45	16.13	16.51	23.69
3	0.49	0.71	0.6	0.74
6	0.33	0.62	0.45	0.5
9	0.41	0.54	0.41	0.6
12	0.65	0.75	0.74	0.75

B40 (40% Biodiesel)

Brake Power (BP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.18	0.06	0.03	0.01
3	1.1	1.02	0.99	0.98
6	1.82	1.73	1.77	1.71
9	2.54	2.55	2.51	2.4
12	2.81	2.53	2.44	2.35

Brake Mean Effective Pressure (BMEP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.21	0.07	0.03	0.01
3	1.29	1.2	1.17	1.16
6	2.16	2.09	2.12	2.09
9	3.07	3.13	3.09	3
12	3.98	3.89	3.87	3.87

Brake Thermal Efficiency (BThEff)

Load	CR 18	CR 16	CR 14	CR 12
0	3.45	0.66	0.42	0.12
3	15.77	11.02	12.2	9.87
6	19.57	12.94	16.06	14.05
9	21.88	13.7	13.94	11.15
12	13.44	11.18	11.2	11.22

Specific Fuel Consumption (SFC)

Load	CR 18	CR 16	CR 14	CR 12
0	2.49	12.9	20.22	72.87
3	0.54	0.78	0.7	0.87
6	0.44	0.66	0.53	0.61
9	0.39	0.63	0.61	0.77
12	0.64	0.77	0.75	0.76

B60 (60% Biodiesel)

Brake Power (BP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.2	0.12	0.09	0.08
3	1.03	0.97	1.13	1.09
6	2.05	1.87	1.7	1.83
9	2.63	2.47	2.29	2.57
12	2.91	2.64	2.58	2.75

Brake Mean Effective Pressure (BMEP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.23	0.14	0.11	0.09
3	1.21	1.14	1.34	1.3
6	2.41	2.24	2.03	2.2
9	3.21	3	2.95	3.14
12	3.99	3.8	4.06	4.03

Brake Thermal Efficiency (BThEff)

Load	CR 18	CR 16	CR 14	CR 12
0	3.51	1.73	1.16	0.53
3	12.63	12.91	12.99	8.55
6	16.05	20.15	15.38	14.31
9	13.31	16.37	8.95	16.37
12	11.39	10.34	11.69	11.25

Specific Fuel Consumption (SFC)

Load	CR 18	CR 16	CR 14	CR 12
0	2.45	4.94	7.4	16.25
3	0.68	0.66	0.66	1
6	0.53	0.43	0.56	0.6
9	0.64	0.52	0.96	0.52
12	0.75	0.83	0.73	0.76

B80 (80% Biodiesel)

Brake Power (BP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.07	0.06	0.14	0.06
3	1.2	0.94	0.9	0.8
6	1.69	1.7	1.81	1.53
9	2.45	2.47	2.49	2.26
12	2.86	2.62	2.46	2.55

Brake Mean Effective Pressure (BMEP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.08	0.07	0.17	0.08
3	1.44	1.12	1.07	1.07
6	2.03	2.05	2.16	2.08
9	2.99	3.03	3.09	3.17
12	4.01	4.08	4.11	3.94

Brake Thermal Efficiency (BThEff)

Load	CR 18	CR 16	CR 14	CR 12
0	0.7	0.61	1.18	0.24
3	9.92	10.15	7	3.53
6	13.56	12.16	14.16	6.73
9	22.31	18.46	11.28	9.99
12	7.79	16.7	12.09	11.55

Specific Fuel Consumption (SFC)

Load	CR 18	CR 16	CR 14	CR 12
0	5.92	14.11	7.24	35.22
3	0.46	0.84	1.22	2.43
6	0.21	0.7	0.61	1.27
9	0.26	0.46	0.76	0.86
12	0.24	0.51	0.71	0.74

B100 (100% Biodiesel)

Brake Power (BP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.05	0.03	0.05	0.15
3	1.04	1.04	0.99	0.83
6	1.79	1.73	1.77	1.45
9	2.53	2.6	2.46	2.24
12	2.77	2.85	2.63	2.51

Brake Mean Effective Pressure (BMEP)

Load	CR 18	CR 16	CR 14	CR 12
0	0.06	0.04	0.07	0.22
3	1.24	1.24	1.18	1.17
6	2.14	2.08	2.12	2.03
9	3.07	3.15	3.01	3.1
12	4.11	3.74	4.03	4.12

Brake Thermal Efficiency (BThEff)

Load	CR 18	CR 16	CR 14	CR 12
0	0.4	0.21	0.34	0.61
3	11.23	8.14	8.52	3.09
6	16.2	14.21	14.5	5.31
9	14.99	15.98	14.11	8.37
12	10.83	10.45	10.51	10.82

