AN INVESTIGATION ON THE PERFORMANCE OF A DIRECT INJECTION DIESEL ENGINE USING ESTERIFIED OILS (BIODIESELS) AS FUEL

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DEDICATED TO

MY PARENTS
MY FAMILY
AND
WELL WISHERS

CERTIFICATE

This is to certify that the thesis entitled **"An Investigation on the Performance of a Direct Injection Diesel Engine using Esterified Oils (Biodiesels) as Fuel"** being submitted by **Mr. H. K. Amarnath** to **THE MAHARAJA SAYAJIRAO UNIVERSITY OF BARODA,** Vadodara in the Faculty of Technology and Engineering for the award of the degree of "Doctor of Philosophy" in Mechanical Engineering, is a record of bonafide research work carried out by him. **Mr. H. K. Amarnath** has worked under my supervision and guidance and has fulfilled all the requirements for the submissions of this thesis, which according to our knowledge has reached requisite standard. The results contained in this thesis have not been submitted, in part of full, to any other degree or diploma at any institute or university.

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Abstract

The aim of the present study is to analyze the performance, emission and combustion characteristics of a variable compression ratio (VCR), compression ignition (CI) engine using a suitable biodiesel as fuel. The biodiesel selected to conduct this experimental investigation is Karanja biodiesel. The selection of Karanja biodiesel is based on an extensive review of literature which indicated that this is relatively unexplored as fuel on a VCR diesel engine. The experimental analysis is followed by a computational study consisting of optimization and modeling for predicting the optimum performance parameters and emission constituents of the engine.

The experimental study is conducted on a four stroke, VCR diesel engine using Karanja biodiesel and its blends with diesel. The thermal performance and emissions characteristics are evaluated by operating the engine at different preset compression ratios (CRs) of 14, 15, 16, 17, 17.5 and 18, different injection pressures (IPs) of 150 bar, 200 bar and 250 bar and varying loads from 0kg to 12 kg in steps of 3kg.The thermal performance parameters evaluated are brake thermal efficiency (BTHE), brake specific fuel consumption (BSFC), brake mean effective pressure (BMEP), volumetric efficiency, heat equivalent of brake power (HBP), heat with exhaust gas (HGas) and exhaust gas temperature (EGT). The emission constituents measured are carbon monoxide (CO), unburnt hydrocarbons (HC), oxides of nitrogen (NO_x), carbon dioxide (CO_2), oxygen (O_2) and SO_x (Oxides of sulphur). Along with emission constituents, the smoke intensity is also measured in terms of Hartridge Smoke Units (HSU). A combustion analysis of the engine running on pure Diesel oil and Karanja biodiesel at full load, IP of 200bar and at different preset CRs of 14, 16 and 18 is also conducted. The combustion parameters analysed are cylinder pressure, net heat release, rate of pressure rise, mass fraction burnt, mean gas temperature, cylinder pressure-volume (PV) plot, cumulative heat release rate and injection pressure.

The experimental study conducted provides large number of results. But, it is impossible to select the input parameters such as CR, IP and blend for obtaining optimum thermal performance and emissions from the engine. Hence, a suitable computational study is required to be carried out in order to meet the objective of finding the optimum combination of the input parameters under different preset priorities.

The problem of finding the optimum working conditions from the point of view of maximising BTHE, minimising BSFC, EGT and proportion of exhaust gas emission constituents poses a multi-objective scenario. Thus, multi-objective optimization of thermal performance and emission characteristics using GA technique for the engine is carried out in order to find optimum CR, IP and blend at full load. In this study MATLAB genetic algorithm (GA) toolbox is selected as the implementation tool for optimization. Further, artificial neural network (ANN) is used to obtain the output parameters using the optimised input parameters. ANN modeling consists of very complex techniques capable of modeling different functions and processes. Various softwares can be used to model ANN. In this study, EASYNN software is selected for the purpose.

From the present study, it can be inferred that blend B20 operates closest to Diesel oil with respect to thermal performance. However, there is not much deviation between the performance of Karanja biodiesel and Diesel oil at higher CRs. Hence higher CRs particularly CR of 18 should be the mode of operation when engine is fuelled with Karanja biodiesel. Also, higher IP of 250bar is preferable for Karanja biodiesel due to its relatively higher viscosity. The emission characteristics evaluation indicates that the blend B100 gives minimum harmful emissions amongst all the blends. Further, at a higher CR of 18 and IP of 250bar fairly reduced exhaust emissions are observed irrespective of blend. Therefore, operating the diesel engine with Karanja biodiesel at a CR of 18 and IP of 250bar results in minimum emissions. It is inferred from the combustion analysis that as the CR increases from 14 to 18 at full load and IP of 200bar, the difference in performance between Karanja biodiesel and diesel reduces i.e. the operation of the engine fuelled with Karanja biodiesel tends towards that fuelled with diesel. The combustion characteristics of diesel engine using Karanja biodiesel is similar to that using pure diesel at higher CR of 18 which is very promising as far as Karanja biodiesel as an alternative fuel in diesel engines is concerned. In short, based on thermal performance, emissions constituents and combustion analysis, it is observed that the optimum CR is 18 which is also suggested by multi-objective optimization. However, the optimum IP and blend are 228bar and B70 respectively. The outputs corresponding to the combination of the optimised CR, IP and blend are not readily available through the hardware experiments conducted. Therefore, an ANN model simulating the actual engine behaviour is built to obtain the outputs corresponding to the optimized CR, IP and blend.

Nomenclature

Symbols

A	Area
D	Engine cylinder diameter
L	Engine cylinder stroke
m_a	Air mass flow rate
m_f	Diesel fuel mass flow rate
N	Speed

Greek

ρ	Density
η	Efficiency

Abbreviations

ANN	Artificial Neural Network
BTHE	Brake Thermal Efficiency
BSFC	Brake Specific Fuel Consumption
BMEP	Brake Mean Effective Pressure
GA	Genetic Algorithm
HBP	Heat Equivalent of Brake Power
HGas	Heat Equivalent of Exhaust Gas
EGT	Exhaust Gas Temperature

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Chapter 1

Introduction

Energy is an essential and vital input for economic activity. It is also the lifeline of modern societies. Building a strong base of energy resources is a pre requisite for sustainable economic and social development of a country. Environmental concerns are also a need of the present times. With increasing trend of motorization & industrialization, the world's energy demand is growing at a faster rate.

World's energy consumption has only increased continuously since decades except for a brief period like the oil crisis in 1970's in which the growth slowed down. Energy consumption has increased by more than 5% in 2010, after a slight decrease in 2009. This strong increase is the result of two converging trends. On one hand, industrialized countries, which experienced sharp decreases in energy demand in 2009, recovered firmly in 2010. Oil, gas, coal, and electricity markets followed the same trend. On the other hand, China and India, which showed no signs of slowing down in 2009, continued their intense demand for all forms of energy.

Energy consumption is not expected to decrease in this century, because the world population is increasing and the economies of developing countries are expanding rapidly. In contrast, the source and supply of primary energy sources like coal, oil and natural gas seem to decrease to a critical point. Although the exact date is debatable, most investigators agree that production peak of oil and natural gas is near. The point of maximum production of oil is called Hubbert Peak. At that point, half of all the recoverable oil that ever existed on our planet has been used. There is still oil left but it is much more difficult and expensive to recover it. Reaching the Hubbert Peak means that production will decrease in future and demand will continue to increase.

In 2008, the energy supply by fossil fuels was nearly 81% of the total world's energy demand. This constitutes 33.5% by oil, 26.8% by coal and 20.8% by gas. The renewable energy sources like hydropower, solar power, wind power, geothermal power and biofuels contributed to about 13% of the world's energy supply and nuclear power contributed to 5.8%. The facts show that oil is the most popular energy fuel. Since their exploration, the petroleum fuels continued as a major conventional energy source. On the other hand, they are limited in reserves and highly concentrated in certain parts of the globe. Those countries not having these resources are facing energy / foreign exchange crisis due to heavy import bill on crude petroleum. Increased extraction and consumption of fossil fuels have led to a fast depletion in the underground based petroleum derived fuels. These factors have contributed to a sharp increase in petroleum prices.

Also, the petroleum fuels are currently the dominant global source of CO_2 emissions, green house gases & global warming. Their combustion is posing a stronger threat to clean environment. The global warming emissions are the most serious environmental problem. Therefore many nations have signed the UN agreement to prevent a dangerous imbalance in the climate system. The extent of dangerous pollutant concentration in the environment is a subject of debate. The global temperature rise by 2% by 2050 is considered as a high risk level by Stockholm Environmental Institute. This means a global temperature rise of more than 2% is harmful to the environment. In order to limit this temperature rise, the carbon emissions should be declined by 75% in the industrial countries by 2050. A 75% decrease in carbon emissions in the next 40 years implies about 2% decrease every year. As of 2011, the warming emissions of energy production continued rising regardless of the consensus on the basic problem. There is a 25–30 years lag in the complete warming effect of emissions. According to Robert Engelman (World Watch Institute) for security of civilization, one has to stop increase of emissions within a decade regardless of economy and population of the state.

Sharp hike in petroleum prices and increase in environmental pollution jointly have necessitated exploring some renewable indigenous alternatives to conventional petroleum fuels. Also, depletion of fossil fuels, vehicular population, increasing industrialisation, extra burden on home economy, growing energy demand, explosion of population, environmental pollution, stringent emission norms (Euro I, II, III, IV), etc emphasize on the need for alternative fuels. The alternative fuels must be technically feasible, environmentally acceptable, readily available and economically competitive. The possible options we have are renewable energies solar power, Compressed Natural Gas (CNG), producer gas, alcohols, hydrogen, bio alcohol such as methanol, ethanol, butanol, chemically stored electricity such as batteries and fuel cells, propane, non fossil methanol, non fossil natural gas, emulsified fuels, biofuels mostly from non edible seed oils, biodiesels i.e. transesterified form of seed based oils and other biomass sources.

Solar powered panels are very costly and solar energy is not available at all times of the day. Solar energy is not evenly distributed at all locations and problems associated with their maintenance do exist. Compressed natural gas is mostly petroleum based and is available only at refinery locations. Alcohols do have lower cetane numbers and high latent heat of vaporization. Emulsified fuels like Ethanol have lower cetane numbers and lower calorific values which are a disadvantage. Hydrogen as a fuel is considered (highly) explosive and its production is expensive and difficult to store and transport.

More so, for developing countries, fuels of bio-origin, such as alcohol, vegetable oils, biomass, biogas, biodiesels, etc. are becoming important because of their renewable and environmental friendly nature. Some of these fuels can be used directly, while others need some sort of modification before they are used as substitute to conventional fuels.

From the point of view of protecting the global environment and the concern for long term supplies of conventional diesel fuels, it becomes necessary to develop alternative fuels comparable to conventional fuels. Diesel fuel is largely utilized in the transport, agriculture, commercial, domestic, and industrial sectors for the generation of power/ mechanical energy, and the substitution of even a small fraction of total consumption by indigenous alternative fuels particularly of bio-origin will have a significant impact on economy, the environment, the development of agro based industries of the region. Of the alternative fuels, biodiesel obtained from vegetable oils holds good promises as an eco-friendly alternative to petroleum diesel fuels.

Bio-diesel which can be used as an alternative diesel fuel is made from renewable biological sources such as vegetable oils and animal fats. It is bio-degradable, non-toxic and possesses low emission profiles. Also, the use of bio-fuels is environment friendly. Biodiesel production increased by 85% making it the fastest growing renewable energy source in 2006. Over 50% of the world's biodiesel is produced in Germany.

The name bio-diesel was introduced in the United States in the year 1992 by the National Bio-diesel Board which has pioneered the commercialization of bio-diesel. Chemically, bio-diesels are methyl/ethyl esters of vegetable oils. Studies indicate that it can be used in compression ignition engines with little or no (engine) hardware modifications.

One hundred years ago, Rudolf diesel first tested vegetable (Peanut) oil as fuel for his engine. With the advent of cheap petroleum, appropriate crude oil fractions were refined to serve as fuels and diesel fuels and diesel engines started evolving together. Later in the 1940s, vegetable oils were used again as fuel in emergency situations during the period of World War II. Because of the increase in crude oil prices, limited resources of fossil fuels and the environmental concerns these days, there has been renewed focus on vegetable oils for the production of bio-diesel fuels. Bio-diesel is an oxygenated fuel and has the potential to reduce the level of pollution and the level of global warming. Biofuels provided 2.7% of the world's transport fuel, as of 2010. According to the International Energy Agency, biofuels have the potential to meet more than a quarter of world demand for transportation fuels by 2050.

Biodiesels have recently been recognized as a potential substitute to Diesel oil. It is produced from oils or fats using a process called transesterification, in which oils are reacted with alcohols in order to form the esters which are called biodiesels. Feedstock for biodiesel include animal fats, vegetable oils like Soy, Rapeseed, Jatropha, Mahua, Sunflower, Palm, Hemp, Pongamia Pinnata (Karanja), Cotton seed, Neem, Rubber seed, Linseed, Corn, Sesame, Cotton seed and Algae. Biodiesel is a liquid closely similar in properties to fossil/mineral diesel. Chemically, it consists mostly of Fatty Acid Methyl (or Ethyl) Esters (FAME). Most of the biodiesels meet the American Society for Testing and Materials (ASTM) biodiesel standards. Several developed countries have introduced policies encouraging the use of bio diesels made from vegetable oils, bio mass etc. in transport, agriculture and other sectors with the idea of achieving the following goals.

- Prevent environmental degradation by using cleaner fuels to reduce the burden on foreign exchange.
- Reduce the dependence on imported finite fossil fuel resources by replacing them with renewable domestic sources and to provide new demand for crops.
- Support the agriculture and thereby uplift the rural economy.
- Utilization of waste lands, promotion of Agro based industries.

The major advantages of biodiesels are

1. Biodiesels are greener to the environment, biodegradable, renewable, indigenous and have properties closer to that of conventional Diesel oils. Hence it can act as a potential diesel fuel supplement in the near future.

2. They help a country to attain energy self sufficiency in transport, power, agriculture & other related sectors and also boosts rural economy by generating employment. The employment is generated as more labour is required to maintain and cultivate trees whose seeds are feedstock for production of biodiesel.

3. In a predominantly vast agricultural country like India, utilization of waste lands for growing non edible seed bearing trees gives a major thrust to agriculture, rural economy & agro based allied industries.

4. Biodiesel does not need exclusively a separate storage infrastructure. Safe storage time would be up to 6 months beyond which it undergoes oxidation forming a gel like substance.

5. The most important advantage of biodiesels is that its mass scale production & implementation on a large scale requires less expenditure in terms of cost and time compared to all other possible alternative energy sources. It is mentioned in the literature that biodiesels are successfully used in the form of blends with diesel in the existing diesels engines with no modifications.

Due to these advantages of biodiesels over other energy sources, it is a thrust area in energy sector, for power production, transport etc.

In a country like India having a huge agricultural potential, vegetable oil proves a promising alternate for petroleum (diesel) fuel. Today, India has 17% of the world's population, and just 0.8% of the world's known fossil fuel and natural gas resources. India’s annual requirement of oil is 120 million metric tonnes. Significant part of this is consumed in the transportation sector. India produces only about 25% of its total requirement. The import cost today of oil and natural gas is over Rs. 2, 00,000 crores. Oil and gas prices are escalating; the cost of a barrel of oil has doubled within a year. We have nearly 60 million hectares of wasteland, of which 30 million hectares are available for energy plantations like Jatropha, Honge (Karanja/pongamia pinnata) etc. Once grown, the crop has a life of 50 years. Each acre will produce about 2 tonnes of bio-diesel at about Rs. 20 per litre. Biodiesel is carbon neutral and many valuable by-products flow from this agro-industry. Carbon neutral is a term used to describe fuels that neither increase nor reduce the amount of carbon in the atmosphere. India has a potential to produce nearly 60 million tonnes of bio-fuel annually, thus making a significant and important contribution to the goal of “Energy Independence” of India, boosting agriculture based industries and uplifting our rural economy and thereby GDP of the country.

India being predominantly an agricultural country requires major attention for the fulfillment of energy demands of a farmer. Irrigation is the bottleneck of Indian agriculture, it has to be developed on a large scale, but at the same time Diesel fuel consumption for these sectors must be kept at a minimum level because of the price of Diesel oil and its scarcity due to fast depletion. Finding an alternative fuel for petroleum diesel fuel is critically important for our nation's economy and security. The complete substitution of oil imports by indigenous alternatives for the transportation and agricultural sectors is the biggest and toughest challenge for India.

In recent years systematic efforts have been made by several investigators to use biodiesels made from vegetable oils like Sunflower, Peanut, Soybean, Rapeseed, Palm , Cotton Seed, Linseed, Jatropha, Corn, Sesame, Karanja (Pongamia Pinnata), Rubber seed oils etc as alternate fuel to Diesel oil. The vegetable oils used to produce biodiesels are made from renewable sources that are potentially inexhaustible, environment friendly, biodegradable, non aromatic & practically have zero sulphur content in them. Many of the vegetable oils are edible in nature, but their use as fuel has limited applications due to higher domestic food requirement. Only few non edible seed oils have been tried on a diesel engine, leaving a lot of scope in this area. The potential non edible seed oils important from Indian perspective are the oils extracted from the seeds of Karanja plant (Pongamia Pinnata), Mahua, Jatropha Carcus (Ratanjyot), Nagachampa and Rubber (Hevca Brasiliensin).

Almost all non edible oils are being used in soap and pharmaceutical industries, but due to dark colour and odour, Karanja oil is less preferable compared to other non edible oil species for these purposes. Hence Karanja oil can be obtained easily for engine applications after transesterifiation. Esterified oil of Karanja is also referred to as Karanja biodiesel throughout in the present study. Karanja oil is easily available in many parts of India and is relatively cheap compared to other non-edible oils. Hence government agencies and non government organisations should be encouraged to take up cultivation of Karanja at a faster pace as compared to other species available in India as a commercial & potential diesel fuel substitute from the future point of view. This point emphasises the selection of Karanja species for the present study. However, details of Karaja species are discussed in Appendix I.

With the use of fuels like Karanja oil and its ester as alternative to diesel, a country can become self-sufficient in its energy requirements and thereby the dependence of the nation on other countries for oil imports will get reduced. Moreover, the farmers of the nation would also be contributing towards a healthier environment. Bio fuel used alone or in blended form with diesel has the advantage of contributing to cleaner exhaust and reduced requirement of fossil fuel resources of petroleum derivatives.

Of all the investigations conducted, Karanja (Pongamia Pinnata) holds promise for further studies since this oil is not much explored as an alternative biofuel for engine applications. Also its production potential in India is about 1,35,000 million metric tonnes and its present utilization being around 200 million metric tonnes per annum, indicating clearly under utilisation of its present potential. The tress can be grown in humid as well as subtropical climates, need no maintenance and yields an oil content of 33%. It is easily available in many parts of India, cost is low and meets all biodiesel standards. Taking into consideration the utility of Karanja oil for all the applications in totality, it is found from the literature that 94% of the oil potential from this species is still underutilised in our country. Also, studies on Karanja biodiesel on a diesel engine at constant preset CR & IP are reported. However, open literature appears to indicate very less details into the studies of thermal performance, emission characteristics and combustion analysis with esterified oil of Karanja (Karanja biodiesel) on a diesel engine particularly at different compression ratios and fuel injection pressures**.**

Development of more efficient compression ignition engines is almost completely based on experimental investigation. Alongside experimental investigations, with the advancement in computing techniques, the theoretical, simulation, modeling and optimization studies applied to engine performance are attempted by few investigators. Simulation, modeling and programming for performance, fuel consumption, exhaust temperature, toxic emissions and other factors are built. Today, many techniques, packages and codes are used to conduct computational studies.

The use of statistics in engineering has increased rapidly. As internal combustion systems become more complex, the problems faced by investigators also increase in complexity. For some design problems, no physical model exists and empirical studies must be developed and executed in order to optimize engineering systems for performance and compliance to regulations. This holds particularly in the field of diesel combustion designs, where typically, the engine is controlled by a great number of parameters, in order to meet multiple performance objectives. To find the optimum solution, one of the famous techniques for this purpose is the Genetic Algorithm (GA). GA is the first evolutionary optimization technique introduced by Holand J. in 1975, which is based on Darwinian principle of the 'survival of fittest' and the natural process of evolution through reproduction. GAs are a family of computational models inspired by evolution. These algorithms encode a potential solution to a specific problem on a simple chromosome like data structure and apply recombination operators to these structures, so as to preserve critical information.

Engine application engineers usually want to know the performance of a CI engine for various proportions of blends, for various compression ratios and at different injection pressures. This requirement can be met either by conducting comprehensive tests or by simulation i.e., modeling the engine operation. Testing the engine under all possible operating conditions and fuel cases are both time consuming and expensive. On the other hand developing an accurate model for the operation of a CI engine fuelled with blends of biodiesel is too difficult due to the complex nature of the processes involved. As an alternative, engine performance and exhaust emissions can be modeled by using artificial neural networks (ANN). ANNs are inspired by the early models of sensory processing by the brain. An ANN can be created by simulating a network of model neurons in a computer. By applying algorithms that mimic the processes of real neurons, we can make the network 'learn' to solve many types of problems.

1.1 Organisation of Thesis

The thesis begins with an introduction giving a brief insight into the importance of biodiesels as an alternative fuel in compression ignition (CI) engine, the application of GA in case of multi-objective optimization problems and modeling using ANN. Chapter 2 gives a systematic and comprehensive review of the literature on use of various vegetable oils and their biodiesels as alternate fuels in CI engines. The review also comprises of few available studies related to multi-objective optimization and ANN. The experimental test rig used to conduct the study is explained in Chapter 3. Chapter 4 deals with the results of the experimental study carried out on the engine using pure Diesel oil, blends of Karanja biodiesel with diesel, Karanja biodiesel as fuels. It also includes a comparison between the results of the present study with the results of similar nature available in the open literature with the same as well as other biodiesels, followed by a summary of the experimental study. The multi-objective optimization through GA and further modeling through ANN for thermal performance and exhaust gas emissions are explained in Chapter 5. Chapter 6 presents the concluding remarks of the study which is followed by appendices and references at the end of the report.

Chapter 2

Review of Literature

This chapter is devoted to a comprehensive review of the open research literature available on the development of a compression ignition engine using diesel as fuel and its subsequent application using esterified form of renewable biofuels popularly known as biodiesels. This review begins, as given in Section 2.1, by throwing some light on the history of development of diesel engines. Section 2.2 gives a review on recent studies in the field of biodiesels. The studies carried out by noted investigators are classified based on the nature of research conducted and is presented in Section 2.3. This review is separated into sub-sections. Section 2.3.1 deals with an exhaustive review of studies conducted on performance and emissions characteristics of different biodiesels. Studies related to combustion analysis using biodiesels are reviewed in Section 2.3.2. Computational studies are separately reviewed in Section 2.4 which includes studies related to multi-objective optimization and artificial neural network is dealt in Sections 2.4.1 and 2.4.2 respectively. Based on the review, the objectives of the present study are identified and are given in Section 2.5.

2.1 History of Diesel Engines

On February 27, 1892, Rudolph Diesel filed for a patent at the Imperial Patent Office in Germany. Within a year, he was granted Patent No. 67207 for a "Working Method and Design for Combustion Engines . . .A New Efficient, Thermal Engine." With contracts from Frederick Krupp and other machine manufacturers, Diesel began experimenting and building working models of his engine. In 1893, the first model ran under its own power with 26% efficiency, remarkably more than double the efficiency of the steam engines of his day. Finally, in February of 1897, he ran the first diesel engine suitable for practical use, which operated at an unbelievable efficiency of 75% (Plate 2.1).

Plate 2.1 First Diesel Engine

Plate 2.2 Rudolph Diesel

Rudolf Diesel (A photograph of Rudolf Diesel is given in Plate 2.2) demonstrated his engine at the Exhibition Fair in Paris, France in 1898 **(K.C. Pandey [1])**. This engine stood as an example of Diesel's vision because it was fuelled by peanut oil - the "original" biodiesel. He thought that the utilization of a biomass fuel was the real future of his engine. He hoped that it would provide a way for the smaller industries, farmers, and common folk a means of competing with the monopolizing industries, which controlled all energy production at that time, as well as serve as an alternative for the inefficient fuel consumption of the steam engine. As a result of Diesel's vision, compression ignited engines were powered by a biomass fuel, vegetable oil, until the 1920's and are being powered again, today, by biodiesel.

The early diesel engines were not small enough or light enough for anything but stationary use due to the size of the fuel injection pump. They were produced primarily for industrial and shipping in the early 1900's. Ships and submarines benefited greatly from the efficiency of this new engine, which was slowly beginning to gain popularity. The 1920's brought a new injection pump design, allowing the metering of fuel as it entered the engine without the need of pressurized air and its accompanying tank. The engine was now small enough to be mobile and utilized in vehicles. 1923-1924 saw the first Lorries built and shown at the Berlin Motor Fair. In 1936, Mercedes Benz built the first automobile with a diesel engine - Type 260D.

Meanwhile, America was developing a diesel industry. It had always been part of Diesel's vision that America would be a good place to use his engines. Size, need, and the access to biomass for fuel were important for invention and were a part of the existing American scene. Adolphus Busch acquired the rights to the American production of the diesel engine. Busch-Zulger Brothers Diesel Engine Company built the first diesel engine in America in 1898. But, not much was done with development and design of the engine here until after World War I.

Clessie L Cummins, a mechanic-inventor who had been set up in business in 1919 by the investment banker William Glanton Irwin, purchased manufacturing rights to the diesel engine from the Dutch licensor Hvid. He immediately began working on the problems, which had been inherent in the engine since its inception - those of size, weight, and instability created by the fuel system. Cummins soon developed a single disk system that measured the fuel injected. Like the other early engines, Cummins' products were stationary engines and his main market was the marine industry.

It was also during the 1920's that diesel engine manufacturers created a major challenge for the bio fuel industry. Diesel engines were altered to utilize the lower viscosity of the fossil fuel residue rather than a biomass based fuel. The petroleum industries were growing and establishing themselves during this period. Their business tactics and the wealth that many of these "oil tycoons" already possessed greatly influenced the development of all engines and machinery. The alteration was first step in the elimination of the production structure for biomass fuels and its competition as well as the first step in forcing the concept of the biomass as a potential fuel base into obscurity, erasing the possibilities from the public awareness.

1929 and the Stock Market crash brought the threat of bankruptcy to Cummins. In an innovative move, he installed a diesel engine in a limousine and took his backer, Irwin, for a ride, assuring further investment. Cummins continued to experiment with the diesel vehicles, setting a speed record in a Duesenberg at Daytona, driving a truck with a Cummins diesel engine coast to coast on $11.22, and establishing an endurance record of 13,535 miles at Indianapolis Speedway in 1931. Cummins' diesel engines were established and trucks as well as other fleets began using them. Over the years, Cummins has continued to improve the efficiency of the diesel engine, providing technological innovations. Their engines have set a high standard for the industry, exceeding the requirements of the Clean Air Act of 1970.

Mercedes Benz began building diesel driven automobiles in the mid 1930's. These were dependable, enduring automobiles that lasted well into the second half of the century. Early American Ford automobiles were not diesel driven, but they were powered by the biomass fuel, ethanol. Henry Ford shared a similar vision with Rudolph Diesel. He believed plant-based fuel to be the basis of the transportation industry. In a partnership with Standard Oil, he helped further the bio-fuel industry in the mid-west, encouraging development of production plants and distribution stations. But, along with biodiesel, this vision was obliterated by the petroleum industry and ethanol disappeared from the scene. Europe remained the leader in the development and production of diesel and biomass fuel engines for automobiles.

The 1970's arrived and the American people, who were firmly dependent on foreign oil, yet, unaware of the depth of their dependence, were suddenly faced with a crisis. In 1973, OPEC (Organization of the Petroleum Exporting Countries), the Middle Eastern organization controlling the majority of the world's oil and India’s main supplier, reduced the supply of oil and raised the price sending the United States into a crisis. This crisis was recreated in 1978. Long lines at the gas pumps occurred. People panicked as they realized their whole infrastructure depended on the consistent supply of oil - foreign oil- to their economy, conservation and alternatives became important.

The American public looked to diesel fuel which was more efficient and economical and they began buying diesel-powered automobiles. These automobiles accounted for 85% of Peugeot's sales, 70% of Mercedes Benz's sales, 58% of Isuzu's sales, 50% Volkswagen's sales, plus a good portion of Audi's, Volvo's and Dotson’s sales during the 1970's. For the first time, an American manufacturer began producing an automobile with a diesel engine. General Motors made and sold diesel automobiles in the late 1970's, accounting for 60% of all diesel sales in the United States. This surge of diesel sales in American ended in the 1980's. The price of oil had been re-stabilized and the immediate need for conservation receded in the American consciousness. Along with this, the automobiles produced by General Motors were basically converted gasoline engines. The higher compression of the diesel combustion caused blocks to crack and crankshafts to wear out prematurely. GM ended production of its diesel automobiles in 1985.

As we entered the 21st Century, only Mercedes Benz and Volkswagen made diesel automobiles for export to the United States. Sales accounted for less than 6% for Mercedes Benz and less than 5% for Volkswagen. A few light trucks utilizing diesel engines were made by American manufacturers, but no automobiles. Diesel engine efficiency and durability kept them the engine of choice for trucks, heavy machinery, and marine engines. The marketing of the petroleum industry, the American desire for immediacy and ease have kept the use of the diesel engine from becoming part of the consciousness of the general population here in America.

Looking to the future, our relationship with the oil industries and dependency on foreign oil, hopefully, will drive us to explore alternatives with a more open mind.

2.2 Evolution of Biodiesel as an Alternative Fuel

Natural resources that our nation relies heavily upon such as oil, petroleum and natural gas are fossil fuels. This means that they will eventually cease to exist. This gives much economic and political power to the nations that have an abundance of these natural resources. The thought of this supply ending also causes a search for renewable resources that would never cease to exist. Petroleum or “black gold” provides the world with nearly half the energy used. 67% of the world’s oil reserves are found in the Middle East. Saudi Arabia alone has one fourth of the world’s oil. One tenth of the world’s oil is found in Abu Dhabi, Iran, Iraq, and Kuwait. Compared with this extensive supply, the U.S. and Canada have only 3% of the world’s reserves. Despite this, the U.S. and Canada consume more than four times the amount of petroleum than the Middle East.

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 a small Diesel engine, which, at the request of the French government was run on arachide (earth-nut or pea-nut) oil. It worked so smoothly 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. The French Government at the time thought of testing the applicability to power the production of the Arachide, or earth-nut, which grows in considerable quantities in their African colonies, and can easily be cultivated there. Diesel himself later conducted related tests and appeared supportive of the idea. 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" **(Bijalwan et al [2])**.

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 1920's and 1930's and later during World War II and 1970's 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 coking of the injectors, combustion chamber and valves **(De Carvalho Macedo [3], Senthilkumar [4])**. 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.

On 31st August 1937, G. Chavanne of the University of Brussels (Belgium) was granted a patent for a "Procedure for the transformation of vegetable oils for their uses as fuels". This patent described the alcoholysis (often referred to as transesterification) of vegetable oils using ethanol in order to separate the fatty acids from the glycerol by replacing the glycerol with short linear alcohols. The transesterfication reaction is the basis for the production of modern biodiesel. **(Korbitz [5])**

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 bio fuel has been validated by the motor industry. Currently, Parente's 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 transesterified 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, including biodiesel. Today 21 countries worldwide produce biodiesel. Throughout the 1990's, plants were opened in many European countries, including the 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.

In 1997, Kyoto Protocol prompted resurgence in the use of biodiesel throughout the world. Under the Protocol, 37 countries commit themselves to a reduction of four greenhouse gases (GHG) (carbon dioxide, methane, nitrous oxide, sulphur hexafluoride) and two groups of gases (hydrofluorocarbons and perfluorocarbons) produced by them, and all member countries give general commitments. During the same period, nations in other parts of the world also saw local production of biodiesel starting up: by 1998, the Austrian Biofuels Institute had identified 21 countries with commercial biodiesel projects. 100% Biodiesel is now available at many normal service stations across Europe.

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

Since biodiesels have different physical and chemical properties compared to petroleum based diesel fuels, the use of biodiesel in the engine will affect the performance and emissions of the engine. To study this effect, more research is required in order to ensure that pure biodiesel can be used in any engine without any major hardware modifications. A large number of experimental studies are reported to study the thermal performance of biodiesel blended fuel used in diesel engine operated at constant compression ratio. Studies on variable compression ratio diesel engine however, are relatively few.

The diesel fuel consumption in India is about five times higher than gasoline fuel (Table 2.1). The demand of high-speed diesel has been estimated to be 66.9Mt for the year 2011–2012, which would be 1.6 times higher than that of current demand. The cost of the diesel fuel increases due to the increase in crude oil price and processing costs such as desulphurization in order to meet stringent emission norms etc. So, it is necessary to take appropriate policy decisions in the country to fulfil future demand of diesel fuel in view of improving fuel quality and stringent emission norms. Therefore, biodiesel and ethanol are being considered to be supplementary fuels to the diesel in the country. In addition to that, these biofuels are being looked to provide employment generation to rural people through plantation of vegetable oils and can be beneficial to sugarcane farmers through the ethanol program.

Table 2.1 Demand of Gasoline and Diesel in India (Courtesy IOC)

Year	Gasoline demand in million tonnes	Diesel demand in million tonnes	Ratio of Diesel/Gasoline
2002-2003	7.62	42.15	5.53
2003-2004	8.20	44.51	5.42
2004-2005	8.81	46.97	5.33
2005-2006	9.42	49.56	5.26
2006-2007	10.07	52.33	5.20
2011-2012	12.85	66.90	5.21

Biodiesel can be derived from vegetable oils and fats. India has about 100 Mha degraded land which can be utilized for producing raw material for biodiesel. Biodiesel has higher flash point temperature, higher cetane number, and lower sulphur content and lower aromatics than that of petroleum diesel fuel **(Sharp [6])**. It could also be expected to reduce exhaust emissions due to fuel containing oxygen. The main problems of current Indian diesel fuel are low flash point (35 ^{0}C against world average of 52 ^{0}C) and high sulphur content which can be improved by blending it with biodiesel. In this regard, a typical data related to trends in improvements in cetane number, flash point and sulphur content by blending biodiesel are given in Table 2.2.

Table 2.2 Properties of Diesel, Biodiesel Blend (B20) and Biodiesel [6]

Properties	Diesel	B20	B100
Cetane number	43.3	46	47.5
Flash Point (^{0}C)	62	90	146
Sulphur wt (%)	0.0476	0.037	0.00

It is well established that significant reductions in emissions can be achieved by use of biodiesel **(Graboski [7], Peterson [8])**. 13–66% reduction in PM with 2–11% increase in NO_x was reported by **Graboski [7]**. CO_2 emission by use of biodiesel in diesel engines will be recycled by the crop plant resulting to no new addition into atmosphere **(Peterson [8])**. The non-regulated emissions like polycyclic aromatic hydrocarbon (PAH), nitrate polycyclic aromatic hydrocarbon (nPAH), sulphate emission etc. are also significantly lower **(National Biodiesel Board [9])**.

India has varied resources for production of ethanol. Government of India has initiated 5% ethanol-blended petrol with effect from 1st January 2003 and 10% ethanol blended petrol is also being envisaged **(Vijayaraghava [10])**. Ethanol–diesel emulsion can also give beneficial results in terms of emission reduction in diesel engines **(Ahmed [11], Ali [12])**. 41% reduction in particulate matter and 5% NOx emission with 15% ethanol - diesel blends are reported **(Ahmed [11])**. Net savings of 20% in CO_2 emissions was achieved in Brazil due to ethanol and bagasse substitution from fossil fuels **(De Carvalho Macedo [3]).** Biodiesel generally causes a decrease in emission of hydrocarbons (HC) and carbon monoxide **(Schumacher et al. [13]).** Therefore, blending of ethanol even in small quantity could give beneficial results.

2.3 Experimental Studies Using Biodiesel as Fuel

This section deals with the brief reviews of studies conducted on thermal performance, emission characteristics and combustion analysis of biodiesel fuelled diesel engines. Studies on thermal performance and emission characteristics are reviewed in Section 2.3.1 and studies on combustion analysis are reviewed in Section 2.3.2.

2.3.1 Thermal Performance and Emissions

Since the time of realisation that there is a need to find alternative fuels, a number of investigators have made attempts to replace diesel with a substitute such as vegetable oils and biodiesels. This section deals with an exhaustive review of the experimental studies conducted by noted investigators on performance and emission characteristics of diesels engines fuelled with variety of biodiesels and vegetable oils. From the review, it is found that generally biodiesels and vegetable oils are used in blended form with diesel to run on the engines. The biodiesels which can be found in this review are based on the seed oils of Rapeseed, Jatropha, Mahua, Karanja, Cottonseed, Tobacco, Ricebran, Tall, Polanga, Soybean, Castor, Sunflower, Sesame, Canola, Palm, Salmon, Rubberseed etc. The studies are generally conducted on the engines by varying load, speed, torque, compression ratio, brake mean effective pressure or brake power in suitable steps and in suitable range. The performances are generally judged based on the specific fuel consumption and brake thermal efficiency and exhaust gas temperature. The exhaust emissions, most investigators considered, are carbon monoxide, unburnt hydrocarbon, oxides of nitrogen and Smoke opacity.

Mc Donnel et al [14] investigated the use of semi-refined rapeseed oil on a diesel engine. They reported that engine performance was better at 25% blend. The use of rapeseed oil over a longer period of time was found to shorten the injector life due to carbon build up even though there was no wear on the engine components or lubricating oil contamination.

Kalam and Masjuki [15] experimented Palm biodiesel on an indirect injection, naturally aspirated, four cylinder, compression ignition engine. The Compression ratio (CR) and rated power of the engine was respectively 21:1 and 39kW at 5000rpm as rated speed. It was found that brake power increases with increase in the concentration of palm biodiesel (POD) in the blend of ordinary diesel (OD) and palm biodiesel. Along with POD, two other fuels were also tested. One of them was fuel A which contained 50ppm (corrosion inhibitor) additive, 7.5% POD and 92.5% OD and the other was fuel B which contained 50ppm (corrosion inhibitor) additive, 15% POD and 85% OD. Fuel B produced maximum brake power (12.4 kW) at 1600 rpm followed by fuel A (11.4 kW) and fuel OD (10.48 kW). The reason for higher brake power was stated as the effect of addition of corrosion inhibitor in POD blends that influences conversion of heat energy to work. It was stated that POD blends with corrosion inhibitor additive could be effective as alternative fuels for diesel engines because they reduce emissions levels such as those of NO_x, CO and HC. A Multi-element oil analyzer was used to measure wear debris and additives depletion. It was observed that wear metals debris such as Fe, Cu, Al, Pb reduced with increasing POD into the blends. The reason for this was stated as the effect of corrosion inhibitor in fuel which controlled corrosion as well as oxidation in lubricating oil. Fuel B produced the lowest level of wear concentration followed by fuel A and OD. It was concluded that using an anti corrosion additive with POD blends is always effective.

Senthil Kumar et al. [16] experimented with various methods of using vegetable oil (Jatropha oil) and methanol such as blending, transesterification and dual fuel operation. A single cylinder direct injection diesel engine was used for this work and was tested at constant speed of 1500 rpm at varying power outputs. The ratio of methanol to Jatropha oil was maintained as 3:7 on the volume basis. The brake thermal efficiency (BTE) increased from 27.4% with neat Jatropha oil to a maximum of 29% with the methyl ester and 28.7% in the dual fuel operation. The smoke emission were 4.4 Bosch Smoke Units (BSU) with neat Jatropha oil, 4.1 BSU with the blend, 4 BSU with methyl ester of Jatropha oil and 3.5 BSU in the dual fuel operation. The nitric oxide (NO) emissions were lower for Jatropha oil compared to diesel. It was further reduced with dual fuel operation and blending with methanol. This was attributed to reduction in charge temperature due to vaporisation of methanol. Dual fuel operation showed higher hydrocarbon and carbon monoxide emissions than the ester and the blend. This was attributed to the quenching of methanol air flame on the walls and also due to low temperature. The final conclusion drawn from the study was that Jatropha oil can be used as fuel in diesel engines directly and by blending it with methanol. Also use of methyl ester of Jatropha oil and dual fuel operation with methanol induction gives better performance and reduced smoke emissions than blend.

Raheman and Phadatare [17] conducted a study on performance and emissions of a diesel engine by using Karanja methyl ester as fuel. Karanja oil was esterifed using the esterification system developed in the laboratory of Agricultural and Food Engineering Department, Indian Institute of Technology, Kharagpur. The system was capable of preparing the oil esters sufficient in quantity for running commonly used farm engines (3:73 kW) for at least 8 hrs. The engine used for conducting the study was a single cylinder, four-stroke, DI, water-cooled diesel engine having a rated output of 7.5 kW at

3000 rpm and a compression ratio of 16:1. The minimum and maximum CO produced were 0.004%, 0.016% for pure Karanja methyl ester resulting in a reduction of 94% and 73%, respectively, as compared to diesel. The minimum and maximum smoke densities produced for B20 to B100 were 1% and 3% with a maximum and minimum reduction of 80% and 20%, respectively, as compared to diesel. On an average, a 26% reduction in NO_x was obtained for biodiesel and its blends as compared to diesel. For B20–B100, the exhaust temperature measured varied between 260°C and 336°C as compared to 262°C and 335°C for diesel indicating no much variation in exhaust temperature. For an average speed of 2525 rpm (±2%), Brake specific fuel consumption (BSFC) for B20 and B40 was found 0.8–7.4% lower than diesel. In case of B60–B100, the BSFC was found 11–48% higher than that of diesel. This reverse trend was observed due to the lower calorific value with an increase in biodiesel percentage in the blends. The maximum BSFCs were found to be 26.79% and 24.62% for B20 and B40, respectively, which were higher than that of diesel (24.62%). The maximum BSFCs found for B60, B80 and B100 were 24.26%, 23.96 and 22.71%, respectively. This lower BTE obtained for B60–B100 could be due to a reduction in the calorific value and an increase in fuel consumption as compared to B20. The maximum BTEs found for B60, B80 and B100 were 24.26% 23.96 %and 22.71%, respectively.

Carraretto et al. [18] presented the results of an investigation carried out on the potentialities of biodiesel as an alternative fuel based on strategic considerations and field experiences in boilers and diesel engines. The operation of a biodiesel fuelled boiler was checked for some months. The engines were bench-tested and then installed on urban buses for normal operation. Distances, fuel consumption and emissions (CO_2, CO, HC and NO_x) were monitored. Also wear and tear on devices, oil and air filters dirtiness and lubricant degradation were checked. The efficiency of the ‘‘biodiesel’’ boiler is always higher than that of the ‘‘diesel’’ one and does not decrease significantly with time. The increased efficiency is connected with the lower temperature of the gases at the stack. The emissions of CO and CO_2 are very similar for the two boilers. The diesel engine used for experimentation was UNIC 8220.12, six cylinder, four stroke, direct injection diesel engine with a compression ratio of 17:1. The engine could produce a maximum power of 158 kW at 2600 rpm and was widely used on local urban buses. The tests were carried out using different blends of biodiesel and diesel oil i.e. 100%, 80%, 70%, 50%, 30%, 20% and 0% volume of biodiesel. The increase of biodiesel percentage in the blend led to a slight decrease of both power and torque over the entire speed range.

A significant increase of SFC over the entire speed range was registered with biodiesel (about +16% average), due to its lower heating value and greater density. However, by reducing the injection advance, it was possible to optimize combustion, improving performances especially at low and medium speed. By reducing the injection angle with respect to nominal injection advance operation, power and torque were found to be increased up to almost the levels of pure diesel oil while SFC was reduced. CO emissions were reduced but NO_x were increased with the use of biodiesel. The reduction in CO was attributed to complete combustion and increase in NO_x was attributed to higher cylinder temperatures.

Nwafor [19] studied the emission characteristics of a diesel engine fuelled with rapeseed oil. The high viscosity of the rapeseed oil was reduced by increasing the inlet temperature of vegetable oil fuel. The results of unheated rapeseed oil, heated rapeseed oil and neat diesel were compared on the same plot. It was found that with the increase in load, the CO emissions increased for unheated oil and diesel while it reduced at low loads for heated and then increased for higher loads. The CO_2 emission plots did not show any major difference between the fuels at low loading conditions but at high loading levels heated fuel showed increase in CO_2 emissions compared to other fuels. The investigator stated that less cone angle of the fuel spray, less viscosity and efficient combustion were the reasons for high CO_2 emissions. The hydrocarbon emissions were highest for diesel fuel by a factor of 2.5 compared to vegetable oil fuels. At low loading conditions, the HC levels were similar for both heated and unheated fuels while at high loading conditions the heated fuel showed higher HC levels compared to unheated fuel. The Exhaust gas temperature (EGT) was highest for heated fuel followed by unheated fuel and diesel. The overall test results showed that the heating of vegetable oil was beneficial at all loading conditions.

Pradeep and Sharma [20] conducted a test on a single cylinder diesel engine running on rubber seed oil and its blends with diesel. BTE was found lower for bio-diesel blends compared to diesel. Higher combustion duration and lower heat release rate were recorded for bio-diesel.

Ramadhas et al [21] conducted a performance and emissions study on a Canon make four stroke, direct injection, naturally aspirated single cylinder diesel engine using methyl esters of rubber seed oil. The rated power of the engine was 5.5kW and the rated speed was 1500 rpm. The means of loading was an electric generator. The maximum

BTE obtained was about 28% for B10, which was around 25% higher than that of diesel. In case of B50 to B100, the BSFC was found to be higher than that of diesel. At maximum load condition, the BSFC of 100% biodiesel was more than 12% than that for diesel. It was noted that, the engine emits more CO using diesel as compared to that of biodiesel blends under all loading conditions. With increasing biodiesel percentage, CO emission level decreased. B20 blends gave smoke density of 28% as compared to 45% in the case of diesel. It was found that biodiesel fuelled engines emit more NOx as compared to that of diesel fuelled engines.

Usta [22] carried out an experimental study on performance and exhaust emissions of a diesel engine fuelled with tobacco seed oil methyl ester (TSOME). A four cylinder, four stroke turbocharged indirect injection diesel engine was used for the experimental study. Maximum efficiency was generated with the D82.5/TSOME17.5 fuel. The TSOME blends resulted in slightly higher torque and power than the diesel fuel at full load due to its slightly higher density and viscosity. The investigators observed that the turbocharged diesel engine used in the experiments supplied more air at the higher speeds resulting in an increase of turbulence intensity in the combustion chamber. This enabled more complete combustion. Therefore, the beneficial effect of TSOME as an oxygenated fuel was partially lost at high speeds. Since TSOME contains about 11.4% oxygen by weight and this oxygen helps to oxidize the combustion products in the cylinder, especially in rich zones, the addition of TSOME decreased CO emission. Also, TSOME has fairly low sulphur, and therefore, the diesel/TSOME fuel blends resulted in significant SO_2 reduction. Although there has not been a significant difference in NO_x emissions at partial loads, NO_x is slightly increased due to the higher combustion temperature and the presence of fuel oxygen with the blend at full load. At partial loads, the TSOME addition resulted in slight decreases in power, torque and thermal efficiency due to the dominant premixed lean combustion with excess oxygen. This indicated that the addition of TSOME is most effective in rich combustion.

Puhan et al. [23] studied the performance of Mahua oil methyl ester (MOME) in comparison with diesel fuel. The compression ignition engine used for the study was Kirloskar, single cylinder, four stroke, constant speed, vertical, water cooled, direct injection type.. The engine was coupled to a swinging field separating, exciting type DC generator and loaded by electrical resistance bank. It was found that BSFC for MOME is higher than diesel. This according to the investigators was due to the fact that esters have lower heating value compared to diesel so more ester-based fuel is needed to maintain constant power output. They also found that the NO_x emissions reduced in case of biodiesels. The reduction in NO_x was around 4% in case of MOME. It was observed that around 11% reduction in smoke number produced in MOME compared with diesel. The specific fuel consumption is higher (20%) than that of diesel and thermal efficiency is lower (13%) than that of diesel. Exhaust pollutant emission are reduced compared to diesel. Carbon monoxide, hydrocarbon, smoke number, oxides of nitrogen were reduced by 30%, 35%, 11%, 4%, respectively, compared to diesel.

Duran et al [24] studied the effect of specific fatty acid methyl esters present in biofuels on particulate matter emissions. The oxygen content of the fuel is taken into account by considering the transesterified oil composition (palmitic, oleic and linoleic acids methyl esters content, which were selected for summing up to 90% of all esters present in oils). The experiments were performed on a Nissan YD2.2 direct injection turbocharged engine. 30% of reduction in particulate emissions is attained when only 8 Wt % of Palmitic acid Methyl Ester (PME) is added to the reference fuel. The effect of Oleic acid Methyl Ester was found to be important only when PME is present in small proportions. The oleic acid methyl ester produced a slighter but more prolonged decrease in particulate formation. Indeed, Cetane Numbers (CNs) for methyl esters of palmitic, oleic and linoleic acids are 85.9, 59.3 and 38.2, respectively. This would imply a better combustion if an increase in PME concentration was made with the consequent decrease in particulate matter emissions. Simulation from NNs equations proved that the amount of palmitic acid methyl ester in fuels is the main factor affecting the amount of insoluble material emitted due to its higher oxygen content and cetane number.

Labeckas and Slavinskas [25] conducted a comparative analysis of the effect of two different biofuels as diesel fuel (80vol%) and rapeseed methyl ester (20vol%) (RME20) as well as diesel fuel (75vol%) and rapeseed oil (25vol%) (RO25) blends on the economical and ecological parameters of a 59KW high-speed direct-injection diesel engine when operating at a wide range of loads and speeds. At the minimum speed of 1400 rpm and maximum load of BMEP 0.77 MPa the BSFC was found to be lower by 1.6% for blend RME20 and higher by 2.3% for RO25 related to diesel fuel where the value was 238 g/kWh. The maximum NO_x emissions generated from a heavy loaded engine run on blend RME20 at speeds 1400, 1800 and 2200 rpm were higher by 1.4%, 23.7% and 44.7% whereas when fuelling it with blend RO25 the top NOx emission levels were found as 37.4%, 26.7% and 11.7% lower related to diesel fuel.

Usta et al. [26] investigated the effects of the methyl ester produced from a hazelnut soap stock/waste sunflower oil mixture using methanol, sulphuric acid and sodium hydroxide in a two stage process on the performance and emissions of a Ford XLD 418T four cycle, four cylinder, turbocharged indirect injection (IDI) Diesel engine at both full and partial loads. From the results, it was observed that even if addition of the biodiesel to the diesel fuel decreases its heating value, higher power was obtained in the experiments. This was due to higher oxygen content of biodiesels according to the investigators. The CO emissions of the blend were higher than those of diesel fuel between 1500 and 2200 rpm speeds at full and 75% loads, while they were lower than those of diesel fuel at higher speeds. The SO_2 emissions of the blend were lower than those of the diesel fuel. The NOx emissions of the blend were slightly higher than those of the diesel fuel at both full and partial loads. The noise measurements were taken 1 m away from the engine in the engine room by using a sound level meter. The biodiesel addition slightly decreased the noise. The reduction was less than 1 dB in the range of engine speeds tested.