Specific Fuel Consumption (SFC)

Load	CR 18	CR 16	CR 14	CR 12
0	21.34	41.76	25.36	14.04
3	0.76	1.05	1.01	2.77
6	0.53	0.6	0.59	1.61
9	0.57	0.54	0.61	1.02
12	0.79	0.82	0.82	0.79

Appendix B: Combustion Results

Net Heat Release

Crank Angle	B0	B20	B40	B60	B80	B100
-10	-6.57999	-1.46893	-2.02581	-2.10721	-2.07765	-2.27776
-9	-5.71203	-1.15955	-2.43161	-1.88589	-2.02919	-1.8364
-8	-4.66029	-1.16763	-2.25749	-1.47127	-1.61576	-1.35051
-7	-3.74102	-1.26175	-1.40545	-0.86568	-0.90746	-1.07532
-6	-2.92438	-0.31231	-0.74539	-0.0147	-0.23358	-0.90352
-5	-1.98187	1.334011	-0.41904	0.9944	0.713208	-0.47392
-4	-0.77314	1.978992	0.605682	2.191265	2.045624	0.249913
-3	0.409352	2.114423	2.500158	7.61775	4.441742	0.950412
-2	1.670393	3.202528	6.443455	24.51713	11.00902	2.075951
-1	5.485539	6.238721	17.03413	44.91689	25.36832	4.108571
0	17.35084	16.9933	35.30848	48.17106	42.10248	8.061817
1	37.29956	41.19534	49.70039	34.79193	46.65376	16.31654
2	52.7728	63.03818	47.6574	23.19314	34.28206	31.08529
3	52.32687	58.43057	32.66955	17.34252	17.23029	47.48297
4	40.46906	34.36252	17.80715	11.95314	7.544894	53.86916
5	27.03101	17.04714	10.44389	7.867843	5.994917	44.83682
6	16.02357	8.586483	9.167673	7.372674	8.030311	30.42429
7	9.480143	3.506203	9.427965	8.72173	9.073482	19.68258
8	7.614659	5.61862	8.782145	10.0783	9.001809	12.03215
9	8.418051	11.84965	7.513774	10.29108	9.119721	5.401937
10	11.37009	14.26951	6.7894	9.096187	9.250125	2.207699
11	15.13086	15.0302	7.196025	7.527761	9.437731	4.161968
12	16.39368	13.34747	9.412906	7.39647	9.421705	7.158303
13	13.89075	10.29105	13.10802	8.379244	9.388743	6.24092
14	10.6748	6.889665	13.38012	9.697397	8.623767	3.538321
15	10.76029	3.622836	10.22528	10.00579	7.610371	4.750959
16	13.17333	3.166705	6.814299	9.232555	6.98236	9.640672
17	13.53701	3.839861	4.989722	7.899974	7.509975	11.69507
18	11.00974	3.930436	5.334953	6.920031	7.973327	8.659498
19	8.948985	5.438904	7.098279	6.783482	8.584277	5.076077
20	10.1796	8.572012	9.921648	7.62419	8.338622	5.581516

Mass Fraction Burned

Crank Angle	B0	B20	B40	B60	B80	B100
-10	0.993368	-5.52085	-5.25725	-3.68499	-5.21743	-4.57977
-9	0.776914	-5.32726	-6.56437	-4.44132	-5.76825	-5.10874
-8	0.842247	-5.77921	-7.03982	-4.74518	-6.23043	-5.00994
-7	1.340309	-6.83233	-7.14089	-4.75866	-6.02898	-5.54649
-6	1.28655	-6.89997	-6.97115	-4.66689	-6.3585	-5.59043
-5	1.177553	-5.51707	-7.83683	-4.12236	-5.7093	-6.06332
-4	1.96861	-5.54313	-6.79667	-3.05346	-4.65976	-5.1856
-3	1.542069	-5.26916	-6.32361	-1.55711	-3.36923	-5.55521
-2	2.599384	-3.91352	-3.26592	5.359923	1.031703	-4.44679
-1	1.870379	-2.02085	0.239816	37.89039	11.01841	-2.76897
0	2.866263	1.556033	20.31132	44.76649	36.05417	-0.70583
1	4.458458	17.3941	40.47201	53.09784	52.77437	6.799593
2	5.090936	52.46247	54.53518	59.19072	58.44518	17.19969
3	6.517919	58.75329	59.88143	64.50101	61.4348	42.16691
4	7.533252	61.75877	62.16954	64.97571	59.95871	59.63517
5	9.476525	69.68662	64.20659	66.73948	63.38976	70.20383
6	10.56566	68.07532	66.39184	68.44614	65.86384	76.24563
7	13.44684	64.71488	69.70727	70.74959	67.79443	81.03581
8	21.73893	72.92587	70.4897	73.43914	70.59089	82.90656
9	41.4802	72.45019	71.80754	76.39564	72.25313	81.47457
10	66.58767	79.51177	74.18745	77.12956	75.29041	80.03813
11	74.33411	83.14497	72.91544	78.76965	76.91386	82.15375
12	76.63609	85.19143	77.69634	79.63108	79.64786	84.98453
13	74.0579	89.17081	81.58768	81.96889	81.62417	84.57489
14	70.64117	89.04168	84.45779	84.17592	83.86236	82.82135
15	69.34873	88.31292	85.99588	86.51929	83.94442	84.47548
16	71.45755	90.09578	85.63202	87.78702	86.55479	88.32879
17	78.86424	91.94648	86.80554	89.29302	86.9013	92.48226
18	86.74236	88.77424	86.67752	89.88566	89.67277	90.9594
19	87.39988	94.37323	88.58007	90.72727	91.24145	90.73625
20	83.7367	95.7209	91.57178	93.07065	92.96494	93.879

Mean Gas Temperature

Crank Angle	B0	B20	B40	B60	B80	B100
-10	292.173	331.9425	368.7858	363.3487	363.3487	378.2278
-9	292.0934	334.8923	364.6302	361.7993	363.0707	377.9976
-8	294.0254	334.5471	364.3004	362.2731	362.974	380.5463
-7	299.1428	331.0716	365.5746	363.9671	365.6521	379.82
-6	299.8387	332.1307	367.9437	365.9519	365.6868	381.2004
-5	299.9177	340.0243	364.9458	369.9595	369.9203	380.2452
-4	306.8017	340.8301	371.1049	376.3549	375.7342	385.5392
-3	304.0096	342.8472	374.1922	384.6544	382.3964	384.5068
-2	312.6038	349.9213	389.8949	420.1456	403.1002	390.3673
-1	306.9934	359.3951	407.5973	585.5898	449.3523	398.7249
0	314.7221	376.9481	508.2164	620.4029	565.0885	408.6831
1	326.955	455.1059	609.162	662.262	642.0752	445.1183
2	331.3894	628.8045	679.019	692.2055	667.2292	495.563
3	341.9099	658.5	704.2576	717.6318	679.3753	617.7954
4	348.9359	671.2675	713.4438	717.5898	670.0976	702.3744
5	363.1423	708.2865	720.8502	723.7215	683.4286	752.0607
6	370.2874	696.7445	728.532	729.0749	691.7828	778.5825
7	391.5721	675.9731	741.5425	737.075	697.1272	798.3033
8	456.0453	713.4448	741.0835	746.6466	706.1617	802.8669
9	612.1972	706.3458	742.9985	757.1889	709.4122	790.3946
10	810.6968	737.2476	750.0326	755.6916	718.8531	777.5864
11	867.7711	750.2225	737.8464	758.6351	721.2053	782.4395
12	880.2093	754.7491	756.7743	757.1866	728.5712	790.5841
13	852.7792	768.8651	770.7201	763.2625	732.0286	782.0171
14	818.4817	761.6612	779.0491	768.3881	736.5045	766.4772
15	801.449	751.2561	780.2074	774.0051	730.4416	768.1477
16	811.9903	753.6397	771.3098	773.7426	736.3921	780.8088
17	865.3663	756.2223	770.3498	774.6078	731.3101	794.7561
18	922.03	732.9222	762.5183	770.5336	737.8074	779.624
19	919.3666	754.746	765.2462	767.7285	738.3934	771.1506
20	881.4613	754.4948	773.5607	772.827	739.6586	779.8135

ANNEX

ANNEX I: BUDGET

MATERIAL	DESCRIPTION	QUANTITY	UNIT COST (Ksh)	TOTAL COST (Ksh)
Diesel fuel	Automotive diesel	150	106.00	15,900.00
Bio-diesel	Croton biodiesel	150	310.00	46,500.00
Return air ticket + other transport costs	Air ticket + transport to and from airport	1	86,027.00	86,027.00
Engine maintenance	Nozzle	2	22,000.00	44,000.00
	Lubrication oil	6	1,800.00	10,800.00
	Oil filters	2	2,000.00	4,000.00
	Scheduled maintenance	1	25,000.00	25,000.00
	One way valve	3	1,500.00	4,500.00
Hiring of emissions analyser		5	10,000.00	50,000.00
Printing		361	10.00	3,610.00
		178	10.00	1,780.00
TOTAL EXPENDITURE				**292,117.00**
Amount awarded	**$3,000.00**			**283,200.00**
Conversion rate	94.4			