Reddy and Ramesh [27] studied the effect of injection timing, injector opening pressure, injection rate and air swirl level on the performance of Jatropha oil fuelled diesel engine. The engine used to conduct the experiments was a single cylinder, constant speed, direct ignition diesel engine. For varying the injection timing the position of the fuel injection pump was changed with respect to the cam. The injection was varied by changing the diameter of the plunger of the fuel injection pump. A masked inlet with proper orientation was used to create a swirl of air. The observations made in this study were as follows. The BTE increased, HC and smoke emissions reduced with advanced injection timing and increased Injection Pressure (IP) whereas enhancing the swirl had only a small effect on emissions.

Yoshimoto [28] conducted an experiment on single cylinder diesel engine to investigate the spray characteristics, engine performance, emissions and combustion characteristics of water emulsions of the blended fuel with equal proportions of rapeseed oil and diesel.

The researcher found that the performance was improved with slightly increased emissions of CO and NO_x with knock free combustion.

Reyes and Sepulveda [29] conducted a study on particulate matter emissions and power of a diesel engine using salmon oil. The fuel tests were performed on a 5959 cm^3, six cylinder, four stroke, water cooled diesel engine Mercedes Benz Model OM-366. The fuels used were diesel-crude biodiesel (B) and diesel-refined biodiesel (M). The maximum engine power was 100 kW at 2200 rpm. It was observed that the maximum power slightly decreased when percentage of crude biodiesel was increased in the fuel blend. Particulate material emission tests were carried out by measuring opacity of the exhaust gases. 100% of refined biodiesel permits reduction of up to 50% of particulate emission and non critical 3.5% of loss in power with a very good specific fuel consumption. In comparison to crude biodiesel from salmon oil, refined biodiesel has a unique economic disadvantage of the cost involved in distillation of the crude biodiesel, condition that will determine its applicability depending on the eventual cost of diesel fuel. From this study, it was found out that the blends M-100, M-40 and B-100 emerge, respectively, as possible advantageous alternatives according to its particulate emission performance. The blends B-40, B-60 and B-80 were critically near the limit of emission requirements normative for the engine tested at maximum power. However all the blends are in accordance with the official requirements under partial load and free acceleration conditions of the tested engine.

Li and Lin [30] used a biodiesel produced using peroxidation process. The fuel properties, engine performance and emission characteristics of ASTM No. 2D diesel, a commercial biodiesel, biodiesel with and without the peroxidation were analyzed and compared. The peroxidised biodiesel gave higher thermal efficiency than diesel. This was due to higher oxygen content and higher cetane number of biodiesel. The emission indices of CO_2 and CO as well as the exhaust gas temperature were lower for biodiesel than compared to diesel. They concluded that the peroxidation technique is capable of effectively improving fuel properties and reducing the emission pollution of biodiesel.

Crookes et al. [31] presented the results of his experiments conducted on spark-ignition engines and compression ignition engines using variety of biofuels. The biofuels tested were biogas containing carbon dioxide, simulated biogas, commercially available seed oil and rape-seed methyl ester (RME). The oxides nitrogen were less for biogas fuelled engine while unburned hydrocarbons were increased compared to natural gas fuelled engine. With the increase in CO_2 fraction in biogas the brake power remained unchanged and BSFC increased. When CR was increased, oxides of nitrogen are reduced. Diesel fuel and edible-grade seed oil (equal volumes of rape seed and soya oils)-base fuels and blends of these containing 25% and 75% seed oil were tested in compression ignition engine. With the increase in vegetable oil fraction in the blend NO_x increased and smoke number reduced. The baseline data obtained for diesel fuel was also compared with the results obtained for RME biodiesel. It was found that RME produced more NO_x and lower smoke reading compared to diesel.

Rakopoulus et al. [32] conducted an extensive study to evaluate and compare the use of biodiesels of various origins as alternatives to conventional diesel fuel at blend ratios of 10/90 (10% biodiesel and 90% diesel) and 20/80 in a Direct injection (DI) diesel engine. The fuels tested were cottonseed oil, soybean oil, sunflower oil and their corresponding methyl esters as well as methyl esters of rapeseed oil, palm oil, corn oil and olive kernel oil. A series of tests were conducted using each of the fuel blends on the engine working at a speed of 2000rpm and at a medium and high load. In each test, exhaust smokiness, exhaust gas emissions such as NO_x, CO and total unburned HC were measured. Brake specific fuel consumption (BSFC) and brake thermal efficiency were also computed. The smoke intensity was significantly reduced with the use of biodiesel blends with respect to that of neat diesel and this reduction was found to be higher with higher percentage of bio-diesels in the blend. On the contrary, the smoke intensity increased with the use of vegetable oils blends and it increased with increase in the percentage of vegetable oil in the blends. The NO_x emissions were slightly reduced with the use of biodiesels or vegetable oils with respect to that of neat diesel fuel and this reduction was found to be higher with higher percentage of biodiesel or vegetable oil in the blend. The CO emissions reduced with the use of biodiesels and this reduction was higher with higher percentage of biodiesel in the blend. On the contrary, the CO emissions increased with the use of vegetable oil and this increase was proportional to the increase in percentage of vegetable oil in the blend. The unburned HC levels observed were very small and they did not show any specific trend for biodiesels or vegetable oils. The performance of the engine operated with biodiesels and vegetable oils was similar to that when operated with neat diesel fuel giving nearly same BTE. However the BSFC was found to be higher in case of high load. For medium load conditions, BSFC was found to be minimum for 10/90 blend. It was concluded that irrespective of the raw feedstock material all tested, biodiesels and vegetable oils can be used safely and advantageously in the diesel engines at least in small blending ratios with normal diesel fuel.

Dwivedi et al. [33] set out a study to characterize particulate emissions from diesel engines fuelled by (i) mineral diesel and (ii) B20 (a blend of 20% biodiesel with diesel); in terms of metals and benzene soluble organic fraction (BSOF), which is an indicator of toxicity and carcinogenicity. A medium duty, transport diesel engine (Mahindra MDI 3000) was operated at idling, 25%, 50%, 75% and 100% rated load at maximum torque with a speed of (1800 rpm). Samples of particulate were collected using a partial flow dilution tunnel for both fuels. Collected particulate samples were analyzed for their metal contents. In addition, metal contents in mineral diesel, biodiesel and lubricating oil were also measured to examine and correlate their (metals present in fuel) impact on particulate characteristics. Results indicated comparatively lower emission of particulate from B20-fuelled engine than diesel engine exhaust. Metals like Cd, Pb, Na, and Ni in particulate of B20 exhaust were lower than those in the exhaust of mineral diesel. However, emissions of Fe, Cr, Ni Zn, and Mg were higher in B20 exhaust. This reduction in particulate and metals in B20 exhaust was attributed to near absence of aromatic compounds, sulphur and relatively low levels of metals in biodiesel. However, benzene soluble organic fraction (BSOF) was found higher in B20 exhaust particulate compared to diesel exhaust particulate.

Pereira and Oliveira [34] presented their experimental investigation concerning the electric energy generation using diesel and soybean bio-diesel as fuel. For all tests conducted, the electric energy generation was assured without problems and it was observed that the emission of CO, HC and SO_X decrease and temperature of exhaust gases and the emissions of NO and NO_X were comparable to or less than that of diesel. The values of exhaust emissions are given in Table 2.3.

Table 2.3 Comparison of Exhaust Emissions Due to Diesel oil and Biodiesel Blends [34]

Parameters	Diesel	B20	B50	B75	B100
O_2 (%)	19.1	19.1	19.2	19.2	19.3
CO_2 (%)	1.38	1.45	1.39	1.77	1.68
CO (ppm)	174	144	102	160	156
SO_2 (ppm)	5.7	3	2	1.3	2.7
NO (ppm)	446	410	438	445	407

C_xH_y (ppm)	103	102	102	84	70
Exhaust gas temperature (^0C)	141.4	129.5	129	132.7	135
Test room temperature (^0C)	32.4	34.4	39.1	28.3	27.8

Altiparmak et al. [35] experimented with tall oil methyl ester–diesel fuel blends as alternative fuels for diesel engines. Low sulphur and aromatic contents are advantages of tall oil fatty acid methyl ester–diesel fuel blends. It was reported that at high engine speeds, torque and engine power output increased by 6.1% and 5.9%, respectively, with blended fuels in comparison with diesel. At low engine speeds, specific fuel consumption increased with blended fuels depending on the amount of tall oil methyl ester. But, relative to diesel fuel, specific fuel consumption for blended fuels did not increase significantly at higher engine speeds. It was concluded that blended fuels can be used without any modifications in diesel engines. With the use of Tall oil methyl ester–diesel fuel blends decreasing CO emissions were decreased up to 38.9%. Higher NO_X concentrations in exhaust gas were obtained with all blended fuels. NO_x concentration was increased up to 30% depending on the amount of TOME. Compared with the diesel fuel, lower smoke opacity was obtained with blended fuels. Amongst the blended fuels, lower opacity was obtained with 30% D2–70% TOME, especially at middle and higher engine speeds.

Sahoo et al. [36] tested high acid value Polanga (Calophyllum inophyllum L.) oil based mono esters produced by triple stage transesterification process and blended with high speed diesel on a small-size water-cooled direct injection diesel engine. The density and viscosity of the Polanga oil methyl ester formed after triple stage transesterification were found to be close to those of petroleum diesel oil. Smoke emissions also reduced by 35% for B60 as compared to neat petro-diesel. Decrease in the exhaust temperature of a biodiesel-fuelled engine led to approximately 4% decrease in NOx emissions for B100 biodiesel at full load. The performance of biodiesel-fuelled engine was marginally better than the diesel-fuelled engine in terms of thermal efficiency, brake specific energy consumption, smoke opacity, and exhaust emissions including NO_x emission for entire range of operations. It was concluded that excess oxygen content of biodiesel played a key role in engine performance. Neat biodiesel was proved to be a potential fuel for complete replacement of petroleum diesel oil.

Karthikeyan and Mahalakshmi [37], tested turpentine (primary fuel) and high-speed diesel (pilot fuel) with sulphur contents 0.04 ppm on a Kirloskar TAF 1, single cylinder, constant speed (1500 rpm), direct injection engine with a bore of 87.5mm and a stroke of 110mm. This engine was modified to operate on a dual fuel (DF) mode. Usually, knock was suppressed in DF operation by adding more pilot-fuel quantity. In the present study, a minimum pilot-fuel quantity was maintained constant throughout the test and a required quantity of diluents (water) was added into the combustion at the time of knocking. The advantages of this method of knock suppression observed were restoration of performance at full load, maintenance of the same pilot quantity through the load range and reduction in the fuel consumption at full load. From the results, it was found that all performance and emission parameters of turpentine, except volumetric efficiency, were better than those of diesel fuel. The BTE of dual fuel engine at 75% load was 1−2% higher than that of diesel baseline. A maximum of 6% drop in volumetric efficiency was reported in DF engine at full load. In DF mode, an increase in fuel consumption was recorded beyond 75% load due to the occurrence of knock. However, proper quantity of diluents admission reduced the SFC and brought it closer to DBL. In this study, it was stated that the diluents admission at the time of knock prepares comparatively lean mixture inside the cylinder by diluting the air fuel mixture with the help of diluents resulting in a sluggish combustion and knock-free operation. The emissions like CO, UBHC were higher for DF operation than those of the diesel baseline (DBL) and around 40−45% reduction of smoke was observed at 100% of full load. The major pollutant of diesel engine, NO_x, was found to be equal to that of DBL. From the above experiment, it was proved that approximately 80% replacement of diesel with turpentine is quite possible.

Raheman and Ghadge [38] presented the performance and emissions of biodiesel obtained from Mahua oil (B100) and its blend with high speed diesel (HSD) in a Ricardo E6 engine. The BSFC, in general, was found to increase with the increasing proportion of Mahua biodiesel in the fuel blends with HSD, whereas it decreased sharply with increase in load for all fuels. The mean BSFC for the blends was higher than that of pure HSD by 4.3%, 18.6%, 19.6%, 31.7% and 41.4%, respectively, for every 20% additional blending of biodiesel in diesel. At full load conditions, the mean brake thermal efficiency of B100 was about 10.1% lower than that of HSD while at lower loads this variation was as high as 17.1%, which showed significantly lower efficiencies of B100 especially at lower loads. The mean temperature was found to increase linearly from 171 ^{0}C at no load to 285 ^{0}C at full load conditions with an average increase of 15% with every 25% increase in load. The reason for this increase in exhaust gas temperature with load was stated that as load increased more amount of fuel was required in the engine to generate that extra power needed to take up the additional loading. The mean EGTs of B20, B40, B60, B80 and B100 were 6%, 10%, 12%, 14% and 16% higher than the mean EGT of HSD, respectively. This, according to the investigators, could be due to the increased heat losses of the higher blends. It was seen that biodiesel and its blends with diesel produced less smoke than pure diesel. The minimum and maximum smoke densities produced for B20 to B100 were 7% and 34% with a maximum and minimum reduction of 46% and 5%, respectively, as compared to HSD. The minimum and maximum CO produced were 0.02–0.2% resulting in a reduction of 81% and 12%, respectively, as compared to diesel. These lower CO emissions of biodiesel blends were according to the author due to their more complete oxidation as compared to diesel. The amount of NO_x produced for B20 to B100 varied between 17 to 50 ppm as compared to 17 to 44 ppm for diesel. From the findings of this study, it was concluded that B100 could be safely blended with HSD up to 20% without significantly affecting the engine performance (BSFC, BTE, EGT) and emissions (Smoke, CO and NO_x) and thus could be a suitable alternative fuel for diesel.

Najafi et al. [39] conducted a comprehensive combustion analysis to evaluate the performance of a commercial DI, water cooled, two cylinders, in-line, naturally aspirated, RD270 Ruggerini diesel engine using waste vegetable cooking oil as an alternative fuel. In order to compare the brake power and the torque values of the engine, it was tested under same operating conditions with diesel fuel and waste cooking oil biodiesel fuel blends. The results were found to be very comparable. The properties of biodiesel produced from waste vegetable oil were measured based on ASTM standards. The total sulphur content of the produced biodiesel fuel was 18 ppm which is 28 times lesser than the existing diesel fuel's sulphur content used in the diesel vehicles (500 ppm). The maximum power produced was 18.2 kW at 3200 rpm using diesel fuel. The maximum torque produced was 64.2 Nm at 2400 rpm also using diesel fuel. By adding 20% of waste vegetable oil methyl ester, it was noticed that the maximum power and torque increased by 2.7 and 2.9% respectively, also the concentration of the CO and HC emissions have significantly decreased when biodiesel was used. It was stated that oxygen content in biodiesels is responsible for reduction in emissions.

Raheman and Ghadge [40] conducted performance tests on diesel engine with CR varied in the range from 18 to 20 and injection timings of 35^o to 40^o before top dead centre (TDC). The biodiesel used for this study was obtained from Mahua oil. The BSFC and EGT increased while BTE decreased slightly with increase in biodiesel proportion in the blend at lower CR and injection timings considered, where as these factors would show a reverse trend at higher CRs and advanced injection timings. It can be noticed that this value for B20 was 4.7% less than that of high speed diesel (HSD). The mean BTE of B20 was 0.9% higher than that of pure HSD. This could be attributed to the presence of increased amount of oxygen in B20, which might have resulted in its improved combustion as compared to pure diesel. Based on the results of this study, the performance of bio-diesel would be similar to diesel at higher CRs and advanced injection timings since bio-diesel is slightly more viscous. The effects of CR, the blend, injection timing and the load were discussed with respect to each performance parameter separately considered.

Roskilly and Nanda [41] investigated experimentally the application of bio-diesel from recycled cooking oil on small Marine craft diesel engines in UK as per the standard test procedure. An analysis of the performance and exhaust emission indicate that performance was comparable to that of diesel with an increase in rate of fuel consumption with a reduction in NO_x and CO emissions acting as contributors to greener environment. The biodiesel consumptions increased by 14.5–20.9%, over the load range compared to that when fuelled with fossil diesel. They also state that reduction in NO_X were due to smaller heating value and higher cetane number which provides shorter delay ensuring smooth combustion. The CO_2 emissions were 0.3% to 3.1% more when fuelled with biodiesel compared to that fuelled with fossil diesel over the load range. However, the CO_2 emissions from the biodiesel fuelled engines can be treated as ‘‘zero net carbon emissions” and this advantage is the most important character of biodiesel compared to that of fossil diesel. This is because of oxygenated nature of biodiesel. The exhaust gas temperatures when fuelled with biodiesel were a little higher than that when fuelled with fossil diesel, ranging from 1.8% to 11.5%. The differences are not very significant. A slight modification in fuel injection system improves atomization with bio-diesel.

Ashok and Saravanan [42] conducted experiments by preparing blends of diesel ethanol emulsified fuel in proportion of 10, 20, 30 and 40 ethanol/diesel by volume and it was used to run a diesel engine at 1500rpm. The BTE is higher at the ratio of 70D: 30E

(70% diesel and 30% ethanol emulsified fuel) when compared with the sole fuel which is pure diesel. There is an increase of 3.45% efficiency of the emulsified fuel when compared with the sole fuel. BSFC is better for the ratio of 90D: 10E, when compared with sole fuel. There is a decrease in consumption of fuel of 0.11 kg/kW-hr for 90D: 10E. The effect of emulsified fuels on the performance and emission characteristics on the engine indicate that on the whole, use of emulsified fuel can significantly increase the BTE and BSFC .It also lowers the smoke density and exhaust gas temperature compared to neat diesel operation. There is rise of NO_X and particulate matter obtained, however this rise can be minimized by the usage of 80D: 20E emulsified fuel by adding the required additives. 80D: 20E was found to give good performance and low emission of the engine as compared to other D/E ratio fuel and neat diesel fuel. There was not much change in the variation of exhaust gas temperature for all ratios of emulsified fuel compared to that of neat diesel fuel which is important from exhaust emissions point of view.

Hasimoglu et al. [43] stated that though esterified fuels gives lower exhaust gas emissions and are biodegradable and renewable as compared to petroleum based diesel oil, viscosity and volatility problems still exist with these fuels. With the concept of a low heat rejection (LHR) engine (the engine that thermal barrier coating is applied is called low heat rejection (LHR) engine & the thermal barrier coated engine parts are piston, cylinder head, cylinder liners and exhaust valves)., the energy of bio-diesel can be released more efficiently thereby improving engine performance. In this experimental study, they used a turbo charged direct injection (DI) diesel engine ,converted to a LHR engine with thermal barrier coating running on sunflower oil bio-diesel as fuel. The results of this study indicated that the SFC and BTE were improved and the EGT was increased. Thermal barrier coatings were used to improve reliability and durability of hot section metal components and enhance engine performance and efficiency in internal combustion engines. Thermal barrier coatings are usually composed of a bond coat (NiCrAl) as an oxidation resistant layer and stabilized zirconia as a top coat that provides thermal insulation toward metallic substrate. Some important advantages of LHR engines are improved fuel economy, reduced engine noise, higher energy in exhaust gases (higher temperature of exhaust gases) and multi-fuel capability of operating low cetane fuels. With the application of the thermal barrier coating, the heating value of the mixture increased further for both fuels. The deterioration of the engine power and torque for biodiesel fuel was caused by the higher viscosity of the biodiesel. By the application of the thermal barrier coating, the engine power and torque were increased mainly due to the increased exhaust gas temperatures before the turbine inlet in LHR engine.

Banapurmath et al [44] conducted an experimental study on performance and emission characteristics of a DI compression ignition engine using methyl esters of Honge (Pongamia pinnata), Jatropha curcas (Ratanjyot) and Sesame (Sesamum indicum) oils. They suggested that the above bio-diesels could be used in its pure form or blended with diesel incorporating no major engine modifications since they have properties close to mineral diesel. Engine performed better with sesame oil bio-diesel as compared to the other two in terms of thermal efficiency, emission and combustion study. The maximum brake thermal efficiency was recorded with SOME and was 30.4% at 80% power output compared to 31.25% for diesel. All the esters resulted in slightly higher smoke emissions than diesel and were attributed to the incomplete combustion because of their lower volatility and higher viscosity. They found that HC and CO emissions were slightly more with karanja and jatropha as compared to sesame oil methyl ester. This was attributed to incomplete combustion because of higher viscosity, lower volatility and increased combustion duration with a comparable rate of heat release.

Velimir and Petrovic [45] states that air pollution in diesel engines is caused mainly by particulate matters and NO_X emissions. They have an adverse effect on environment and health. This study presents information on defining the best procedure and methodology for measurement of diesel engine particulate emissions. Euro 4 vehicle has 7.10^{13} particles/km and particulate matter of 18 mg/km under standard test conditions.

Utlu and Kocak [46] reported that waste frying oil methyl ester (WFOME) can be used as an alternative diesel engine fuel to operate a turbocharged direct injection diesel engine without modifications to engine or injection system. The lower heating value of WFOME is lower than that of diesel fuel. Since WFOME has a lower heating value, higher density and viscosity, WFOME’s specific fuel consumption is increased by 14.34%. For both the fuels tested, minimum brake specific fuel consumption was obtained at 1750 rpm as 229.59 g/kWh for diesel fuel, and 258.66 g/kWh for WFOME? The emission values were decreased by 17.14% for CO, 1.45% for NOx. Smoke intensity is increased by about 22.46% for the utilization of WFOME compared to diesel fuel. Exhaust temperatures of WFOME were decreased on an average by 6.5% than diesel fuel. WFOME's power value was lower than diesel fuel. The decrease in average power was 4.5% with the use of WFOME. The investigators stressed that long period tests must be done for determination of WFOME's effect on fuel store, fuel systems' elements, engine oil & wear, injectors and burning combustion, pistons, manifolds and valves. This lower engine power obtained for WFOME could be due to fuel flow problems, as higher density, higher viscosity, and decreasing combustion efficiency as bad fuel injection may lead to reduced atomization and lower thermal efficiency than diesel fuel.

Ballesteros et al. [47] investigated important emissions such as total hydrocarbons (HC), particulate matter (PM), volatile organic fraction (VOF) and mean particle diameter (Dm) of PM for typical low sulphur diesel fuel available in Spanish petrol stations and for biodiesel fuel which was obtained from sunflower oil. Total HC emissions were determined in a regulated emissions bench from environment equipped with a flame ionization detector (FID) which, in addition to the total HC, also measures the amount of methane. The particulate matter was collected in a partial dilution mini-tunnel conditioned for the determination of PM emissions in diesel engines & the particle size distributions were determined in a scanning mobility particle sizer.

Gumus [48] used hazelnut kernel oil of Turkish origin to evaluate the performance of a DI diesel engine. Addition of Hazelnut Oil Methyl Ester (HOME) content in the blend, the BTE initially increased until it reached a maximum value at B20 blend and then decreased with increase of the HOME content in blend. Minimum BTE is obtained when neat HOME is used. Although addition of the HOME to the diesel fuel decreased its heating value, higher BTE was obtained for B5 and B20. The reason for this was that the biodiesel included approximately 10% (in weight) higher oxygen that could be used in combustion, especially in the fuel rich zone. This could be a possible reason for more complete combustion, thereby increasing the BTE. However, with more increase of HOME content in blend lower BTE was obtained for B50 and B100 due to the lower heating value and the higher viscosity, which resulted in slightly poorer atomization and poorer combustion. With increasing of HOME content in blend, negative effect of the lower heating value and the higher viscosity of HOME were more effective than positive effect of oxygen concentration included in HOME. The BSFC initially decreased slightly with the addition of HOME content in the blend until it reached 20% value but it increased with more enhancement of the HOME content owing to the lower calorific value and the higher viscosity of HOME. The EGT of B5 and B20 were found to be higher than diesel fuel in the high load owing to the higher viscosity which resulted in poorer atomization, poorer evaporation, and extended combustion which goes on at exhaust stroke. When HOME concentration is increased (B50, B100), the viscosity of blends increased even more and as a result of this, the EGT of blends were lower than EGT of diesel fuel due to more deterioration of combustion and oxidizing of more fuel. It was also found that with higher percentage of HOME blends low amount of CO_2 emissions took place as a consequence of higher viscosity of HOME. According to the researcher, due to the presence of oxygen in biodiesel the formation of smoke is restricted.

Kegl and Pehan [49] discussed the influence of biodiesel produced from rapeseed oil on the injection, spray, and engine characteristics with an aim to reduce harmful emissions. It was found that for biodiesel spray angle was narrower and the penetration length was larger. This was due to low fuel vaporization, worse atomization, and higher injection pressure of B100. Worse atomization was a consequence of high surface tension and viscosity of B100. This led to higher spray tip penetration. By using the engine without any modifications, biodiesel had a positive effect on CO and smoke emissions and on exhaust gas temperature at rated and peak torque conditions. The HC emission was increased at peak torque condition only. The thermal efficiency was practically unaffected, meanwhile the specific fuel consumption increases. Regarding the smoke and NO_x emissions, it can be concluded that B100 reduces smoke to a great extent, but increases the NO_x emission by about 5% at both tested regimes. The use of B100 increased the pump plunger surface roughness. However, this should not worsen the sliding conditions at the pump plunger walls. After biodiesel usage, the average value of root mean square roughness decreased which could be an indicator for even better lubrication conditions. In order to reduce all harmful emissions (of the considered engine), the injection pump timing has to be retarded from 23 to 19°CA BTDC. It has to be pointed out that by this modification the specific fuel consumption and other engine performances remained within acceptable limits.

Altun et al. [50] used a blend of 50% sesame oil and 50% diesel fuel as an alternative fuel in a direct injection diesel engine. Engine performance and exhaust emissions were investigated and compared with the ordinary diesel fuel in a diesel engine. The experiments were performed on a Lombardini 6 LD 400, one cylinder, four-stroke, air-cooled, direct injection diesel engine. It was found that the power produced by the blend of the sesame oil and diesel fuel was close to that of ordinary diesel fuel. The specific fuel consumption was higher for blended fuel than that of the ordinary diesel fuel. It was seen that the blended fuel emitted low CO values and slightly low NO_x values when compared with an ordinary diesel fuel. It was concluded that sesame oil and diesel fuel can be used as an alternative fuel successfully in a diesel engine without any modification and also it is an environmental friendly fuel in terms of emission parameters.

Correa and Arbilla [51] evaluated seven carbonyl emissions (formaldehyde, acetaldehyde, acrolein, acetone, propionaldehyde, butyraldehyde, and benzaldehyde) by a heavy-duty diesel engine fuelled with pure diesel (D) and biodiesel blends (v/v) of 2% (B2), 5% (B5), 10% (B10), and 20% (B20). The tests were conducted using a six cylinder heavy-duty engine, typical of the Brazilian fleet of urban buses, in a steady-state condition under 1000, 1500, and 2000 rpm. The exhaust gases were diluted nearly 20 times and the carbonyls were sampled with SiO_2-C_{18} cartridges, impregnated with acid solution of 2,4-dinitrophenylhydrazine. The chemical analyses were performed by high performance liquid chromatography using UV detection. From the experimental results obtained it was found that formaldehyde was the most abundant carbonyl in the exhaust, followed by butyraldehyde, acetone + acrolein, acetaldehyde, propionaldehyde and benzaldehyde. In all biodiesel blends, it was found that total carbonyls emissions were higher in 1500 rpm than 1000 and 2000 rpm. Using average values for the three modes of operation (1000, 1500, and 2000 rpm) benzaldehyde showed a reduction on the emission (3.4% for B2, 5.3% for B5, 5.7% for B10, and 6.9% for B20) and all other carbonyls showed a significant increase: 2.6, 7.3, 17.6, and 35.5% for formaldehyde; 1.4, 2.5, 5.4, and 15.8% for acetaldehyde; 2.1, 5.4, 11.1, and 22.0% for acrolein+acetone; 0.8, 2.7, 4.6, and 10.0% for propionaldehyde; 3.3, 7.8, 16.0, and 26.0% for butyraldehyde.

Sureshkumar et al. [52] presented the results of performance and emission analyses carried out in an unmodified diesel engine fuelled with Pongamia pinnata methyl ester (PPME) and its blends with diesel. The engine used to carry out the experiments was a single cylinder, four-stroke, water-cooled and constant-speed (1500 rpm) compression ignition engine. Engine tests were conducted to get the comparative measures of brake specific fuel consumption (BSFC), brake specific energy consumption (BSEC) and emissions such as CO, CO_2, HC, NO_x to evaluate the behaviour of PPME and diesel in varying proportions. For the blends B20 and B40, the BSFC was lower than and equal to that of diesel, respectively, and the BSEC was found less than that of diesel at all loads. However, as the PPME concentration in the blend increased, the BSFC increased at all loads. The BSEC for all the fuels tested increases initially at low loads and at higher load conditions; its value is less than that of diesel for all the blends and more than that of diesel for PPME. The CO concentration was totally absent for the blends of B40 and B60 for all loading conditions and as the PPME concentration in the blend increases above 60%, the presence of CO was observed. However, engine emits more CO for diesel as compared to PPME blends under all loading conditions. The lower percentage of PPME blends emits less amount of CO_2 in comparison with diesel. Blends B40 and B60 emit very low emissions. This was attributed to the fact that biodiesel in general is a low carbon fuel and has a lower elemental carbon to hydrogen ratio than diesel fuel. The hydrocarbon emissions were almost zero for all PPME blends except for B20 where some traces were seen at no load and full load. As the PPME content of the fuel increased, a corresponding reduction in NO_x emission was noted and the reduction was remarkable for B40 and B60. The maximum and minimum amount of NO_x produced were 230 and 48 ppm corresponding to B20 and B60. From the experimental studies, the investigators concluded that blends of PPME with diesel up to 40% by volume (B40) could replace the diesel for diesel engine applications for getting less emissions and better performance and will thus help in achieving energy economy, environmental protection and rural economic development.

Zheng et al [53] compared the performance and emissions of the engine fuelled with soy, Canola and yellow grease derived B100 biodiesel fuels and an ultra-low sulphur diesel fuel under high load engine operating conditions. Conditions and the performance of the test fuels were examined. For the high load, a brake mean effective pressure (BMEP) level of 8 bar was used, representing about 65% of the maximum rated power. The tests were conducted on a naturally-aspirated, four stroke, single cylinder, direct injection diesel engine coupled to a DC motoring dynamometer. In the high load operating condition, the engine-out NO_x emissions were dependent on the fuel cetane number, for the same SOI. The biodiesel fuel with a cetane number similar to the diesel fuel produced higher NO_x emissions than the diesel fuel. The biodiesel fuels with a higher cetane number, however, had comparable NO_x emissions with the diesel fuel. A higher cetane number would result in a shortened ignition delay period thereby allowing less time for the air/fuel mixing before the premixed burning phase. Consequently, a weaker mixture would be generated and burnt during the premixed phase resulting in relatively reduced NO_x formation. Generally the emissions of soot, CO and HC were lower for the engine fuelled with biodiesels.

Banapurmath et al. [54] reported the results of research conducted in a four-stroke, single cylinder, water-cooled, direct injection, compression-ignition (CI) engine using Honge oil and blends of its ester. Experiments were conducted with injection timings of 19, 23 and 27^0 BTDC at various loads and at a constant rated speed of 1500 rev min^{-1}. The performance and emission characteristics of Honge oil and Honge oil methyl ester (HOME) blended with diesel, to produce blends designated B20, B40 and B80, were studied. The brake thermal efficiency with B20 operation was closer to diesel operation. Increasing the proportion of HOME in the blend decreases the thermal efficiency. This decrease was attributed by the investigators to the poor combustion characteristics of the blends due to their relatively high viscosity and poor volatility that overcomes the advantage of the excess oxygen present in the biodiesel. The maximum brake thermal efficiency value observed with HOME operation was 29.51% at 80% load whereas it was 31.25% with diesel. With increasing blend ratio, the exhaust gas temperature slightly increases. This was attributed to the lower calorific value of HOME. The smoke level at maximum power was 78.5 HSU for HOME, 75 HSU for the B20 blend and 84 HSU for the B80 blend and 70 HSU for diesel at full load. The HC emissions were found to be 75, 84 and 65 ppm for B20, HOME and diesel respectively and the CO emissions were 0.26, 0.32 and 0.20% for B20, HOME and diesel respectively. The NO_x emissions for B20, B40, B80 and B100 fuels were measured as 1150, 1066, 1055 and 1070 ppm in comparison with 1250 ppm for diesel operation. Of the HOME blends tested, B20 gave the best performance with reduced emissions. The BTE of the engine with the B20 blend at 80% power output was 30.00 % which is the closest to diesel operation. Hence the investigators recommended B20 blend for existing diesel engines.

Keskin et al. [55] studied the usability of cotton oil soap stock biodiesel-diesel fuel blends as an alternative fuel for diesel engines. The biodiesel used in this study was produced by reacting cotton oil soap-stock with methyl alcohol at determined optimum condition. The cotton oil biodiesel-diesel fuel blends were tested in a single cylinder direct injection diesel engine. There were no noticeable differences in the measured engine power output between diesel fuel and the blends fuels at lower speeds. However, at higher engine speeds, a slight decrease in power output of the engine was obtained with B40 and B60 compared to diesel fuel. Comparing with diesel fuel, increasing of specific fuel consumption with blend fuels ranged from 0% to 10.5%, depending on the amount of biodiesel. The lowest percent heat losses to exhaust were obtained with blend of B20. The results showed that engine thermal efficiency of the engine with B20 increased. However, there were no significant differences in percent heat loss to exhaust of engine between other blend fuels and diesel fuel. Particulate Matter with blend fuels decreased at maximum torque speed by 46.6% depending on the amount of biodiesel. The properties of blend fuels, such as oxygen content, higher cetane number and low levels of sulphur content, improved combustion process. It was concluded based that blends of cotton oil soap stock biodiesel and diesel fuel can be used as alternative fuels in conventional diesel engines without any major changes. It was also stated that high calorific value, high cetane number, low sulphur content and low aromatic content, are advantages of cotton oil soap-stock biodiesel–diesel fuel blends. Moreover, cotton oil soap-stock as raw material is cheaper than other vegetable oils.

Agarwal et al. [56] carried out a study to investigate the performance and emission characteristics of Mahua oil, Rice bran oil and Linseed oil methyl ester (LOME), in a Kirloskar stationary single cylinder, four stroke diesel engine and compare it with mineral diesel. The linseed oil, Mahua oil, Rice bran oil and LOME were blended with diesel in different proportions. Baseline data for diesel fuel was collected. The performance data were analyzed based on the data recorded for BTE, brake-spec ific energy consumption (BSFEC), and smoke density for all fuels. The performance and emission parameter for different fuel blends were found to be very close to diesel. 50% linseed oil blend and 30% Mahua oil blend with diesel showed minimum thermal efficiencies compared to their respective blends. Smoke density and BSFEC were slightly higher for vegetable oil blends compared to diesel. However, BSFEC for all oil blends was found to be lower than diesel. BSFEC was found minimum for 20% biodiesel blend. Also 20% rice bran oil blend showed minimum BSFEC than other blends. However, BSFEC was slightly higher for biodiesel blends than mineral diesel. 20% LOME showed improved smoke emission performance than other blends. However, at lower loads, 100% LOME showed slightly higher smoke density than other blends. Economic analysis was also conducted to find out cost of biodiesel after transesterification. Comparative study of cost for different vegetable oils, biodiesel and mineral diesel showed that cost per unit energy produced is almost similar for all fuels.

Korres et al. [57] made an effort to evaluate JP-5 along with diesel and biodiesel for use in a diesel engine. Naval aviation turbine fuel, JP-5, has been accepted as alternative to JP-8 in the frame of the Single Fuel Policy. In this study, JP-5 was used along with diesel and/or biodiesel fuel on a diesel engine, in order to evaluate its behaviour in the frame of the Single Fuel Policy. The fuels used were typical diesel and JP-5 fuels of the Greek market, satisfying the EN-590 specification for 2005 and the MIL-DTL-5624U specification, respectively, and biodiesel derived from animal fats. The base fuels were tested for compliance to the specifications and then used, alone and in various blends, on a stationary single-cylinder diesel engine. Emissions were measured under various loads, along with the volumetric fuel consumption. The diesel fuel used served as the reference. The addition of biodiesel in the blend caused an increase in NO_x emissions as compared to the reference fuel case, while JP-5 reduced these emissions when used in percentages up to 40% by volume. However, these emissions strongly depend on the engine type and the cycle used for their measurement. Particulate matter (PM) emissions were significantly lowered with the addition of biodiesel, as expected. JP-5 initially reduced PM emissions, but when used in percentage over 60% by volume, JP-5 slightly increased the emissions, probably due to its lower cetane number. However, PM emissions with JP-5 remained lower than diesel fuel emissions. Fuel sulphur content seemed to have an undesirable effect on smoke opacity, as observed by the filter luminosity results. The biodiesel increased fuel consumption due to its chemically bound oxygen content. In comparison with the petroleum derived fuels biodiesels showed about the same consumption results. Overall, JP-5 was found to be an attractive alternative for diesel fuel in the frame of Single Fuel Policy.

Kalam and Masjuki [58] presented the experimental test results of a diesel engine using additive-added Palm biodiesel (it is also known as Palm diesel) obtained from palm oil. The test results obtained are brake power, SFC, exhaust emissions and anti-wear characteristics of fuel's contaminated lubricants. The diesel engine used for this study was Isuzu make, of four cylinder, four stroke, water cooled, indirect injection. The test fuels chosen were (1) 100% conventional diesel fuel (B0) supplied by the Malaysian petroleum company (Petronas), (2) B20 as 20% POD blended with 80% B0 and (3) B20X as B20 with X% additive. 4-Nonyl phenoxy acetic acid (NPAA) (CAS number-3115-49-9) was used as additive in biodiesel fuel. The maximum brake power obtained at 2500rpm was 12.28kW fromB20X fuel followed by 11.93kW (B0) and 11.8kW (B20). They attributed this to the effect of fuel additive in B20 blend which influences the conversion of thermal energy to work or increases the fuel conversion efficiency by improving the fuel ignition and combustion quality. The lowest SFC was obtained from B20X fuel followed by B0 and B20 fuels. The average SFC values all over the speed range were 405, 426.69 and 505.38 g/kWh for B20X, B0 and B20 fuels, respectively. It was found that NO_x concentration decreases with B20X fuel (92ppm), which was lower than B20 (119ppm) and B0(115ppm) fuels. It can be examined that NO_x is produced from high combustion temperature. 1% additive was found to be helpful to reduce combustion temperature by allowing high fuel conversion as compared to B20 fuel. Hence, the additive was effective in B20 fuel. It was also found that among all the fuels, fuel B20X produced the lowest level of CO emissions, which was 0.1%, followed by B20 (0.2%) and B0 (0.35%). 1% additive in biodiesel-blended fuel produced complete combustion as compared to B0 fuel. B20X produced lowest HC emission (29ppm) followed by B20 (34 ppm) and B0 (41ppm). From the lubricant test results, it was found that from 1% to 3% of B20X contaminated into lubricant acts as anti- wear additive by reducing Wear Scar Diameter and the coefficient of friction and increasing Flash Temperature Parameter. Hence, the additive was found to be effective in B20 fuel used in diesel engine.

Karabectus et al. [59] investigated the performance and exhaust emissions of a diesel engine fuelled with diesel fuel and a biodiesel, namely Cottonseed oil methyl ester (COME). COME was prepared using cottonseed oil, methyl alcohol and potassium hydroxide as a catalyst. The fuel properties of COME were determined according to ASTM standard test procedure in Petroleum Research Centre of Middle East Technical University. Tests were carried out at full load conditions in a single cylinder, four-stroke, direct injection diesel engine. Before supplied to the engine, COME was preheated to four different temperatures, namely 30 ^{0}C, 60 ^{0}C, 90 ^{0}C and 120 ^{0}C. Preheating of COME was carried out using fuel heating equipment mounted just upstream of the fuel pump. A resistance controlled by a thermostat was installed around a fuel reservoir. The thermostat was used for keeping the temperature of the COME in the reservoir at the required value by energising or de-energising the resistance. Preheating of the COME caused a considerable decrease in its kinematic viscosity and a slight decrease in its specific gravity, thus causing them to approach the values of diesel fuel. COME preheated to 30 ^{0}C, 60 ^{0}C, 90 ^{0}C and 120 ^{0}C are indicated by COME30, COME60, COME90 and COME120, respectively. The test data were used for evaluating the brake power and BTE together with CO and NO_x emissions. The results revealed that preheating COME up to 90 ^{0}C leads to favourable effects on the BTE and CO emissions but causes higher NO_x emissions. Compared to diesel fuel, the brake power obtained with COME30, COME60 and COME90 were moderately decreased, whereas it was significantly decreased with COME120 because of excessive fuel leakage. A higher BTE was determined with the preheated COME mainly due to its lower heating value and improved combustion compared to diesel fuel. Particularly, COME90 and COME120 yielded with a high improvement in the BTE. The use of preheated COME usually yielded a significant decrease in CO emissions, while NO_x emissions were increased due to higher combustion temperatures caused by preheating and oxygen content of COME. The results suggest that COME preheated up to 90 ^{0}C can be used as a substitute for diesel fuel without any significant modification at the expense of increased NO_x emissions.

Keskin et al. [60] investigated the effect of tall oil biodiesel with Mg and Mo based fuel additives on diesel engine performance and emission. Tall oil resinic acids were reacted with MgO and MoO_2 stoichiometrically for the production of metal-based fuel additives (combustion catalysts). The metal-based additives were added into tall oil biodiesel (B60) at the rate of 4 µmol/l, 8 µmol/l and 12 µmol/l for preparing test fuels. In general, both the metal-based additives improved flash point, pour point and viscosity of the biodiesel fuel, depending on the rate of additives. The measured SFC of B60 with metal-based additives slightly decreased. The values of SFC obtained with B60–8Mo were lower than the other test fuels. The catalyst effect of metal-based additives and better fuel properties of biodiesel increased the thermal efficiency of the engine by reducing SFC. CO emission of biodiesel fuel decreased with Mg and Mo based additives. In comparison with B60, maximum reduction with Mg and Mo based additives was 36.21% with B60–12Mg at 2600 rpm and 24.12% with B60–12Mo at 2400 rpm, respectively. In general, when compared to Mo-based additive, relatively lower CO emission was measured with Mg-based additive. The CO emissions decreased mainly because of the catalyst effect as stated in the study. At low engine speed, although lower NO_x emissions were measured with B60–8Mg and B60–12Mg, the higher NO_x emissions were recorded with other biodiesel fuels. In addition, in comparison with Mo-based additives, lower CO_2 concentration was measured with Mg-based additives. Due to oxygen content of biodiesel fuels and catalyst effect of the metal-based additives, smoke opacity of biodiesel fuels reduced at all engine speeds when compared with diesel fuel.

Meng et al. [61] studied the impact of biodiesel/diesel blend fuels on aYC6M220G turbo-charge diesel engine & exhaust emissions were evaluated compared with diesel. The biodiesel used was produced from waste cooking oil. The tested results showed that without any modification to diesel engine, under all conditions, the performance of the engine kept normal, and the B20, B50 blend fuels (include 20%, 50% crude biodiesel respectively) led to unsatisfactory emissions while the B'20 blend fuel (include 20% refined biodiesel) significantly reduced emissions. In general, the reference fuel B0 resulted in smaller fuel consumption per unit energy output compared to all other blends fuels which may be, according to the investigators, due to the lower calorific value in the blend fuels. The minimum values of specific fuel consumption for B0, B20 and B50 was 203.4 g/(kWh), 210.7 g/(kWh) and 222.8 g/(kWh) respectively. The emissions of carbon monoxide, unburned hydrocarbons and nitrogen oxide were examined. B20, B50 blend fuels both were inferior to the reference fuel as far as carbon monoxide and unburned hydrocarbons were concerned and better in nitrogen oxide emission.

Banapurmath et al. [62] evaluated the feasibility of popular alternative fuels in the form of Honge oil/Honge oil methyl ester and producer gas as a total replacement for fossil fuels. Experiments were conducted on a single cylinder four-stroke CI engine operated on single and dual fuel modes at three injection timings of 19^0, 23^0 and 27^0 BTDC and three injection pressures of 190, 200 and 210 bar. Earlier, in single fuel mode of operation, optimum conditions in terms of injection timings and injection pressures for Honge oil and Honge oil methyl ester were determined. The results obtained indicated BTE in dual fuel mode of operation to be lesser than single fuel mode of operation at all the injection timings investigated. However, it was observed that the BTE improved marginally when the injection timing was advanced for both Honge oil and Honge oil methyl ester. The BTE with dual fuel operation at injection timings of 19^0, 23^0 and 27^0 BTDC is 17.25%, 18.25% and 19.00%, respectively, as compared to 28.5%, 27% and 26% for neat Honge oil operation. EGT is found to be higher for dual fuel mode as compared to single fuel mode at all fuel injection timings. The smoke emission for Honge oil- producer gas was found to be more than producer gas-diesel oil. This was due to comparatively higher viscosity of Honge oil. However, with producer gas-Honge oil methyl ester higher brake thermal efficiency and reduced emissions were obtained. With dual fuel operation, smoke and NO_x emissions were considerably reduced with increase in CO emissions. On the whole, it is seen that operation of the engine is smooth on Honge oil–producer gas, Honge oil methyl ester–producer gas dual fuel operation.

Honge oil producer gas results in a slightly reduced thermal efficiency and increased smoke, HC and CO levels.

Nadar and Reddy [63] modified a single cylinder, four stroke, diesel engine to work in dual fuel mode. To study the feasibility of using methyl ester of Mahua oil (MEMO) as pilot fuel, it was used as pilot fuel and liquefied petroleum gas was used as primary fuel. In dual fuel mode, pilot fuel quantity and injector opening pressure were varied as they affected the performance and emissions of the engine. The values of injector opening pressure selected were 180, 200 and 220 bar and pilot fuel quantity were 5, 6, 11 and 7.22 mg per cycle. The diesel engine was modified to work in the dual fuel mode by attaching an LPG line to the intake manifold. The pilot MEMO flow rate and LPG flow rate were varied by adjusting the fuel injection pump and the flow control valve, respectively. The pilot fuel quantity of 5 mg per cycle resulted in higher brake thermal efficiency at the injector opening pressure of 200 bar. The injector opening pressure of 180 bar resulted in EGT. The reason for this was stated that due to injection of pilot fuel at low pressure a higher ignition delay resulted in slow combustion of the fuel. This slow combustion further resulted in higher combustion temperature and hence caused higher EGT. It was observed that the injector opening pressure of 200 bar resulted in lower CO emission at the pilot fuel quantity of 5 mg per cycle due to complete combustion of fuel. The injector opening pressure of 200 bar resulted in lower HC emission. The lowest HC emission was observed between the pilot fuel quantity of 5 mg per cycle and 6.11 mg per cycle. The injector opening pressure of 220 bar resulted in lower NO_x emission. This could be due to the poor combustion of the fuel which results in lower combustion temperature and hence lower NO_x emission. It was also found that the injector opening pressure of 200 bar results in lower smoke emission at the pilot fuel quantity of 5 mg per cycle. Higher heat content of LPG, higher combustion temperature, extended duration of combustion and rapid flame propagations were the reasons stated for reduced smoke level. From the experimental results it was concluded that MEMO can be used as a substitute for diesel in dual fuel engine with the pilot fuel quantity of 5 mg per cycle and at the injector opening pressure of 200 bar.

Devan and Mahalakshmi [64] evaluated experimentally the performance and emissions of a direct injection (DI) diesel engine using Poon oil-based bio-diesels of B20 and B40 blends. The reductions in smoke, hydrocarbon and CO emissions were found to reduce but the NO_X was found to slightly increase. B40 blend was investigated to be better performance wise. This is due to the fact that ignition delay is shorter because of higher cetane number, the combustion was complete and cleaner since the fuel contains oxygen in its molecular structure. Smoke, percentage is less with bio-diesel compared to neat oil due to better combustion. Air fuel ratio is higher for bio-diesels. Esterified oils were better in their heat release characteristics as compared to neat oil. Diesel has slightly higher heat release rate and higher peak pressure in the cylinder during combustion.

Anand and Kannan [65] conducted experimental study of performance and emissions characteristics of a variable compression ratio (VCR) diesel engine running at constant speed on cotton seed oil based bio diesel (COME) in blend proportion of 5% to 20% with diesel. Experimental investigation revealed that higher BTE and lower SFC were observed when biodiesel proportions were less and compression ratios were 15 and 17. At 15: 1 compression ratio (CR), the BSFC is increased by 2.5% and 4.92% for the B5 and B20 blends. At 17:1 CR, it is increased by 3.5% for both the B5 and B20 blends, whereas at 19:1 CR, the B20 blend suggests slightly better fuel economy. The maximum BTE varied between 27.37-29.28% for COME-diesel blends and 26.65-27.92% for Diesel fuel. Higher the blend proportion, a higher CR was required to give similar performance hence they concluded that bio-diesel running needed a higher CR. NO_X emissions were higher as the CR increased. The maximum NO_x emissions were found to be 205 ppm at a CR of 17. The emission of CO and CO_2 were slightly higher for lower blends. The emission of unburned hydrocarbons HC for all fuels were low, 15–80 ppm, showing slightly milder values for COME-diesel blends compared with diesel fuel. All bio-diesel blends tested revealed that they can be safely used in the engine requiring no hardware modifications as such.

Jindal et al. [66] conducted an experimental investigation on the effect of CR and IP in a DI diesel engine running on Jatropha methyl ester. The study was targeted at finding the suitability of bio-diesel with respect to the standard design parameters of diesel engine. Results revealed that increase in CR up to 18.5 improves the performance of bio-diesel as far as BSFC and BTE were concerned as against diesel at preset CR of 17.5 during the trials three injection pressure of 150, 200 and 250 bar were considered. The BSFC was lowest at 200 bar for 50% load and at 250 bar for higher loads where as it was high at 150 bar. This can be attributed to more efficient utilization fuel at higher IP because of better atomization and slight delay in admission of fuel and hence lesser fuel going into the engine cylinder. There is 10% improvement over the standard setting of 17.5 CR/210

IP. BTE increases by about 8.9% with load as the losses encountered are less at higher loads. At CR of 18, there is 5.5% increase in BTE because of better combustion and higher lubricity. Higher compression ratio generally ensures better combustion due to proper atomization and higher surface area available for proper fuel air mixing. Emissions from engine like particulate hydrocarbon, NO_X, EGT and smoke opacity are found lower compared to diesel with slight increase observed in CO and CO_2 emissions. NO_X emissions were found slightly higher at 250 bar and higher CRs because of higher exhaust temperature. All comparisons are made for bio-diesel against base line data of diesel.

Bajpai et al. [67] undertook an experimental study to assess the feasibility of blending Karanja vegetable oil with diesel oil and utilization in a DI diesel engine. Pre heating of oil and blending with diesel was found to reduce the viscosity and with higher flash point, it was safe to store. Engine operation for short duration test studies were satisfactory requiring no major engine hardware modification and can suitably substitute petro-diesel. Performances were comparable and emissions were less for entire range of operations. Self lubricity and free oxygen content of the fuel suggested that, Karanja as an optimum test fuel with slight modification in injection timing and duration to offset its higher viscosity effect.

Banapurmath and Tewari [68] investigated at the prospects of using a dual fuel mode with producer gas and bio-diesel combination towards reducing petro-diesel consumption by CI engine. They suggest the use of non edible oil esters application to save the domestic food requirement. In this context, Honge oil methyl ester was used. Also vegetable oil based bio-diesel has higher smoke emission but duel fuel operation with producer gas/ biodiesel could reduce this problem with improved performance. This requires the use of certain modification in the form of a gas carburettor in the engine hardware. This would improve the performance due to higher calorific value and reduced viscosity. The BTE values of 24.25%, 22.25% and 23% were obtained with producer gas–diesel, producer gas–Honge oil and producer gas–Honge oil methyl ester, respectively. Smoke and NO_X emissions were lower whereas CO and HC emissions were considerably higher due to higher carbon content of bio-gas. Nitrogen oxide emissions in dual fuel mode of operation were 130 ppm, 195 ppm and 175 ppm with diesel, Honge oil and HOME as injected fuel, respectively.

Kandasamy and Thangavelu [69] investigated on the performance of diesel engine using various bio fuels. In this study the bio fuels were blended with diesel and preheated before it was injected into the cylinder. The preheating ensures the enhancement of combustion efficiency and the overall performance of the engine. The investigators concluded that the highest engine efficiency was obtained for the B60 of Pongamia oil and also that the efficiency increased with increase in the temperature. For the rice bran oil, the variation in the engine efficiency was very minimum with change in temperature and the performance of the engine was found to be good for the B40 blend. However the overall performance of the engine was found to be good when the oil and diesel blend is supplied to the cylinder after pre heating.

Baiju et al [70] investigated the scope of utilizing biodiesel developed from both the methyl as well as ethyl alcohol route (methyl and ethyl ester) from Karanja oil as an alternative to diesel fuel. The physical and chemical properties of ethyl esters were comparable with that of methyl esters. B20 of Karanja Oil Ethyl Ester (KOEE) showed minimum efficiency. The low efficiency obtained might be due to low volatility, slightly higher viscosity and higher density of the ethyl ester of Karanja oil, which affected mixture formation of the fuel and thus leads to slow combustion. At 100% load, B20 Karanja Oil Methyl Ester (KOME) showed the lowest fuel consumption. According to them, it could be due to the fact that engine consumes more fuel with diesel-biodiesel blend fuels than with neat diesel fuel to develop the same power output (due to the lower calorific value of diesel-biodiesel blend fuel). Smoke is lower for B100KOME for all loads. This was due to the complete combustion of KOME due to the presence of more oxygen. Smoke emission from ethyl ester was found to be more than that of methyl ester. Methyl esters emitted less CO compared to ethyl esters. It was observed that the enrichment of oxygen owing to biodiesel addition resulted in better combustion. Among the esters, ethyl esters emitted more NO_x than methyl esters. This might be due to higher bulk modulus of ethyl ester than methyl ester.

Haldar et al [71] studied the performance and emission characteristics of Putranjiva roxburghii. In the Tropic of Cancer, these plants are abundantly available. The investigators observed that million tons of seeds of Putranjiva oil go a waste annually which villagers in remote areas can use in pure form or blended with diesel oil to operate engines for running irrigation pumps, grinding mills or straw choppers for cattle feed for shorter duration at the time of fuel crisis or emergency period. In this study, the Ricardo Variable Compression Diesel Engine was run with 10%, 20%, 30% and 40% blends of pure Putranjiva oil with diesel at different loads (0– 2.7 kW), and different injection timings (45^0, 40^0, 35^0 CA BTDC) at constant compression ratio 20:1. It was found that cetane number of Putranjiva oil is less than diesel which meant that ignition timing is crucial for better combustion and hence improves the engine performance and emissions. It appeared that Putranjiva oil blends yield better in performance at 45^0 CA BTDC injection timing in comparison to 40^0 CA BTDC timing for diesel. The volatile nature of Putranjiva oil increases the hydrocarbon emission in exhaust gas.

Mani and Nagarajan [72] used waste plastic oil as fuel in diesel engine. The properties of waste plastic oil was compared with the petroleum products and found that it can also be used as fuel in compression ignition engines. They studied the influence of injection timing on the performance and emission characteristics of a single cylinder, four stroke, direct injection diesel engine using waste plastic oil as a fuel. Tests were performed at four injection timings (23°,20^o,17° and 14° before top dead centre -BTDC). When compared to the standard injection timing of 23° BTDC, the retarded injection timing of 14° BTDC resulted in decreased oxides of nitrogen, carbon monoxide and unburned hydrocarbon while the brake thermal efficiency, carbon dioxide and smoke increased under all the test conditions.

Haldar et al. [73] investigated vegetable oils of Putranjiva, Jatropha and Karanja to find out the most suitable alternative diesel by a chemical processing. These vegetable oils were subjected to degumming in order to improve their properties by removing impurities (gum particles). In degumming process, the vegetable oil to be degummed was mixed with concentrated acid and was vigorously stirred for a few minutes. The mixture was kept for a week so that reaction gets completed and gum particles settle down. The gum can be used as a soil fertilizer and the degummed oil was used after washing it thrice with acid to remove the remaining gum. Degumming is an easy, simple and less expensive process than transesterification where reduction of vegetable oil is little. In this study, Putranjiva vegetable oil was used in a diesel engine for the first time. It was found that out of the three non edible oils tested, Jatropha yields best performance and emission at high loads. The emission levels of CO, CO_2, NO_x, HC, smoke and particulates for Jatropha were encouraging. The parameters varied during experimentation were injection timing and brake power. The experiments were conducted on Ricardo Variable Compression engine using 10%, 20%, 30% and 40% blends of degummed Karanja, Jatropha and Putranjiva oils with diesel. It was found that the oils of Karanja, Jatropha and Putranjiva gave more efficiency at 45^0 BTDC timing than diesel and at 40^0 BTDC timing diesel gives more. The reason stated was less cetane numbers for straight vegetable oils which demanded ignition timing to be more for better combustion. The BSFC and BTE of the engine were satisfactory for 29% blends but blends above 20% did not give effective performance due to their high viscosity. The non edible oils gave lesser emissions due to their higher ignition temperature and better combustion compared to diesel. Overall, it was concluded in this study that any diesel engine used for agricultural purpose can be run with 20% blend of degummed vegetable oil and diesel.

Bhale et al. [74] studied the performance and emission with ethanol blended Mahua biodiesel fuel and ethanol–diesel blended Mahua biodiesel fuel. The tests were conducted on a Kirloskar TV1 four stroke, compression ignition, water cooled, single cylinder engine at the rated speed of 1500 rpm at various loads. The Brake Mean Effective Pressure (BMEP) was varied from 0 to 650 kPa for each blend and observations were taken. The test fuels considered were Mahua methyl ester (MME), MME with 20% ethanol (MME E20), MME with 10% ethanol (MME E10) and MME with 10% diesel and 10% ethanol (MME E10 D10). A considerable reduction in emission was obtained. A 20 volume % ethanol blending into MME achieved improved combustion with reduction in CO by almost 50% on an average without affecting the thermal efficiency. The NO_x emission level was highest for MME. However, with 20% (by volume) ethanol blended MME, it was the lowest. It was stated that low NO_x level was mainly because of the high latent heat of ethanol which reduced the overall combustion temperature which is one of the key parameter for NO_x formation. Smoke emissions from diesel combustion of the MME/ethanol blended fuel decreased strongly with increasing percentage of ethanol in MME. The smoke reduction was attributed to an improved fuel–air mixing by an increased ignition delay due to the lower cetane number as a result of the ethanol addition and also to an increase in the oxygen content in the blended fuel. The quite low boiling point of ethanol also promotes the atomization of the fuel spray and reduces soot by a flash boiling effect. Ethanol blended biodiesel is totally a renewable, viable alternative fuel for improved cold flow behaviour and better emission characteristics without affecting the engine performance.

Nagaprasad et al [75] conducted experiments to determine the performance of castor non-edible vegetable oil and its blend with diesel on a single cylinder, 4 stroke, naturally aspirated, direct injection, water cooled, eddy current dynamometer Kirloskar Diesel Engine at 1500 rpm for variable loads. Initially, neat castor oil and their blends were chosen. The BTE, BSFC of castor oil were found to be 33.45% lower and 54.76% higher compared to those of diesel. This was attributed to higher viscosity and lower calorific value of the fuel. However at rated load, the neat castor oil emissions viz. CO, UHC, smoke were 56.41%, 20.27%, 31.32% respectively higher and NO_x were 44% lower compared to those of diesel. This was attributed to incomplete combustion of the fuel and delay in the ignition process. The Exhaust gas temperature (E.T) of 25% blend of castor oil has lower values compared with all other blends and is well comparable with diesel. The E.T of all blends and diesel increased with increase of operating loads. The 25% blend of castor oil has higher performance than other blends due to reduction in exhaust gas temperature. From the analysis, it was observed that 25% of neat Castor oil mixed with 75% of diesel was the best suited blend for diesel engine without any engine modifications. It was concluded that castor non-edible oil can be used as an alternate to diesel, which is of low cost. The usage of neat bio-diesel will have a great impact in reducing the dependency of India on oil imports, they opined.

Karabectas [76] investigated the effect of turbocharger on the performance and emissions of a diesel engine fuelled with rapeseed oil methyl ester. The performance of the engine was measured in terms of brake power, torque, BSFC and BTE and the emissions measured were CO and NO_x. A naturally aspirated (NA) four stroke direct injection diesel engine was used to carry out the experiments. The experiments were carried out at full load and at speeds between 1200 rpm and 2400 rpm with intervals of 200 rpm. To record the second set of readings, a turbocharger (TU) with boost pressure of 0.7 bar was installed on the engine and readings were recorded at same conditions as done in the earlier case. The brake power and torque of the engine with diesel as fuel were higher than those with biodiesel for both NA and TU operations. However, the use of turbocharger caused considerable increase in brake power and torque with biodiesel when compared to diesel fuel. Also, the values of brake power and torque in the cases of diesel fuel and biodiesel approached each other in TU operation. Under both NA and TU operating conditions, biodiesel showed slightly higher BSFC in comparison to diesel fuel. According to the investigator, this was due to higher fuel density and lower calorific value of biodiesel. It was also found that BSFC decreased in case of TU operation compared to NA operation. The BTE for biodiesel was higher in case of NA operation while in TU operation, it improved further compared to the BTE when engine was fuelled with diesel fuel. NO_x emissions were higher with biodiesel than that with diesel, while CO emissions were less with biodiesels. The higher oxygen content in biodiesels was stated as the reason for higher NO_x emissions. Better combustion of biodiesel due to its higher cetane number and oxygen content resulted in less CO emissions. In the case of TU operation a noticeable increase in NO_x emissions was observed for both diesel and biodiesel. The TU operation with biodiesel resulted in higher ratio of decrease in CO emissions compared to diesel fuel. Overall, it was concluded that the use of biodiesel improves the performance and exhaust emissions of the turbocharged engine better compared with the use of diesel fuel.

Qi et al. [77] conducted an experimental study on the performance characteristics of a direct injection engine fueled with biodiesel/diesel blends. In this study, the biodiesel produced from soybean crude oil was prepared by a method of alkaline-catalyzed transesterification. The effects of biodiesel addition to diesel fuel on the performance, emissions and combustion characteristics of a naturally aspirated DI compression ignition engine were examined. Biodiesel has different properties from diesel fuel. A minor increase in BSFC and decrease in BTE for biodiesel and its blends were observed compared with diesel fuel. The significant improvement in reduction of CO and smoke were found for biodiesel and its blends at high engine loads. HC had no evident variation for all tested fuels. NO_x were slightly higher for biodiesel and its blends. Biodiesel and its blends exhibited similar combustion stages to that of diesel fuel. The use of transesterified soybean crude oil can be partially substituted for the diesel fuel at most operating conditions in terms of the performance parameters and emissions without any engine modification.

Panwar et al. [78] conducted a study for Performance evaluation of a diesel engine fuelled with methyl ester of castor seed oil. In this investigation, castor methyl ester (CME) was prepared by transesterification using potassium hydroxide (KOH) as catalyst and was used in four stroke, single cylinder variable compression ratio type diesel engine. Tests were carried out at a rated speed of 1500 rpm at different loads. Straight vegetable oils pose operational and durability problems when subjected to long term usages in diesel engines. These problems were attributed to high viscosity, low volatility and polyunsaturated character of vegetable oils. The process of transesterification was found to be an effective method of reducing vegetable oil viscosity and eliminating operational and durability problems. The important properties of methyl ester of castor oil were compared with diesel fuel. The engine performance was analyzed with different blends of biodiesel and was compared with mineral diesel. It was concluded that the lower blends of biodiesel increased the break thermal efficiency and reduced the fuel consumption. The exhaust gas temperature increased with increasing biodiesel concentration indicating that Biodiesel is a viable alternative to diesel.

Aydin and Ilkilic [79] used ethanol as an additive to research the possible use of higher percentages of biodiesel in an unmodified diesel engine. Commercial diesel fuel (DF), 20% biodiesel and 80% diesel fuel (B20), and 80% biodiesel and 20% ethanol (BE20), were used in a single cylinder, four strokes direct injection diesel engine. The effect of test fuels on engine torque, power, brake specific fuel consumption, brake thermal efficiency, exhaust gas temperature, and CO, CO_2, NO_x and SO_2 emissions were investigated. The engine torque that obtained for BE20 was higher than both those obtained for diesel and B20 fuels. Average increase of torque values for BE20 was 1.2% and 1.3% when compared to diesel fuel and B20, respectively. The obtained powers for DF and BE20 were almost similar, however the power obtained from B20 was lower than that of other fuels. According to the authors, the higher values of viscosity, density and higher cetane number of B20 were responsible for this power reduction. Average brake-specific fuel consumption for usage of B20 was 22.32% higher than that of diesel fuel and 20.13% higher than that of BE20. It was observed that brake thermal efficiency was 31.71% at 2500 rpm for BE20, while those of DF and B20 were 28.15% and 25.95% respectively. The EGT with BE20 was higher when compared to those of diesel and B20 fuels. The main reason for large difference between BE20 and diesel fuel was stated as the improved combustion of BE20 due to the ethanol added to biodiesel. The NO_x emissions were slightly increased with the use of both biodiesel–ethanol blend and standard diesel fuel with respect to those of the biodiesel–diesel blend. But as far as CO_2 emissions are concerned, a drastic decrease was obtained for B20 fuel when compared with those of both diesel and BE20 fuels. For B20 the average CO_2 decrease was about 67% and 67.5% when compared to diesel and BE20 fuels, respectively. The CO and SO_2 emissions were reduced with the use of both biodiesel–ethanol and biodiesel–diesel blends with respect to those of the neat diesel fuel. The main conclusion derived by this research is that the use of ethanol with biodiesel can potentially remove serious problem revealed with the use of high percentages of biodiesel in operation of unmodified diesel engines.

Sayin and Gumus [80] investigated the influence of compression ratio (CR) and injection parameters such injection timing (IT) and injection pressure (IP) on the performance and emissions of a Direct Injection diesel engine using biodiesel blended-diesel fuel. The increased IP gave the better results for BSFC, BSEC and BTE. Finer breakup of fuel droplets obtained with increased IP provide more surface area and better mixing with air and this effect improved combustion. It was also found that by increasing CR density of air charge was enhanced in the cylinder. The more density and the higher angles of spray cone resulted in increase of amount of air entrainment in the spray. Enough air in the fuel spray contributes to the completion of combustion.

Zhu et al. [81] conducted a study using diesel fuel, biodiesel and biodiesel-ethanol (BE) blends on a 4-cylinder direct-injection diesel engine to investigate the performance and emission characteristics of the engine under five engine loads at the maximum torque condition and engine speed of 1800 rpm. In comparison with Euro V diesel fuel, the biodiesel and BE blends have higher brake thermal efficiency. On the whole, compared with Euro V diesel fuel, the BE blends led to reduction of both NO_x and particulate emissions of the diesel engine. The degree of reduction of NO_x and particulate matter reductions increased with increase in ethanol in the blends. With high percentage of ethanol in the BE blends, the HC, CO emissions could increase. But the use of BE5 could reduce the HC and CO emissions as well.

The studies conducted on performance and emissions of biodiesels from 2000 to 2011 are shown in Table 2.4 in a chronological order. The studies are categorized into the thermal performance, emission characteristics and both the thermal performance & emissions characteristics. Also, the type of fuels used for the respective studies are identified.

Table 2.4 Summary of Studies Based on Thermal Performance and Emission Characteristics

Sr. No	Investigators	Year	Fuel	Type of study (*TP/EC/ TP&EC)	Remarks
1	Mc Donnel et al.	2000	Semi-Refined rapeseed oil	TP	The use of rapeseed oil over a longer period of time was found to shorten the injector life due to carbon build up even though there was no wear on the engine components or lubricating oil contamination.
2	Kalam and Masjuki	2002	Palm oil biodiesel(POD) and its	EC	It was stated that POD blends with corrosion inhibitor additive could be

			blends with diesel		effective as alternative fuels for diesel engines because they reduce emissions levels such as those of NO_x, CO and HC.
3	Senthil Kumar et al.	2003	Jatropha oil with methanol	TP & EC	Jatropha oil can be used as fuel in diesel engines directly and by blending it with methanol. Also use of methyl ester of Jatropha oil and dual fuel operation with methanol induction gives better performance and reduced smoke emissions than blend.
4	Raheman and Phadatare	2004	Karanja biodiesel	TP & EC	A 26% reduction in NOx was obtained for biodiesel and its blends as compared to diesel. CO and smoke was less for Karanja. BTE was less and BSFC was higher for biodiesel.
5	Carraretto et al.	2004	Commercial biodiesel	TP & EC	By reducing the injection angle with respect to nominal injection advance operation, power and torque were found to be increased up to almost the levels of pure diesel oil while SFC was reduced. CO emissions were reduced but NO_x were increased with the use of biodiesel.
6	Nwafor	2004	Rapeseed oil	EC	The investigator stated that less cone angle of the fuel spray, less viscosity and efficient combustion had a considerable effect on CO_2 emissions.
7	Pradeep and Sharma	2005	Rubber seed oil and its blends with diesel	TP & EC	BTE was found lower for bio-diesel blends compare to diesel.
8	Ramadhas et al	2005	Methyl esters of rubber seed oil	TP & EC	It was found that biodiesel fuelled engines emit more NO_x as compared to that of diesel fuelled engines. For biodiesel, the BSFC was found to be higher than that of diesel
9	Usta	2005	Tobacco seed oil methyl ester	TP & EC	It was observed that the turbocharged diesel engine used in the experiments supplied more air at the higher speeds resulting in an increase of turbulence intensity in the combustion chamber. This enabled more complete combustion. Therefore, the beneficial effect of TSOME as an oxygenated fuel was partially lost at high speeds.
10	Sukumar Puhan et al.	2005	Mahua oil methyl ester	TP	It was found that BSFC for MOME is higher than diesel. This according to the investigators was due to the fact that esters have lower heating value compared to diesel so more ester-based fuel is needed to maintain constant power output.
11	Duran et al	2005	Palmitic, oleic and linoleic acids methyl esters	EC	The amount of palmitic acid methyl ester in fuels is the main factor affecting the particulate matter emissions.
12	Labeckas and Slavinskas	2005	Rapeseed methyl ester and its blends with diesel	TP & EC	For the engine fuelled with rapeseed methyl ester NO_x emissions were higher than diesel fuelled engine
13	Usta et al.	2005	Hazelnut soap	TP & EC	The noise measurements were taken 1

			stock/waste sunflower oil mixture		m away from the engine in the engine room by using a sound level meter. The biodiesel addition slightly decreased the noise. The reduction amount was less than 1 dB in the range of engine speeds tested.
14	Narayana Reddy and Ramesh	2006	Jatropha oil	TP & EC	The BTE increased, HC and smoke emissions reduced with advanced injection timing and increased IP whereas enhancing the swirl had only a small effect on emissions.
15	Yoshimoto	2006	Rapeseed oil	TP & EC	It was found that the performance was improved with rapeseed oil slightly increased emissions of CO and NO_x with knock free combustion
16	Reyes and Sepulveda	2006	Salmon oil biodiesel	TP & EC	It was observed that the maximum power slightly decreased when percentage of crude biodiesel was increased in the fuel blend. Particulate material emission tests were carried out by measuring opacity of the exhaust gases. 100% of refined biodiesel permits reduction of up to 50% of particulate emission and non critical 3.5% of loss in power with a very good specific fuel consumption.
17	Li and Lin	2006	Commercial Biodiesel produced using peroxidation process	TP &EC	They used a biodiesel produced using peroxidation process which was capable of effectively improving fuel properties and reducing the emission pollution of biodiesel.
18	Crookes et al.	2006	Biogas containing carbon dioxide, simulated biogas, commercially available seed oil and rape-seed methyl ester	EC	The oxides nitrogen were less for biogas fuelled engine while unburnt hydrocarbons were increased compared to natural gas fuelled engine.
19	Rakopoulus et al**.**	2006	Cottonseed oil, soybean oil, sunflower oil and their corresponding methyl esters as well as methyl esters of rapeseed oil, palm oil, corn oil and olive kernel oil	TP & EC	The smoke intensity was significantly reduced with the use of biodiesel blends with respect to that of neat diesel and this reduction was found to be higher with higher percentage of bio-diesels in the blend.
20	D. Dwivedi et al	2006	Commercial Biodiesel	EC	Samples of particulate were collected using a partial flow dilution tunnel for both fuels. Collected particulate samples were analyzed for their metal contents. Metals like Cd, Pb, Na, and Ni in particulate of B20 exhaust were lower than those in the exhaust of mineral diesel.
21	Pereira and Oliveira	2007	Soybean bio-diesel	EC	It was observed that the emission of CO, HC and SO_X decrease and temperature of exhaust gases and the emissions of NO and NO_X were comparable to or less than that of diesel.

22	Altiparmak et al.	2007	Tall oil methyl ester–diesel fuel blends	TP & EC	It was concluded that blended fuels can be used without any modifications in diesel engines. With the use of Tall oil methyl ester–diesel fuel blends decreasing CO emissions were decreased up to 38.9%.
23	Sahoo et al	2007	Polanga oil based methyl esters	TP & EC	The performance of biodiesel-fuelled engine was marginally better than the diesel-fuelled engine in terms of thermal efficiency, brake specific energy consumption, smoke opacity, and exhaust emissions including NO_x emission for entire range of operations.
24	Karthikeyan and Mahalakshmi	2007	Turpentine oil	TP & EC	It was found that all performance and emission parameters of turpentine, except volumetric efficiency, were better than those of diesel fuel.
25	Raheman and Ghadge	2007	Mahua biodiesel and its blends with diesel	TP & EC	From the findings of this study, it was concluded that pure biodiesel could be safely blended with diesel up to 20% without significantly affecting the engine performance (BSFC, BTE, EGT) and emissions (Smoke, CO and NO_x) and thus could be a suitable alternative fuel for diesel.
26	Nazafi et al.	2007	Waste vegetable cooking oil methyl ester and its blends with diesel	TP & EC	By adding 20% of waste vegetable oil methyl ester, it was noticed that the maximum power and torque increased by 2.7 and 2.9% respectively, also the concentration of the CO and HC emissions have significantly decreased when biodiesel was used.
27	Raheman and Ghadge	2008	Mahua oil biodiesel	TP & EC	Based on the results of this study it was concluded that, the performance of bio-diesel would be similar to diesel at higher CRs and advanced injection timings since bio-diesel is slightly more viscous.
28	Roskilly and Nanda	2008	Bio-diesel from recycled cooking oil	TP & EC	This analysis of the performance and exhaust emission indicate that performance was comparable to that of diesel with an increase in rate of fuel consumption with a reduction in NO_x and CO emissions acting as contributors to greener environment.
29	Ashok and Saravanan	2008	Blends of diesel and ethanol	TP & EC	There was an increase of 3.45% efficiency of the blended fuel when compared with the sole diesel fuel. Specific fuel consumption was better for the ratio of 90/10 (Diesel / Ethanol) when compared with sole diesel fuel.
30	Hasimoglu et al.	2008	Sunflower oil based biodiesel	TP	In this experimental study, they used a turbo charged direct injection (DI) diesel engine converted to a LHR engine with thermal barrier coating. The results of this study indicated that the SFC and BTE were improved and the EGT was increased
31	Banapurmath et	2008	Methyl esters of	TP & EC	Engine performed better with sesame

	al		Honge, Jatropha curcas and sesame oils		oil bio-diesel as compared to the other two in terms of thermal efficiency, emission and combustion study.
32	Velimir and Petrovic	2008	Commercial biodiesel	EC	This study presents information on defining the best procedure and methodology for measurement of diesel engine particulate emissions.
33	Utlu and Kocak	2008	Waste frying oil methyl ester (WFOME)	TP & EC	It was stated that lower engine power obtained for WFOME could be due to fuel flow problems, as higher density, higher viscosity, and decreasing combustion efficiency as bad fuel injection may lead to reduced atomization and lower thermal efficiency than diesel fuel.
34	Ballesteros et al.	2008	Sunflower oil biodiesel	EC	Total HC emissions were determined in a regulated emissions bench from environment equipped with a flame ionization detector (FID) which, in addition to the total HC, also measures the amount of methane.
35	Gumus	2008	Hazelnut Oil Methyl Ester	TP & EC	When HOME concentration is increased (B50, B100), the viscosity of blends increased even more and as a result of this, the EGT of blends were lower than EGT of diesel fuel due to more deterioration of combustion and oxidizing of more fuel.
36	Kegl and Pehan	2008	Rapeseed oil biodiesel	EC	It was found that for biodiesel spray angle was narrower and the penetration length was larger. This was due to low fuel vaporization, worse atomization, and higher injection pressure of biodiesel
37	Altun et al.	2008	Sesame oil blended with diesel	EC	It was seen that the blended fuel emitted low CO values and slightly low NO_x values when compared with an ordinary diesel fuel.
38	Correa and Arbilla	2008	Commercial biodiesel	EC	In all biodiesel blends, it was found that total carbonyls emissions were higher at lower speeds than at higher speeds.
39	Sureshkumar et al.	2008	Karanja biodiesel	TP & EC	The investigators concluded that blends of PPME with diesel up to 40% by volume (B40) could replace the diesel for diesel engine applications for getting less emissions and better performance and will thus help in achieving energy economy, environmental protection and rural economic development.
40	Zheng et al	2008	Soy, Canola and yellow grease derived biodiesel fuels	TP & EC	The biodiesel fuel with a cetane number similar to the diesel fuel produced higher NO_x emissions than the diesel fuel. The biodiesel fuels with a higher cetane number, however, had comparable NO_x emissions with the diesel fuel.
41	Banapurmath et	2008	Honge oil, its methyl	TP & EC	The BTE of the engine with the B20

	al		ester and its blends with diesel		blend at 80% power output was 30.00 % which is the closest to diesel operation. Hence the investigators recommended B20 blend for existing diesel engines.
42	Keskin et al.	2008	Cotton oil soap stock biodiesel-diesel fuel blends	TP	It was concluded that blends of cotton oil soapstock biodiesel and diesel fuel can be used as alternative fuels in conventional diesel engines without any major changes. It was also stated that high calorific value, high cetane number, low sulphur content and low aromatic content, are advantages of cotton oil soap-stock biodiesel−diesel fuel blends.
43	Agarwal et al.	2008	Linseed oil, Mahua oil, rice bran oil and linseed oil methyl ester blended with diesel	TP & EC	50% linseed oil blend and 30% Mahua oil blend with diesel showed minimum thermal efficiencies compared to their respective blends. Smoke density and BSFEC were slightly higher for vegetable oil blends compared to diesel.
44	Korres et al.	2008	Naval aviation turbine fuel, JP-5	EC	JP-5 initially reduced PM emissions, but when used in percentage over 60% by volume, JP-5 slightly increased the emissions, probably due to its lower cetane number.
45	Kalam and Masjuki	2008	Palm biodiesel with 4-Nonyl phenoxy acetic acid as additive	TP & EC	1% additive was found to be helpful to reduce combustion temperature by allowing high fuel conversion as compared to B20 fuel.
46	Karabectus et al.	2008	Cottonseed oil methyl ester (COME)	TP & EC	The use of preheated COME usually yielded a significant decrease in CO emissions, while NO_x emissions were increased due to higher combustion temperatures caused by preheating and oxygen content of COME
47	Keskin et al.	2008	Tall oil biodiesel with metal based (Mg and Mo) additives	TP & EC	Due to oxygen content of biodiesel fuels and catalyst effect of the metal-based additives, smoke opacity of biodiesel fuels reduced at all engine speeds when compared with diesel fuel.
48	Meng et al.	2008	Waste cooking oil biodiesel	TP & EC	The tested results showed that without any modification to diesel engine, under all conditions, the performance of the engine kept normal, and the B20, B50 blend fuels (include 20%, 50% crude biodiesel respectively) led to unsatisfactory emissions while the B’20 blend fuel (include 20% refined biodiesel) significantly reduced emissions.
49	Banapurmath et al.	2008	Karanja oil and its methyl ester with producer gas	TP & EC	It was observed that the Brake thermal efficiency improved marginally when the injection timing was advanced for both Karanja oil and Karanja oil methyl ester. The operation of the engine is smooth on Karanja oil−producer gas, Karanja oil methyl ester−producer gas dual fuel

					operation. Karanja oil producer gas results in a slightly reduced thermal efficiency and increased smoke, HC and CO levels.
50	Nadar and Reddy	2008	Mahua oil methyl ester (MEMO) and liquefied petroleum gas (LPG)	EC	The diesel engine was modified to work in the dual fuel mode by attaching an LPG line to the intake manifold. The pilot MEMO flow rate and LPG flow rate were varied by adjusting the fuel injection pump and the flow control valve, respectively. From the experimental results it was concluded that MEMO can be used as a substitute for diesel in dual fuel engine with the pilot fuel quantity of 5 mg per cycle and at the injector opening pressure of 200 bar.
51	Devan and Mahalakshmi	2009	Poon oil-based bio-diesel and its blends with diesel	EC	The smoke, hydrocarbon and CO emissions were found to reduce but the NO_x was found to slightly increase for biodiesel blends.
52	Anand and Kannan	2009	Cottonseed oil methyl ester	TP & EC	At compression ratio of 17:1, ignition delay is shorter for all biodiesel blends than neat diesel due to higher cetane number
53	Jindal and Nandwana	2009	Jatropha oil methyl ester	TP & EC	Demonstrated that increase in compression ratio associated with increase in injection pressure improves the performance of the engine. Proposed that Smoke and CO emission reduce with use of biodiesel
54	Bajpai et al.	2009	Karanja oil and its blends with diesel	TP & EC	Pre heating of oil and blending with diesel was found to reduce the viscosity and with higher flash point, it was safe to store.
55	Banapurmath and Tewari	2009	Karanja oil and its methyl ester with producer gas	TP & EC	A gas carburetor was suitably designed to maximize the engine performance in dual fuel mode with Karanja oil–producer gas and Karanja oil Methyl Ester –producer gas respectively. Dual fuel mode of operation with carburetor resulted in better performance with reduced emissions.
56	Kandasamy and Thangavelu	2009	Rice bran oil and Pongamia oil	TP	Fuel tank with heater, temperature regulator and stirrer unit were the accessories used in the experimental set up for blending and heating the oil. It was observed that engine efficiency increases with increase in the temperature of biodiesel.
57	Baiju et al	2009	Methyl and ethyl esters of Karanja oil	TP & EC	Among the esters, ethyl esters emitted more NOx than methyl esters. This might be due to higher bulk modulus of ethyl ester than methyl ester.
58	Haldar et al	2009	Methyl ester of Putranjiva roxburghii oil	TP & EC	It was found that cetane number of Putranjiva oil is less than diesel which meant that ignition timing is crucial for better combustion and hence

					improves the engine performance and emissions.
59	Mani and Nagarajan	2009	Waste plastic oil	TP & EC	When compared to the standard injection timing of 23° BTDC, the retarded injection timing of 14° BTDC resulted in decreased oxides of nitrogen, carbon monoxide and unburned hydrocarbon while the brake thermal efficiency, carbon dioxide and smoke increased under all the test conditions.
60	Haldar et al.	2009	Oils of Putranjiva, Jatropha and Karanja	TP & EC	It was found that out of the three non edible oils tested, Jatropha yields best performance and emission at high loads
61	Bhale et al.	2009	Mahua oil methyl ester	TP & EC	It was noted that the fuel filter was clogged 30% more for biodiesel than that for diesel after 200 hours of operation. Blending Mahua methyl ester with ethanol and kerosene has improved the cold flow performance
62	Nagaprasad et al	2009	Castor oil and its blends with diesel	TP & EC	It was concluded that castor non-edible oil can be used as an alternate to diesel, which is of low cost. The investigators opined that usage of neat bio-diesel will have a great impact in reducing the dependency of India on oil imports.
63	Karabectas	2009	Rapeseed oil methyl ester	TP & EC	NO_x emissions were higher with biodiesel than that with diesel, while CO emissions were less with biodiesels. The higher oxygen content in biodiesels was stated as the reason for higher NO_x emissions
64	Qi et al.	2010	Soybean oil methyl ester	TP & EC	A minor increase in BSFC and decrease in BTE for biodiesel and its blends were observed compared with diesel fuel. The significant improvement in reduction of CO and smoke were found for biodiesel and its blends at high engine loads.
65	Panwar et al.	2010	Methyl ester of Castor seed oil	TP	It was concluded that the lower blends of biodiesel increased the break thermal efficiency and reduced the fuel consumption. The exhaust gas temperature increased with increasing biodiesel concentration indicating that Biodiesel is a viable alternative to diesel.
66	Aydin and Ilkilic	2010	Sunflower oil biodiesel	TP & EC	Blends of biodiesel and ethanol fuel can be used as alternative fuels in conventional diesel engines without any major changes. The NOx emissions were slightly increased with the use of both biodiesel–ethanol blend and standard diesel fuel with respect to those of the biodiesel–diesel blend
67	Sayin and Gumus	2011	Commercial biodiesel	TP	Finer breakup of fuel droplets obtained with increased IP provide

					more surface area and better mixing with air and this effect improved combustion.
68	Zhu et al.	2011	Commercial biodiesel blended with ethanol	TP & EC	The degree of reduction of NO_x and particulate matter reductions increased with increase in ethanol in the blends.

*TP- Thermal Performance, EC- Emission Characteristics, TP & EC – Thermal Performance and Emission Characteristics

Among the sixty-eight (68) studies reported in the Table 2.4, seven (7) studies are on thermal performance, fourteen (14) studies are on emission characteristics and remaining forty-seven (47) studies are on both thermal performance and emission characteristics.

There are various observations made by investigators in their respective studies. Some general observations which can be identified from the open literature are as follows:

- Biodiesels can be successfully used in existing diesel engines without any hardware modifications.
- The thermal efficiency of a biodiesel fuelled engine is observed to be slightly lesser than that of a diesel fuelled engine.
- The harmful exhaust emissions are generally lesser for all biodiesels as compared to pure diesel.

The different biodiesels which are considered by earlier investigators for studying the thermal performance and emission characteristics are Rapeseed, Polanga, Soybean, Palm, Jatropha, Sunflower, Cottonseed, Karanja, Putranjiva, Castor, Waste plastic oil, Ricebran, Mahua, Poon, Tall, Linseed, Canola, Sesame, Turpentine, Tobacco seed oil and Rubberseed oil. Most of the studies are conducted by blending biodiesels with pure diesel.

Very few studies appear to be reported in the open literature using Karanja biodiesel and Karanja oil as a fuel on diesel engine (Refer Table 2.5). Among 9 studies reported, 6 are conducted using Karanja biodiesel as fuel and the remaining using Karanja oil. It is found that all of them are studies related to the effect of load, blend, speed, injection timing, brake power, etc on thermal performance and emission characteristics.

Table 2.5 Studies Reported with Karanja Biodiesel On Thermal Performance and Emissions at Constant Compression Ratio

Sr. No	Investigators	Year	Fuel		Type of study (*TP/ EC/ TP & EC)	Parameters Varied
			Biodiesel	Oil		
1	Raheman and Phadatare	2004	Karanja biodiesel		TP & EC	Load, Blend
2	Banapurmath et al.	2008	Biodiesels from Karanja, Jatropha and sesame oil		TP & EC	Brake power, Blend, Injection Timing
3	Sureshkumar et al.	2008	Karanja biodiesel and its blends with diesel	Karanja oil	TP & EC	Load, Blend
4	Banapurmath and Tewari	2008	Karanja biodiesel	Karanja oil	TP & EC	Brake power, Blend, Injection Timing
5	Banapurmath et al.	2008	Karanja biodiesel with producer gas	Karanja oil	TP & EC	Brake power, Blend, Injection Timing
6	Bajpai et al.	2009		Karanja oil and its blends with diesel	TP & EC	Blend, Load
7	Murugu Mohan Kumar Kandasamy, Mohanraj Thangavelu	2009		Rice bran and Karanja oil	TP	Load, Blend
8	Baiju et al.	2009	Methyl and ethyl esters of Karanja oil		TP & EC	Load, Blend
9	Haldar et al.	2009		Oils of Putranjiva, Jatropha and Karanja	TP & EC	Load, Blend, Injection timing

*TP- Thermal Performance, EC- Emission Characteristics, TP & EC – Thermal Performance and Emission Characteristics

It is also observed that all the studies are conducted on a constant compression ratio engine at constant preset injection pressure (close to standard CR of 17.5, IP of 210 bar). Among the studies using Karanja biodiesel as fuel, there is no study reported for the evaluation of thermal performance and emission characteristics under different preset compression ratios and varied injection pressures.

Table 2.6 refers to the various studies conducted at different preset compression ratios using biodiesels other than Karanja.

Table 2.6 Studies with Varying Compression Ratios For Different Biodiesels Other Than Karanja

Sr. No	Investigators	Year	Fuel	Engine Variables
1	Raheman and Ghadge	2008	Mahua biodiesel	Load, Injection timing, Compression ratio, blend
2	Jindal et al.	2009	Jatropha biodiesel	Load, compression ratio, injection pressure
3	Anand et al	2009	Cottonseed oil methyl	Brake Mean Effective Pressure, Compression

			ester	Ratio, Blend
4	Sayin and Gumus,	2011	Commercial Biodiesel	Compression ratio, injection timing and injection pressure

2.3.2 Combustion Analysis

In this section, the review of studies based on combustion analysis of biodiesel fuelled engines is dealt with. Generally, the parameters generally considered to analyse the combustion processes of biodiesels in a CI engine are heat release rate, cylinder pressure, ignition delay, combustion duration, rate of pressure rise.

Senthil Kumar et al. [16] experimented with various methods of using vegetable oil (Jatropha oil) and methanol such as blending, transesterification and dual fuel operation. A single cylinder direct injection diesel engine was used for this work and was tested at constant speed of 1500 rpm at varying power outputs. The ratio of methanol to Jatropha oil was maintained as 3:7 on the volume basis. Jatropha oil and its methyl ester showed longer ignition delays as compared to diesel. According to the investigators, this was due to lower cetane numbers of jatropha oil and its methyl esters. The ignition delay was 11^0 CA with Jatropha oil, 10^0 CA with methyl ester, 12^0 CA with blend and 13^0 CA BTDC with dual fuel operation. The increase in the ignition delay with blend and dual fuel operation was attributed to the cooling effect produced by vaporisation of methanol. The peak cylinder pressure and maximum rate of pressure rise are minimum for Jatropha oil due to its high viscosity which leads to poor atomisation. However, the peak pressure with ester and blend were found to be higher due to the improvement in the preparation of air fuel mixture as a result of lower viscosity. Standard diesel showed highest peak pressure and rate of pressure rise. The duration of combustion was lowest for diesel and it increased in the ascending order for Jatropha methyl ester, blend, Jatropha oil, dual fuel operation. The highest combustion duration for dual fuel operation were attributed to the burning of inducted methanol by flame propagation

Banapurmath et al. [54] reported the results of research conducted in a four-stroke, single cylinder, water-cooled, direct injection, compression-ignition (CI) engine using Honge oil and blends of its ester. Experiments were conducted with injection timings of 19, 23 and 27^0 BTDC at various loads and at a constant rated speed of 1500 rev min^{-1}. The heat release rates, maximum rate of pressure rise, ignition delay, and combustion duration for Honge oil and Honge oil methyl ester (HOME) blended with diesel, to produce blends designated B20, B40 and B80, were determined. From the study, it was found that blends of HOME showed longer ignition delays when compared with diesel. However, the B20 blend exhibits a shorter ignition delay than the other blends. Peak pressure and maximum rate of pressure rise were highest for diesel followed by the B20 HOME blend. The peak pressures for B20, B40, B80 and B100 were measured at 78.50, 71.00, 69.25, 75.00 bar respectively compared with diesel which was 81.75 bar at full load. The combustion duration increased with an increase in the power output for all of the fuels. According to the investigators, the increase in power output was due to an increase in the quantity of fuel injected. Longer combustion duration was observed with HOME blends than for diesel due to the longer diffusion combustion phase.

Nadar and Reddy [63] modified a single cylinder, four stroke, diesel engine to work in dual fuel mode. To study the feasibility of using methyl ester of Mahua oil (MEMO) as pilot fuel, it was used as pilot fuel and liquefied petroleum gas was used as primary fuel. In dual fuel mode, pilot fuel quantity and injector opening pressure were varied as they affected the performance and emission of the engine. The values of injector opening

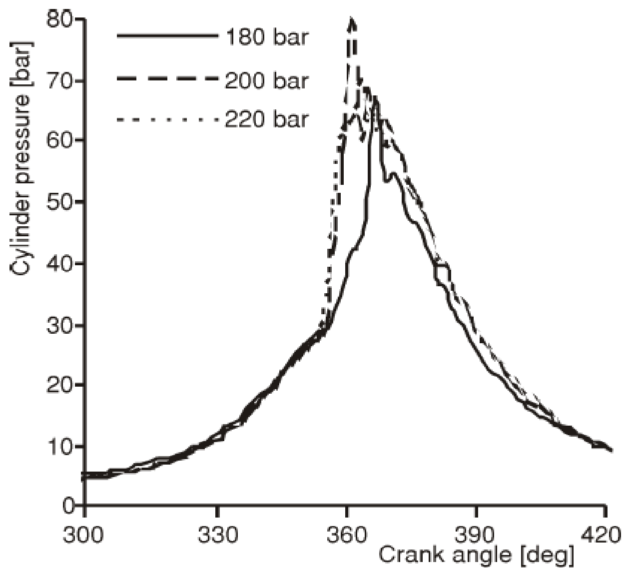


Figure 2.1 Pressure vs. Crank Angle at 3.88mg per cycle [63]

pressure selected were 180, 200 and 220 bar and pilot fuel quantity were 3.88, 5, 6.11 and 7.22 mg per cycle. The diesel engine was modified to work in the dual fuel mode by attaching an LPG line to the intake manifold. The pilot MEMO flow rate and LPG flow rate were varied by adjusting the fuel injection pump and the flow control valve, respectively. It was observed that the fuel injector opening pressure of 220 bar results in sudden pressure rise (Refer Figure 2.1). According to the investigators, the sudden rise in pressure was due to poor penetration of small quantity of pilot fuel, which results in accumulation of fuel nearer to the injector. The pilot fuel quantity of 5 mg per cycle resulted in a smooth pressure rise (Refer Figure 2.2).

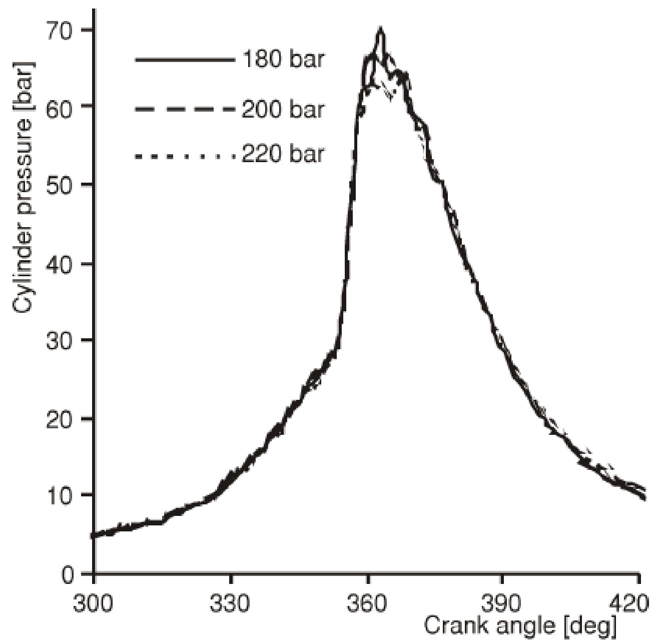


Figure 2.2 Pressure vs. Crank Angle at 5 mg per cycle [63]

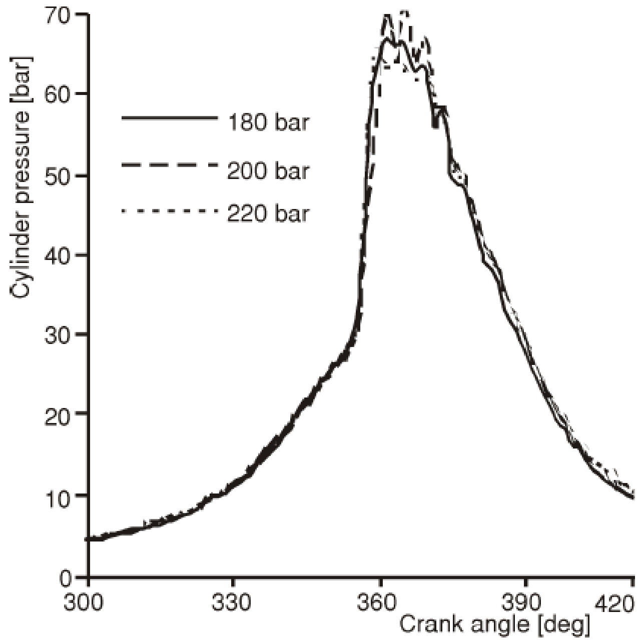


Figure 2.3 Pressure vs Crank Angle for 6.11 mg per cycle [63]

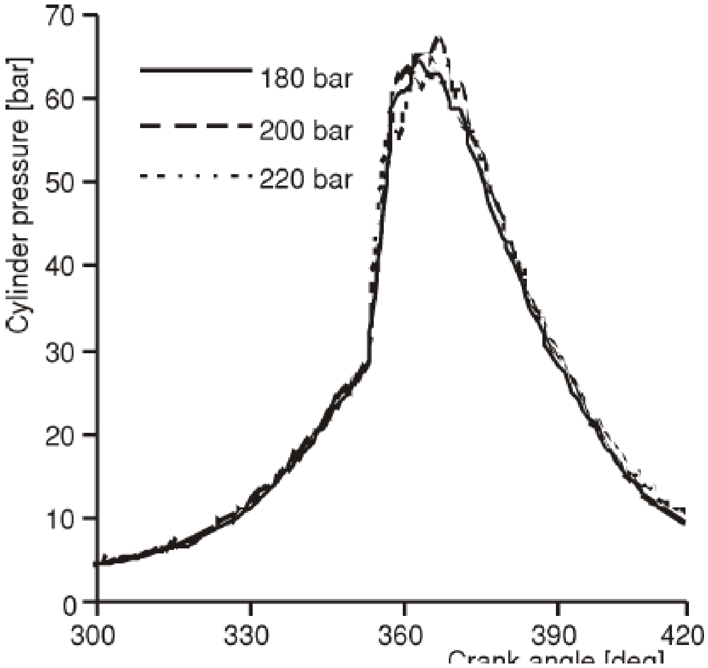


Figure 2.4 Pressure vs Crank Angle at 7.22 mg per cycle [63]

Devan and Mahalakshmi [64] analysed combustion of poon oil and poon oil based methyl ester (POME) in a single cylinder four stroke direct injection diesel engine. The blends were prepared with 20% Poon oil and 40% Poon oil methyl ester separately with standard diesel on volume basis. It was observed that the cylinder peak pressure for neat Poon oil and its blends with diesel was lower than that for standard diesel. This was attributed to more fuel-air mixture being burnt inside the cylinder before the peak pressure being achieved. However, the cylinder peak pressure at full load for Poon oil methyl ester and its diesel blend was higher than that of neat POME and its diesel blends. According to the investigators, this was due to the improvement in the preparation of air fuel mixture as a result of low fuel viscosity in case of POME – diesel blends. Standard diesel operation showed the highest peak pressure. Neat Poon oil did not show a pronounced heat release due to its higher viscosity which caused reduction in air entrainment and fuel/air mixing rates. For neat20 blend, the heat release was higher than that for neat Poon due to reduced viscosity and better spray formation. Thermal cracking of the double bond carbon chains during the injection process resulted in the breakdown unsaturated fatty acids of higher molecular weight compounds. These volatile compounds contributed to the better ignition quality of the vegetable oil despite the fact that vegetable oils had much higher viscosities than standard diesel.

Tashtoush et al. [82] conducted a study in which they used waste vegetable oil as a replacement to conventional diesel fuel in CI engines. They tested the performance and emission parameters of the engine by varying the air fuel ratio between10:1 to 20:1. About 100 litres of Waste Vegetable Oil (WVO) were collected from different local restaurants and brought into the labs of the Department of Applied Chemistry at Jordan University of Science and Technology (JUST) for treatment. The samples of WVO were decanted and then transesterified to produce the ethyl ester using acidic catalyst, HCl. All the esterified WVO samples used in this study were from used Palm oil since most local restaurants used palm oil.

In this study, it was observed that the combustion efficiency for both fuels (diesel and biodiesel) increased up to the stoichiometric A/F ratio (about 15) then started to decrease as the A/F ratio became leaner (Refer Figure 2.5). O_2 concentration in the case of biodiesel was higher at the two energy levels (Refer Figure 2.6). This is due to the addition of alcohol during the transesterification process in addition to the higher fuel oxygen present in biomass-based fuels including biodiesel. The general trend of the variation of O_2 with A/F ratio is in agreement with that of combustion efficiency as more O_2 is consumed with better combustion. The emission of CO is shown in figures which indicate that CO concentration declined with increasing A/F ratio for the two fuels at the two energy input levels.

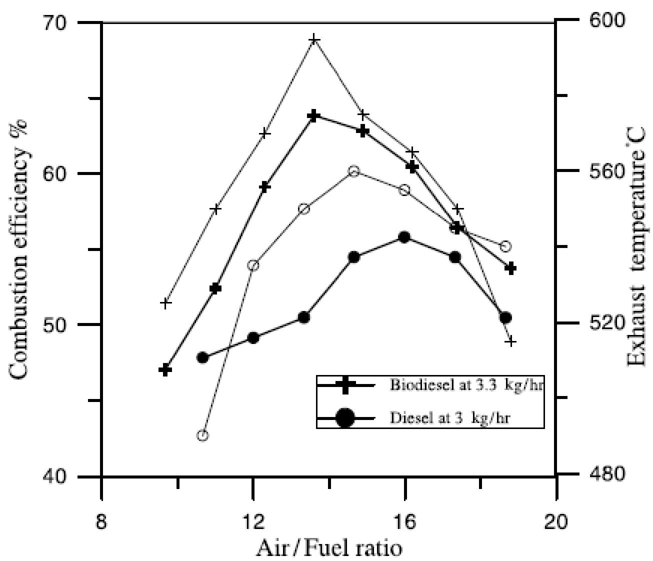


Figure 2.5 Variation of Combustion Efficiency and Exhaust Temperature with Air-Fuel Ratio [82]

It was also found that at the higher energy input, biodiesel combustion performance deteriorated and was inferior to diesel fuel due to its high viscosity, density and low volatility.

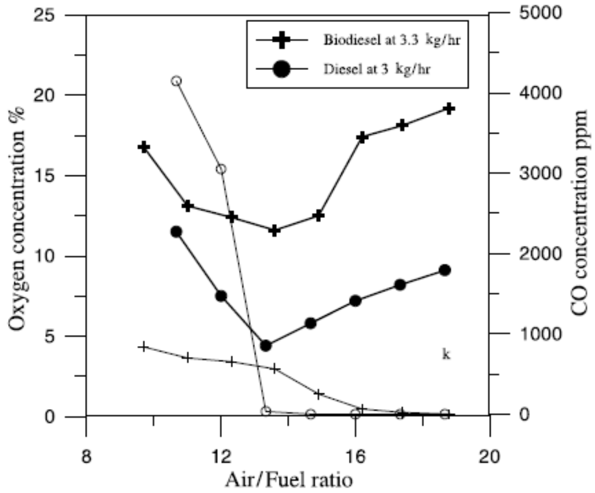


Figure 2.6 Variation of Oxygen Concentration and CO with Air-Fuel Ratio [82]

Tsolakis et al. [83] conducted the experiments on rape-seed oil on a diesel engine equipped with exhaust gas recirculation. The experiments were carried out on a Lister-Petter TR1 engine. The engine is a 773-cm^{3,} naturally aspirated, air cooled and single-cylinder direct injection diesel engine. The combustion of RME, B20 and B50 in the unmodified engine with pump–line–nozzle injection system resulted in advanced combustion compared to Ultra Low Sulphur Diesel (ULSD). The ignition delay was reduced while the initial uncontrolled premixed combustion phase (uncontrolled heat release phase) was increased. This resulted in increased cylinder pressure and temperature and hence early fuel ignition. The use of Exhaust Gas Recirculation (EGR) was more effective in the case of biodiesel blends combustion compared to ULSD combustion.

Canakci [84] tested biodiesel from soybean oil and compared the results with that of diesel in a turbocharged direct injection diesel engine. Commercially procured No. 1 and No. 2 diesel fuels tested as pure fuel (B100) and as 20% blend with No. 2 diesel fuel (B20). No. 2 diesel fuel had lesser cetane number than No. 1 diesel fuel. It was observed that for the B100, the start of combustion was earlier than for the No. 2 diesel fuel and this can be observed in the heat release rate profiles shown in Figure 2.7. B100 fuel was found to be start to burn at about 3.33^0 earlier than No. 2 diesel fuel while No. 1 diesel fuel was observed to start burning at about 0.25^0 later than No. 2 diesel fuel. It was found that the ignition delay for the B100 and No. 1 diesel fuel were shorter than for the No. 2 diesel fuel. B100 and No. 1 diesel fuel had about 1.06^0, 0.35^0 shorter ignition delay than No. 2 diesel fuel, respectively.

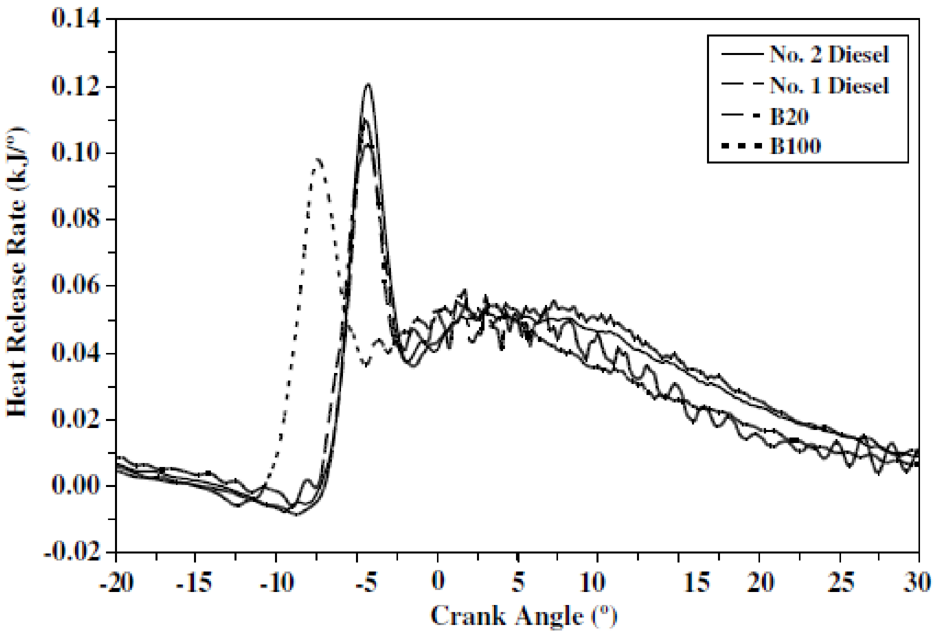


Figure 2.7 Comparison of Heat Release Rates for Different Fuels [84]

Gattamaneni et al. [85] used rice bran methyl ester(RBME) to conduct a study on combustion and emission performance on a 4.4 kW, constant speed, single cylinder, four-stroke, naturally aspirated, air-cooled, direct injection Kirloskar make diesel engine coupled to an electrical dynamometer for engine loading. They analyzed the performance and emission characteristics of Rice Bran Methyl Ester at variable loads whose properties were comparable to ASTM bio-diesel standards. It was observed that all the fuels exhibit a general trend of decrease in ignition delay with increase in load. According to the investigators, this was due to the rise in the cylinder gas temperature at the time of injection with increase in load. It was observed that at no load condition, the delay period for RBME decreased by 8% while its diesel blends showed almost 2% decrease for every 20% addition of RBME when compared to diesel. At 50 and 75% of the rated load, the delay period was decreased by 10 and 20%, respectively, for RBME compared to diesel. It was also observed that for every 20% addition of RBME in the RBME diesel blend the delay period decreased by almost 5% at all loads. This was attributed to the oxygen present in RBME and the breakdown of higher molecules of RBME to lower molecules of volatile compounds during injection and this advances the start of combustion resulting in decrease in ignition delay. The cylinder pressures for RBME and its diesel blends were found to be higher compared to diesel at all crank angles. This, according to the investigators, was due to earlier ignition of RBME and its diesel blends, which resulted in earlier start of combustion and hence higher pressure values compared to diesel. It was also observed that, the maximum rate of pressure rise decreased with increase of RBME in the fuel. This, according to the investigators, was a consequence of the decrease in ignition delay with increase in percentage of RBME in the fuel. They, further stated that reduced ignition delay implied, the quantity of accumulated fuel during ignition delay was lesser than during higher ignition delay. Hence, the pressure rise was not as drastic as in the case of diesel. Finally, they concluded that RBME and its diesel blends were suitable substitute for diesel as they showed satisfactory combustion and characteristics compared to diesel.

Sahoo and Das [86] undertook a study of combustion analysis of three different bio-diesels from non-edible oils namely Karanja, Jatropha and Polanga based bio-diesels for applications in generating sets and agricultural applications. Blends of B100, B20 and B50 were used at variable loads of 0%, 50% and 100% of full loads. The engine combustion parameters such as peak pressure, time of occurrence of peak pressure, heat release rate and ignition delay were computed. Combustion analysis revealed that neat Polanga bio-diesel that results in maximum peak cylinder pressure was the optimum fuel as far as the peak cylinder pressure was concerned. The ignition delays were consistently shorter for neat Jatropha bio-diesel, varying between 5.9^o and 4.2^o crank angles lower than diesel with the difference increasing with the load. Similarly, delays were shorter for neat Karanja and Polanga biodiesel when compared with diesel. Characterization of bio-diesels and effects of blends on cylinder pressure, heat release rate and ignition delay indicate that cylinder pressure is around 6.6 bar higher than HSD, delay is consistently shorter 4.2^o to 5.9^o crank angle and heat release is just closer to diesel and all bio-diesels considered above are quite suitable as fuels for diesel engine operations.

Buyukkaya [87] studied on the effect of biodiesel on the combustion characteristics of DI diesel engine. The biodiesel used for this purpose was the rapeseed oil methyl ester. It was found that ignition delay was shorter for neat rapeseed oil and its blends tested compared to that of standard diesel. The maximum heat release rates of standard diesel, B5, B20, B70 and B100 are 84, 79.7, 77.50, 74.9 and 72.2 J/ ^{0}CA, respectively. This was according to the author a consequence of the shorter ignition delay the premix combustion phase for neat rapeseed oil and its blends is less intense. The ignition delay slightly decreased with the use of biodiesels. According to the investigator, the chemical reactions during the injection of biodiesel at high temperature resulted in the breakdown of the high molecular weight esters. These complex chemical reactions led to the formation of gases of low molecular weight. Rapid gasification of this lighter oil in the fringe of the spray spreads out the jet, and thus volatile combustion compounds ignited earlier and reduced the delay period.

Table 2.7 compares the various studies conducted on combustion analysis using different biodiesels during the period from 2003 to 2010. The biodiesels include those made from Jatropha oil, Waste vegetable oil, Rapeseed oil, Soybean oil, Ricebran oil, Karanja oil, Mahua oil, Poon oil and Polanga oil. Most of the investigators have considered net heat release rate, cylinder pressure and ignition delay as parameters for analyzing the combustion characteristics.

Table 2.7 Summary of Studies On Combustion Analysis

Sr. No	Investigators	Year	Fuel	Combustion parameters studied	Remarks
1	Senthil Kumar et al.	2003	Jatropha oil blended with methanol	Peak cylinder pressure, Rate of pressure rise	Jatropha oil and its methyl ester showed longer ignition delays as compared to diesel. The peak pressure with ester and blend were found to be higher due to the improvement in the preparation of air fuel mixture as a result of lower viscosity.
2	Tashtoush et al.	2003	Waste vegetable oil	Combustion efficiency	It was observed that the combustion efficiency for both fuels (diesel and biodiesel) increased up to the stoichiometric A/F ratio (about 15) then started to decrease as the A/F ratio became leaner. It was also found that at the higher energy input, biodiesel combustion performance deteriorated and was inferior to diesel fuel due to its high viscosity, density and low volatility.
3	Tsolakis et al.	2007	Rapeseed oil methyl ester	Cylinder pressure, net heat release	With the use of biodiesel the ignition delay was reduced while the initial uncontrolled premixed combustion phase (uncontrolled heat release phase) was increased. This resulted in increased cylinder pressure and temperature and hence early fuel ignition.
4	Canakci	2007	Soybean oil methyl ester	Injection line pressure, Heat release rate	Biodiesel fuel was found to be start to burn at about 3.33^0 earlier than diesel fuel. It was also found that the ignition delay for the biodiesel was shorter than for diesel fuel.
5	Gattamaneni et al.	2008	Rice bran oil methyl ester	Ignition delay, peak pressure,	The cylinder pressures for RBME and its diesel blends were found to be higher compared to diesel at all crank angles. This, according to the

				rate of pressure rise, heat release rate,	investigators, was due to earlier ignition of RBME and its diesel blends, which resulted in earlier start of combustion and hence higher pressure values compared to diesel.
6	Banapurmath et al	2008	Karanja oil and its blends with ester	Ignition delay, Rate of pressure rise, Combustion duration, Heat release rate	Blends of Karanja methyl ester showed longer ignition delays when compared with diesel. Peak pressure and maximum rate of pressure rise were highest for diesel followed by the B20 blend
7	Nadar and Reddy	2008	Mahua methyl ester	Cylinder pressure	It was observed that the fuel injector opening pressure of 220 bar results in sudden pressure rise. According to the investigators, the sudden rise in pressure was due to poor penetration of small quantity of Mahua methyl ester, which results in accumulation of fuel nearer to the injector.
8	Devan and Mahalakshmi	2009	Poon oil methyl ester	Cylinder pressure,	It was observed that the cylinder peak pressure for neat Poon oil and its blends with diesel was lower than that for standard diesel. This was attributed to more fuel-air mixture being burnt inside the cylinder before the peak pressure being achieved.
9	Sahoo and Das	2009	Karanja, Jatropha and Polanga based bio-diesels	Cylinder pressure, heat release rate, ignition delay	Combustion analysis revealed that neat Polanga bio-diesel that results in maximum peak cylinder pressure was the optimum fuel as far as the peak cylinder pressure was concerned.
10	Buyukkaya	2010	Rapeseed oil methyl ester	Cylinder pressure, heat release rate	It was found that ignition delay was shorter for neat rapeseed oil and its blends tested compared to that of standard diesel. This was according to the author a consequence of the shorter ignition delay the premix combustion phase for neat rapeseed oil and its blends is less intense.

Among the 10 studies reported, only 2 are using Karanja biodiesel as fuel. In these studies Ignition delay, Rate of pressure rise, Combustion duration, Heat release rate and Cylinder pressure at constant compression ratio are only discussed.

From the exhaustive review carried out it is observed that no study on thermal performance and emission characteristics for Karanja biodiesel at different preset compression ratios with selected injection pressures appears to be reported. Further, it can also be noted that, no studies on combustion analysis using Karanja biodiesel as fuel wherein combustion parameters such as mass fraction burnt, net heat release rate, cumulative heat release rate, pressure volume plot, mean gas temperature and injection pressure are reported.

All studies made before the year 2000 are reviewed by A.S.Ramadhas et al. [96]. Only 11 studies refer to the thermal performance, emission characteristics of biodiesels as fuels on diesel engines at constant compression ratio. It appears from the open literature that the bulk of studies are conducted on the use of biodiesels as fuels only after 2000. Hence the studies conducted after 2000 have been focussed at present. Review related studies reported in the open literature are given in Appendix II.

2.4 Computational Study

Various computational based studies are also reported in the literature concerned with performance evaluation and emissions using pure diesel, biodiesel and biodiesel-pure diesel blends. They are either optimizsation studies using various techniques to optimise the thermal performance and emission constituents or using Artificial Neural Network (ANN) for modeling using experimental data to interpolate the performance and emission predictions. The studies related to multi-objective optimization are reviewed in Section 2.4.1 and that related to neural network modeling using ANN are reviewed in Section 2.4.2.

2.4.1 Optimization

Many techniques are available for optimization. Multi-objective optimization has gained momentum since the beginning of 21st century in almost all engineering fields. A review of literature related to optimization using multi-objective optimization applied to the thermal performance and emission characteristics of internal combustion engines show that there are only a few studies reported in general and diesel engines in particular. The studies are as follows:

Kesgin [88] developed a computer program which calculated the NO_x emissions with the input of engine speed and fuel data as well as the following data: zero dimensionally determined pressure, temperature, equivalence ratio, volume and mass depending on time in the burned zone, which are calculated by a two zone engine cycle simulation program. The validity of this program was verified by measurements from a turbocharged, lean-burn, natural gas engine. Using the results from this program, the effects of operational and design parameters of the engine were investigated. Then a wide range of engine parameters were optimised using a simple Genetic Algorithm regarding both efficiency and NO_x emissions. The results show an increase in efficiency as well as the amount of NO_x emissions being kept under the constraint value of 250

mg/Nm^3 in O_2 of 5%, which is the limit value for stationary internal combustion engines used in combined heat and power plants.

Shi [89] investigated optimal injection strategies for a heavy duty CI engine fuelled with diesel and gasoline like fuels. A CFD (Computational Fluid Dynamics) tool with detailed fuel chemistry was used to evaluate engine performance and pollutant emissions. The CFD tools feature a recently developed efficient chemistry solver that allowed the optimization tasks to be completed in practical computer times. A non-dominated sorting genetic algorithm II (NSGA II) was coupled with the CFD tool to seek optimal combinations of injection system variables to achieve clean and efficient combustion. This optimization study identified several key factors that play an important role in engine performance. It was found that the fuel volatility and reactivity play an important role in mid load condition, while at high load condition the performance of the engine is less sensitive to fuel reactivity. The study indicated that high volatility fuels such as gasoline and E10 are beneficial at high load giving good fuel economy.

Maheshwari et al. [90] reported the results of an experimental investigation of the performance of an IC engine fuelled with a Karanja biodiesel blends, followed by multi-objective optimization with respect to engine emissions and fuel economy, in order to determine the optimum biodiesel blend and injection timings complying with Bharat Stage II emission norms. Nonlinear regression was used to regress the experimentally obtained data to predict the brake thermal efficiency, NO_x, HC and smoke emissions based on injection timing, blend ratio and power output. AXUM software was used to determine the coefficients of the proposed functional relationship. To acquire the data, experimental studies had been conducted on a single cylinder, constant speed (1500 rpm), direct injection diesel engine under variable load conditions and injection timings for neat diesel and various Karanja biodiesel blends (5%, 10%, 15%, 20%, 50% and 100%). Retarding the injection timing for neat Karanja biodiesel resulted in an improved efficiency and lower HC emissions. The functional relationship developed between the correlating variables using nonlinear regression was able to predict the performance and emission characteristics with a correlation coefficient (R) in the range of 0.95-0.99 and very low root mean square errors. The outputs obtained using these functions were used to evaluate the multi-objective function of optimization process in the 0-20% blend range. The overall optimum was found to be 13% biodiesel-diesel blend with an injection timing of 24^0 BTDC, when equal weightage is given to emissions and efficiency.

A summary of the studies carried out on multi-objective optimization is shown in Table 2.8. It can be observed that only three (3) studies with multi-objective optimization are reported out of which two studies are based on Genetic Algorithm.

Table 2.8 Summary of Studies Carried on Multi-objective Optimization

Sr. No	Investigators	Year	Optimization technique used	Fuel	Remarks
1	Kesgin	2004	Genetic algorithm	Natural gas	Developed a computer program which calculated the NO_x emissions with the input of engine speed and fuel data
2	Shi	2010	Non-dominated sorting genetic algorithm II (NSGA II) was coupled with the CFD (Computational Fluid Dynamics) tool	Diesel and Gasoline	The CFD tools feature a recently developed efficient chemistry solver that allowed the optimization tasks to be completed in practical computer times. A Non-dominated sorting genetic algorithm II (NSGA II) was coupled with the CFD tool to seek optimal combinations of injection system variables to achieve clean and efficient combustion.
3	Maheshwari et al.	2011	Genetic algorithm	Karanja biodiesel and its blends with diesel	Nonlinear regression had been used to regress the experimentally obtained data to predict the brake thermal efficiency, NO_x, HC and smoke emissions based on injection timing, blend ratio and power output. AXUM software was used to determine the coefficients of the proposed functional relationship.

It can be observed from the table that only one (1) study **(Maheshwari et al. [105])** using Karanja biodiesel is conducted which involves optimizing the performance and emissions based on injection timing, blend ratio and power output.The engine used is at constant CR of 17.5 & IP of 205 bar.

2.4.2 Artificial Neural Network

ANN is used in predicting the output parameters of the engine with some input data available. It is a powerful modeling technique that investigators have employed in many engineering research studies. Few studies concerned with the application of ANN are reported predicting the thermal performance and emissions for conventional diesel engines. This section includes brief reviews of the studies related to ANN.

Najafi et al. [91] developed ANN based on their experimental work on an ethanol blended gasoline fuelled spark ignition engine. The results showed that the training algorithm of Back Propagation was sufficient for predicting engine torque, brake power, Brake Thermal Efficiency(BTE), volumetric efficiency, Specific Fuel Consumption(SFC)and exhaust gas components for different engine speeds and different fuel blend ratios. The ANN predictions for the brake power, engine torque, BTE, volumetric efficiency and BSFC yield a correlation coefficient (R) of 0.999, 0.995, 0.981, 0.985 and 0.986, respectively. It was found the root mean square error (RMSE) values were 0.74 kW, 0.49 N m, 0.59%, 0.48% and 0.003 kg/kWh for the brake power, engine torque, BTE, volumetric efficiency and BSFC, respectively. The ANN predictions for the brake power and torque yield a mean relative error (MRE) of 2.32% and 0.46% respectively. MRE of BTE was 1.28%. The ANN predictions for the volumetric efficiency yield an MRE of 0.48%. The ANN predictions for the BSFC yield MRE of 0.85%. The ANN predictions for the CO yield a correlation coefficient (R) of 0.994, RMSE of 0.18%V and MRE of 4.21%. Similarly R, RMSE, MRE for CO_2 emission were 0.985, 0.24%V and 1.54% respectively. The ANN predictions for the HC yield R of 0.987, RMSE of 5.41 ppm and MRE of 2.23%. The ANN predictions for the NO_x yield R of 0.973, RMSE of 89.85 pm and MRE of 5.57%. It was observed that the ANN model predicted engine performance and exhaust emissions with R in the range of 0.97−1. MRE values were in the range of 0.46−5.57%, while RMSE were found to be very low. There is a good correlation between the simulations from ANN and the measured data. Therefore, it was concluded that ANN is a useful method for simulating engine parameters.

Kiani et al. [92] modelled ANN to predict the engine brake power, output torque and exhaust emissions (CO, CO_2, NO_x and HC) of the engine. To acquire data for training and testing of the proposed ANN, a four-cylinder, four-stroke spark ignition engine was fuelled with ethanol-gasoline blended fuels with various percentages of ethanol (0, 5, 10,15 and 20%), and operated at different engine speeds and loads. An ANN model based on standard back propagation algorithm was developed using some of the experimental data for training. The back propagation neural networks (BPNN) were trained using the training sets formed by including 80 percent of data. After training, the BPNNs were tested using the testing datasets including 40 samples. There were three input and six output parameters in the experimental tests. The input variables are engine speed in rpm and the percentage of ethanol blending with the conventional gasoline fuel and engine load as percentage. The six outputs for evaluating engine performance include engine torque and brake power and emission parameters (CO, CO_2, HC, NO_x). The performance of the network can be evaluated by comparing the error obtained from converged neural network runs and the measured data. Error was calculated at the end of training and testing processes based on the differences between targeted and calculated outputs. The correlation coefficient between output and targets were all close to 1 which indicated that the performance can be predicted using this network. It was also observed that the ANN provided the best accuracy in modeling the emission indices with correlation coefficient of 0.98, 0.96, 0.90 and 0.71 for CO, CO_2, HC and NO_x, respectively.

Yusaf [93] used ANN modeling to predict brake power, torque, BSFC, and exhaust emissions of a diesel engine modified to operate with a combination of both compressed natural gas (CNG) and diesel fuels. The structure of ANN model used was as shown in the Fig. 2.8. Approximately 70% of the total experimental data (220 values) was selected at random and was used for training purpose, while the 30% was reserved for testing. The experimental data set for every output parameter includes 20 values, of which 14 values were used for training the network and six values were selected randomly to test the performance of the trained network. Simulations were performed using MATLAB. A multi-layer perception network (MLP) was used for non-linear mapping between the input and the output variable. To improve the modeling, several architectures were evaluated and trained using the experimental data. The back-propagation algorithm was utilized in training of all ANN models. This algorithm uses the supervised training technique where the network weights and biases are initialized randomly at the beginning of the training phase. The error minimization process is achieved using gradient descent rule.

The ANN predictions for the brake power, engine torque, brake thermal efficiency, brake specific fuel consumption and exhaust temperature yield a correlation coefficient (R) of 0.9808, 0.9884, 0.92897, 0.9838, and 0.9934 respectively.

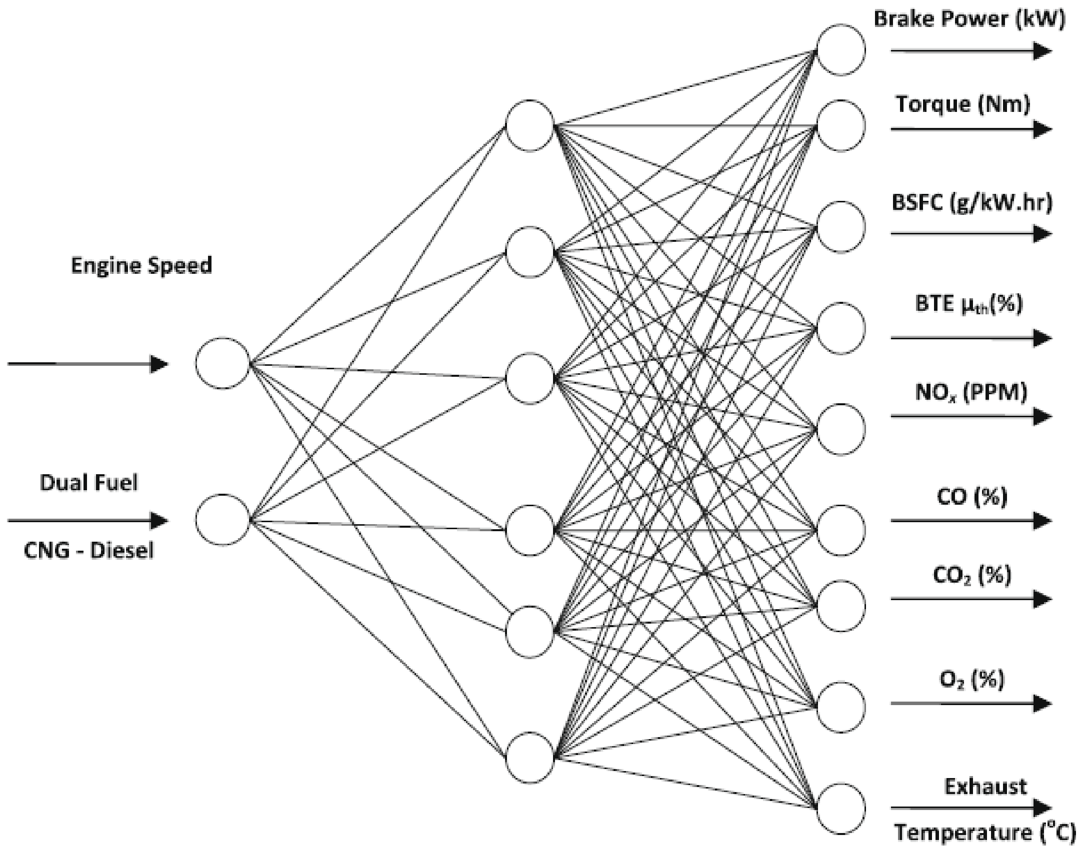


Figure 2.8 Structure of ANN for Dual Fuel (CNG-Diesel) Operated Diesel Engine [93]

Shivakumar et al. [94] used ANN was used to predict the engine performance and emission characteristics of an engine. Separate models were developed for performance parameters as well as emission characteristics. To train the network, compression ratio, injection timing, blend percentage, percentage load, were used as the input parameters where as engine performance parameters like BTE, BSEC, EGT ($T_{exh\ gas}$) were used as the output parameters for the performance model and engine exhaust emissions such as NO_x, smoke and (UBHC) values were used as the output parameters for the emission model. Out of 225 patterns, 70% (161 patterns) were used in the training set, 15% (32 patterns) in the validation set and remaining 15% (32 patterns) were employed for testing for both the models. For the performance and emission characteristics, the Mean Relative Error (MRE) was within 5% and 8% respectively which were found to be within the acceptable limits. For the combined model based on both the parameters, MRE was slightly more than the individual parameters and the accuracy of prediction slightly reduced. Hence, they suggested that individual models, rather than a combined model should be preferred. It was concluded that use of ANN reduces the experimental efforts and hence can serve as an effective tool for predicting the performance of the engine and emission characteristics under various operating conditions with different biodiesel blends.

Table 2.9 gives the summary of various studies conducted using ANN modeling.

Table 2.9 Summary Of Studies Carried Using ANN

Sr. No	Investigators	Year	Fuel	Engine type	(Fuel, engine type) Remarks
1	Najafi et al.	2009	Ethanol blended gasoline	Spark ignition engine with constant compression ratio	Developed ANN based on their experimental work on an ethanol blended gasoline fuelled spark ignition engine. The results showed that the training algorithm of Back Propagation was sufficient for predicting performance and emission parameters for different engine speeds and different fuel blend ratios.
2	Kiani et al.	2010	Ethanol blended gasoline	Spark ignition engine at constant compression ratio	An ANN model based on standard back propagation algorithm was developed using some of the experimental data for training. It was also observed that the ANN provided the best accuracy in modeling the emission indices with correlation coefficient of 0.98, 0.96, 0.90 and 0.71 for CO, CO_2, HC and NO_x, respectively.
3	Yusaf	2010	Compressed Natural gas	Diesel Engine at constant compression ratio	The back-propagation algorithm was utilized in training of all ANN models. This algorithm uses the supervised training technique where the network weights and biases are initialized randomly at the beginning of the training phase. The error minimization process is achieved using gradient descent rule.
4	Shivakumar et al.	2011	Biodiesel from Waste cooking oil	Diesel engine at variable compression ratio	Separate models were developed for performance parameters as well as emission characteristics.

Only few investigators seem to have developed neural networks based on their data obtained during experimental work. It appears that the field of ANN has emerged as a field of research in recent years and hence very few studies of its application in compression ignition engines are reported. Among four (4) studies reported, only one (1) study is conducted on a variable compression ratio diesel engine using biodiesel made from waste cooking oil as fuel.

It can be observed that very few computational studies involving multi-objective optimization and ANN are reported. From the review of various optimization studies it is identified that no study using Karanja biodiesel is reported where optimization is done based on compression ratio, injection pressure, load and blend. Also, according to the review on studies of ANN, no modeling using ANN for Karanja biodiesel fuelled diesel engine operated at different preset compression ratios and at different injection pressures appears to have been reported so far.

2.5 Scope of Present Study

From a systematic and exhaustive review of the earlier studies the scope of the present study is identified as follows. A large number of experimental studies are reported as seen in Section 2.3.1 to study the thermal performance of biodiesels used in diesel engines operated at constant compression ratio and constant injection pressure. In these studies the effect of load, blend, speed, injection timing, brake power, etc on thermal performance and emission characteristics are studied. In Section 2.3.2, studies related to combustion analysis using different biodiesels where combustion parameters like Ignition delay, Rate of pressure rise, Combustion duration, Heat release rate, Cylinder pressure, Combustion efficiency, Injection line pressure are presented. The fields of multi-objective optimization and ANN modeling on IC engine performance have emerged as areas of research in recent times. It is found that very few studies are reported in these areas.

The studies conducted on thermal performance, emissions characteristics, combustion analysis, multi-objective optimization using Karanja biodiesel on a diesel engine are listed in Table 2.10.

Table 2.10 Various types of studies using Karanja biodiesel at constant compression ratio

Sr. No	Investigators	Year	Fuel	Nature of work	Parameters varied / combustion parameters studied / optimized parameters
1	Raheman & Phadatare	2004	Karanja biodiesel	Thermal performance and Emission Characteristics	Load, Blend
2	Banapurmath et al.	2008	Karanja, Jatropha	Thermal performance	Brake power, Blend,

			and sesame oil methyl esters	and Emission Characteristics	Injection Timing
3	Sureshkumar et al.	2008	Karanja biodiesel	Thermal performance and Emission Characteristics	Load, Blend
4	Banapurmath et al.	2008	Karanja oil and its biodiesel with producer gas	Thermal performance and Emission Characteristics	Brake power, Blend, Injection Timing
5	Banapurmath and Tewari	2008	Karanja oil, its biodiesel and its blends with diesel	Thermal performance and Emission Characteristics	Brake power, Blend, Injection Timing
6	Bajpai et al.	2009	Karanja oil and its blends with diesel	Thermal performance and Emissions Characteristics	Blend, Load
7	Murugu Mohan Kumar Kandasamy, Mohanraj Thangavelu	2009	Rice bran and Karanja oil	Thermal performance and Emission Characteristics	Load, Blend
8	Baiju et al.	2009	Methyl and ethyl esters of Karanja oil	Thermal performance and Emission Characteristics	Load, Blend
9	Haldar et al.	2009	Oils of Putranjiva, Jatropha and Karanja	Thermal performance and Emission Characteristics	Load, Blend, Injection timing
10	Banapurmath et al	2008	Karanja oil and its blends with ester	Combustion analysis	Ignition delay, Rate of pressure rise, Combustion duration, Heat release rate
11	Sahoo and Das	2009	Karanja, Jatropha and Polanga based bio-diesels	Combustion analysis	Cylinder pressure, heat release rate, ignition delay
12	Maheshwari et al.	2011	Karanja biodiesel and its blends with diesel	Multi Objective Optimization using Genetic Algorithm	Injection timing, blend, power output
13	---	---	---	ANN	---

The following observations can be made from Table 2.10,

- It is clear that no studies on thermal performance and emission characteristics at different preset compression ratios and at different injection pressures using Karanja biodiesel as fuel are reported, although many studies have been conducted at a constant compression ratio.
- There is no study reported with Karanja biodiesel as fuel where combustion parameters like mass fraction burnt, net heat release rate, cumulative heat release rate, pressure volume plot, mean gas temperature and injection pressure are considered.
- It can be observed that in multi-objective optimization using genetic algorithm, only one (1) study using Karanja biodiesel is conducted which involves the optimization of performance and emissions based on injection timing, blend ratio

and power output. There appears to be no study reported where optimization is done for compression ratio and injection pressure.

- No neural network is modelled for Karanja biodiesel fuelled variable compression ratio CI engine.

Hence, the objectives of the proposed study are as under:

1. To experimentally evaluate the thermal performance and the exhaust gas emission characteristics of a diesel engine fuelled with Karanja biodiesel and its blends with diesel at different preset compression ratios and at different injection pressures in addition to varying loads.
2. To analyze the combustion parameters namely mass fraction burnt, net heat release rate, cumulative heat release rate, pressure volume plot, mean gas temperature, injection pressure in addition to rate of pressure rise and cylinder pressure.
3. To carry out multi-objective optimization on thermal performance and engine exhaust emissions based on compression ratio, load, injection pressure, blend using genetic algorithm technique to find the optimum operating conditions.
4. To model an ANN for predicting the engine performance and exhaust emissions of the variable compression ratio diesel engine used for the experimentation.

A flowchart depicting the proposed experimental and computational study is given in Figure 2.9.

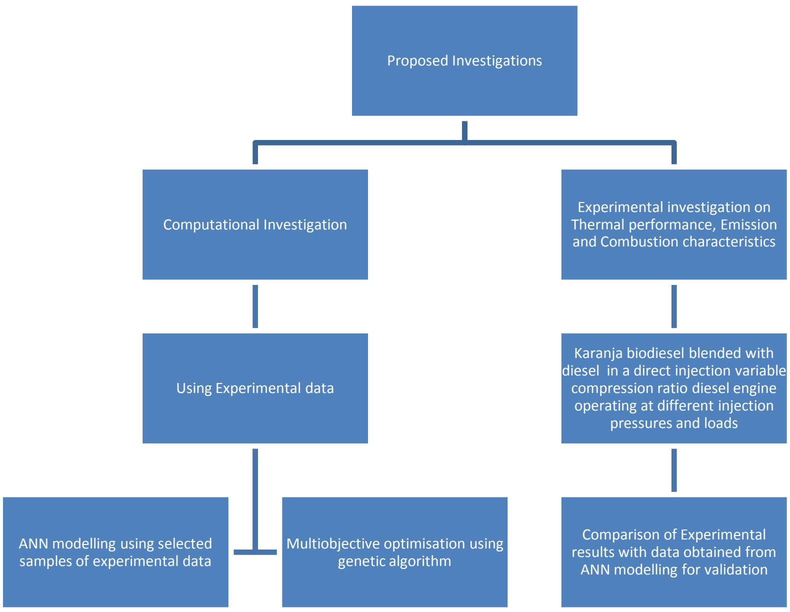


Figure 2.9 Flow chart for the proposed study

The proposed study consists of two parts.

1. Experimental study
2. Computational study

The experimental study is to be conducted on a four stroke, variable compression ratio diesel engine using Karanja biodiesel and its blends with diesel. The thermal performance and emissions characteristics are to be evaluated by running the engine at different preset compression ratios, different injection pressures and varying loads. The experimental study also consists of combustion analysis of the engine running on pure diesel and Karanja biodiesel at different preset compression ratios and full load.

The computational study consists of multi-objective optimization of thermal performance and emission characteristics and modeling an ANN for the engine. The multi-objective optimization is to be conducted by using genetic algorithm technique in order to get optimum performance and emissions of the engine when fuelled with blends of Karanja biodiesel and diesel. The ANN is modelled by using a set of selected results of the experimental study. The accuracy with which this neural network works will be judged by comparing the outputs from the network with the experimental data.

Chapter 3

Experimental Test Rig

An experimental test rig is developed to undertake the thermal performance evaluation and emission characteristics evaluation of a variable compression ratio compression ignition engine fuelled with Karanja biodiesel and its blends with Diesel oil. The experimental test rig is suitably developed to conduct various test runs under different working conditions to evaluate the thermal performance and emission constituents of a bio-diesel run engine in comparison with that of a conventional diesel operated engine. This chapter is organized in two sections. Section 3.1 describes the hardware used in the main experimental test rig. Various measurement systems incorporating sensors and instruments are described in Section 3.2.

3.1 Description of Experimental Test Rig

The experimental test rig consists of a variable compression ratio compression ignition engine, eddy current dynamometer as loading system, fuel supply system for both Diesel oil oil supply and biodiesel supply, water cooling system, lubrication system and various sensors and instruments integrated with computerized data acquisition system for online measurement of load, air and fuel flow rate, instantaneous cylinder pressure, injection pressure, position of crank angle, exhaust emissions and smoke opacity. Plate 3.1 is the photographic image of the experimental setup used in the laboratory to conduct the present study and Figure. 3.1 represents the schematic representation of the experimental test setup. Table 3.1 gives the technical specifications of different components used in the test rig. The setup enables the evaluation of thermal performance and emission constituents of the VCR engine. The thermal performance parameters include brake power, brake mean effective pressure, brake thermal efficiency, volumetric efficiency, brake specific fuel consumption, exhaust gas temperature, heat equivalent of brake power and heat equivalent of exhaust gas. Commercially available labview based Engine Performance Analysis software package “EnginesoftLV” is used for on line performance evaluation. The exhaust emissions of the engine are analysed using an exhaust gas analyser. The constituents of the exhaust gas measured are CO (% and ppm), CO_2 (%), ,

O_2 (%), HC (ppm), NO_x (ppm) and SO_x (ppm). The smoke intensity is measured in terms of Hartridge Smoke Unit (HSU in %) /K(the light absorption coefficient (m^{-1}).

Table 3.1 Technical Specifications of Experimental Test Rig

Engine	Type - single cylinder, four stroke Diesel, water cooled, rated power 3.5 kW at 1500 rpm, stroke 110 mm, bore 87.5 mm. 661 cc, CR 17.5, Modified to VCR engine CR range 12 to 18
Dynamometer	Type eddy current, water cooled, with loading unit
Piezo sensor	Range 5000 psi, with low noise cable
Crank angle sensor	Resolution 1 Deg, Speed 5500 RPM with TDC pulse.
Data acquisition device	NI USB-6210, 16-bit, 250kS/s.
Piezo powering unit	Make-Cuadra, Model AX-409.
Temperature sensor	Type RTD, PT100 and Thermocouple, Type K
Temperature transmitter	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
Load indicator	Digital, Range 0-50 Kg, Supply 230VAC
Load sensor	Load cell, type strain gauge, range 0-50 Kg
Fuel flow transmitter	DP transmitter, Range 0-500 mm WC
Air flow transmitter	Pressure transmitter, Range (-) 250 mm WC
Software	“EnginesoftLV” Engine performance analysis software
Rotameter	Engine cooling 40-400 LPH; Calorimeter 25-250 LPH
Exhaust gas analyzer	Make – Indus Scientific, Five gas analyzer
Smoke meter	Make – Indus Scientific, Range – 0 to 100% HSU

Plate 3.1 Experimental Test Rig

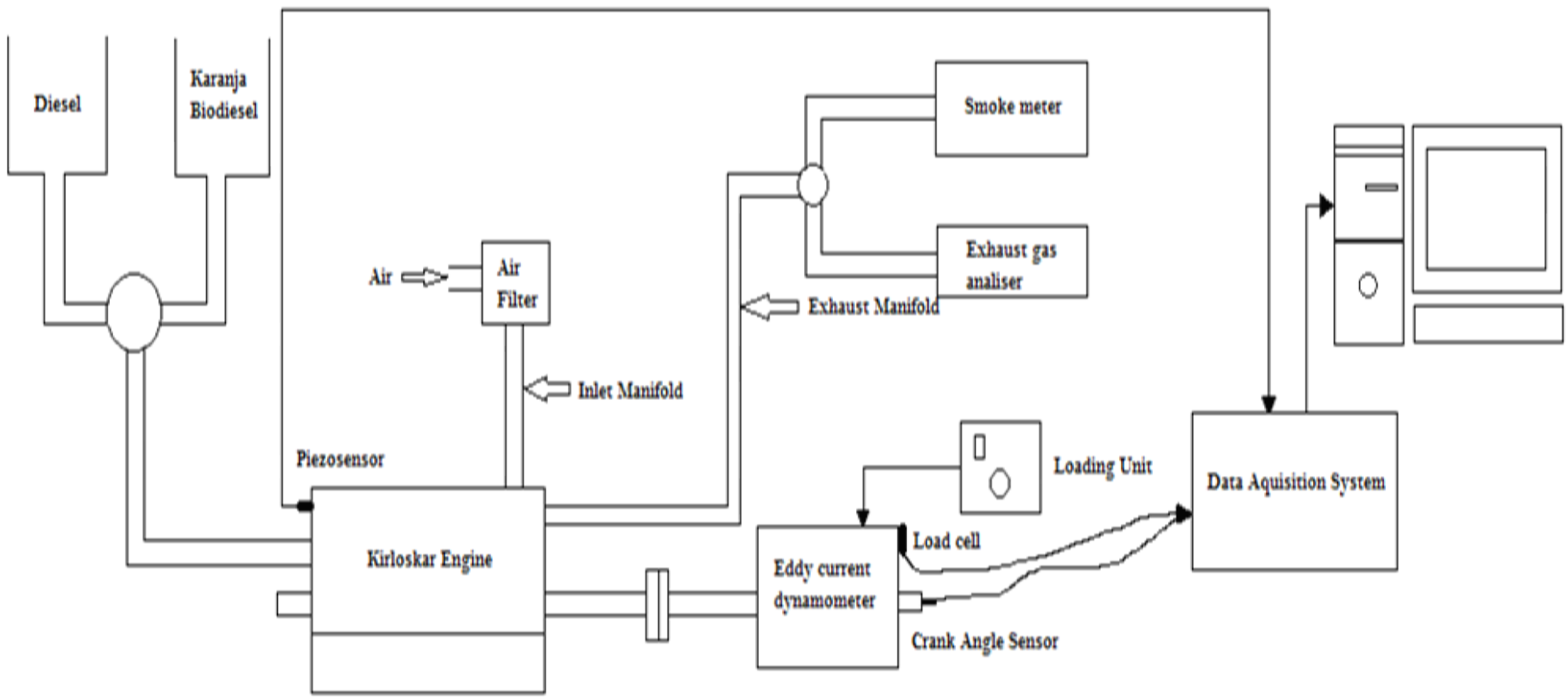


Figure. 3.1 Schematic Diagram of Experimental Test Rig

3.1.1 Variable Compression Ratio (VCR) Diesel Engine

The variable compression ratio (VCR) diesel engine used to conduct the experiments is a single cylinder, four stroke, water cooled, direct injection engine. The technical specifications of the engine are given in Table 3.2. The engine is mounted on a stationary frame with a suitable cooling system. The lubricating system is inbuilt in the engine. Two different photographic images of the engine are given in Plates 3.2 and 3.3.

Table 3.2 Technical Specifications of VCR Diesel Engine

Make & Model	Kirloskar Oil Engine, TV1
Type	Four stroke, Water cooled, Diesel
Number of cylinder	One
Bore and Stroke	87.5 mm and 110 mm
Combustion principle	Compression ignition
Cubic capacity	0.661 litres
Compression ratio	17.5 :1 (modified to work at 12, 13, 14, 15, 16, 17.5 and 18 compression ratios)
Peak cylinder pressure	77.5 kg/cm^2
Direction of rotation	Clockwise
Maximum speed	2000 rpm
Minimum idle speed	750 rpm
Minimum operating speed	1200 rpm
Fuel injection timing for standard engine	23^0 BTDC
Valve clearances at inlet and exhaust	0.18 mm and 0.20 mm
Bumping clearance	0.046”-0.052”
Lubricating system	Force feed system
Power rating 1. Continuous 2. Intermittent	 7/1500 hp/rpm 7.7/1500 hp/rpm
Brake mean effective pressure at 1500	6.35 kg/cm^2

rpm	
Lubricating oil pump	Gear type
Lubricating oil pump delivery	6.50 lit/min
Sump capacity	2.70 litres
Lubricating oil sump	1.5% normally exceed of fuel
Connecting rod length	234 mm
Overall dimensions	617L × 504W × 877H
Weight	160 kgs

3.1.2 Compression Ratio Setting

The engine with fixed compression ratio can be modified by providing additional variable combustion space. There are different arrangements by which this can be achieved. Tilting cylinder block method is one of the arrangements which can be used to vary the combustion space volume. A photographic image of the tilting cylinder block installed on the engine cylinder is given in Plate 3.4. The engine is made to operate as a variable compression ratio (VCR) engine by providing a tilting block arrangement to suitably change the compression ratio (CR) to the desired value in the given range without stopping the engine and without altering the combustion chamber geometry.

The tilting cylinder block arrangement consists of a tilting block with six allen bolts, a compression ratio adjuster with lock nut and compression ratio indicator. For setting a chosen compression ratio, the allen bolts are to be slightly loosened (refer Plate 3.5). Then, the lock nut on the adjuster is to be loosened and the adjuster is to be rotated to set a chosen compression ratio by referring to the compression ratio indicator and to be locked using lock nut. Finally all the allen bolts are to be tightened gently. The compression ratios considered for conducting the experiments are 14, 15, 16, 17, 17.5 and 18. Due to rough running of the engine and greater vibration, the compression ratio below 14 is not set though there is a provision to set the CR value up to 12.

The basic principle of the tilting cylinder block assembly is as shown in Figure 3.2 and Plate 3.5. When the CR is to be reduced the block is tilted so that the clearance volume increases and swept volume remains a constant (indicated by red colour in the Figure 3.2).

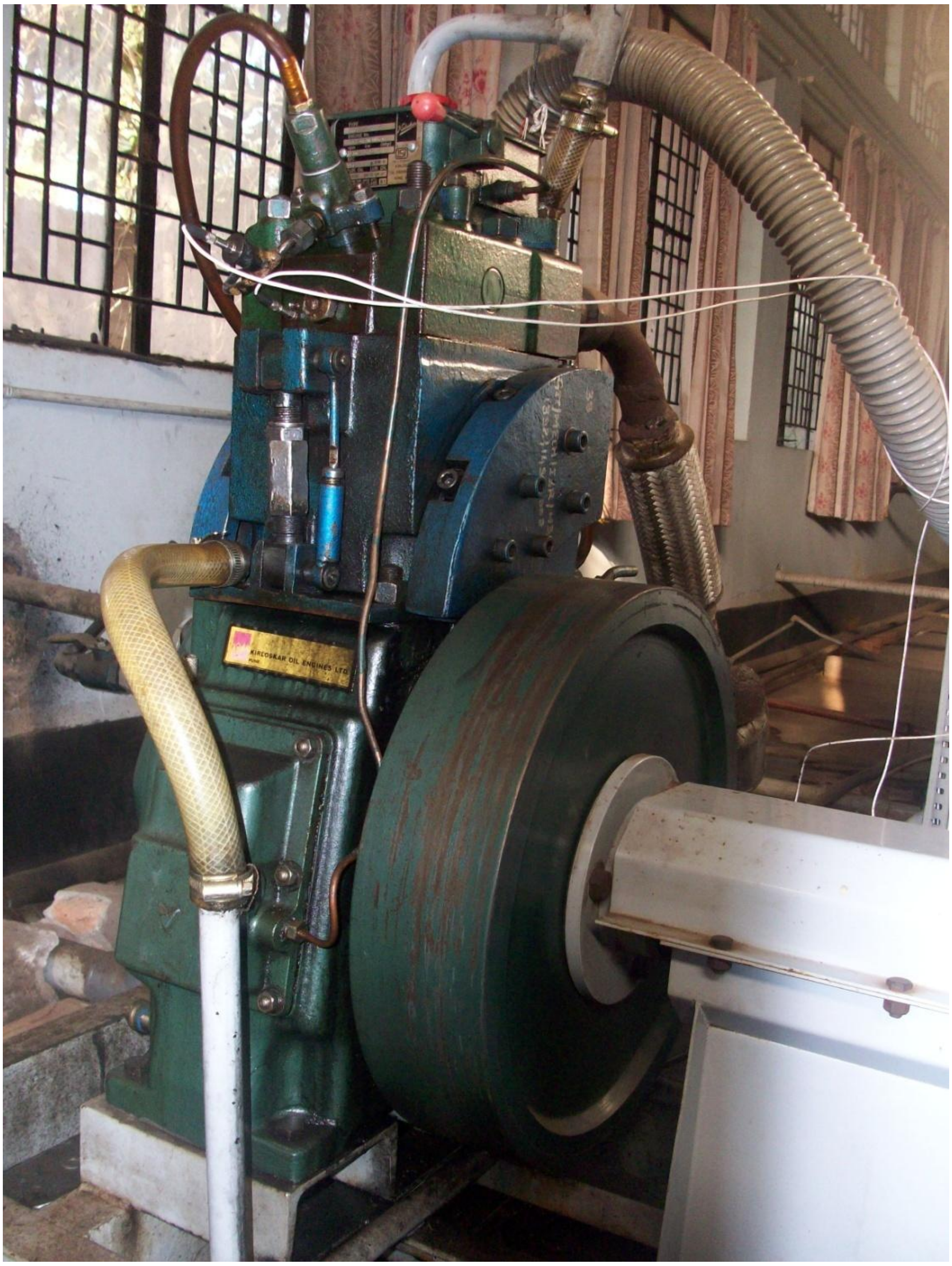


Plate 3.2 Variable Compression Ratio Diesel Engine

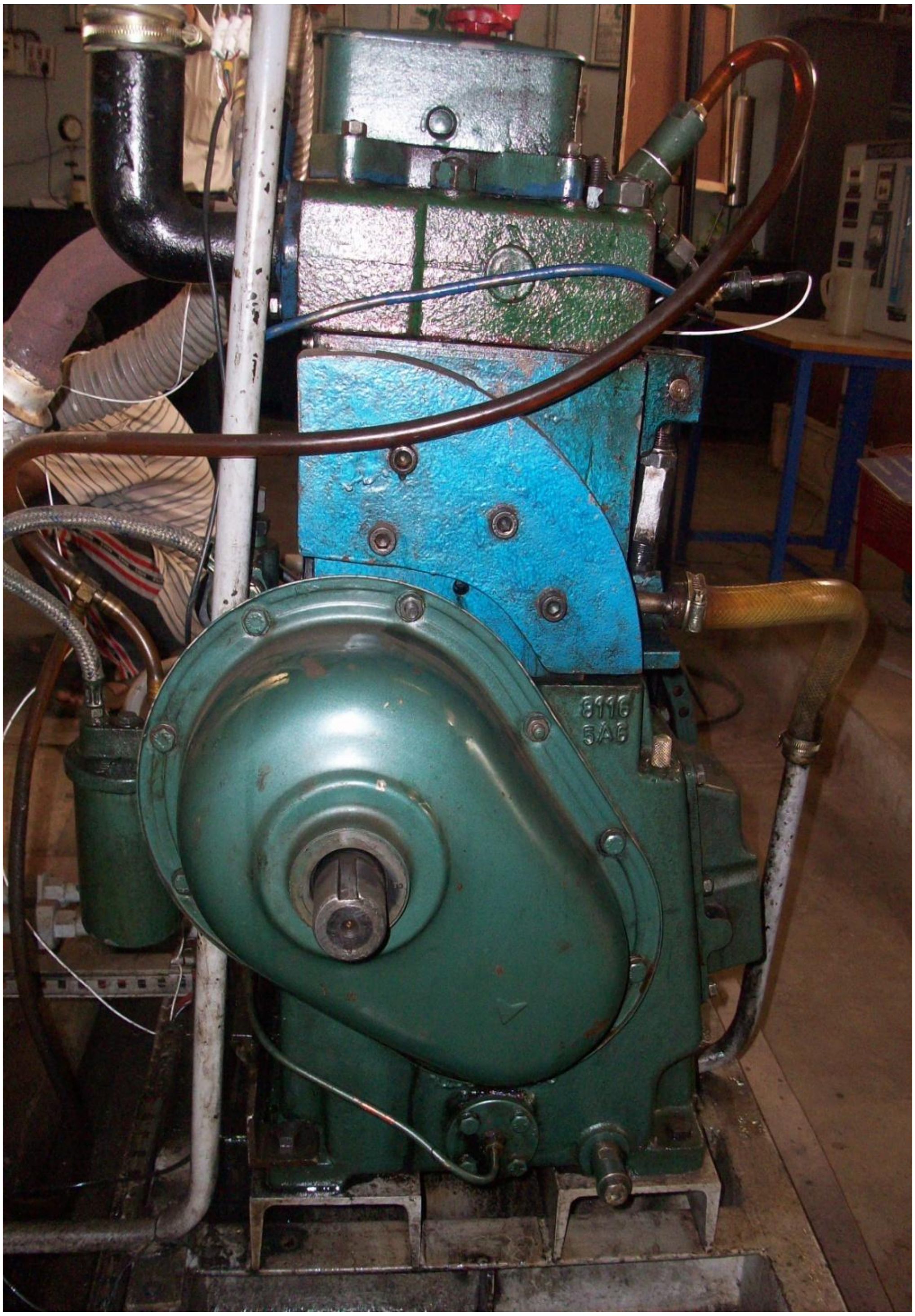


Plate 3.3 Rear View of the Engine

Figure 3.2 Principle of Tilting Cylinder Block Assembly

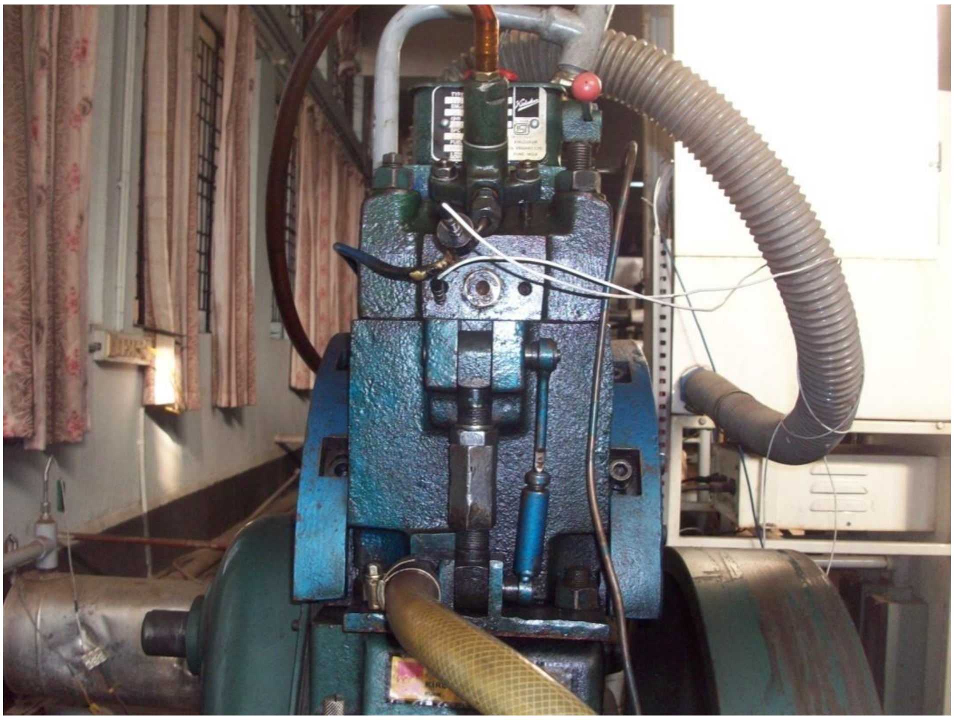


Plate 3.4 Tilting Cylinder Block Arrangement

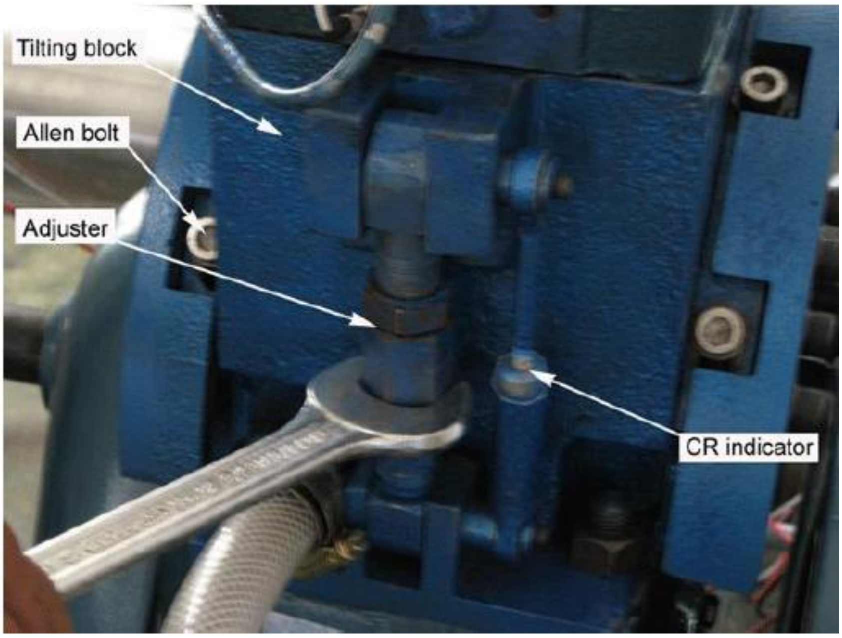


Plate 3.5 Compression Ratio Setting

3.1.3 Fuel Injection Pump

The fuel injection pump manufactured by MICO BOSCH is used for injecting Diesel oil or bio-diesel in to the VCR engine (Plate 3.6). The fuel injection pump is operated by the cam shaft and the fuel injection timing can be varied by adding/removing shims placed beneath the pump and pump mounting bracket.

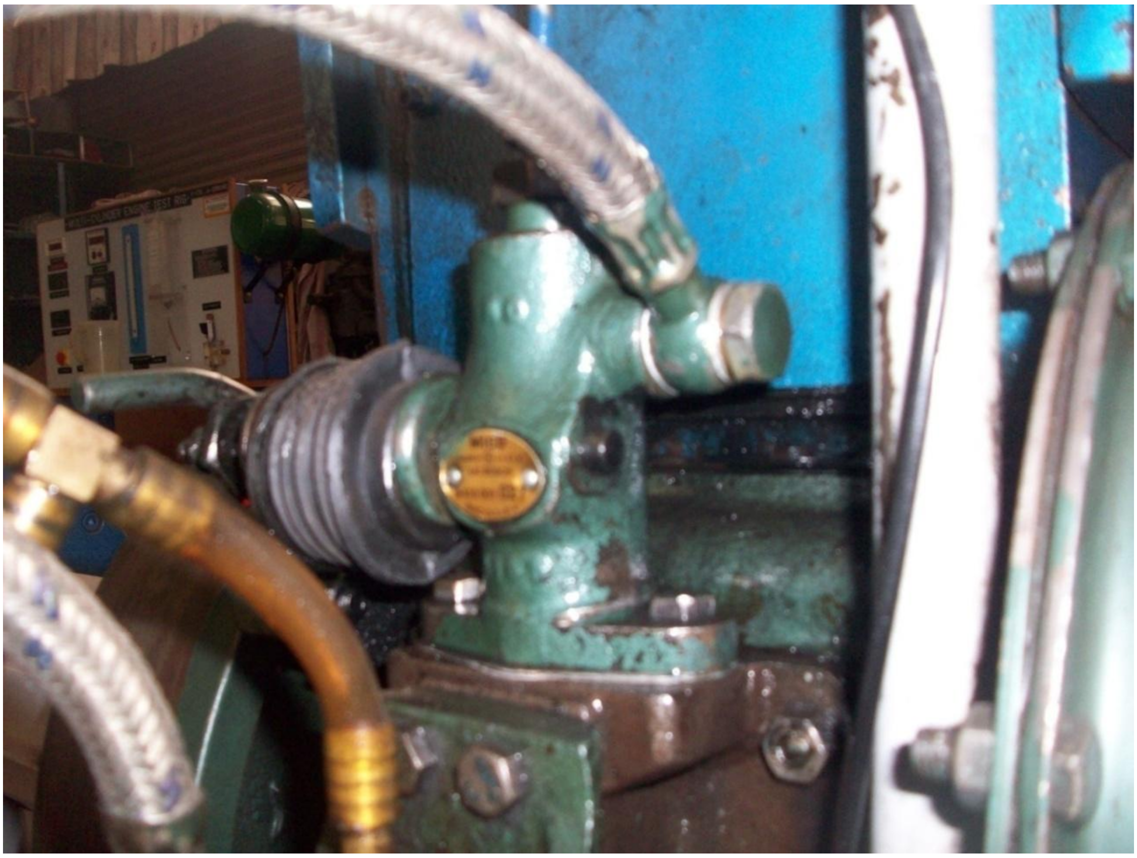


Plate 3.6 Fuel Injection Pump

3.2 Measurement Systems

Various measurement systems used to capture the experimental data used in the test rig are load measurement system, fuel injection pressure measurement system, cylinder pressure measurement system, emission measurement system and data acquisition system.

3.2.1 Load Measurement System

The experimental study is conducted at various loads and hence an accurate and reliable load measuring system is a must. The load measuring system of this experimental test rig consists of a dynamometer of eddy current type, a load cell of strain gauge type and a loading unit. The load is applied by supplying current to the dynamometer using a loading unit. The load applied to the engine is measured by a load cell. The dynamometer, load cell and loading unit are discussed in the following paragraphs.

A dynamometer is a device which is used for measuring force, torque or power produced by an engine. It can also be used to apply load or torque on the engine. The dynamometer used in this study is an eddy current type with a water cooling system. The eddy current dynamometers provide an advantage of quicker rate of load change for rapid load setting. The VCR diesel engine is directly coupled to the eddy current dynamometer with a loading unit using which desired loads up to 12kg can be applied. The load measurement is made using a strain gauge load cell and the speed measurement is done using a shaft mounted on a crank angle sensor. Plate 3.7 shows the image of the eddy current dynamometer used in this study. Plate 3.8 depicts the assembly of the eddy current dynamometer and the engine. The technical specifications of the dynamometer are given in Table 3.3.

The eddy current dynamometer unit basically comprises of a rotor, shaft, bearings, casing and bed plate. The rotor is mounted on the shaft which runs in the bearings. The bearings rotate within the casing supported in ball bearing trunnions which form a part of the bed plate of the machine. Inside the casing, there are two field coils connected in series. When a direct current is supplied to these coils using a loading unit, a magnetic field is created in the casing across the air gap on either side of the rotor. The details of the loading unit are given later in this section.

Plate 3.7 Eddy Current Dynamometer

Plate 3.8 Assembly of Eddy Current Dynamometer and Engine

Table 3.3 Technical Specifications of Eddy Current Dynamometer

Model	AG-10
Make	SAJ Test Plant Pvt. Ltd
End flanges both side	Carbon shaft model 1260 type A
Water inlet	1.6bar
Minimum kPa	160
Pressure lbf/in^2	23
Air gap mm	0.77/0.63
Torque Nm	11.5
Hot coil voltage max	60
Continuous current amps	5
Cold resistance ohms	9.8
Speed max.	10000 rpm
Load	3.5 kg
Bolt size	M12×1.75
Weight	130 kg
Arm Length mm	185

When the rotor turns in this magnetic field, eddy current gets induced creating a braking effect between the rotor and the casing. The rotational torque exerted on the casing is measured by a strain gauge load cell incorporated in the restraining linkage between the casing and the dynamometer. To prevent over heating of the dynamometer, cooling water is circulated with the help of a water pump through the cooling passages of the casing. Water passes from the inlet to the casing through a flexible connection permitting movement of the casing.

Water passes through loss (Grooved) plates in the casing positioned on either side of the rotor and absorbs the heat generated. Heated water discharges from the casing through a flexible connection to an outlet flange on the bed plate. Different components connected to the eddy current dynamometer are shown in Plate 3.9.

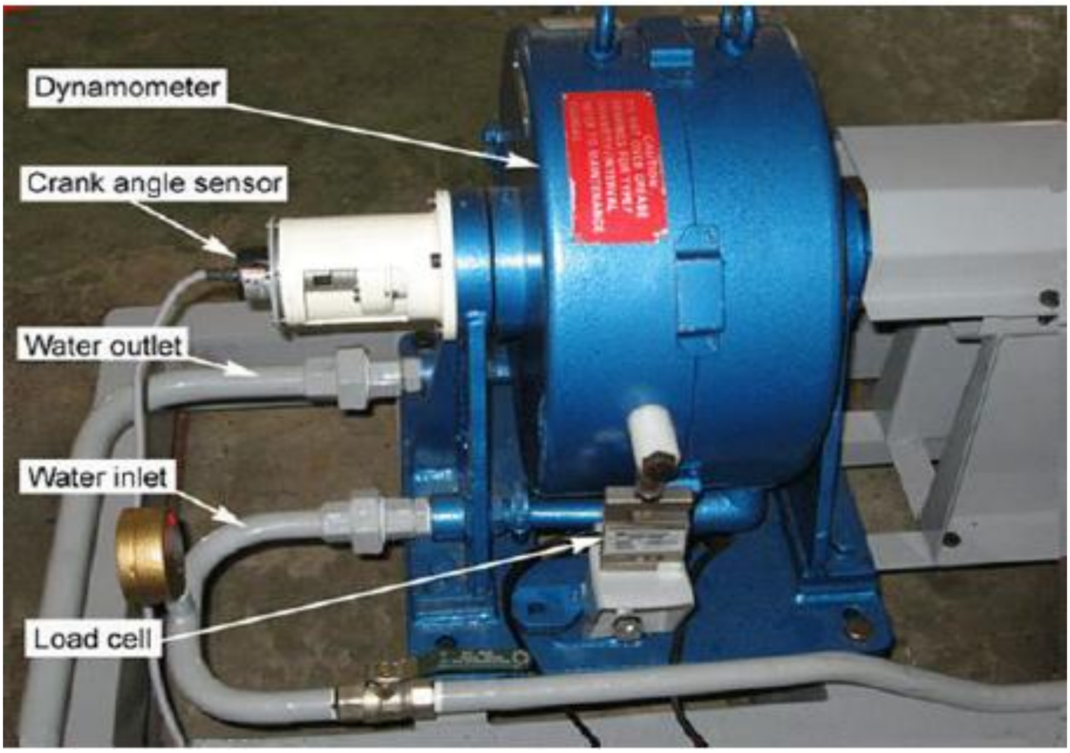


Plate 3.9 Components Connected to the Eddy Current Dynamometer

A load cell is a transducer that is used to convert a mechanical signal (force) into an analogous electrical signal. This conversion of the signals from the mechanical to electrical is indirect and happens in two stages. A mechanical arrangement is provided due to which the force being applied deforms a strain gauge. The strain gauge consisting of an electrical wire measures the deformation (strain) as an electrical signal, because the strain changes the effective electrical resistance of the wire. A load cell usually consists of four strain gauges in a Wheatstone bridge configuration. The electrical signal output is typically in the order of a few millivolts and requires amplification by an instrumentation amplifier before it can be used. The output of the transducer is plugged into an algorithm to calculate the force applied to the transducer. A photographic image of the load cell used and its technical specifications are given in Plate 3.10 and Table 3.4 respectively.

Plate 3.10 Load Cell

Table 3.4 Technical Specifications of the Load Cell Used

Make	Sensotronics
Model	60001
Type	S – Beam Universal
Capacity	0 – 50 kg
Mounting thread	M10 × 1.25 mm
Full scale output (mV/V)	3.00
Tolerance on output (FSO)	+/-0.25%
Zero balance (FSO)	+/-0.1mV/V
Non-linearity (FSO)	<+/-0.025%
Hysteresis (FSO)	<+/-0.020%
Non-repeatability	<+/-0.010%
Creep (FSO) in 30 min	<+/-0.020%

Operating temperature range	-20 ^{0}C to +70 ^{0}C
Rated excitation	10V AC/DC
Maximum excitation	15V AC/DC
Bridge resistance	350 Ohms (Nominal)
Insulation resistance	>1000 Meg ohm @ 50VDC
Span / ^{0}C (of load)	+/-0.001%
Zero / ^{0}C (of FSO)	+/-0.002%
Combined error (FSO)	<+/-0.025%
Safe overload (FSO)	150%
Ultimate overload (FSO)	300%
Protection class	IP 67
Overall dimensions	51 L x 20 W x 76 H mm
Weight	380 gm

The loading unit consists of a dimmerstat to control the magnitude of the direct current flowing into the dynamometer and a switch to ON/OFF the loading unit. The current is supplied into the loading unit through the main power supply. The loading unit used in this experimental test rig is of make Apex, Model AX-155 and constant speed type. The load values used to conduct the experiment are 0kg, 3kg, 6kg, 9kg and 12kg which correspond to the torque values of 0Nm, 5Nm, 10Nm, 15Nm and 20Nm respectively. The loading unit used in this experiment is shown in Plate 3.11. The assembly of the same into the engine panel box can be seen in Plate 3.12.

Plate 3.11 Dynamometer Loading Unit

Plate 3.12 Engine Panel Box

The schematic diagram of the dimmerstat used in the loading unit is given in Figure. 3.3. Figure. 3.4 shows the circuit diagram of dimmerstat. It consists of a high conductivity insulated copper wire wound over an insulated toroidal core made of magnetic material. The free ends of the wire are marked A and C in Figure 3.4. A carbon brush (marked E) traverses over this coil. The carbon brush attached to a manually rotating handle makes contact with different number of turns. The number of turns between E & C, can be varied from zero to a maximum of total number of turns between A & C. Therefore the load applied to the engine which is proportional to output voltage can be varied smoothly from zero to the value of the input voltage simply by rotating the handle in the clockwise direction.

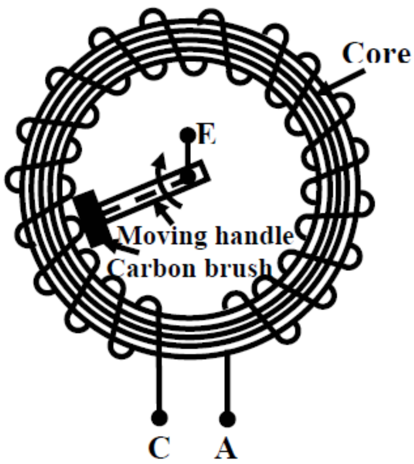


Figure 3.3 Schematic Diagram of the Loading Dimmerstat

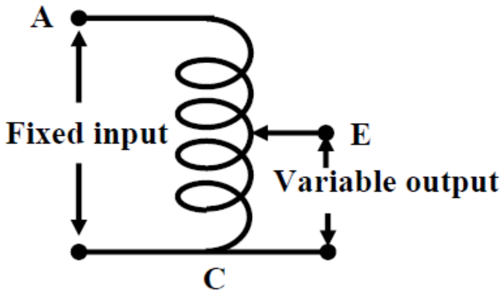


Figure 3.4 Circuit Diagram of the Loading Dimmerstat

3.2.2 Fuel Injection Pressure Measurement System

Fuel injection system is a system for admitting the fuel into an internal combustion engine. Fuel injection pressure (also called as fuel inline pressure) is the pressure at which fuel is injected into the engine cylinder. In the present experimental study, a Piezo Sensor, Make PCB Piezotronics, Model HSM111A22, Range 5000 psi (345 bar), is used to record the fuel injection pressure (Refer Plate 3.13). The technical details of this sensor are given in Table 3.5. The sensor consists of a diaphragm made of stainless steel and of hermetically sealed type. The sensor is installed in a specially made high pressure pipe line. The location of fuel injection pressure sensor is indicated as No. 1 in Plate 3.14. Initially the default value of pressure is 210 bar. During experimentation it is adjusted approximately for about 150bar, 200bar and 250bar. The injection pressure is changed by adjusting the fuel injector spring tension which is carried out by tightening or loosening the nut for higher or lower injection pressures respectively (Refer Plate 3.14). The nut is tightened by rotating it clockwise and loosened by rotating it anticlockwise.

Plate 3.13 Piezosensor

Table 3.5 Technical details of Injection pressure sensor

Make	PCB Piezotronics
Model	SM111A22
Serial No	15345
Description	Pressure Sensor
Type	ICP

Sensitivity	1.001 mV/PSI
Linearity	0.4 % FS
Uncertainity	+/- 1%
Bias	10.67 V DC

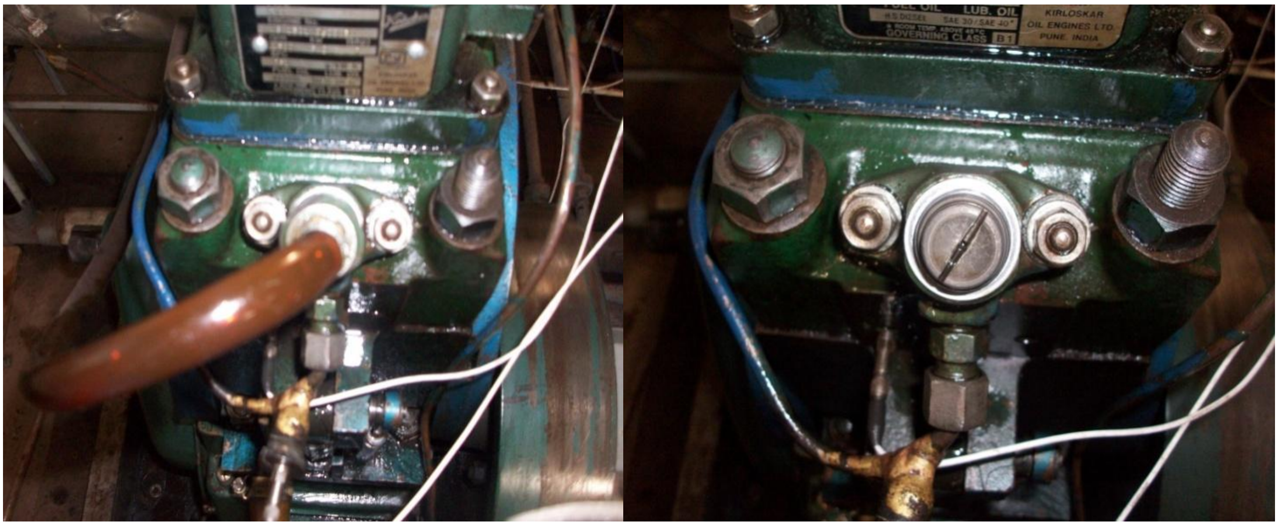


Plate 3.14 The Nut Adjustment for Setting Injection Pressure

The piezosensor is interfaced with a data acquisition system (discussed later in the chapter) so that the pressure is displayed on the computer screen. The data acquisition system reads the pressure signals from the sensor and sends it to the computer which is installed with commercial software (Enginesoft LV). The software displays the value of pressure in bar on the monitor.

3.2.3 Cylinder Pressure Measurement System

The cylinder pressure is measured using a Piezo sensor of Make PCB Piezotronics, Model HSM111A22, Range 5000 psi (345 bar), and diaphragm stainless steel & hermetically sealed type, by mounting it on the cylinder head. The piezo sensor is mounted on engine head. Its location is indicated as No. 2 in the Plate 3.15. The piezoelectric transducer produces a charge output, which is proportional to the in-cylinder pressure. This charge output is supplied to a piezo powering unit which is of Make Cuadra AX- 104 Model.

The piezo sensor consists of a quartz crystal. One end of the sensor is exposed to the cylinder pressure through the diaphragm. As the pressure inside the cylinder increases the crystal is compressed. Since the piezoelectric crystals have a tendency to generate electric charge when deformed, the sensor generates electric charge proportional to the pressure. The charge generated is smaller in magnitude and difficult to measure. Hence a charge amplifier is incorporated in the sensor to produce an output voltage proportional to the charge.

Plate 3.15 Location of Pressure Sensors

3.2.4 Data Acquisition System

For acquiring in-cylinder pressure changes with respect to the crank angle, a high speed data acquisition system is required. Plate 3.16 shows the circuit where different sensors are interfaced to the system. This is used for analyzing the measured cylinder pressure and injection pressure data. The pressure signals from the pressure sensors are converted into digital form considering the transducer sensitivity and the charge amplifier setting during experimentation. Transducers normally provide relative pressures. Therefore, it is necessary to have means of determining the absolute pressure at some point in the cycle, which is to be taken as reference.

The inlet manifold pressure is used as a reference pressure. The mean intake manifold pressure is usually an accurate indicator of the cylinder pressure when the piston is at the Bottom Dead Center (BDC). The cylinder pressure variation with respect to piston displacement in terms of pressure and crank angle is logged into the computer via a data acquisition system. Further to get the valuable information from experimentation, the data is to be analyzed after verifying whether the data is properly recorded or not. Therefore processing of data is an important step in the experimental investigation. The large data which is collected during the experimentation has to be systematically processed as discussed in the following paragraph.

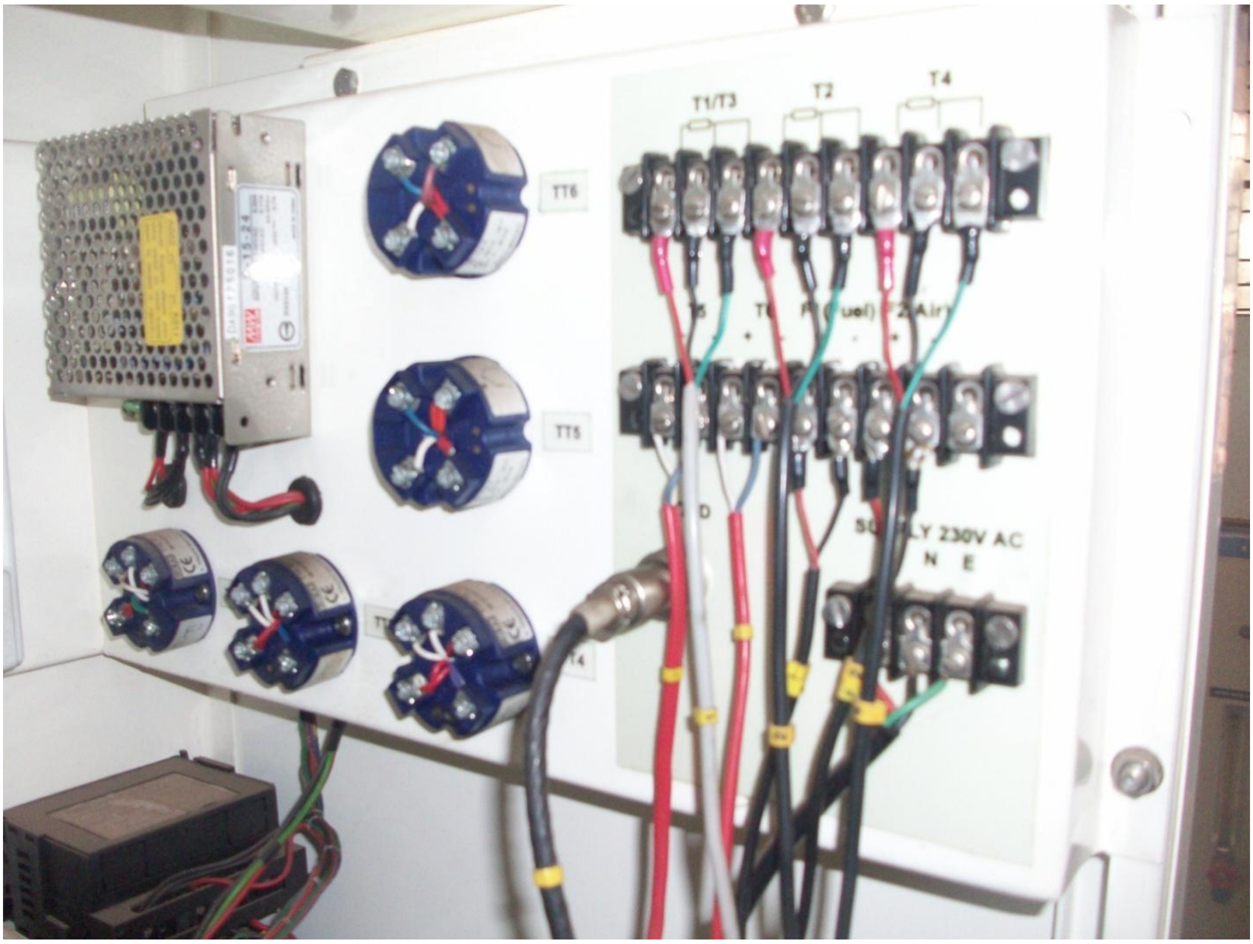


Plate 3.16 Sensors Interface Circuit

The cylinder pressure and injection pressure with respect to crank angle generally has large cycle to cycle variations and hence one such cycle data cannot be used to represent the particular operating condition. Usually an average of 100 cycles of pressure vs crank angle data is chosen for quantitative analysis. The number of cycles to be averaged depends upon the repeatability of the pressure data. An appropriate calibration factor is calculated for conversion of the pressure signals in voltages to the conventional unit. Now the calibration factor is applied to the average cycle and the relative pressures are calculated. These relative pressures are again calibrated with the reference pressure, which is set equal to the pressure in the intake manifold while the piston is at bottom dead center and the absolute pressure values for the average cycle are obtained. Now the pressure volume phasing can be easily done for the average cycle having the absolute pressure values. The experimental data captured is given in Appendix III and sample calculations are given in Appendix IV.

3.2.5 Emission Measurement System

The emission measurement system is used to measure the constituents of exhaust gas and its opacity (smoke number). This system consists of an exhaust gas analyzer and a smoke meter. The exhaust gas analyzer measures the exhaust gas constituents of Carbon dioxide (CO_2), Carbon monoxide (CO), Oxides of nitrogen (NO_x), Unburnt Hydrocarbons (HC), Oxygen (O_2) and Oxides of sulphur (SO_x). The smoke meter is used to measure the intensity of exhaust smoke and it is measured in terms of Hartrigde Smoke Unit (%) and light absorption coefficient (K expressed in m^{-1}). A photographic image of the assembly of the emission measurement systems used in the experiment is given in Plate 3.17. The range, data resolution and accuracy of the exhaust measurement systems are given in Table 3.6. The calibration certificate of exhaust gas analyser and smoke meter are given in Appendix V.

Table 3.6 Range, Resolution and Accuracy of Exhaust Measurement Systems

Gas	Range	Data Resolution	Accuracy
CO	0-15.00%, 0-4000ppm	0.01%, 1ppm	$\pm$ 0.06%, $\pm$ 5%
CO_2	0-20.00%	0.01%	$\pm$ 0.5%
HC	0-30000 ppm	0.01%	$\pm$ 12 ppm
O_2	0-25.00%	1 ppm	$\pm$ 0.1%
NO_x	0-5000 ppm	1 ppm	$\pm$ 3 ppm
SO_x	0-5000 ppm	1 ppm	$\pm$ 5%
Smoke	0-100% HSU	0.1%	$\pm$ 0.1%

3.2.5.1 Exhaust Gas Analyzer

An instrument used to analyze the chemical composition of the exhaust gas released by a reciprocating engine is called exhaust gas analyzer. An image of the exhaust gas analyzer used in this study is given in Plate 3.18. The analyser (Model PEA205) is of make INDUS *Scientific* Pvt Ltd, Bengaluru. The instrument measures the concentrations of Carbon monoxide (CO in % & ppm), Carbon Dioxide (CO_2) and Oxygen (O_2) in percentage, Hydrocarbons (HC), Nitric Oxide (NO_x) and Oxides of Sulphur (SO_x) in ppm in the engine exhaust gas. The technical specifications of the exhaust gas analyser are given in the Table 3.7.

Table 3.7 Technical Specifications of Exhaust Gas Analyzer

Gases Measured	Carbon Monoxide, Hydrocarbon, Carbon dioxide, Oxygen, NO_x and SO_x
Principle	Non-Dispersive Infrared Sensors for CO, CO_2, HC and Electrochemical sensors for O_2, NO_x and SO_x
Data Resolution, Accuracy, Range	Given in Table 3.6
Startup Time	< 2 minutes from power ON. Full accuracy in 3 minutes
Auto Zero	Every 24 minutes with automatic fresh air intake
Gas Flow Rate	500 – 1000 ml per minute
Sample Handling System	S.S. Probe, PU Tubing with easily detachable connectors, water separator cum filter, disposable particulate fine filter.
Operating conditions	Temperature : 5 to 45 ^{0}C Pressure : 813 to 1060mbar Humidity: 0-90%

Plate 3.17 Assembly of Emission Measurement Systems

Plate 3.18 Exhaust Gas Analyzer

The analyzer uses the principle of Non-Dispersive Infra Red (NDIR) for measurement as shown in Figure. 3.5. In this technique, an infrared light is passed through the exhaust gas. Most molecules of gas can absorb the infrared light, causing it to bend, stretch or twist. The amount of infrared light absorbed by the gas molecules is proportional to their concentration in the exhaust gas. This method of detection does not cause ionization of gas molecules because the energy of the photons is not high enough. The source of infrared light is an incandescent bulb. The type of molecule absorbing the light depends on the wavelength of light absorbed by the molecule.

CO, HC & CO_2 are sensed measured by NDIR principle while O_2 ,NO_x, SO_x use Electro Chemical(EC) sensors, for their measurement.

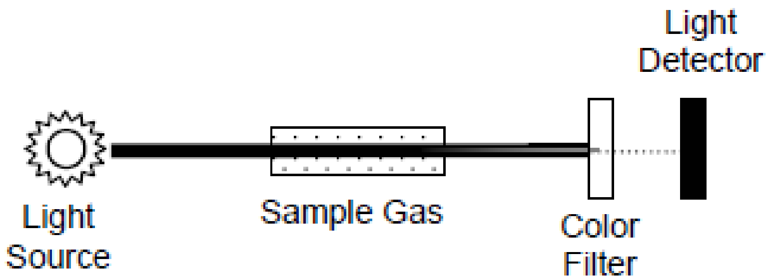


Figure. 3.5 Principle of Non-Dispersive Infra Red Technique

When the probe is inserted into the exhaust pipe (Plate 3.19) of the engine the exhaust gas is passed through a metal mesh screen. The screen filters the soot and dust particles after which it is allowed to pass through a fine fibre element which filters the entire gas for any foreign particles. After this, the clean and cool sample gas enters the direct sensor measurement through a filter arrangement and the readings are displayed on the screen and are recorded. The emission measurements are carried out on dry basis.

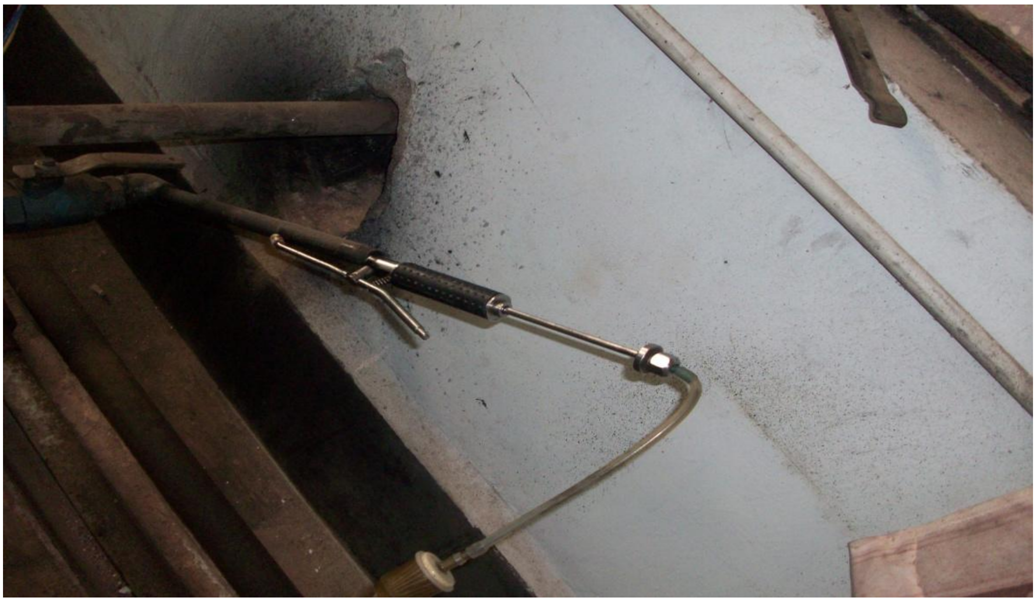


Plate 3.19 Measurement of Exhaust Gas Constituents

3.2.5.2 Smoke Meter

Photographic image of the Smoke Meter (Automotive Exhaust Monitor) given in Plate 3.20 is of make INDUS *Scientific* Pvt Ltd, Bengaluru. The smoke meter is used for measuring the opacity of the exhaust gas from the engine. The instrument measures the smoke opacity in terms of Hartridge smoke unit (HSU) which is measured in % and in terms of light absorption coefficient K (1/m).

The exhaust monitor consists of a smoke chamber which contains the smoke column through which the smoke from exhaust pipe of the engine is passed and smoke density is measured. The gas to be measured is fed into the smoke chamber. The gas enters the smoke column at its center. The smoke column is a tube, which has a light source and a detector placed at one end. The opacity of smoke is directly proportional to the attenuation of light between a light source and a detector. The technical details of Smoke Meter are given in Table 3.8.

Plate 3.20 Smoke Meter

Table 3.8 Technical Details of the Smoke Meter

Principle of Operation	Attenuation of light beam
Geometry	Folded Hartridge Geometry
Measurement	Smoke density in Hartridge smoke units (HSU) & K (m^{-1})
Range	0 to 100% in HSU, 0.01 m^{-1} in K
Resolution	Given in Table 3.6
Time Constant	Physical 0.4 second, Electrical 1m second
Temperature Sensor	RTD (PT-100) or Thermocouple
Light Source	LED, Green Spectrum (567 nm)
Detector	Photocell
Probe	A steel probe with synthetic rubber connecting hose
Warmup Time	20 minutes
Operating Temperature	5 to 50 ^{0}C
Measuring Temperature	80 ^{0}C

The working of smoke meter based on the principle of folded geometry is illustrated in Figure. 3.6. The light source is a green LED, marked as '**S'.** The light beam from **'S'** falls on a partially coated mirror **'PM',** gets reflected to the right and passes through the smoke column in the pipe **'P'.** The beam hits a mirror **'M'** located at the end of the smoke column and gets reflected in the opposite direction. The beam passes through the lens **'L'** for the second time to get focused on the detector **'D'** after transiting the partially coated mirror. The net result of this beam folding is that the beam travels through the smoke column twice, thus making the traversed length twice the length of the smoke column.

A heater is placed around the smoke pipe in order to raise the temperature of the smoke. This will prevent condensation of smoke inside the smoke column. A centrifugal fan is mounted at the either end of smoke column to drive out the smoke after measurement. The smoke opacity is measured by inserting the probe of the smoke meter in the exhaust pipe of the engine (Refer Plate 3.21).

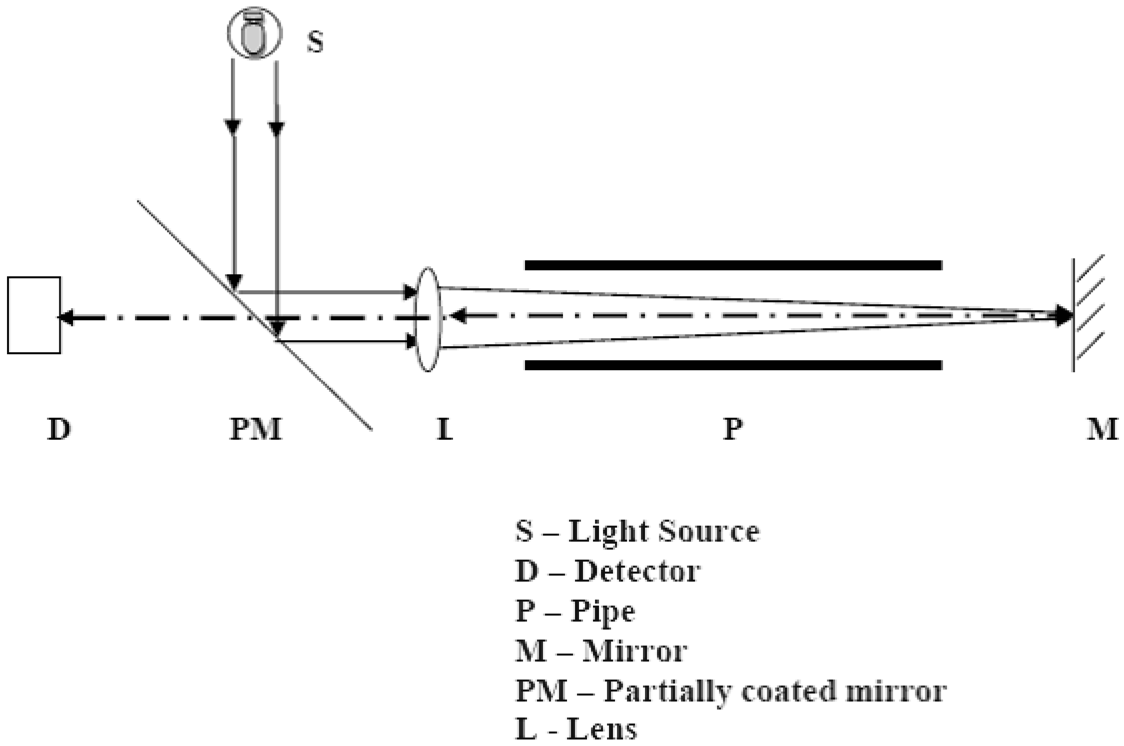


Figure 3.6 Principle of Folded Geometry

Plate 3.21 Measurement of Exhaust Smoke

3.2.6 Commercial Software - EnginesoftLV

Labview based Engine Performance Analysis software package “Enginesoft LV” is used for the on line performance evaluation. Plate 3.22 gives an image of a typical menu during interface with Enginesoft LV. EngineSoftLV can serve most of the engine testing application needs including monitoring, reporting, data entry, data logging. The software evaluates power, efficiencies, fuel consumption and heat release. It is configurable as per engine set up. Various graphs are obtained at different operating conditions. While on line testing of the engine is in RUN mode necessary signals are scanned, stored and presented in the form of graphs. Stored data file is accessed to view the data graphical and tabular formats. The results and graphs can be printed. The data in excel format can be used for further analysis.

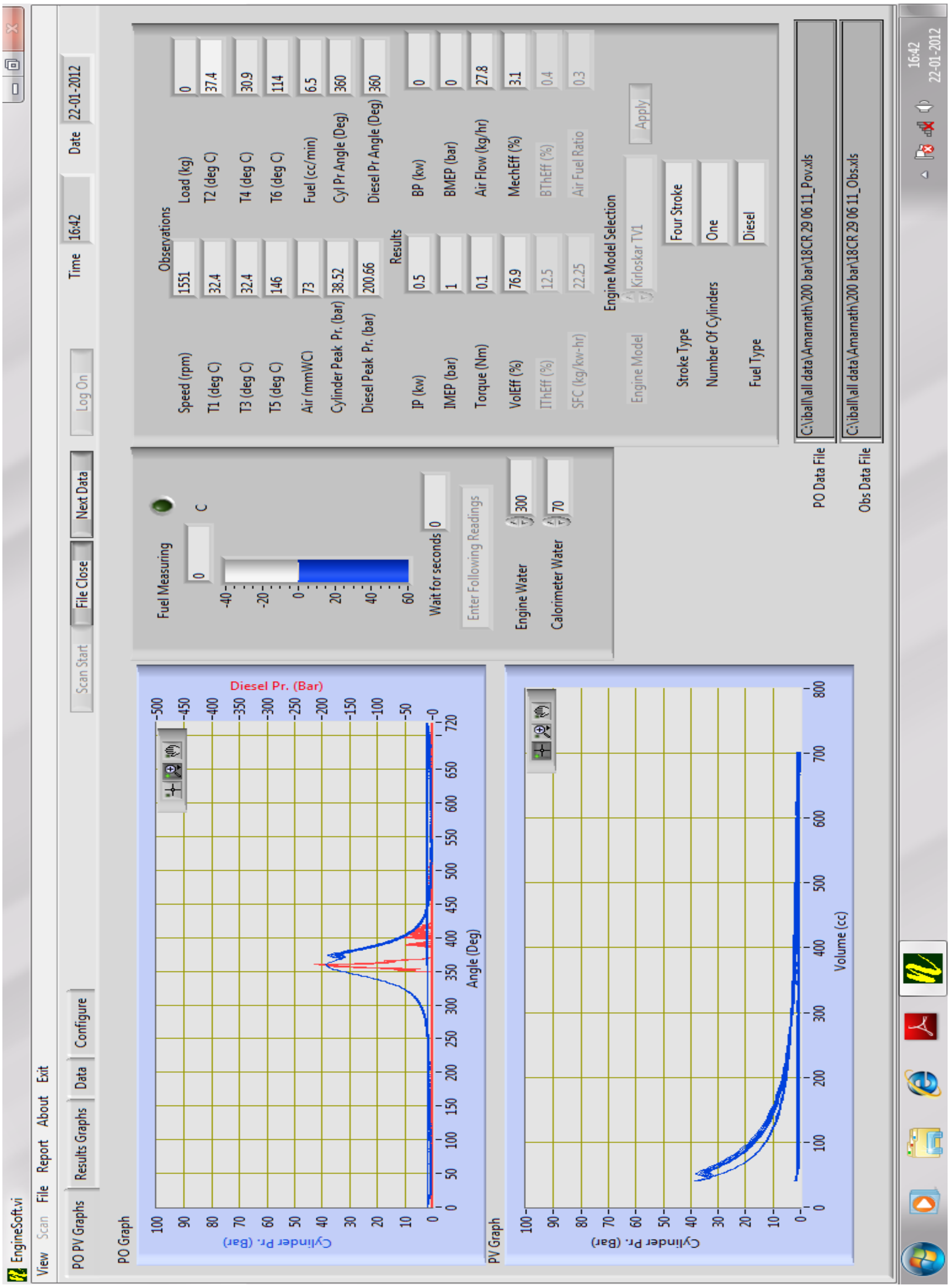


Plate 3.22 Interface of EnginesoftLV

Chapter 4

Experimental Study

An experimental study is conducted to evaluate the thermal performance and emission constituents and combustion analysis of a Karanja biodiesel fuelled variable compression ratio diesel engine. Section 4.1 includes the procedure for evaluation of thermal performance emission constituents and combustion analysis. The results and discussions are presented in Section 4.2. A brief of summary of the experimental investigations is given in section 4.3. In the discussions to follow Diesel, diesel refers to Diesel oil only as fuel.

4.1 Experimental Procedure

The experimental procedure followed in the present study is divided into three different programs viz. thermal performance evaluation, emission characteristics evaluation and Combustion Analysis. The experimental procedure followed for thermal performance evaluation is explained in Section 4.1.1 which is followed by emission characteristic evaluation in Section 4.1.2. The procedure followed for conducting combustion analysis is dealt in Section 4.1.3. Sections 4.1.1 and 4.1.2 are further subdivided into three sections which include the procedures for validation of the test set up using Diesel oil alone as fuel, followed by the procedures using Karanja biodiesel , blends of Karanja biodiesel and Diesel oil as fuels. Section 4.1.3 consists of two more sections which explain procedures for Combustion Analysis using first Diesel oil and then using Karanja biodiesel as fuels. Readings are recorded with repeat trials to ensure the accuracy of the data recorded.

4.1.1 Thermal Performance

Thermal Performance Evaluation is carried out in three different experimental programs:

1. With Diesel oil as fuel at different preset compression ratios, Loads and injection pressures each time keeping any two of the parameters a constant and varying the other.
2. With Karanja biodiesel as fuel following similar condition as in 1.

3. With blends of Karanja biodiesel and diesel as fuel following similar conditions as in 1. The blend proportions used to conduct the experiments are B20, B40, B60 and B80. B20 corresponds to a blend with 20% of Karanja biodiesel and 80% Diesel oil by volume. The blends are prepared by direct mixing of both the fuels in required proportions.

The first experimental program is a validation test(for ensuring the preparedness of the experimental test setup). The experimental procedure undertaken for conducting the validation test, thermal performance evaluation using Karanja biodiesel and blends of Karanja biodiesel with diesel are explained in detail in Sections 4.1.1.1, 4.1.1.2 and 4.1.1.3 respectively. The Diesel oil and Karanja biodiesel used during the study are obtained from the same lot acquired for the purpose and from the same source to ensure consistency in their properties.

4.1.1.1 Validation Test

A validation test is conducted using Diesel oil as fuel. The following step by step procedure is adopted for the test:

1. Check the lubrication, cooling and fuel systems of the engine for their adequacy.
2. Switch ON the electric supply and ensure that all digital and electric instruments are ON.
3. Start the engine and run under idling condition (no load) for 10 minutes to ensure warm and steady operating conditions.
4. Set the compression ratio at a selected value (say 14) using the tilting block arrangement for the engine.
5. Set the injection pressure at a selected value (say 150 bar) using the nut provided on the cylinder head near fuel injection line.
6. Record all the thermal performance parameters for no load condition through a data acquisition system interfaced with the computer. A commercial software, “Enginesoft LV” is used to interface, record and analyze the data.
7. Adjust the load for 3kg using the loading unit dimmerstat and wait for 3 minutes for engine to get stabilized. Repeat step 6 to ensure correctness & reliability/ repeatability of the data recorded.
8. Repeat step 6 for different loads viz. 6kg, 9kg and 12kg.
9. Repeat the steps 4 to 8 for different compression ratios viz. 15, 16, 17, 17.5 and 18.

10. Repeat the steps 4 to 9 for different injection pressures viz. 200 bar and 250 bar.
11. After all readings are recorded, bring down the loading condition to 'no load' before stopping the engine. The water is allowed to circulate for about 5 minutes for engine cooling and then the pump is stopped.

4.1.1.2 Karanja Biodiesel

The procedure followed for conducting the experiment using Karanja biodiesel is the same as explained in Section 4.1.1.1 except that Karanja biodiesel is to replace Diesel oil as fuel. The Karanja biodiesel used to conduct the experiments is commercially procured from MINT BIOFUELS, Pune. The properties of the Karanja biodiesel provided by the manufacturer are shown in Table 4.1. It compares the properties of Karanja biodiesel with different standards. It is clear that the properties of Karanja biodiesel used for the present study confirm to the different set standards. Various methods used to synthesize biodiesels developed by various investigators are explained in Appendix VI.

Table 4.1 Properties of Karanja Biodiesel (Courtesy Mint Biofuels)

Property	Unit	Value	ASTM D 6751 – 09 Standards	EN Standard	BIS Standards
Kinematic Viscosity @ 40 ^{0}C	Cst	5.21	1.9 - 6.0	3.5 – 5	2.5 – 6
Flash point	^{0}C	160	130 min	120 min	120 min
Density @ 15 ^{0}C	Kg/cu m	874	Not mentioned	860 – 900	860 – 900
Water and sediments	% volume	< 0.05	0.05 max	0.05 max	0.05 max
Free Glycerin	% mass	0.012	0.02 max	0.02 max	0.02 max
Total Glycerin	% mass	0.18	0.24 max	0.25 max	0.25 max
Copper strip corrosion		Class 1	No 3 max	Class 1	Class 1
Carbon Residue (100%) Sample	% mass	0.028	0.05 max	0.3 max	0.05 max
Acid Number	mgKOH/gm	< 0.5	0.5 max	0.5 max	0.5 max
Calorific Value	kJ/kg	38874 – 39710	Not Mentioned	Not mentioned	Not mentioned
Cetane number		55*	47 min	Not mentioned	Not mentioned

No definite colour for biodiesel is specified by International standards. Colour is feed dependent.
*Cetane number for Diesel oil is 47.

4.1.1.3 Blends of Karanja Biodiesel and Diesel

The procedure followed for conducting the experiment using blends of Karanja biodiesel and diesel is same as explained in Section 4.1.1.1 except that blends of Karanja biodiesel and diesel is to replace Karanja biodiesel as fuel. The blends selected for the experimental study are B20, B40, B60 or B80. Table 4.2 gives the properties of different tested fuels.

Table 4.2 Properties of Different Tested Fuels

Tested fuels	Calorific Value (kJ/kg)	Density (kg/cu m)
Diesel oil	42000	830
B20	41458	839
B40	40917	847
B60	40375	856
B80	39834	865
B100	39292	874

Properties for Diesel oil are measured and for B100 are given by the supplier of biodiesel. Other properties of blends are determined using a mixture rule model.

4.1.2 Emission Characteristics

The experimental evaluations of emission characteristics are conducted in following three programs:

1. With Diesel oil as fuel at different preset compression ratios ,Loads and injection pressures each time keeping any two of the parameters a constant (Validation Test).
2. With Karanja biodiesel as fuel following similar condition as in 1.
3. With blends of Karanja biodiesel and diesel as fuel following similar conditions as in 1.

The first experimental program is a validation test . The experimental procedure for conducting the validation test, emission characteristic evaluation using Karanja biodiesel and blends of Karanja biodiesel with diesel are explained in detail in Sections 4.1.2.1, 4.1.2.2 and 4.1.2.3.

4.1.2.1 Validation Test

A validation test is conducted using Diesel oil as fuel. The following step by step procedure is adopted for the test:

1) Repeat the steps 1 to 5 as in Section 4.1.1.1.
2) Record all the constituents of exhaust gas for no load condition using the exhaust gas analyzer and smoke meter.
3) Adjust the load at 3kg using the loading unit (dimmerstat) and wait for 3 minutes for engine to get stabilized. Repeat step 2.
4) Repeat step 2 for different loads viz. 6kg, 9kg and 12kg.

5) Repeat the steps 1 to 4 for the different compression ratios viz. 15, 16, 17, 17.5 and 18.
6) Repeat the steps 1 to 5 for different injection pressures viz. 200 bar, 250 bar.
7) After all readings are recorded, bring down the loading condition to ‘no load’ before stopping the engine. The water is allowed to circulate for about 5 minutes for engine cooling and then the pump is stopped.

4.1.2.2 Karanja Biodiesel

The procedure followed for conducting the experiment using Karanja biodiesel is the same as explained in Section 4.1.2.1 except that Karanja biodiesel is to replace Diesel oil as fuel.

4.1.2.3 Blends of Karanja Biodiesel and Diesel

The procedure followed for conducting the experiment using blends of Karanja biodiesel & diesel is the same as explained in Section 4.1.2.1 except that blends of Karanja biodiesel and diesel is to replace Karanja biodiesel as fuel.

4.1.3 Combustion Analysis

The Combustion Analysis is conducted in following two programs:

1. With Diesel oil as fuel at full Load, at different preset compression ratios and at standard preset injection pressure.
2. With Karanja biodiesel as fuel at full load, at different preset compression ratios and at standard preset injection pressure.

The experimental procedures undertaken for conducting combustion analysis using Diesel oil as fuel and Karanja biodiesel as fuel are explained in detail in Sections 4.1.3.1 and 4.1.3.2 respectively.

4.1.3.1 Diesel oil

The procedure followed for conducting the experiment using Diesel oil as fuel for combustion analysis is as follows:

1) Repeat the steps 1 to 5 given in Section 4.1.1.1.

2) Adjust the load for 12kg using the loading unit dimmerstat and wait for 3 minutes for engine to get stabilized.
3) Set compression ratio at a selected value (say 14) using the tilting block arrangement for the engine.
4) Set the injection pressure to 200 bar using the nut provided on the cylinder head near fuel injection line.
5) Record all the parameters of combustion for full load (12 kg) condition and set compression ratio. Commercial software, “Enginesoft LV” is used to interface, record and analyze the data.
6) Repeat step 5 for other compression ratios viz. 16 and 18
7) After all readings are recorded, bring down the loading condition to ‘no load’ before stopping the engine. The water is allowed to circulate for about 5 minutes for engine cooling and then the pump is stopped.

4.1.3.2 Karanja Biodiesel

The procedure followed for conducting the experiment using Karanja biodiesel as fuel is the same as explained in Section 4.1.3.1 except that Karanja biodiesel is to replace Diesel oil as fuel.

4.2 Results and Discussions

The results obtained from the experiments conducted for thermal performance evaluation, emission characteristics evaluation and combustion analysis are interpreted in this section. Before carrying out the series of experiments, the engine’s readiness for the test is validated by running the engine with Diesel oil alone as fuel and is presented in Section 4.2.1. The interpretations to the results of thermal performance parameters and emission characteristics evaluation are explained in Sections 4.2.2 and 4.2.3 respectively. Sections 4.2.2.1, 4.2.2.2 and 4.2.2.3 include the interpretations of the variation of thermal performance parameters with respect to, different preset compression ratios, different loads and different injection pressures. Similar studies are carried out with respect to emission constituents in Section 4.2.3 where emission constituents are studied at different preset compression ratios, Loads and injection pressures. Section 4.2.4 presents the interpretations for analysis of combustion parameters.

The uncertainties for maximum and minimum loads for the thermal performance parameters are represented in Table 4.3. The calculations of uncertainty analyses are given in Appendix VII.

Table 4.3 Uncertainty of Performance Parameters at Maximum and Minimum Loads

Quantity	Uncertainty at Maximum Load (%)	Uncertainty at Minimum Load (%)
Brake Mean Effective Pressure	4.4	4.1
Brake Thermal Efficiency	2.3	4.1
Brake Specific Fuel Consumption	2.2	4.2
Volumetric Efficiency	1.5	1.5
Heat Equivalent of Brake Power	2.1	3.8
Heat Equivalent of Exhaust Gas Temperature	1.0	1.8

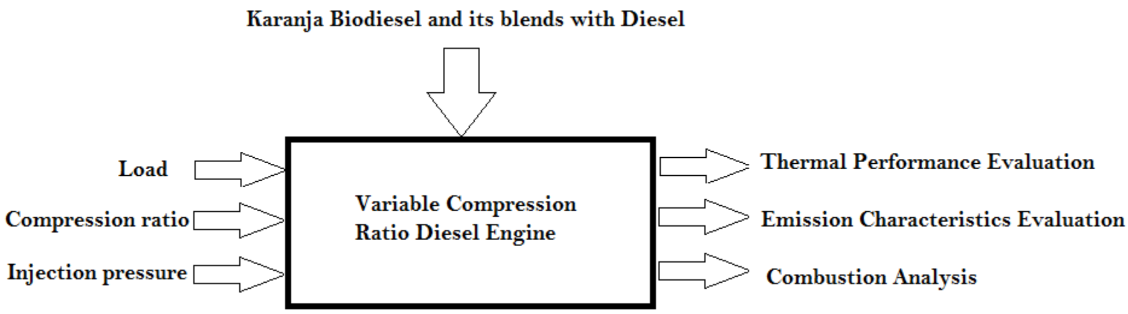


Figure 4.1 Input and Output Variables of the Engine System

Figure 4.1 indicates the Input (Control) variables and Output (Response) variables for the engine system.

4.2.1 Validation Test

The validation test is conducted for thermal performance evaluation of the engine running on Diesel oil alone as fuel. The tests are conducted at different loads (from 0kg to 12kg in steps of 3kg), at different preset compression ratios (CR) (14, 15, 16, 17, 17.5, 18) and injection pressure (IP) of 200bar. The thermal performance parameters considered for validation of the engine setup are Brake Thermal Efficiency (BTHE) and Brake Specific Fuel Consumption (BSFC). Similar tests are also considered for evaluation of emission constituents to capture a baseline data using Diesel oil alone as fuel for comparison with Karanja biodiesel-diesel blends for further studies. The emission constituents considered validation of the engine setup are Unburnt Hydrocarbon (HC), Oxygen (O_2) and Oxides of Nitrogen (NO_x).

Figures 4.2 and 4.3 give the comparison of effect of the variation of load on selected thermal performance parameters such as BTHE and BSFC respectively. The test is conducted at a rated CR of 17.5, IP of 200bar and at different loads. Although the standard IP is 210 bar, the pressure setting varied close to 190-210 bar and the mean i.e. 200bar is taken as standard value of IP. The trends of the results closely match with that of Jindal et al. [66] for load variation at standard settings of the engine. The results of Jindal et al. [66] are considered for comparison since the engine used in the present study is of the same type as that used by them. The magnitudes of BTHE and BSFC may not match on comparison as properties of and quality of Diesel oils used for the studies may vary from region to region. Further, the aging of the engine also matters.

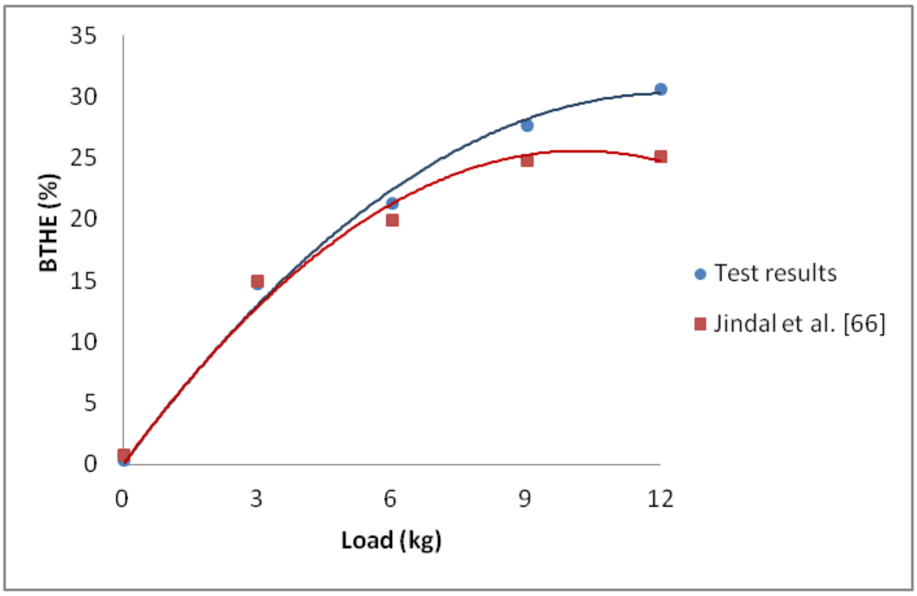


Figure 4.2 Comparison of Variation of BTHE with Load at CR of 17.5 and IP of 200bar

The validation test is needed to be conducted at different preset CRs and IPs to ascertain the preparedness of the engine. Since there appears to be no valid reference available in literature for variation of BTHE and BSFC with CR, the value of BTHE and BSFC at rated load (full load), CR of 17.5 and IP 200 bar from the present study is compared with results of Jindal et al. [66] at the same conditions in Figure 4.4 and Figure 4.5.

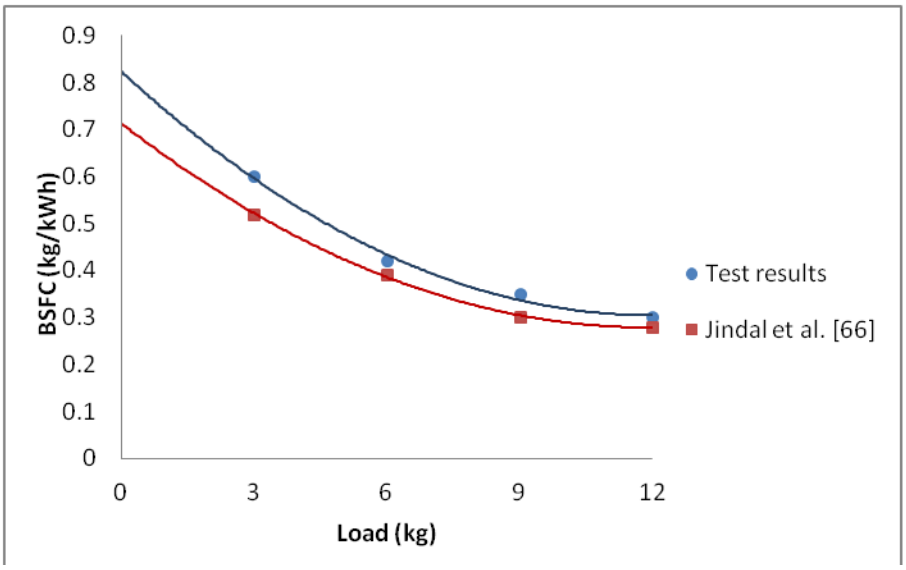


Figure 4.3 Comparison of Variation of BSFC with Load at CR of 17.5 and IP of 200bar

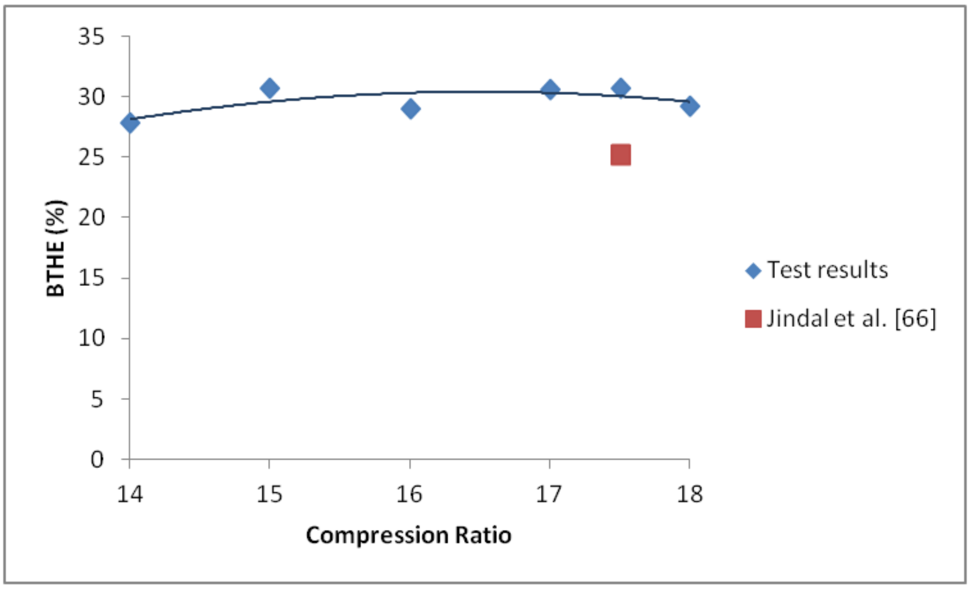


Figure 4.4 Comparison of BTHE at Rated Load, CR of 17.5 and IP of 200bar

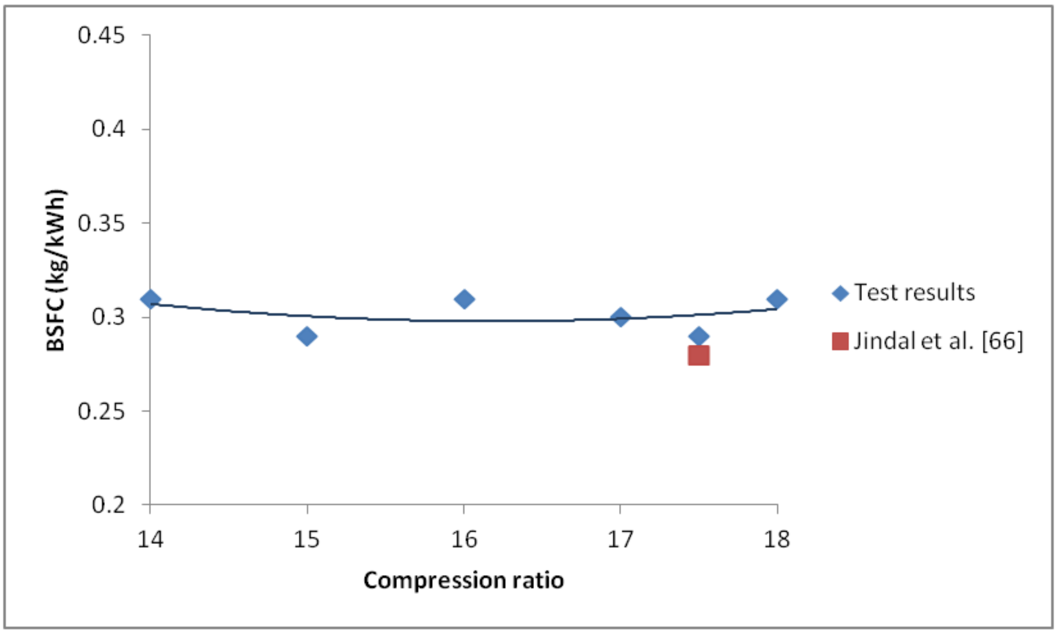


Figure 4.5 Comparison of BSFC at Rated Load, CR of 17.5 and IP of 200bar

From the comparisons made in Figures 4.4 and 4.5, it is observed that the values of BTHE and BSFC at rated load, CR of 17.5 and IP 200 bar are in close agreement with each other. Since, the comparisons made in Figures 4.2-4.5 are at an IP of 200 bar, the preparedness of the engine at varying IPs is ascertained.

The engine tests are conducted to capture emission characteristics at different loads (0 to 12kg in steps of 3kg), different preset CRs (14, 15, 16, 17, 17.5, 18) and different IPs (150, 200, 250bar) keeping any two of them a constant each time. Figures 4.6 to 4.14 give the effect of the variation in load, compression ratio and injection pressure on selected emission constituents such as HC (ppm), NO_x (ppm) and O_2 (%). It should be noted that the emission test carried out are standardized with respect to Diesel oil as fuel on the engine where experimental studies are proposed to be carried out and this is taken as baseline data for further comparisons of various emissions with other tested fuel combinations. Also, the quality of Diesel varies from region to region and hence validation of emissions is not possible.

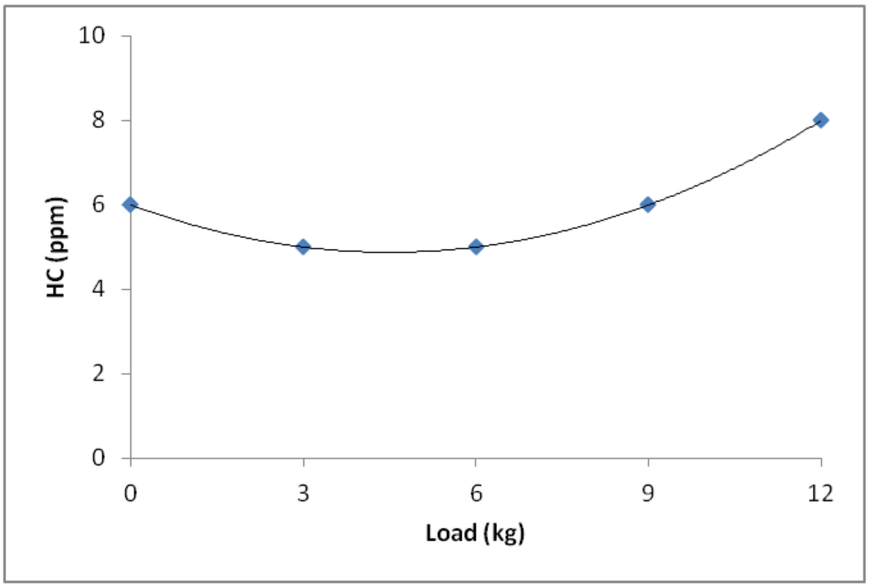


Figure 4.6 Variation of HC with Load at CR of 17.5 and IP of 200bar

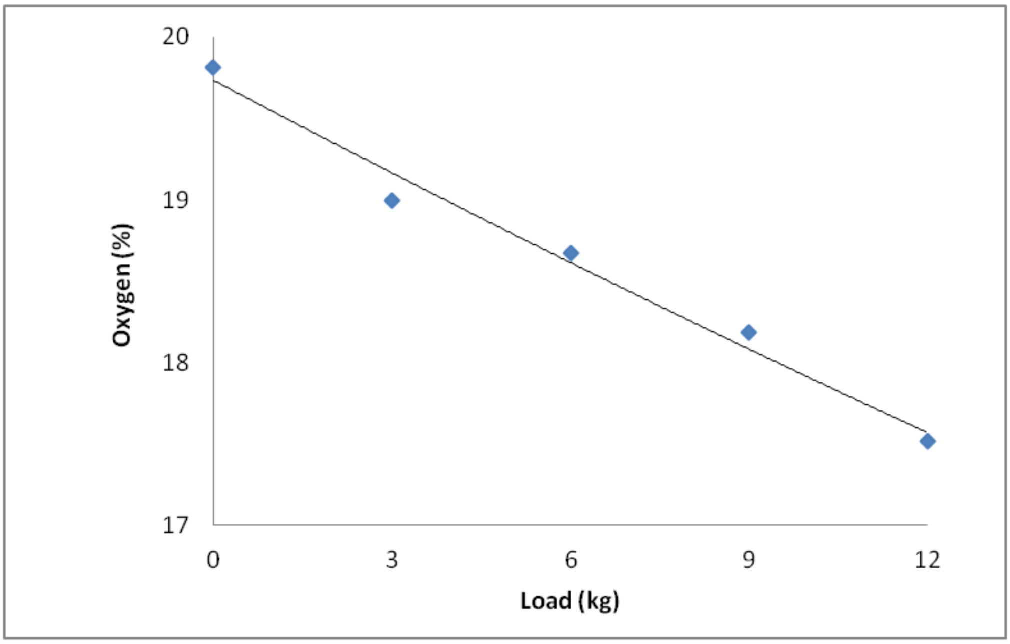


Figure 4.7 Variation of O_2 with Load at CR of 17.5 and IP of 200bar

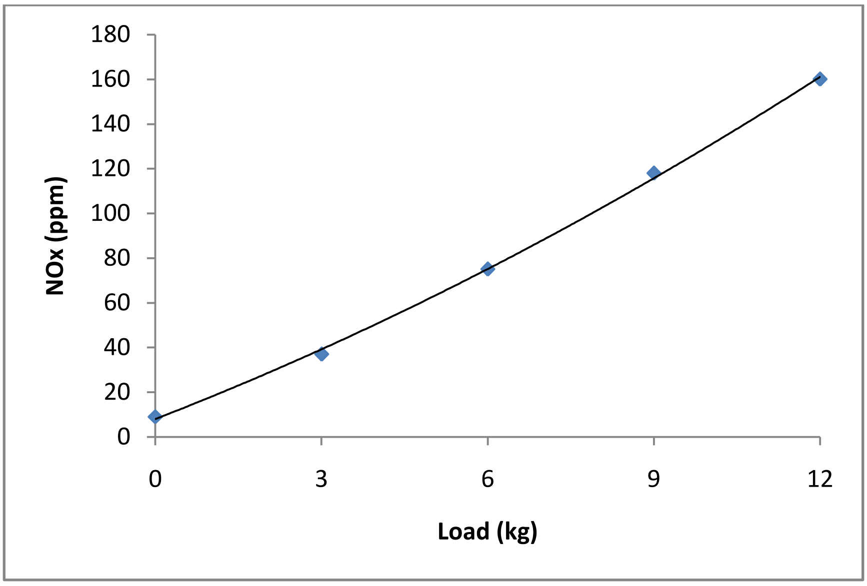


Figure 4.8 Variation of NO_x with Load at CR of 17.5 and IP of 200bar

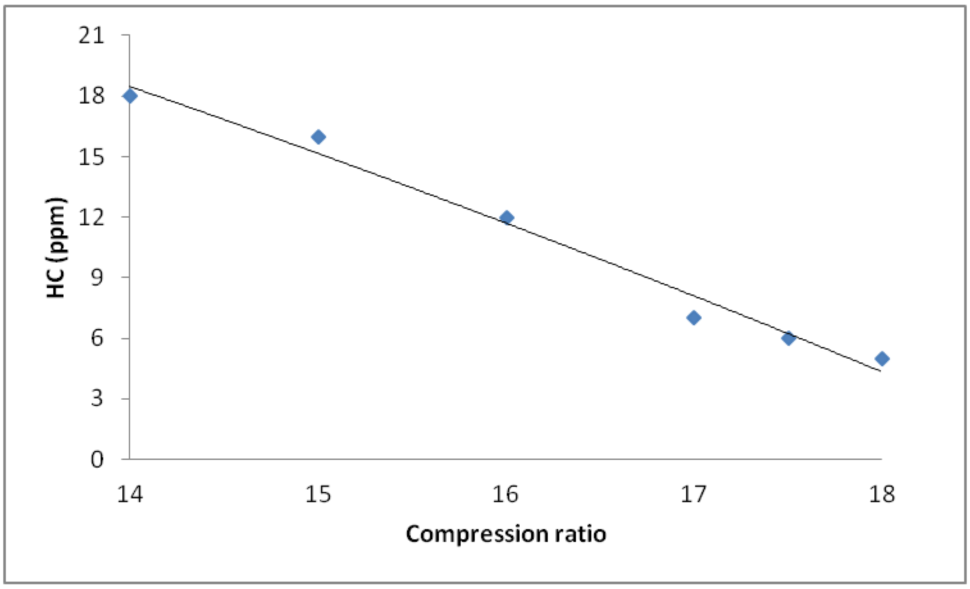


Figure 4.9 Variation of HC with CR at Load of 12kg and IP of 200bar

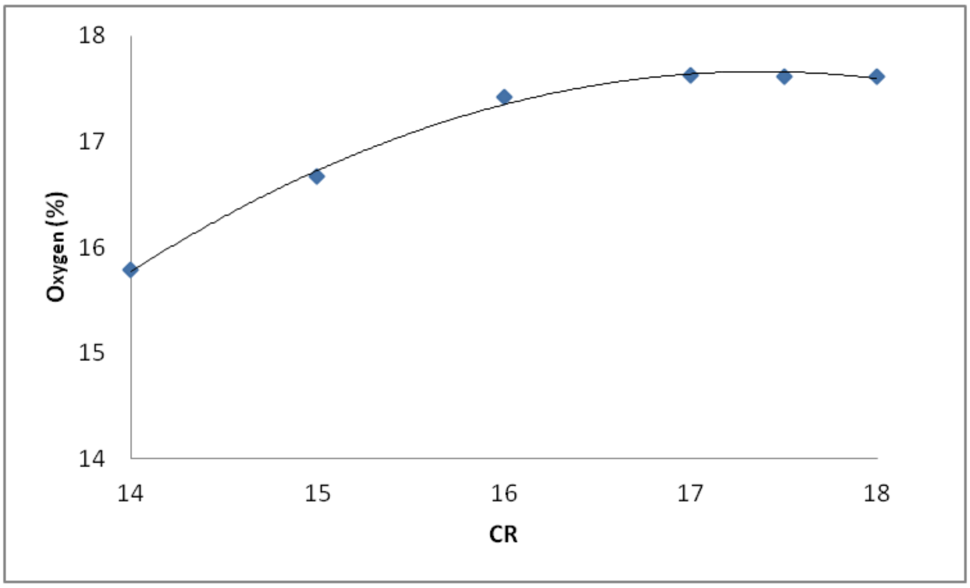


Figure 4.10 Variation of O_2 with CR at Load of 12kg and IP of 200bar

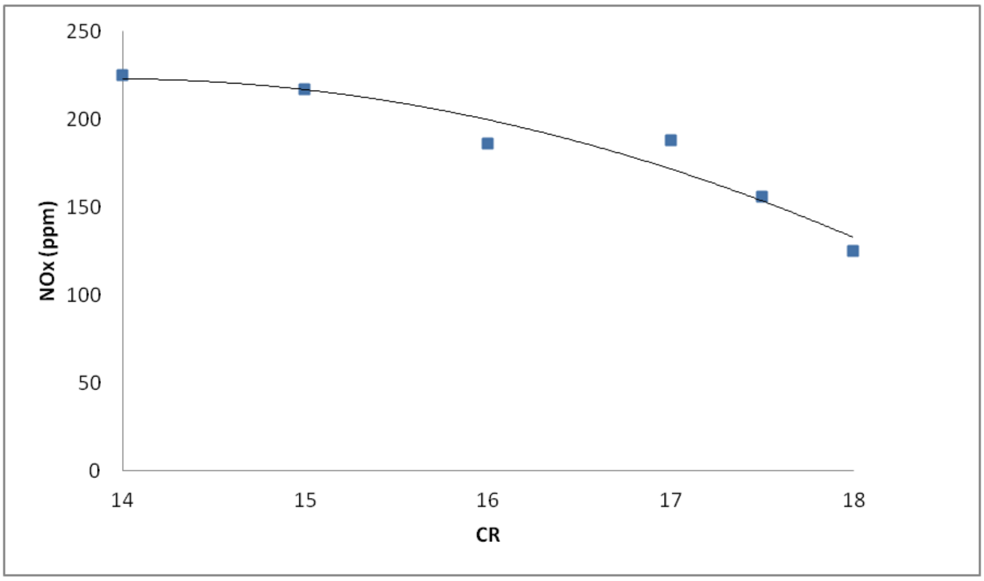


Figure 4.11 Variation of NO_x with CR at Load of 12kg and IP of 200bar

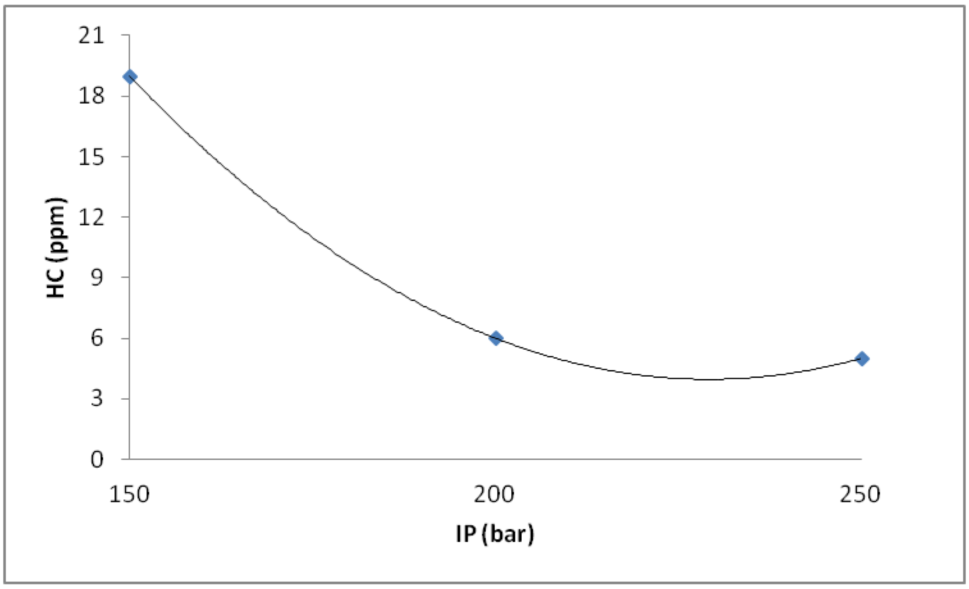


Figure 4.12 Variation of HC with IP at Load of 12kg and CR of 17.5

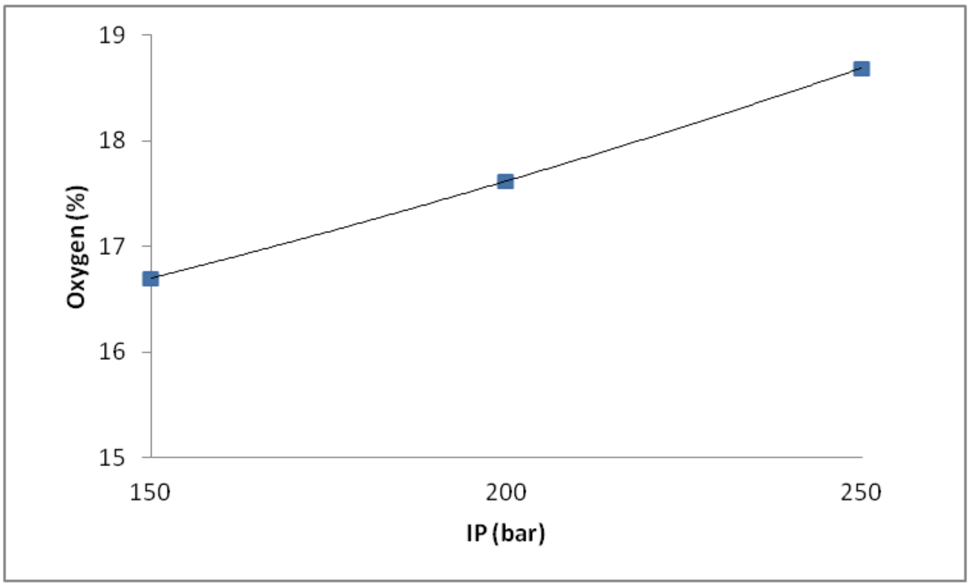


Figure 4.13 Variation of O_2 with IP at Load of 12kg and CR of 17.5

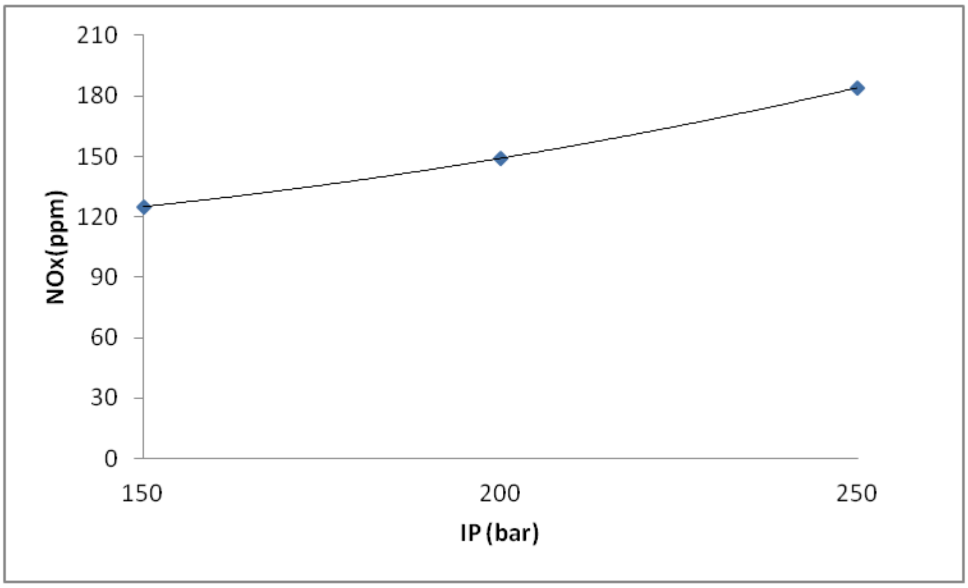


Figure 4.14 Variation of NO_x with IP at Load of 12kg and CR of 17.5

4.2.2 Thermal Performance

The thermal performance parameters considered in the present study are BTHE BSFC, EGT, Brake Mean Effective Pressure (BMEP), Volumetric Efficiency, Heat Equivalent of Brake Power (HBP), Heat Equivalent of Exhaust Gas (HGas). The variations of the performance parameters with CRs, Load and IPs are presented in three different sections i.e., 4.2.2.1, 4.2.2.2 and 4.2.2.3 respectively. Section 4.2.2.1 describes the results of the experiment carried out on the thermal performance parameters when CR is varied at values of 14, 15, 16, 17, 17.5 and 18. The purpose of this study is to find out whether the rated CR of 17.5 is or not to be selected for further analysis with Karanja biodiesel and its blends. For each of the parameters, the variations in the performance for Diesel oil, Karanja biodiesel and blends of Karanja biodiesel with Diesel are superposed and compared.

4.2.2.1 Compression Ratio

The variations of thermal performance parameters at different values of CRs at 14, 15, 16, 17, 17.5 and 18 with a constant rated load of 12kg and IP of 200bar is presented in Figures 4.15 to 4.23.

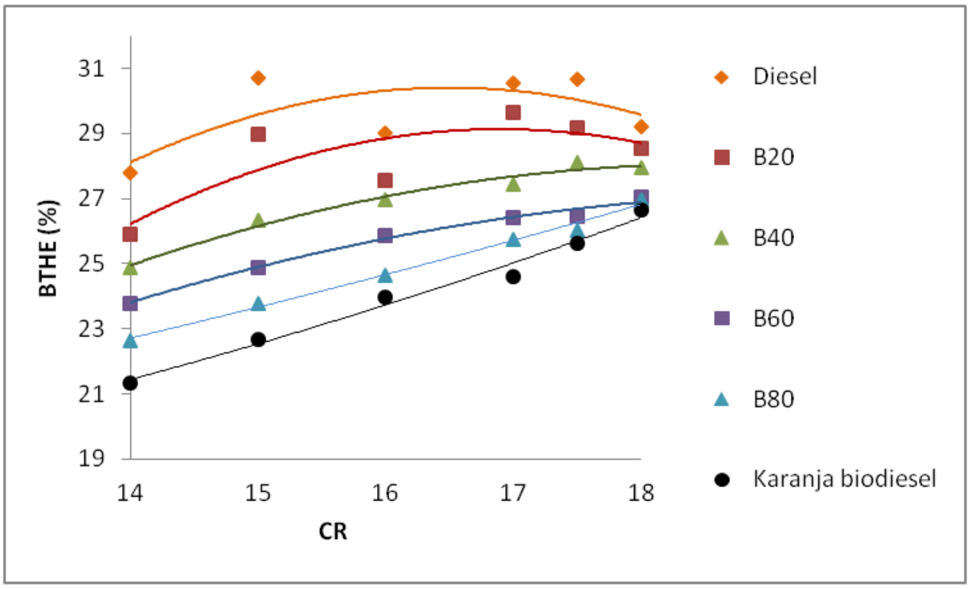


Figure 4.15 Comparison of Variation of BTHE with CR at a Load of 12kg and IP of 200bar

A comparison of the variation of BTHE with CR for Karanja biodiesel and its blends with Diesel oil at a load 12kg and IP of 200bar is presented in Figure 4.15. It can be observed that BTHE increases continuously with increase in CR for all the fuels tested except for B20 and Diesel oil. The increasing trend is due to higher air temperature achieved at higher CR which results in better combustion of fuel. It is also observed that BTHE is decreased with increase in biodiesel content in the blend at a constant CR. The decrease may be due to higher viscosity of biodiesel which hinders the fuel evaporation due to poor atomization during combustion process.

With the increase in CR from 14 to 18, the BTHE increased upto 6% for all fuels. The values of BTHE at full load (12kg) and 18CR are 26.64%, 26.98%, 27.03%, 27.95%, 28.54% and 29.2% for Karanja biodiesel, B80, B60, B40, B20 and Diesel oil respectively.

It can be observed in Figure 4.15 that with the increase in CR the performance with Karanja biodiesel and its blends with diesel approaches that of Diesel oil. At 18 CR the BTHE of the engine operated with Karanja biodiesel is 9.6% less than Diesel oil. Due to this reason CR of 18 is selected for comparison of variation of other thermal performance parameters with load and IP for further analysis.

A comparison of the variation of BSFC with CR for Karanja biodiesel and its blends with Diesel oil at a load 12kg and IP of 200bar is shown in Figure 4.16. It can be observed that as the CR of the engine increases, there is a decrease of 5% to 13% in

BSFC for all the fuels as expected. The BSFC is found to be the least at 18CR because at higher CRs complete combustion of fuel takes place due to high heat (temperature) of compressed air. The complete combustion produces higher brake power. Therefore, load demand is met with less fuel consumption.

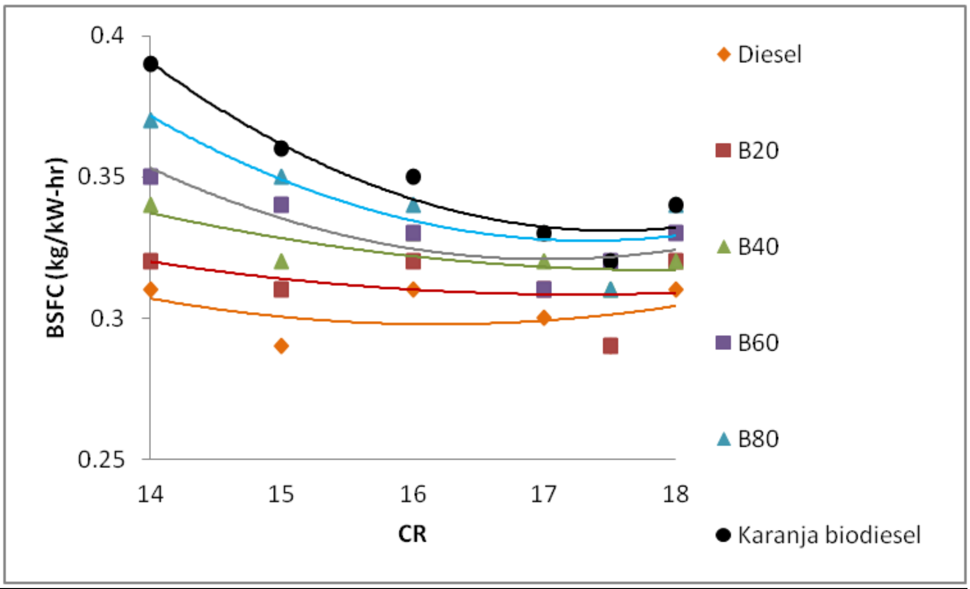


Figure 4.16 Comparison of Variation of BSFC with CR at a Load of 12kg and IP of 200bar

It is also seen that BSFC increases with the increase in percentage of Karanja biodiesel in the blend. This may be due to the reason that fuel burning rate required is more with Karanja biodiesel because of its lower calorific values. The reduced delay due to higher cetane number during injection accounts for increased rate of fuel injection.

As CR increases from 14 to 18, BSFC decreases by about 5% to 13% for biodiesel blends B40, B60, B80 and B100. It is seen that the consumption of pure Karanja biodiesel is about 25% more at CR of 14 while that at CR of 18 is about 10% more. At CR of 18 the thermal performance of Karanja biodiesel is close to that of Diesel oil. The values of BSFC for Karanja biodiesel at CRs of 14, 15, 16, 17, 17.5 and 18 are 0.39, 0.36, 0.35, 0.33, 0.32 and 0.31 kg/kWh respectively.

Figure 4.17 shows the comparison of variation of BTHE and BSFC with CR at a load of 12kg and IP 200bar of the present study with that of earlier investigations of Sureshkumar et al. [52], Banapurmath et al. [44], Jindal et al. [66], Raheman & Ghadge [40] and Raheman & Phadatare [17].

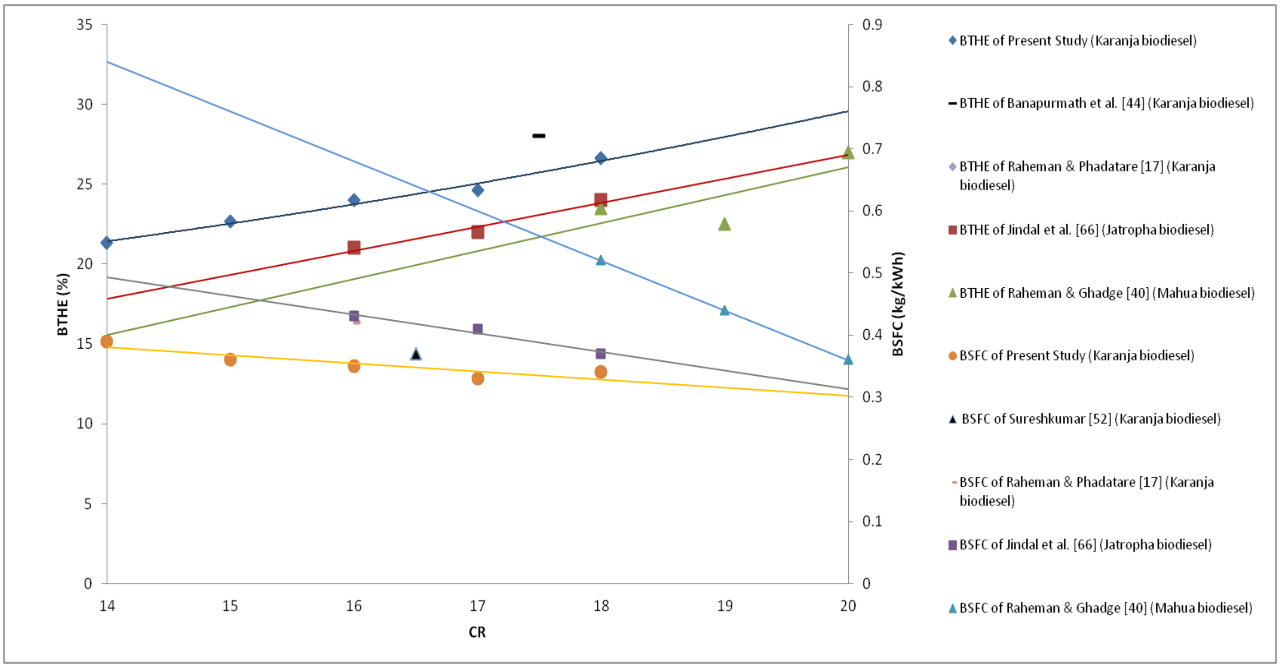


Figure 4.17 Comparison of Variation of BTHE and BSFC with CR at a Load of 12kg and IP of 200bar with Earlier Studies

Similar increasing trends of BTHE with CR as found in the present study are observed by Jindal et al. [66] and Raheman & Ghadge [40] while using Jatropha and Mahua biodiesels respectively. It is seen that the value of BTHE at 18 CR is less by 10% for Jatropha and 12% for Mahua as compared to that of Karanja biodiesel used in the present study. It should be noted that the values of BTHE with Karanja biodiesel are available only at a given single value of CR from the earlier studies. Banapurmath [44] has reported a 10% higher value of BTHE at a constant CR of 17.5 while 12% lesser BTHE has been reported by Raheman and Phadatare [17] at a constant CR of 16 using Karanja biodiesel as compared to the present study.

Similar decreasing trends of BSFC with CR as found in the present study are observed by Jindal et al. [66] and Raheman & Ghadge [40] while using Jatropha and Mahua biodiesels respectively. It is seen that the value of BSFC at 18 CR is more by 9% for Jatropha and 52% for Mahua as compared to that of Karanja biodiesel used in the present study. It should be noted that the values of BSFC with Karanja biodiesel are available only at a given single value of CR from the earlier studies. Sureshkumar [52] has reported a 3% higher value of BSFC at a constant CR of 16.5 while 21% more BSFC has been reported by Raheman and Phadatare at a constant CR of 16 using Karanja biodiesel as compared to the present study.

Figure 4.18 shows a comparison of the variation of BMEP with CR for Karanja biodiesel and its blends with Diesel oil at a load of 12kg and IP of 200bar. Brake Mean Effective Pressure (BMEP) is the external shaft work per unit volume displacement of the engine. From the figure, it can be observed that CR does not have a significant effect on BMEP.

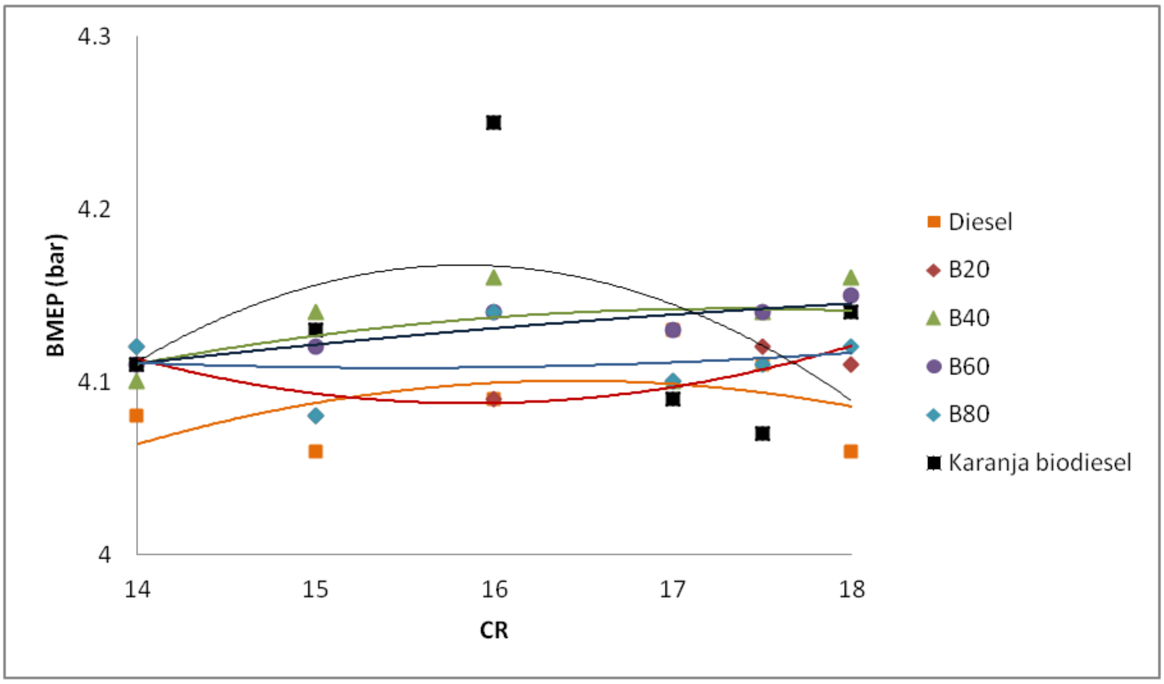


Figure 4.18 Comparison of Variation of BMEP with CR at a Load of 12kg and IP of 200bar

It is also observed that the use of different Karanja biodiesel-diesel blends does not have much effect on BMEP. Hence, it is inferred that though the cylinder peak pressure attained is slightly less with biodiesel, the effective pressure on the engine piston remains fairly the same. It may probably be due to complete burning of the fuel of biodiesels in the premixed combustion phase which takes place during the diffusion phase with Diesel oil combustion.

A comparison of the variation of volumetric efficiency with CR for Karanja biodiesel and its blends with Diesel oil at a load 12kg and IP of 200bar is presented in Figure 4.19. The volumetric efficiency is an important performance parameter which is also referred to as the breathing capacity of engine. It is basically a mass ratio and not a volume ratio because it is the ratio of the mass of charge actually admitted into the cylinder to the mass of the charge corresponding to the swept volume. It is desirable to maximize the volumetric efficiency since the amount of fuel that can be burnt and power produced for a given engine displacement are maximized with higher rate of air inflow at suction conditions. Although it does not influence in any way directly the thermal efficiency of the engine, it will influence the efficiency of the system since the fuel economy will go up with an engine having higher volumetric efficiency (Heywood [96], Ferguson & Kirkpatrick [98]).

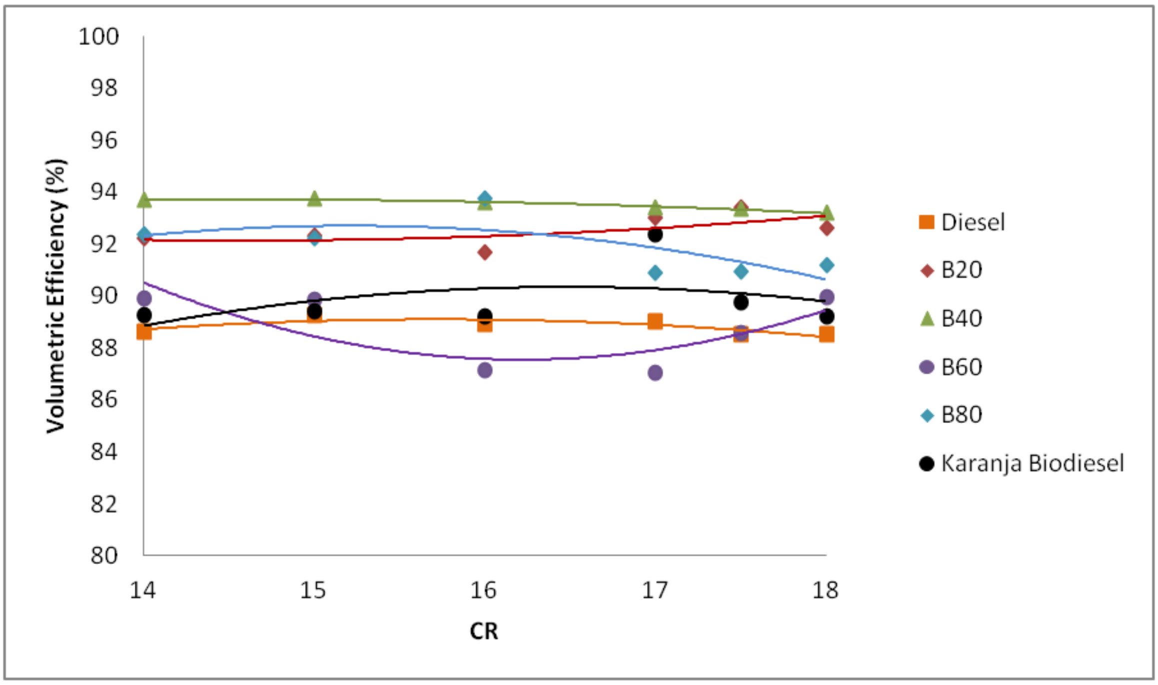


Figure 4.19 Comparison of Variation of Volumetric Efficiency with CR at a Load of 12kg and IP of 200bar

It is observed that change in CR does not affect the volumetric efficiency of the engine (Figure 4.19). Therefore, the engine could be operated at any CR without hampering the volumetric efficiency of the engine. It can also be noted that this does not depend on blend proportion. However, it is marginally higher for Karanja biodiesel than Diesel oil. For Karanja biodiesel, the volumetric efficiencies at tested CRs are 89.23%, 89.41%, 89.2%, 92.34%, 89.76% and 87.6%.

Figure 4.20 shows the comparison of variation of HBP with CR for Karanja biodiesel and its blends with Diesel oil at a load 12kg and IP of 200bar. HBP represents the amount of input percent thermal energy associated with the useful power output of an engine. It is evident from the figure that the HBP increases with increase in CR for all the fuels tested. The trend observed is due to higher brake power at higher CR resulting in better combustion of fuel which in turn is due to high heat of compressed air. It is observed that HBP decreases with increase in content of Karanja biodiesel in the blend at constant CR. The trend is due to lower calorific value of Karanja biodiesel. It can also be noted that the increase in HBP with CR is continuous for Karanja biodiesel while it increases up to CR of 16 then remains almost constant upto CR of 17 and subsequently decreases for other tested fuels. Therefore, it can be inferred that combustion is better at CR of 16 for the blends and at CR of 18 for Karanja biodiesel.

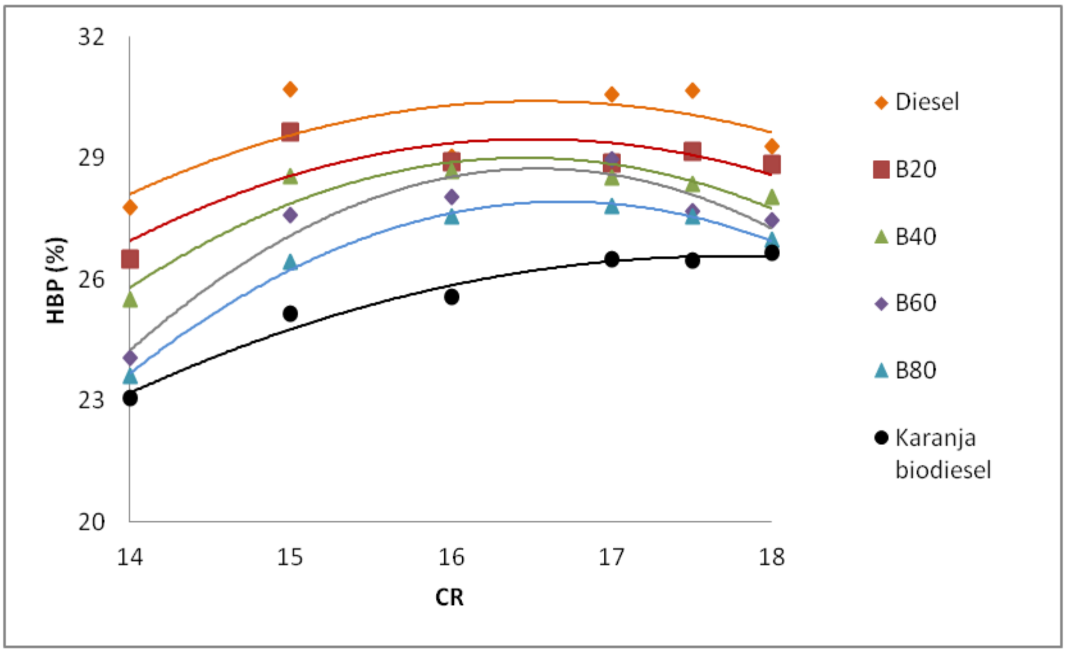


Figure 4.20 Comparison of Variation of HBP with CR at a Load of 12kg and IP of 200bar

The values of HBP for Diesel oil for the respective tested CRs are 27.76%, 30.69%, 29.01%, 30.56%, 30.66% and 29.29% and the same for Karanja biodiesel are 23.04%, 25.13%, 25.57%, 26.48%, 26.46% and 26.64%. It can be noted that HBP for Karanja biodiesel is closer to that of Diesel oil particularly at higher CRs.

Figure 4.21 presents the comparison of variation of HGas with CR for Karanja biodiesel and its blends with Diesel oil at a load of 12kg and IP of 200bar. HGas represents the amount of total thermal energy carried with the engine exhaust. Higher exhaust temperature represents the effectiveness of combustion but greater heat loss and hence the attainment of lesser value of thermal efficiency. It can be observed from the figure that the change in CR does not have much effect on the HGas of the engine. It can also be observed that there is no specific trend of variation of HGas with Karanja biodiesel content in the blend. However, at most of the CRs it is less for Karanja biodiesel than Diesel oil. The trend may be due to lower heat released during combustion of Karanja biodiesel which probably may be due to its lower calorific value.

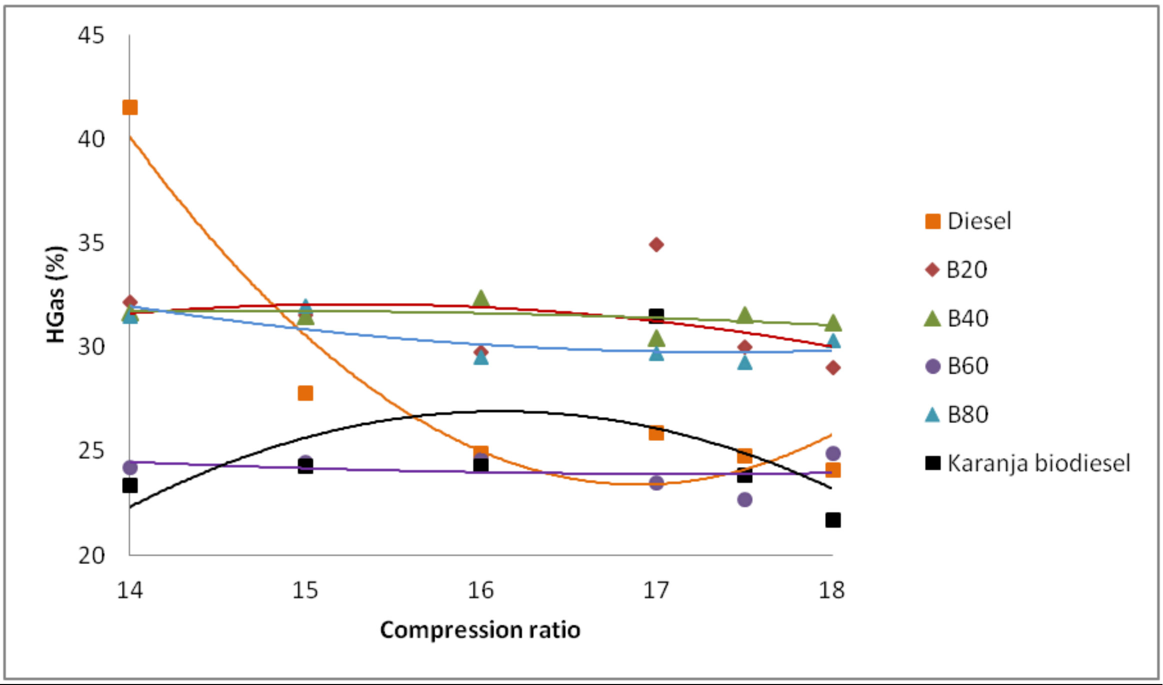


Figure 4.21 Comparison of Variation of HGas with CR at a Load of 12kg and IP of 200bar

It can also be noted that at a CR of 14, the HGas for Diesel oil is higher compared to that of Karanja biodiesel-diesel blends. It may be due to a fact that at lower CR combustion is incomplete due to less heat of compressed air. Therefore, higher calorific value of Diesel oil results in higher HGas. The values of HGas for Diesel oil at various tested CRs are 41.57%, 27.8%, 24.92%, 25.92%, 24.79 and 24.08% and for Karanja biodiesel are 23.39%, 24.31%, 24.32%, 31.48%, 23.87% and 21.71%.

A comparison of the variation of EGT with CR for Karanja biodiesel and its blends with Diesel oil at a load of 12kg and IP of 200bar is given in Figure 4.22. EGT represents the temperature of flue gases that come out of the engine and indicates the effectiveness of combustion and engine thermal energy conversion. Lower EGT is always desirable for any engine. It can be seen from the figure that EGT decreases with the increase in CR for all the fuels tested. The reduction in EGT can be attributed to high temperature of compressed air inside the cylinder at higher CRs due to which the fuel is burnt more completely.

It is observed that EGT reduces with the increase in blend proportion. Also, the rate of EGT drop is almost the same at different CRs for all fuels tested. The EGT reduces upto 18% for all fuels as the CR increases from 14 to 18. At CR of 18, the values of EGT for Karanja biodiesel, B80, B60, B40, B20 and Diesel oil are found to be

343.77^{0}C, 346.99^{0}C, 347.25^{0}C,345.67^{0}C, 347.96^{0}C and 350.75^{0}C respectively indicating that EGT at higher CRs for blends are closer to Diesel oil.

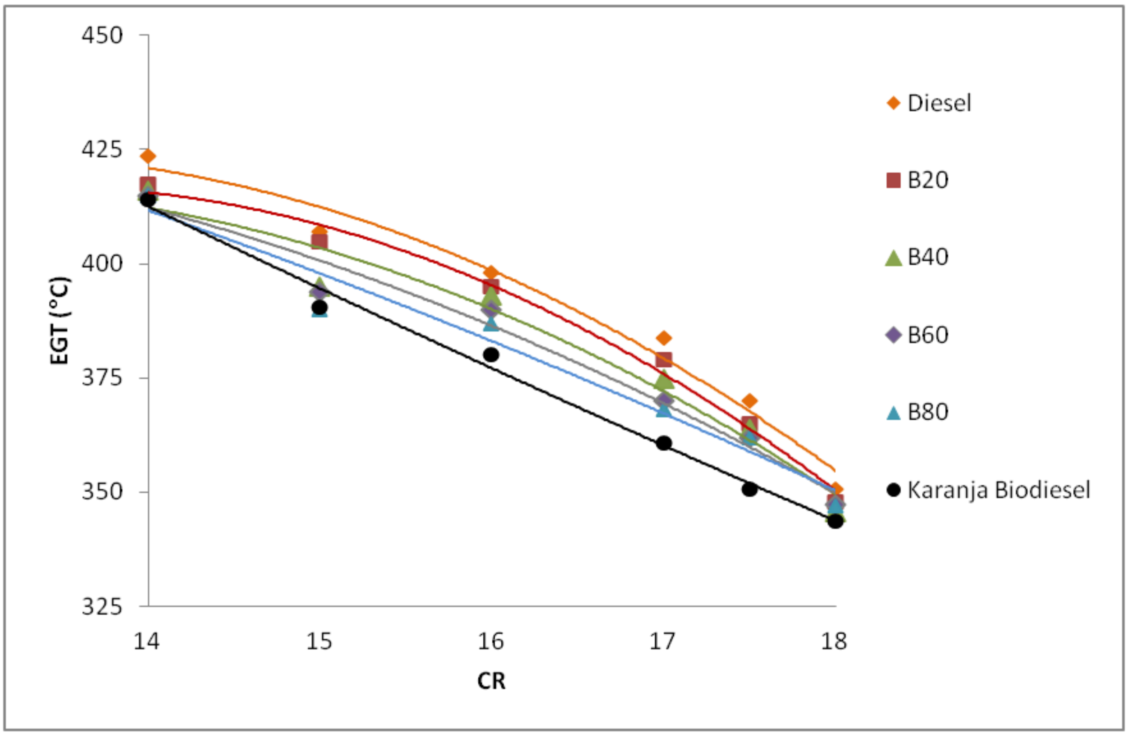


Figure 4.22 Comparison of Variation of EGT with CR at a Load of 12kg and IP of 200bar

Figure 4.23 shows the comparison of variation of EGT with CR of the present study at a load of 12kg and IP of 200bar with that of earlier investigations of Sureshkumar [52], S. Jindal et al. [66] and Raheman & Ghadge [40]. The fuels used by the investigators are Karanja biodiesel, Jatropha biodiesel and Mahua biodiesel respectively. It can be seen that, similar decreasing trend of EGT with CR as found in the present study is observed by Raheman & Ghadge [40] while an opposite trend is seen by Jindal et al. [66]. Further it is observed that the value of EGT at 18 CR is less by 4% (Raheman & Ghadge [40]) and more by 17% (Jindal et al. [66]) as compared to that of the present study. It is also observed that the value of EGT for the study of Sureshkumar [52] with Karanja biodiesel at a constant CR of 16.5 is not comparable with that of the present study. It can however be noted that the EGT values lie in range of 350-400^{o}C for the present study and Jindal et al [66].

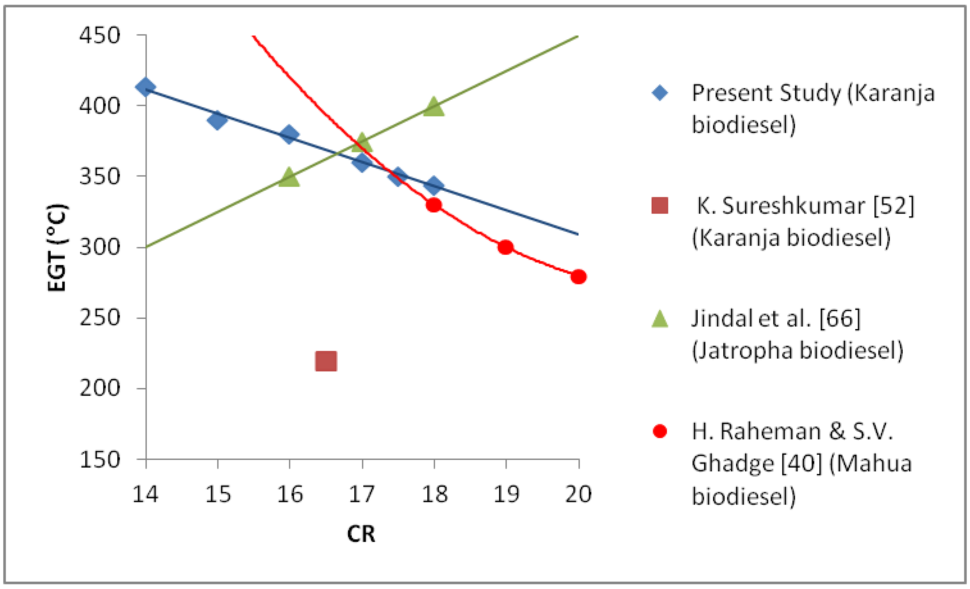


Figure 4.23 Comparison of Variation of EGT with CR at a Load of 12kg and IP of 200bar with Earlier Studies

4.2.2.2 Load

The variations of thermal performance parameters at different loads from 0kg and 12kg in steps of 3kg are presented in Figures 4.24 to 4.32. The load is varied maintaining a constant CR and IP at 18 and 200bar respectively.

Figure 4.24 presents the comparison of variation of BTHE with load for Karanja biodiesel and its blends with Diesel oil at CR of 18 and IP of 200bar. It is observed that BTHE increases significantly with increase in load for all the blends tested. The trend may be due to the reason that relatively less portion of power is lost with increasing load. It can be observed that BTHE decreases with the increase in blend proportion at constant load. The reduction in BTHE can be attributed to lower heating value of the blends. Also, the higher viscosity of the blend may result in slightly reduced atomization and poorer combustion. The early initiation of combustion for Karanja biodiesel and significant early pressure rise before TDC contributes to increased compression work and heat loss resulting in a decrease in BTHE.

At full load (12kg), BTHE for Karanja biodiesel is lower by 8% as compared to Diesel oil. The values of BTHE at full load are 28.65%, 28.75%, 29.32%, 30.05%, 30.5% and 31.25% for Karanja biodiesel, B80, B60, B40, B20 and Diesel oil respectively which indicate that BTHE for Diesel oil and Karanja biodiesel are close to each other.

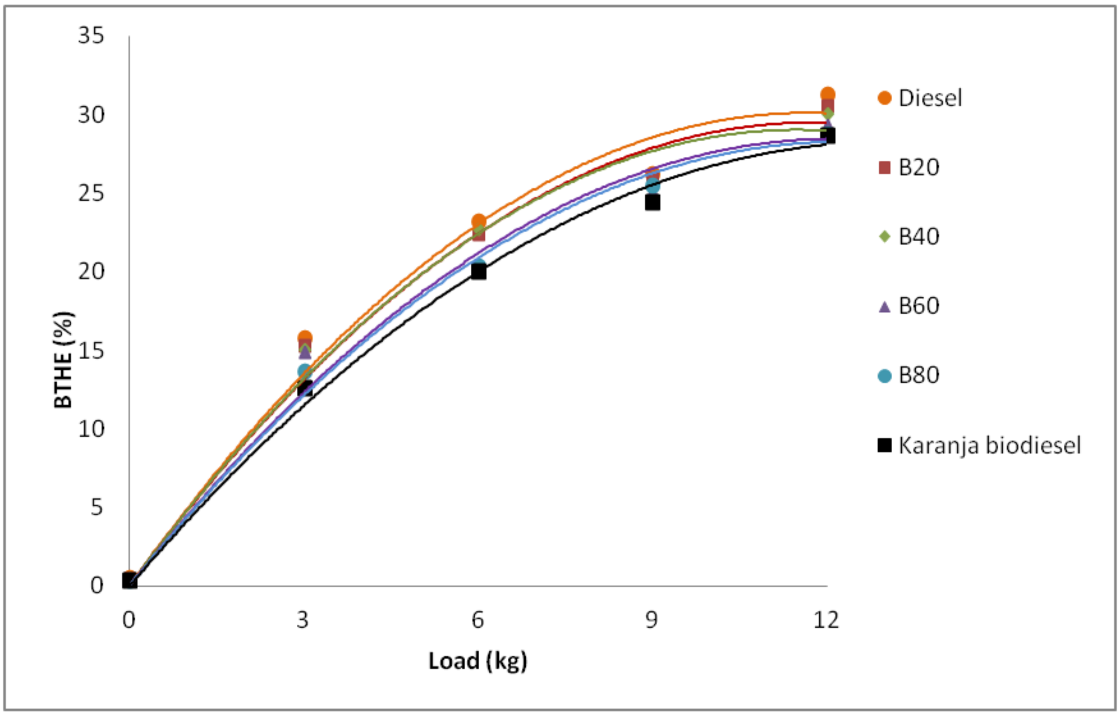


Figure 4.24 Comparison of Variation of BTHE with Load at a CR of 18 and IP of 200bar

The comparison of variation of BSFC with load for Karanja biodiesel and its blends with Diesel oil at CR of 18 and IP of 200bar is shown in Figure 4.25. It is observed that BSFC decreases significantly with the increase in load for all blends as expected. The trend observed may be attributed to the percent increase in fuel required to operate the engine is less than the percent increase in brake power due to relatively less portion of the heat losses at higher loads. It can also be due to increase in fuel consumption with load because the mixture must be enriched beyond stoichiometric. BSFC is higher at low loads because the friction becomes a larger portion of the indicated work and engine starts on rich mixture (Ferguson and Kirkpatrick [98]).

It can be observed that BSFC increases with increase in blend proportion at constant load. The trend observed may be due to lower heating value and relatively higher viscosity of blends due to which the fuel consumption rate is higher. It is observed that BSFC for Karanja biodiesel and its blends are close to that of Diesel oil.

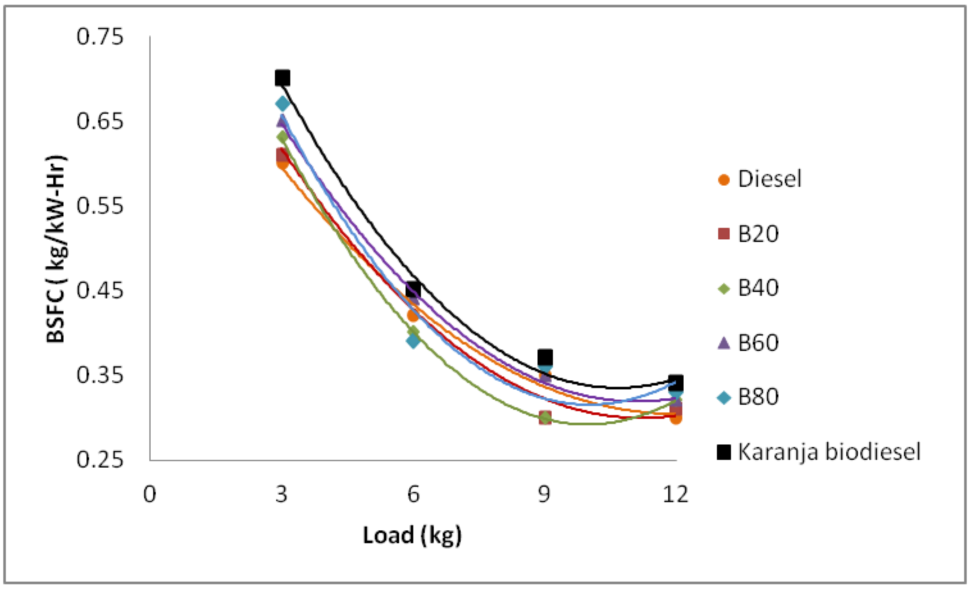


Figure 4.25 Comparison of Variation of BSFC with Load at a CR of 18 and IP of 200bar

At full load, BSFC for Karanja biodiesel is higher by about 18% as compared to Diesel oil. At full load, BSFC for Diesel oil, B20, B40, B60, B80 and Karanja biodiesel are 0.3, 0.31, 0.32, 0.32, 0.33 and 0.34 kg/kWh respectively.

Figure 4.26 shows the comparison of variation of BTHE and BSFC with load at a CR of 18 and IP of 200bar of the present study with that of earlier investigations of Sureshkumar et al. [52], Banapurmath et al. [44], Jindal et al. [66], Raheman & Ghadge [40] and Raheman & Phadatare [17]. Banapurmath et al. [44] and Raheman & Phadatare [17] used Karanja biodiesel and Jindal et al. [66] and Raheman & Ghadge [40] used Jatropha and Mahua biodiesels respectively.

Similar increasing trends of BTHE with load as found in the present study are observed by Banapurmath et al. [44] and Raheman & Phadatare [17] using Karanja biodiesel and Jindal et al. [66] and Raheman & Ghadge [40] using Jatropha biodiesel. At a constant load of 12 kg, preset CR of 18 and IP of 200bar, it is seen that Banapurmath et al. [44] reported a 3% higher value of BTHE while that reported by Raheman & Phadatare [17] was 26% lower as compared to that of present study. Further, the performance in terms of BTHE at a constant load of 12kg, CR of 18 and IP of 200bar when compared using different biodiesel, is found to be 30% and 39% less for Jatropha and Mahua biodiesel respectively with Karanja biodiesel used in the present study.

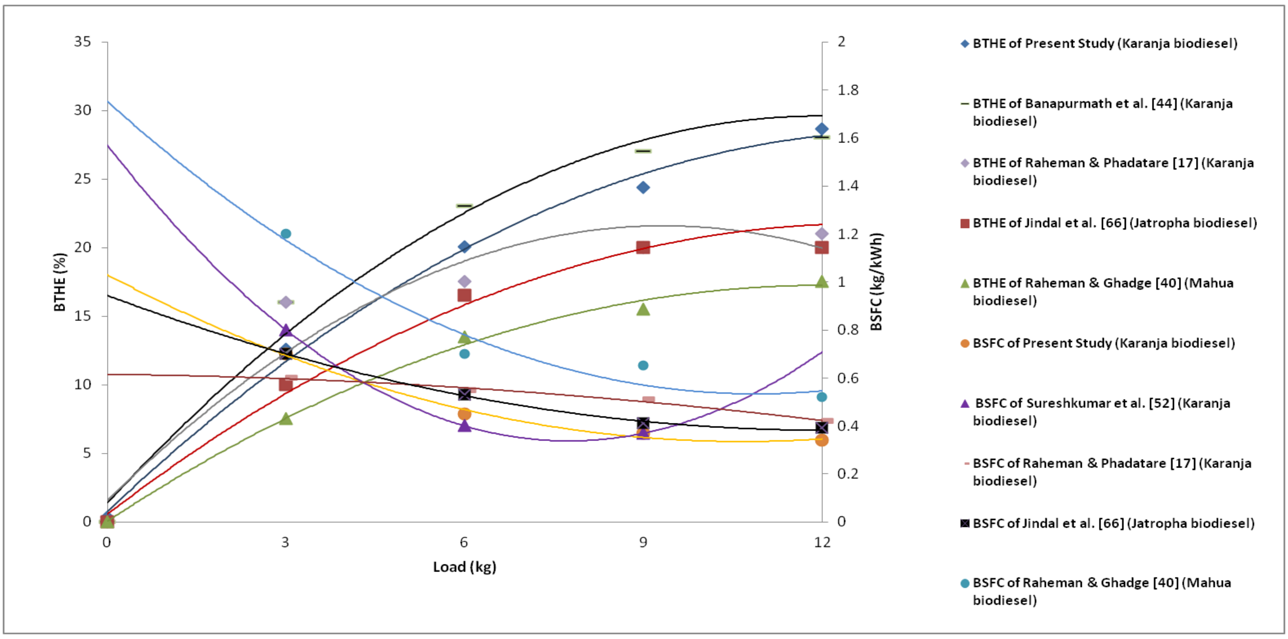


Figure 4.26 Comparison of Variation of BTHE and BSFC with Load at a CR of 18 and IP of 200bar with Earlier Studies

Similar decreasing trends of BSFC with load as found in the present study are observed by S. Banapurmath et al. [44], Raheman & Phadatare [17], Sureshkumar et al. [52], Jindal et al. [66] and Raheman & Ghadge [40]. Sureshkumar et al. [52] used Karanja biodiesel has reported almost same value of BSFC at a load of 9kg as that of the present study. 24% more BSFC as compared to the present study has been reported by Raheman & Phadatare [17] at a load of 12kg using Karanja biodiesel. It is seen that the value of BSFC at a load of 12kg is more by 14% and 50% for Jatropha and Mahua biodiesels respectively as compared to that of Karanja biodiesel used in the present study.

The comparison of variation of BMEP with load for Karanja biodiesel and its blends with Diesel oil at CR of 18 and IP of 200bar is shown in Figure 4.27. It can be observed that BMEP increases linearly with load for all the blends tested. As the load on the engine increases, the rate of fuel consumption increases resulting in greater thermal energy release and hence effective pressure on the piston increases to develop the required brake power.

It is also seen that BMEP is higher for blends than that for pure Diesel oil. However, the trend of variation of BMEP is same for all blends. The higher BMEP for blends may probably be due to higher rate of fuel consumption and greater overall heat release with complete burning of fuel due to the oxygenated nature of biodiesel.

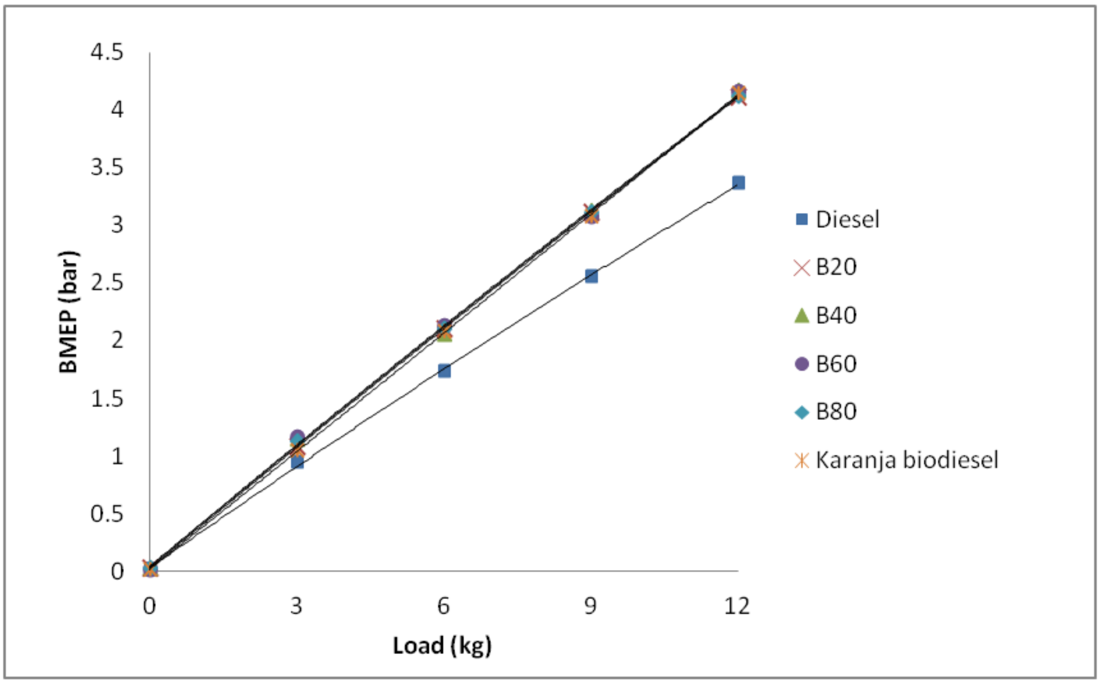


Figure 4.27 Comparison of Variation of BMEP with Load at a CR of 18 and IP of 200bar

The BMEP at full load for Diesel oil, B20, B40, B60, B80 and B100 are 3.36, 4.11, 4.16, 4.15, 4.12 and 4.14 bar respectively. With every increment of 3kg in load, there is an increase of about 1 bar of BMEP for the blends tested.

Figure 4.28 presents comparison of variation of volumetric efficiency with load for Karanja biodiesel and its blends with Diesel oil at CR of 18 and IP of 200bar. It can be seen that load has no significant effect on volumetric efficiency of the engine. At all tested loads, the engine runs with a higher and almost constant volumetric efficiency.

The performance with Karanja biodiesel is better than Diesel oil as fuel in terms of volumetric efficiency. The better performance may be due to greater mass of fuel actually inducted in case of Karanja biodiesel as compared to Diesel oil as fuels.

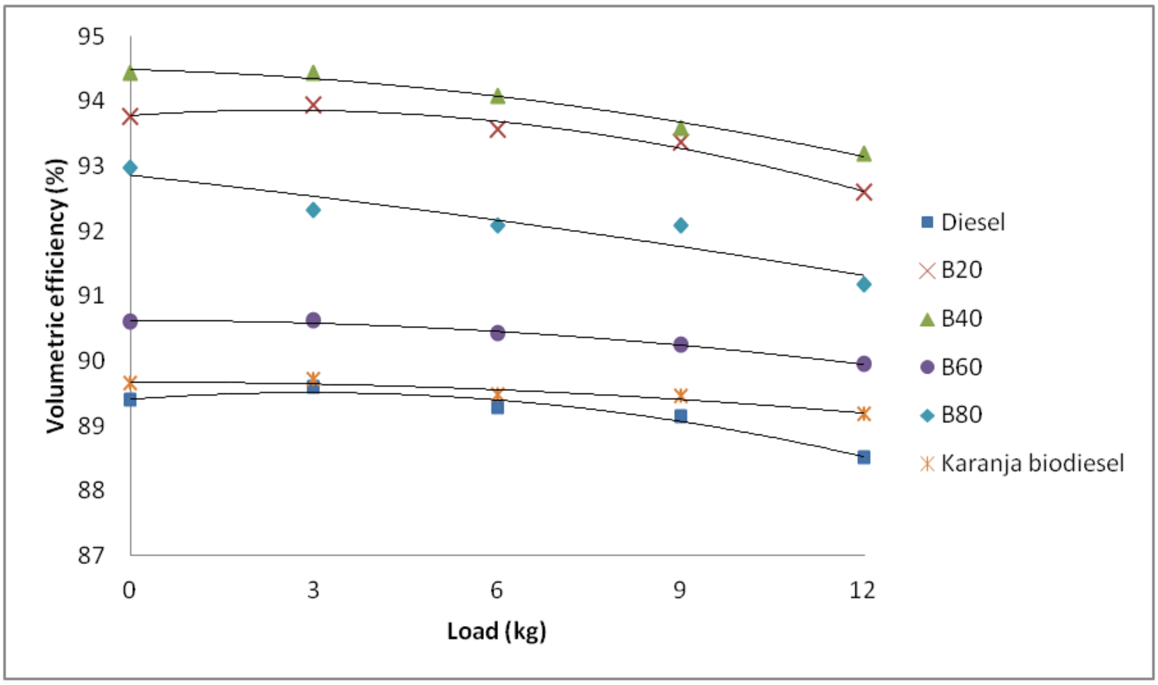


Figure 4.28 Comparison of Variation of Volumetric Efficiency with Load at a CR of 18 and IP of 200bar

Figure 4.29 presents the comparison of variation of HBP with load for Karanja biodiesel and its blends with Diesel oil at CR of 18 and IP of 200bar. It can be observed that HBP increases with increase in load for all the tested fuels. The trend observed is because at higher load fuel consumption increases which results in higher HBP. It can also be observed that HBP reduces with increase in blend proportion at a constant load which may be due to lower calorific value of Karanja biodiesel compared to Diesel oil.

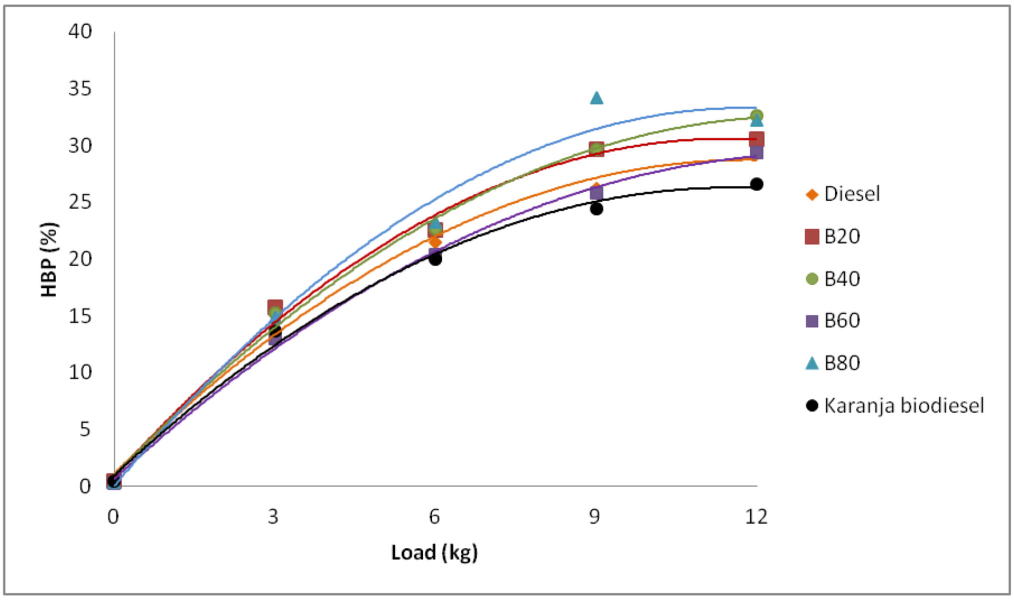


Figure 4.29 Comparison of Variation of HBP with Load at a CR of 18 and IP of 200bar

The values of HBP for Diesel oil are 0.33%, 15.04%, 21.46%, 26.17% and 29.29% and for Karanja biodiesel are 0.41%, 13.63%, 20.03%, 24.39% and 26.64% for the tested loads of 0kg, 3kg, 6kg, 9kg and 12kg. It can be noted that HBP for Karanja biodiesel is close to that of Diesel oil (just about 2% less for Karanja).

Figure 4.30 shows the comparison of variation of HGas with load for Karanja biodiesel and its blends with Diesel oil at CR of 18 and IP of 200bar. It is observed that change in load does not have much effect on the HGas of the engine. It can also be observed that HGas is less for blends compared to Diesel oil. The trend can be attributed to the lower calorific value of Karanja biodiesel. The trend can also probably be due to the result of complete oxidation of the fuel constituents due to inherent oxygen present in Karanja biodiesel.

It is always desirable to have less heat carried away by exhaust from the engine. The values of HGas for highest load tested are 24.08%, 29.03%, 31.18%, 24.92%, 30.35% and 21.71% for Diesel oil, B20, B40, B60, B80 and Karanja biodiesel respectively. HGas for Karanja biodiesel as compared to Diesel oil is 9.8% lower at a load of 12kg.

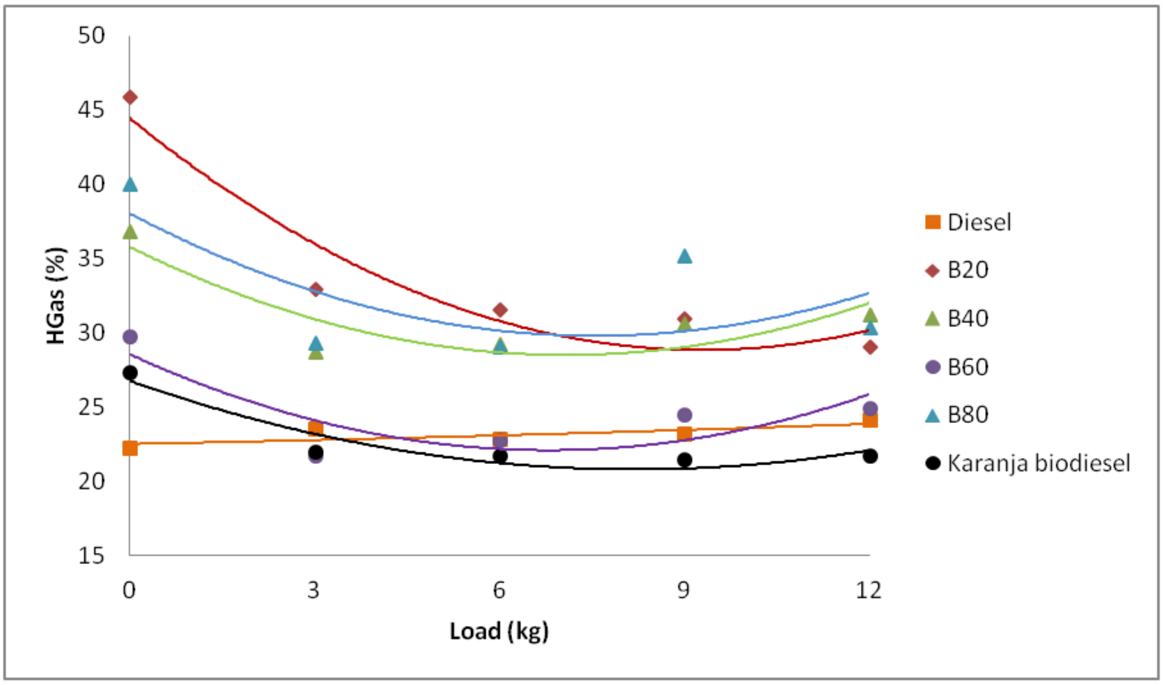


Figure 4.30 Comparison of Variation of HGas with Load at a CR of 18 and IP of 200bar

Figure 4.31 represents the comparison of variation of EGT with load for Karanja biodiesel and its blends with Diesel oil at CR of 18 and IP of 200bar. It can be observed that with the EGT increases with the increase in load. The trend may be due to higher temperature inside the engine cylinder as more fuel is burnt to meet the higher load demand.

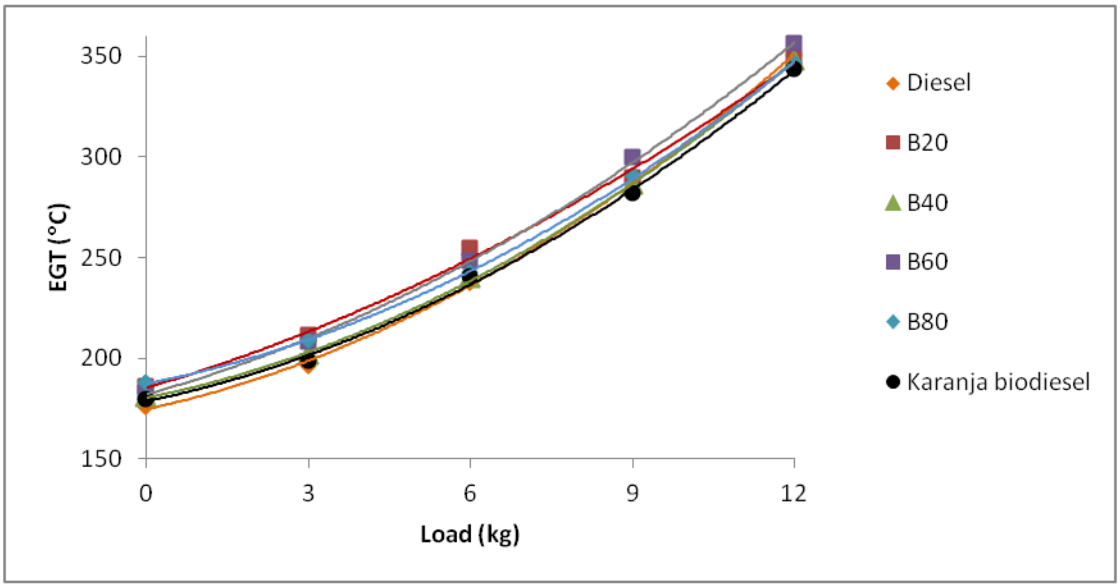


Figure 4.31 Comparison of Variation of EGT with Load at a CR of 18 and IP of 200bar

It is observed that EGT decreases with increase in blend proportion however the decrease is not significant. The trend may be attributed to the lower calorific value and complete combustion of blends due to their inherent oxygen content. The values of EGT at full load for Diesel oil, B20, B40, B60, B80 and Karanja biodiesel are 350.75 ^{0}C, 348.56 ^{0}C, 347.96 ^{0}C, 356.04 ^{0}C, 346.99 ^{0}C and 343.77 ^{0}C respectively.

Figure 4.32 shows the comparison of variation of EGT with Load at a CR of 18 and IP of 200bar of the present study with that of earlier investigations of Sureshkumar et al. [52], Raheman & Phadatare [17], Jindal et al. [66], Raheman & Ghadge [40].

Similar increasing trends of EGT with load as found in the present study are observed by Sureshkumar et al. [52], Raheman & Phadatare [17] and Raheman & Ghadge [40]. Sureshkumar et al. [52] and Raheman & Phadatare [17] used Karanja biodiesel and Raheman & Ghadge [40] used Mahua biodiesel for the study. Sureshkumar et al. [52] has reported 22% less EGT at a constant load of 9kg while 9% less EGT has been reported by Raheman & Phadatare at a constant load of 12kg using Karanja biodiesel as compared to the present study. It is seen that the EGT at 12kg load is more by 11% for Jatropha and less by 2% for Mahua as compared to that of Karanja biodiesel used in the present study.

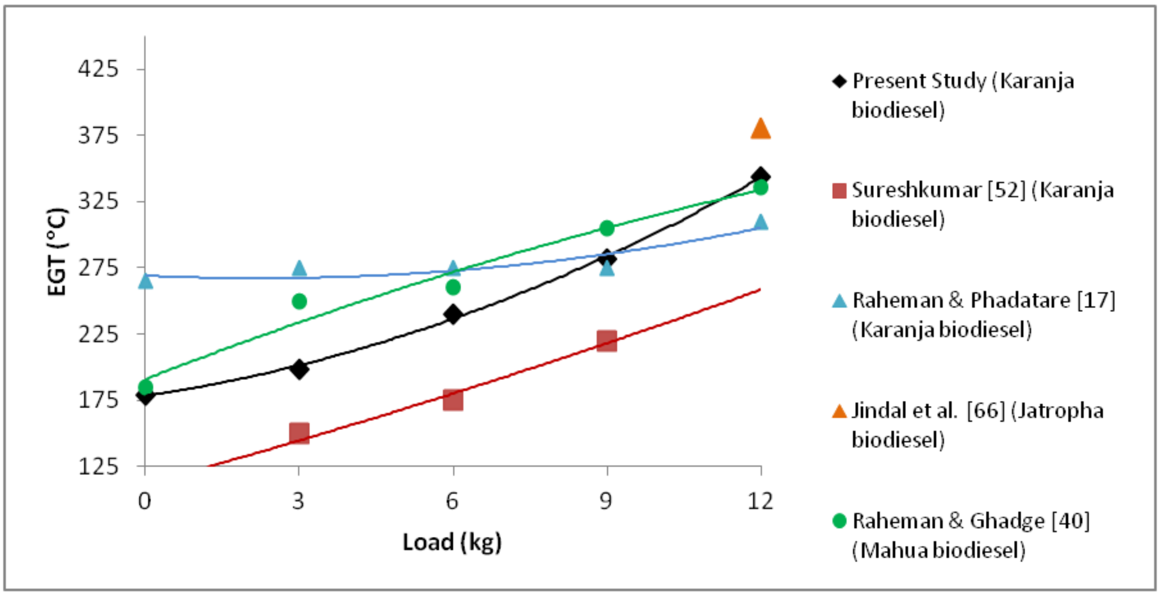


Figure 4.32 Comparison of Variation of EGT with Load at a CR of 18 and IP of 200bar with Earlier Studies

4.2.2.3 Injection Pressure

The variations of thermal performance parameters at different IPs from 150bar and 250bar in steps of 50bar are presented in Figures 4.33 to 4.41. The IP is varied maintaining at a constant CR and load at 18 and 12kg respectively.

The comparison of variation of BTHE with IP for Karanja biodiesel and its blends with Diesel oil at CR of 18 and load of 12kg is presented in Figure 4.33. It is observed that BTHE increases with the increase in IP for all the fuels tested. The trend observed is mainly due to higher degree of atomisation at higher IPs due to which more surface area of fuel droplets is available to be surrounded by oxygen ensuring complete combustion though the amount of fuel being injected is the same across different injection pressures.It is also observed that BTHE increases linearly for Karanja biodiesel, B40, B20 and Diesel oil fuel whereas the variation is non linear for B60 and B80. This can be attributed to experimental perturbations. BTHE is found to increase from 31.34%to 32.7% for Diesel oil and from 27.46% to 28.33% for Karanja biodiesel when the IP increased from 150bar to 250bar. BTHE variation for B60 and B80 has been shown with bestfit trend line.

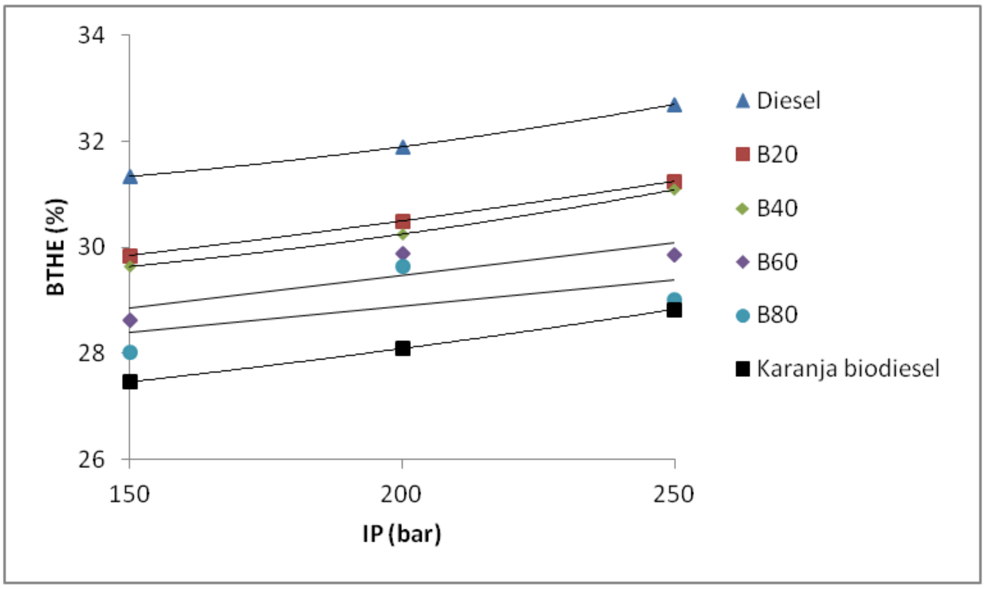


Figure 4.33 Comparison of Variation of BTHE with IP at CR of 18 and Load of 12kg

Figure 4.34 presents comparison of variation of BSFC with IP for Karanja biodiesel and its blends with Diesel oil at CR of 18 and load of 12kg. It can be observed that BSFC reduces with the increase in IP for all the fuels tested. The trend may be due to higher degree of atomisation at higher IP which exposes larger surface area of fuel droplet to the high temperature air inside the cylinder leading to complete combustion of fuel. Hence the power demand is met with less amount of fuel.

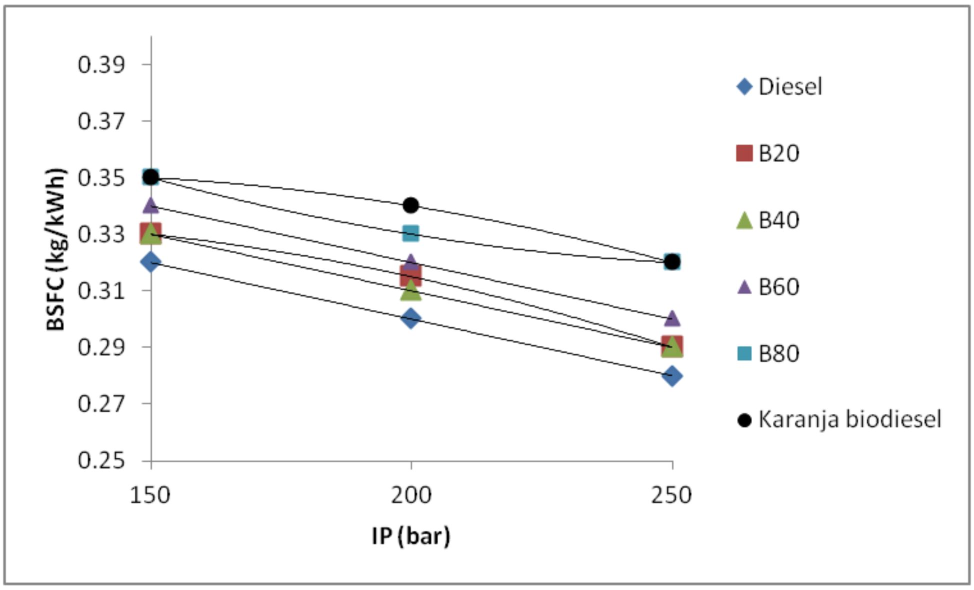


Figure 4.34 Comparison of Variation of BSFC with IP at CR of 18 and Load of 12kg

BSFC is higher for higher blend proportion at a constant IP. The trend is mainly due to greater requirement of fuel flow into the engine because of relatively higher viscosity and lower calorific value of Karanja biodiesel. BSFC for Karanja biodiesel is higher by 0.04 Kg/kW-hr as compared to that of Diesel oil at IP of 250 bar.

Figure 4.35 shows the comparison of variation of BTHE and BSFC with IP at a CR of 18 and load of 12kg of the present study with that of earlier investigations of Jindal et al. [66] using Jatropha biodiesel. BTHE and BSFC variations are inversely proportional.

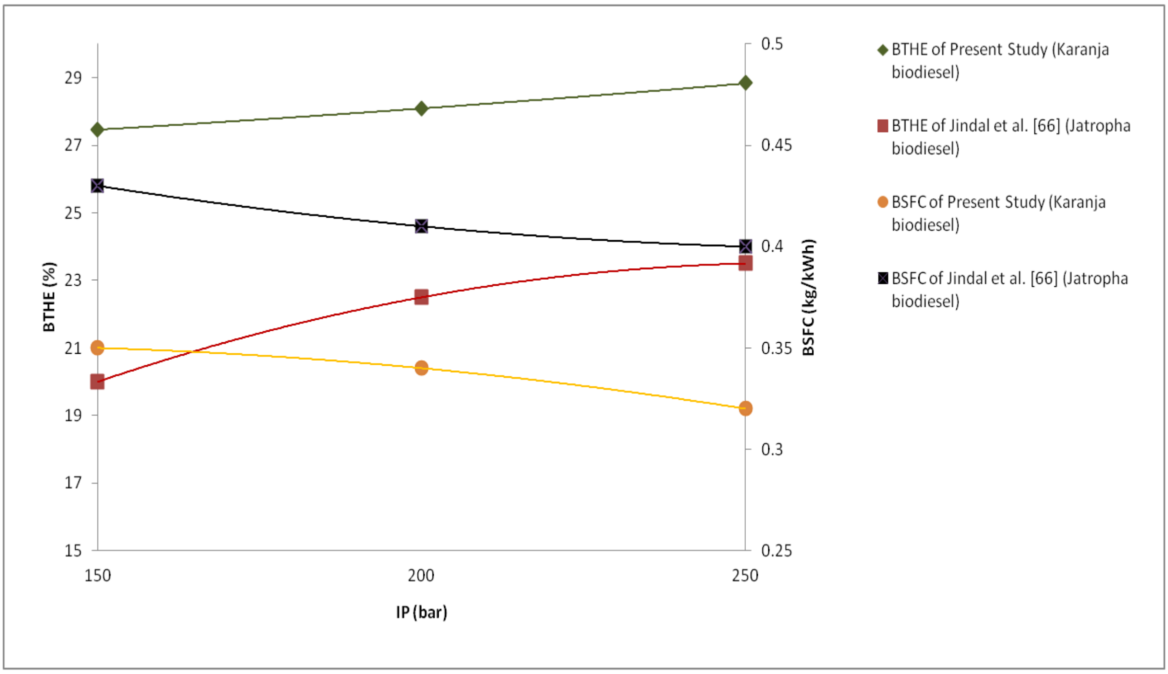


Figure 4.35 Comparison of Variation of BTHE and BSFC with IP at a CR of 18 and IP of 200bar with Earlier Studies

Similar increasing trend of BTHE with load as found in the present study is observed by Jindal et al. [66]. It is seen that the value of BTHE at an IP of 250bar is less by 18% for Jatropha as compared to that of Karanja biodiesel used in the present study. Also, similar decreasing trend of BSFC with load as found in the present study is observed by Jindal et al. [66]. It is seen that the value of BSFC at an IP of 250bar is more by 25% for Jatropha as compared to that of Karanja biodiesel used in the present study.

Figure 4.36 shows the comparison of variation of BMEP with IP for Karanja biodiesel and its blends with Diesel oil at CR of 18 and load of 12kg. It can be observed that the effect of IP on BMEP is not significant. However, a marginal increase in BMEP with increase in IP is observed for all the fuels tested. The trend may probably be due to finer degree of atomisation of fuel at higher IP, leading to proper mixing of fuel with the hot compressed air causing efficient combustion. The trend can also be attributed to more surface area of fuel droplets exposed to hot air in the engine cylinder leading to complete combustion. It can be observed that no specific trend is followed for the variation of BMEP with blend proportion. It can be inferred that BMEP is more a function of the injection pressure but not the blend.

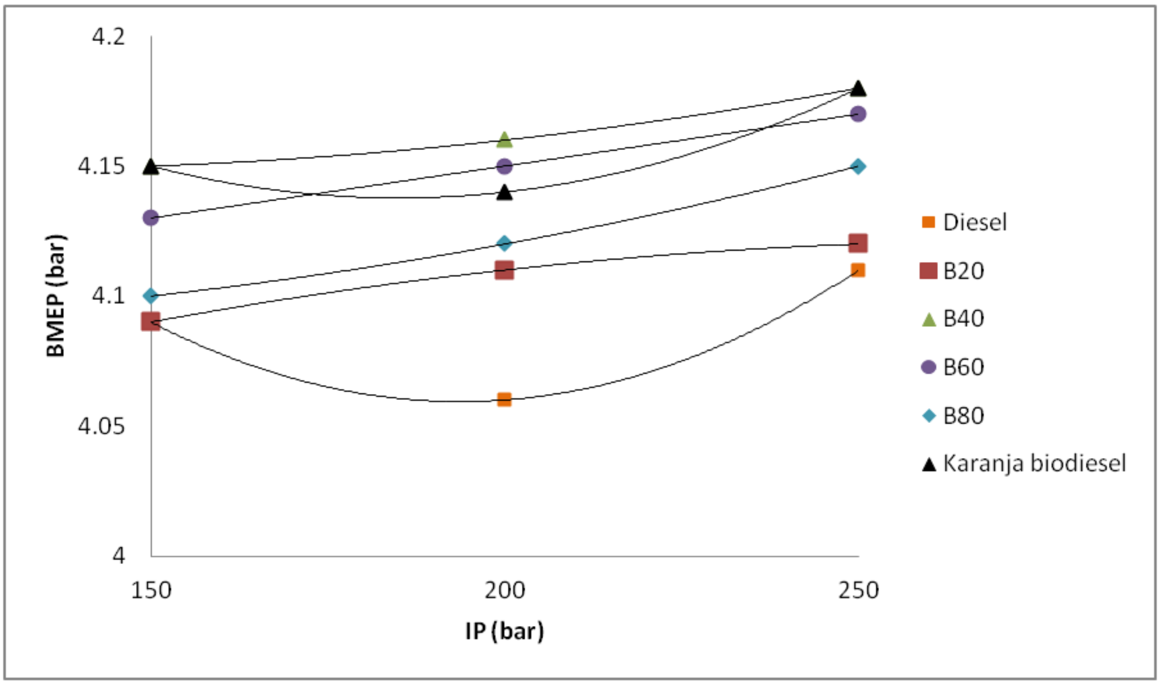


Figure 4.36 Comparison of Variation of BMEP with IP at CR of 18 and Load of 12kg

Figure 4.37 shows the comparison of variation of volumetric efficiency with IP for Karanja biodiesel and its blends with Diesel oil at CR of 18 and load of 12kg. It is observed that volumetric efficiency remains constant with IP for Diesel oil, B60 and Karanja biodiesel.

Although the blends B20, B40 and B80 show some variation in volumetric efficiency, the trend cannot be attributed to nature of fuel or change in IP. It can only be due to experimental perturbations. It is observed that volumetric efficiency for blends is more compared to that of Diesel oil. The trend may be due to more dilution of charge with residual gases with Diesel oil compared to Karanja biodiesel.

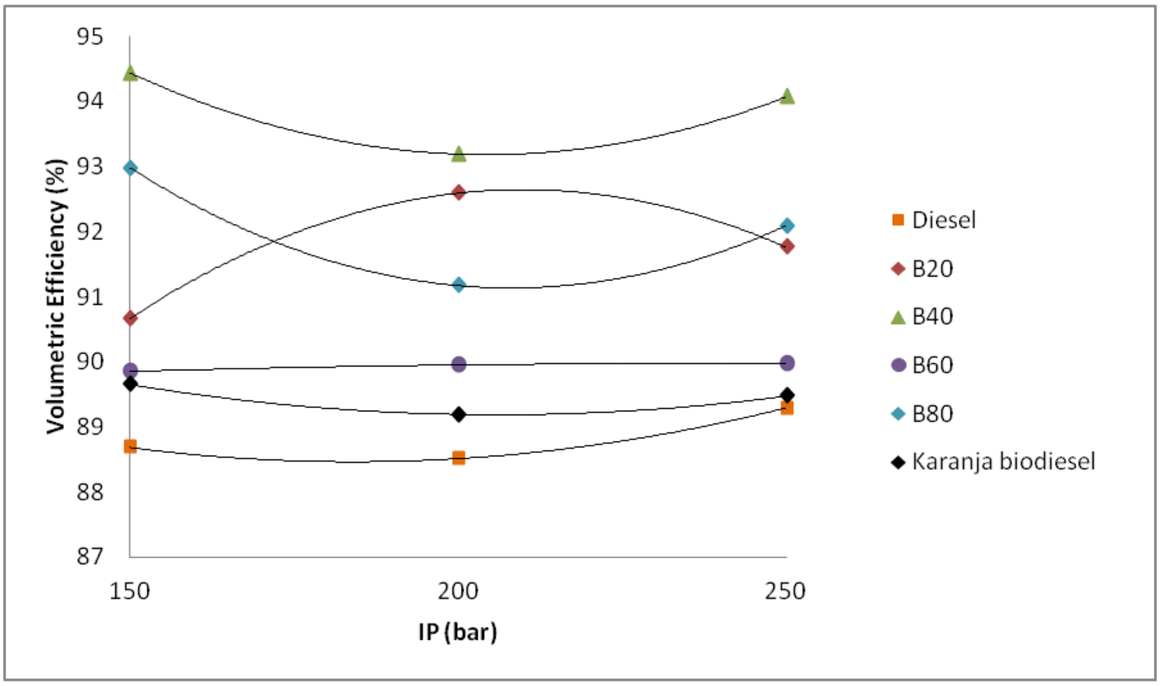


Figure 4.37 Comparison of Variation of Volumetric Efficiency with IP at CR of 18 and Load of 12kg

Figure 4.38 presents the comparison of variation of HBP with IP for Karanja biodiesel and its blends with Diesel oil at CR of 18 and load of 12kg. It is seen that the HBP increases with increase in IP for all the fuels tested. The trend is due to a reason that

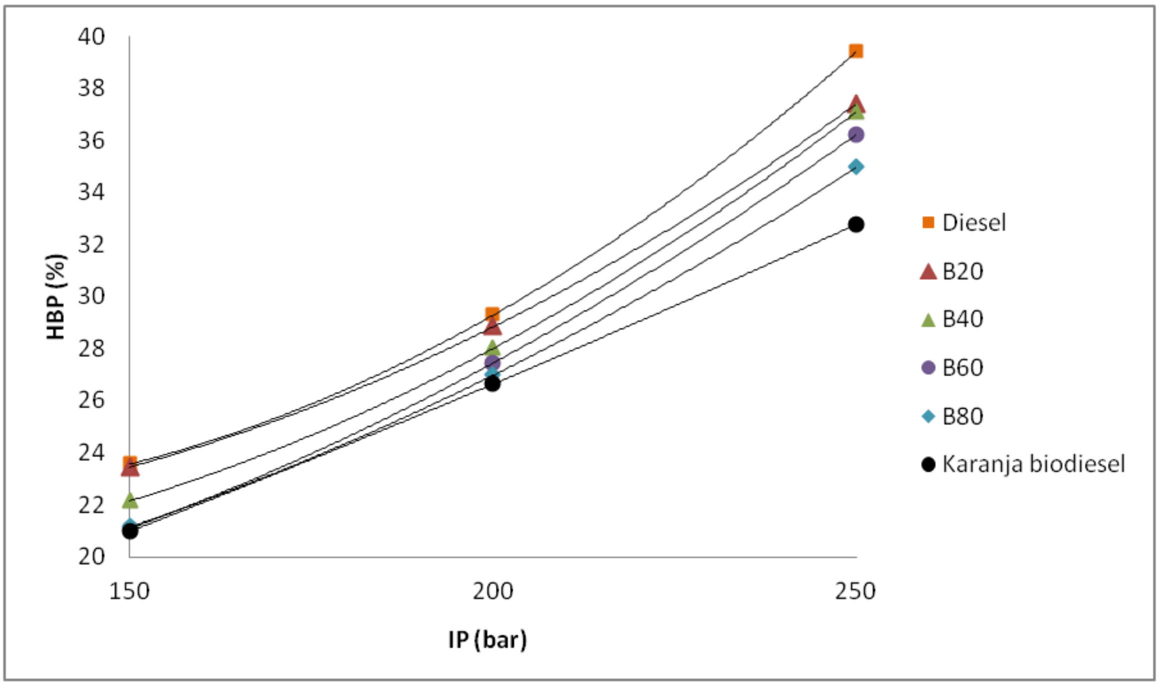


Figure 4.38 Comparison of Variation of HBP with IP at CR of 18 and Load of 12kg

at higher IP, the fuel is atomized to a higher degree which exposes larger surface area of fuel droplet to the high temperature air inside the cylinder which ensures complete combustion of fuel hence more heat is released.

It is also observed that HBP decreases with blend proportion at both lower and higher IPs. It may be due to lower calorific value of Karanja biodiesel. The HBP for Diesel oil for the respective IPs tested are 23.56%, 29.29% and 39.41% and for Karanja biodiesel are 20.98%, 26.64% and 32.78% respectively. It can be noted that HBP for Karanja biodiesel is close to that of Diesel oil at all IPs tested.

Figure 4.39 presents the comparison of variation of HGas with IP for Karanja biodiesel and its blends with Diesel oil at CR of 18 and load of 12kg. It is observed that HGas increases with increase in IP for blends B40, B60, B80 and Karanja biodiesel. The increase is linear for B60, B80 and Karanja biodiesel while it is initially decreasing and subsequently increasing for B40. The increasing trend may due to complete combustion of fuel at higher IP which in turn is due to finer atomisation. It is also observed that HGas for Diesel oil and B20 decrease with increase in IP. The trend may be due to experimental perturbations.

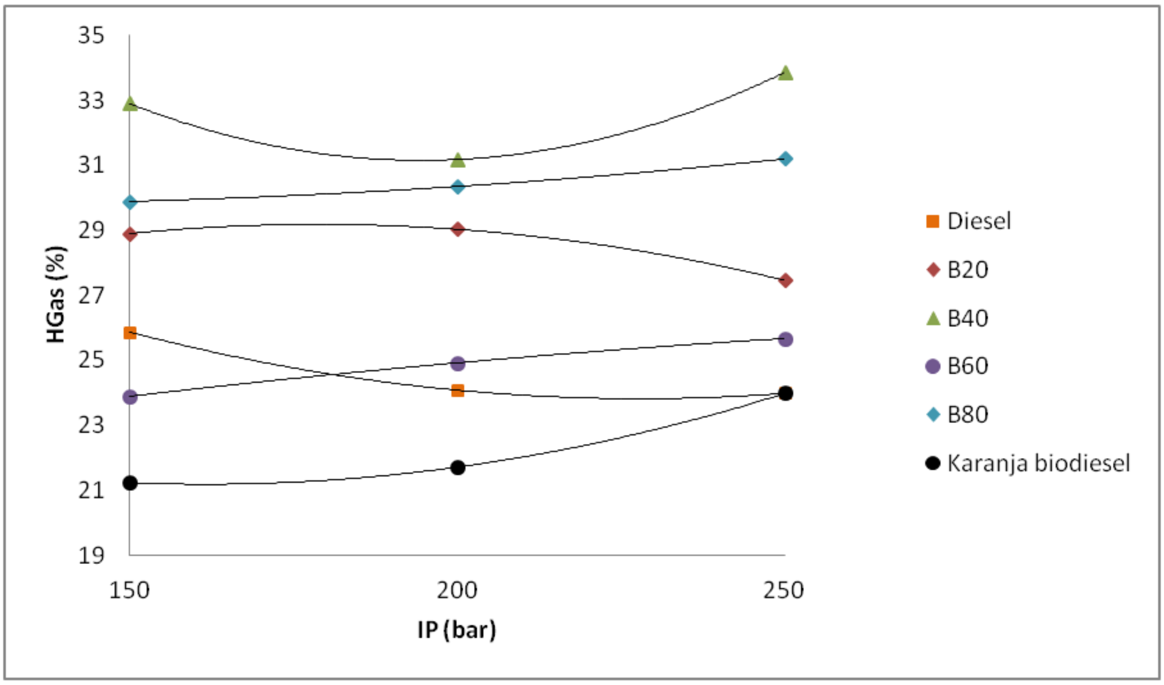


Figure 4.39 Comparison of Variation of HGas with IP at CR of 18 and Load of 12kg

The HGas at tested IPs for Diesel oil are 25.87%, 24.08% and 23.98% and Karanja biodiesel are 21.23%, 21.71% and 23.98% respectively. HGas is observed to be less for Karanja biodiesel compared to Diesel oil. The reason may be complete combustion of Karanja biodiesel as it is an oxygenated fuel.At IP of 200 bar the values of HGas are identical for Karanja biodiesel & Diesel oil.

The comparison of variation of EGT with IP for Karanja biodiesel and its blends with Diesel oil at CR 18 and Load of 12kg is presented in Figure 4.40. It is observed that EGT increases linearly with the increase in IP for all the fuels tested. The trend is due to complete combustion of fuel at higher IP due to which more heat is generated in the exhaust. The complete combustion is obviously due to better atomisation at higher IP. It is also observed that EGT is less for blends compared to Diesel oil. The trend may be due to lower calorific value of Karanja biodiesel.

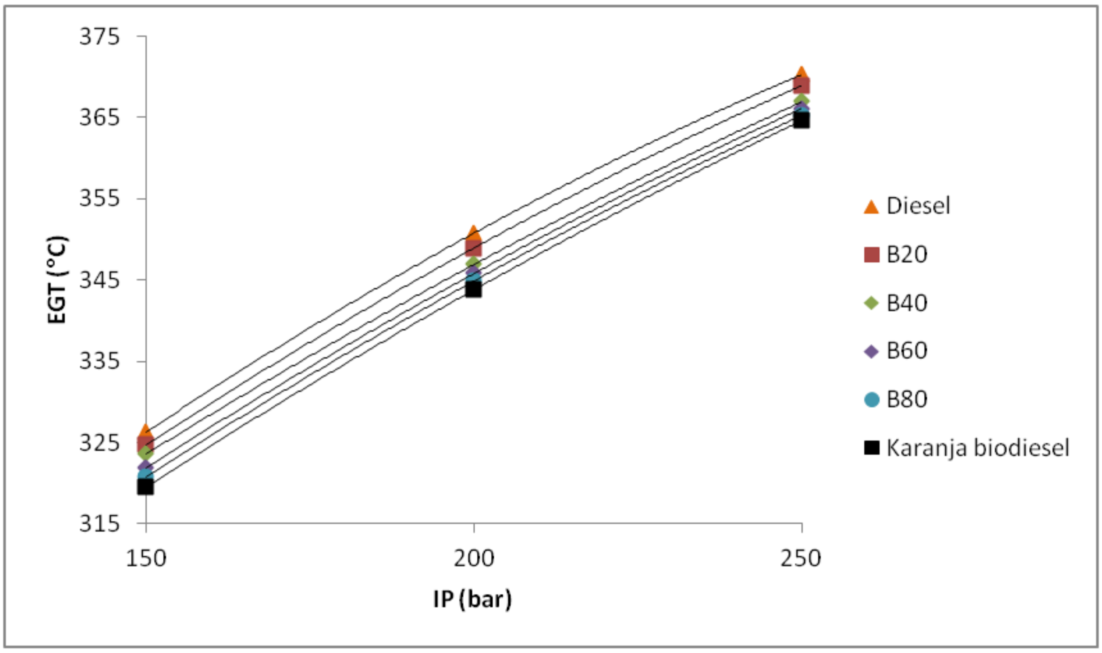


Figure 4.40 Comparison of Variation of EGT with IP at CR of 18 and Load of 12kg

The increase in EGT around 15% (50^0C) as the IP is increases from 150 to 250 bar. At IP of 250 bar, the values of EGT for Karanja biodiesel, B80, B60, B40, B20 and Diesel oil are found to be 364.59 ^{0}C, 365.23 ^{0}C, 366.09 ^{0}C, 366.9 ^{0}C, 368.9 ^{0}C and 370.26 ^{0}C respectively. It can be noted that EGT for Karanja biodiesel is about 2% less than that for Diesel oil.

A comparison of variation of EGT with IP of present study with Jindal et al. [66] is shown in Figure 4.41. They studied the variation of EGT with IP using Jatropha biodiesel. The EGT for Jatropha biodiesel is observed to be higher than Karanja biodiesel at all IPs. It is seen that the EGT at an IP of 250bar is 13% more for Jatropha biodiesel as compared to that of Karanja biodiesel used in the present study.

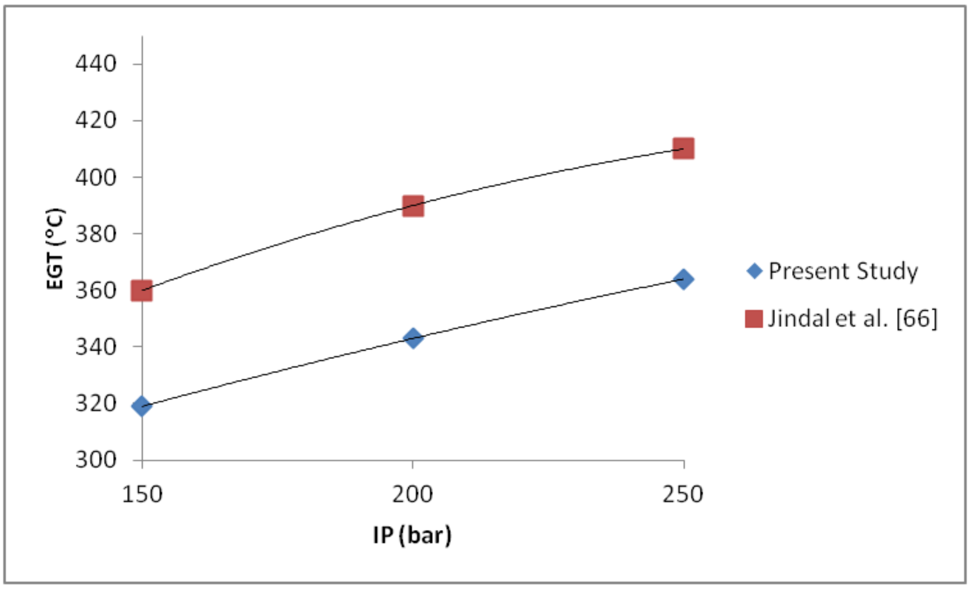


Figure 4.41 Comparison of Variation of EGT with IP at a CR of 18 and Load of 12kg of Present Study with Jindal et al. [66]

4.2.3 Emission Characteristics

The combustion product emissions from diesel engines are hydrocarbon emissions and form a significant part of the engine exhaust. Hydrocarbons in the exhaust may also condense to form white smoke during engine starting and warm up. Specific hydrocarbon compounds in exhaust gases are a source of diesel odour. Diesel engines are also a source of different particulate emissions. Between 0.2 to 0.5 % of the fuel mass is emitted as small particles which consist primarily of soot with some additional hydrocarbon material. Diesel engines are not a very significant source of carbon monoxide emission comparatively. The other exhaust emissions include NO_x i.e nitrogen oxides and aldehydes which are a significant source of air pollution. Half of NO_x, CO and HC pollutants in air are primarily because of IC engines. NO_x may react with solar radiation to form ozone. Hydrocarbons cause cellular mutations and are responsible for ground level ozone formation. SO_x formation is significant in diesel engines. However with the use of biofuels as fuels, the harmful constituents of HC, CO, SO_x can significantly be brought down.

The emission constituents considered for evaluation are Carbon dioxide (CO_2), Carbon monoxide (CO), Oxides of Nitrogen (NO_x), Unburned Hydrocarbon (HC), Oxygen (O_2), Smoke intensity (HSU) and Oxides of Sulphur (SO_x). The variations of the emission constituents with CR, Load and IP are interpreted in three different sections 4.2.3.1, 4.2.3.2 and 4.2.3.3 respectively. For each constituent, the variations for Diesel oil, Karanja biodiesel and blends of Karanja biodiesel with diesel as fuels are superposed and analysed.

4.2.3.1 Compression Ratio

The variations of exhaust emission constituents at different values of CRs at 14, 15, 16, 17, 17.5 and 18 with a constant rated load of 12kg and IP of 200bar is presented in Figures 4.42 to 4.50.

Figure 4.42 presents the comparison of variation of CO with CR for Karanja biodiesel and its blends with Diesel oil at load of 12kg and IP of 200 bar. It can be observed that CO emission decreases with increase in CR.

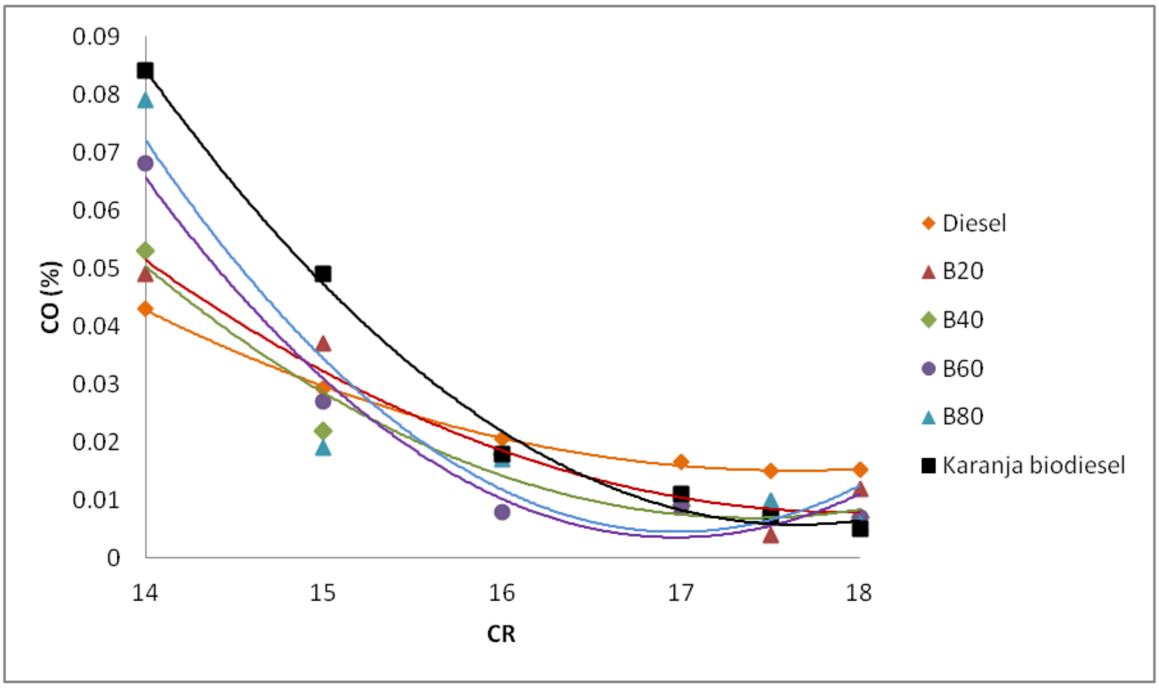


Figure 4.42 Comparison of Variation of CO with CR at Load of 12kg and IP of 200bar

The trend may be due to better combustion of fuel at higher CR which in turn is due to high air temperature inside the cylinder. It can also be observed that at low CRs, CO emissions are higher and at higher CRs, it is lesser for blends compared to Diesel oil. The lower CO emissions for blends may be due to the oxygenated nature of Karanja biodiesel due to which more of carbon gets oxygenated forming CO. The higher CO emissions for the blends at low CR may be due to higher viscosity of biodiesel which leads to poor atomization and incomplete combustion.

The values of CO emissions for Diesel oil at tested CRs are 0.043%, 0.0293%, 0.0205%, 0.0166%, 0.015% and 0.0152% and for Karanja biodiesel 0.084%, 0.049%, 0.018%, 0.011%, 0.007% and 0.005% respectively. It can be noted that CO emissions are 66% less for pure Karanja biodiesel as compared to Diesel oil at CR of 18.

The comparison of variation of HC with CR for Karanja biodiesel and its blends with Diesel oil at a load of 12kg and IP of 200 bar is shown in Figure 4.43. The unburnt hydrocarbons (HC) are generated in the exhaust as the result of incomplete combustion of fuel. Hydrocarbons cause eye irritation and choking sensations. They are major contributors to the characteristic diesel exhaust smell and also have a negative environmental effect, being an important component of smog. Hydrocarbon from diesel engines come primarily from (i) the fuel trapped in the injector at the injection that later diffuses out, (ii) the fuel mixed into the air surrounding the burning spray so lean that it cannot burn, (iii) the fuel trapped along the walls by crevices, deposits, or oil due to impingement by the spray.

It can be observed from the figure that HC emissions decrease with increase in CR for all the fuels tested. The trend observed may be due to complete combustion of fuel which is due to high heat of compressed air at higher CR.

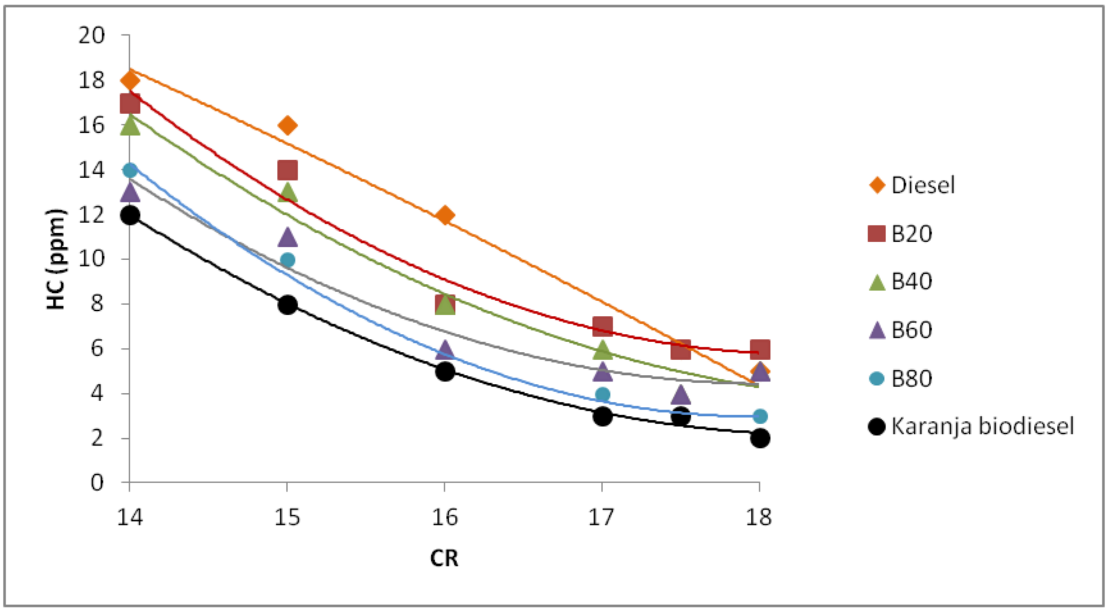


Figure 4.43 Comparison of Variation of HC with CR at Load of 12kg and IP of 200bar

It is also observed that HC decreases with the increase in blend proportion. The trend may be due to better combustion of Karanja biodiesel due to its oxygenated nature. The mean percentage decrease in HC emission with Karanja biodiesel as compared to Diesel oil is of the order of 60% except at 14 CR where it is around 33%.

Figure 4.44 shows the comparison of variation of HC emissions with CR at a load of 12kg and IP of 200bar of the present study with that of earlier investigations of Banapurmath et al. [44] and Jindal et al. [66]. The biodiesel used by the investigators are Karanja and Jatropha respectively.

Banapurmath et al. [44] studied HC emissions at a constant CR of 17.5 using Karanja biodiesel. The investigators reported 59ppm higher HC emissions as compared to that of the present study. The trends of HC emissions with CR observed by Jindal et al. [66] using Jatropha biodiesel are not as found in the present study. The HC emissions are almost constant at all the CRs for Jatropha biodiesel. It is seen that the HC emissions at a CR of 18 is more by 13ppm for Jatropha as compared to that of Karanja biodiesel used in the present study.

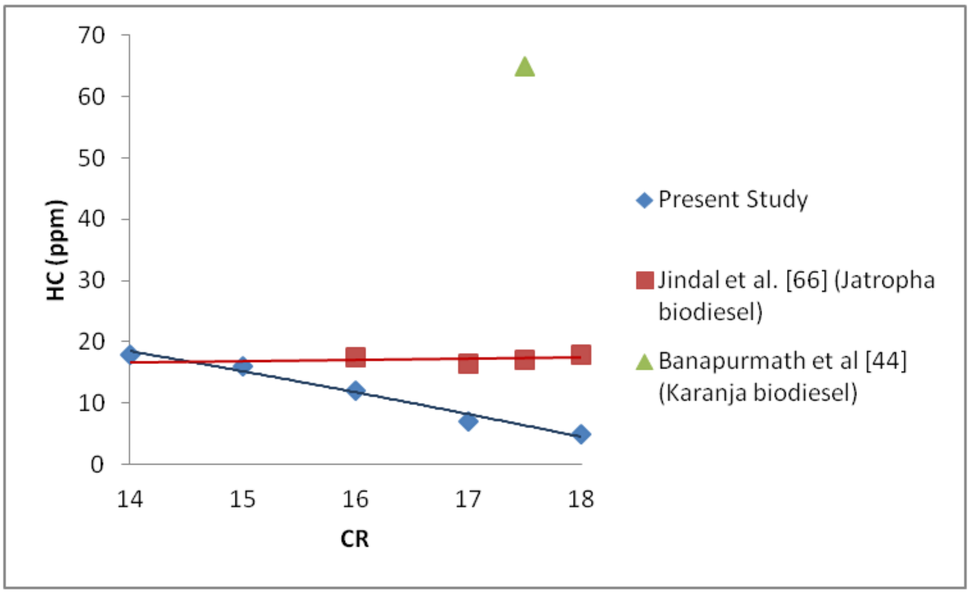


Figure 4.44 Comparison of Variation of HC with CR at a Load of 12kg and IP of 200bar Present Study with Earlier Studies

Figure 4.45 shows comparison of variation of NO_x with CR for Karanja biodiesel and its blends with Diesel oil at load of 12kg and IP of 200 bar. Nitrogen oxides are formed throughout the combustion chamber during the combustion process due to the reaction of atomic oxygen and nitrogen. Reactions forming NO_x are very temperature dependent and NO_x emissions from the engine are low at lesser loads. NO_x emissions consist of nitrogen dioxide (NO_2) and nitric oxide (NO).

It can be observed from the figure that NO_x emissions increases for blends while it decreases for Diesel oil with increase in CR. The trend observed for blends is because at lower CR, less oxygen is available from blends to form NO_x due to less heat of compressed air, but at higher CR greater availability of oxygen and higher heat of compressed air initiates early combustion ensuring complete oxidation of fuel.

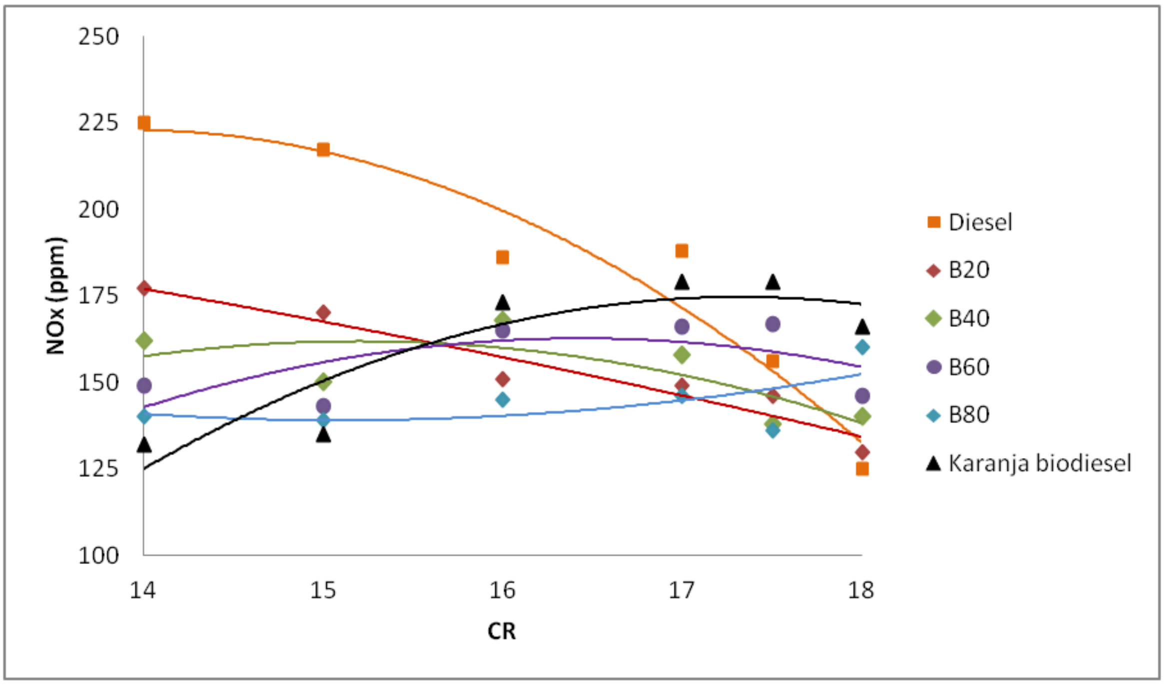


Figure 4.45 Comparison of Variation of NO_X with CR at Load of 12kg and IP of 200bar

The values of NO_x emissions in ppm for Diesel oil at CRs of 14, 15, 16, 17, 17.5 and 18 are 225, 217, 186, 188, 156, and 125 and for pure Karanja biodiesel are 132, 135, 173, 179, 179, and 166 respectively. At a CR of 14 the NO_X emissions for Diesel oil are higher by 70.4% and at CR 18 they are lower by 24.6% as compared to Karanja biodiesel. At higher CRs of 17 to 18 NO_x emission is found to be higher as compared to that of Diesel oil

Figure 4.46 presents the comparison of variation of CO_2 with CR for Karanja biodiesel and its blends with Diesel oil at load of 12kg and IP of 200bar. The amount of

CO_2 in the exhaust is an indication of degree of complete oxidation of the carbon constituent of the fuel and hence indicates the extent of conversion of chemical energy into thermal energy. It is observed from the figure that CO_2 emission initially decrease, reach the lowest and subsequently increase with the increase in CR for all the fuels tested. It is also observed that CO_2 emission is higher for Karanja biodiesel compared to Diesel oil at all CRs. The trend may be due to complete oxidation of carbon present in the biodiesel due to its inherent oxygen content.

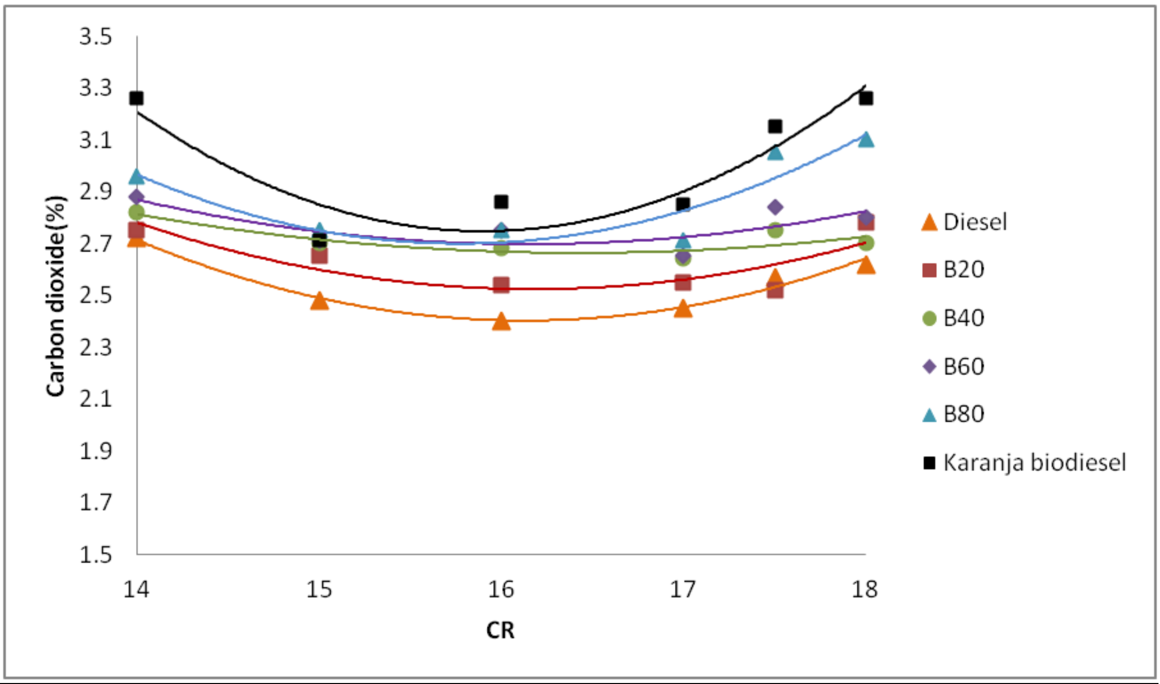


Figure 4.46 Comparison of Variation of CO_2 with CR at Load of 12kg and IP of 200bar

As the CR increases from 14 to 18 the values of CO_2 emission for Diesel oil are 2.72%, 2.48%, 2.4%, 2.45%, 2.57% and 2.62% and for Karanja biodiesel are 3.26%, 2.71%, 2.86%, 2.85%, 3.15% and 3.26%. It can be noted that the percentage decrease in CO_2 emissions between Karanja Biodiesel and Diesel oil over the range of CRs is 22%.

The comparison of variation of O_2 emission with CR for Karanja biodiesel and its blends with Diesel oil at load of 12kg and IP of 200bar is presented in Figure 4.47. It can be observed that O_2 content in the exhaust increases with increase in CR. At low CRs, O_2 emission is less with Diesel oil as compared to Karanja biodiesel and its blends. However, at higher CRs of 16, 17 and 18, O_2 emission is found to be almost the same for all the tested fuels existing in the range of 17 to 18% indicating only a marginal increase at higher CRs which in turn indicates that the O_2 emission is not affected by CR except for Diesel oil at CR of 14 where it is around 1% less. This is inconfirmity with the variation of A/F with CR. The trend observed may be due to high temperature inside the cylinder at higher CR which results in better combustion of fuel releasing more oxygen.

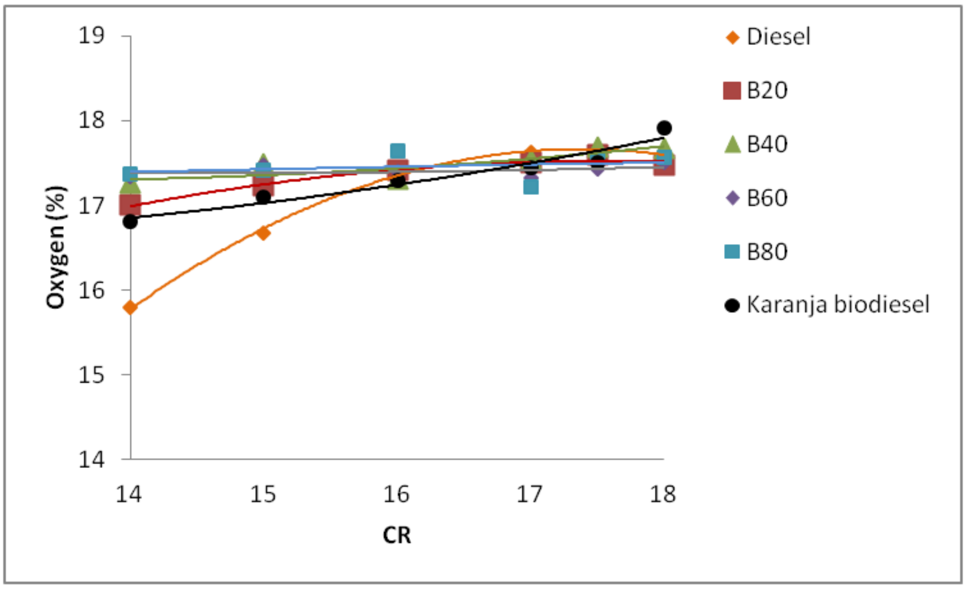


Figure 4.47 Comparison of Variation of O_2 with CR at Load of 12kg and IP of 200bar

It is also seen that O_2 is more for Karanja biodiesel than Diesel oil at all CRs. The trend may be due to inherent presence of oxygen in the biodiesel. The O_2 in the exhaust increases by 12%, 2.76%, 2.37%, 0.92%, 1.1% and 6.54% for Diesel oil, B20, B40, B60, B80 and Karanja biodiesel respectively when the CR is increased from 14 to 18. The O_2 emission is 2.8% more for Karanja biodiesel than Diesel oil.

Figure 4.48 gives the comparison of variation of SO_x with CR for Karanja biodiesel and its blends with Diesel oil at load of 12kg and IP of 200bar. Sulphur dioxide is a colourless toxic gas with a characteristic of irritating odour. Oxidation of sulphur dioxide produces sulphur trioxide which is the precursor of sulphuric acid which, in turn, is responsible for the sulphate particulate matter emissions. Sulphur oxides have a profound impact on environment being the major cause of acid rains. Sulphur dioxide is generated from the sulphur present in Diesel oil fuel. The concentration of SO_2 in the exhaust gas depends on the sulphur content of the fuel.

It can be observed from the figure that SO_x decreases with the increase in CR for all the fuels tested. It can also be seen that SO_x emissions are highest for Karanja biodiesel at low CRs and it reduces with increase in CR and becomes minimum at highest CR. At lower CR, there is a chance of incomplete combustion. Hence formation of carbon dioxide is less due to which the oxygen present in fuel as well as that in air reacts with sulphur present in air in case of Karanja biodiesel and sulphur present in fuel in case of blends forming oxides of sulphur. With the increase in CR the oxides of sulphur reduce by 67, 51, 59, 55, 47 and 127 ppm for Diesel oil, B20, B40, B60, B80 and Karanja biodiesel respectively. This is consistent with lower air fuel ratio at higher CR for the fuels tested though oxygen present in the exhaust is same at higher CRs (Refer Appendinx III, P291-a for the results of variation of A/F ratio with CR, load and IP).

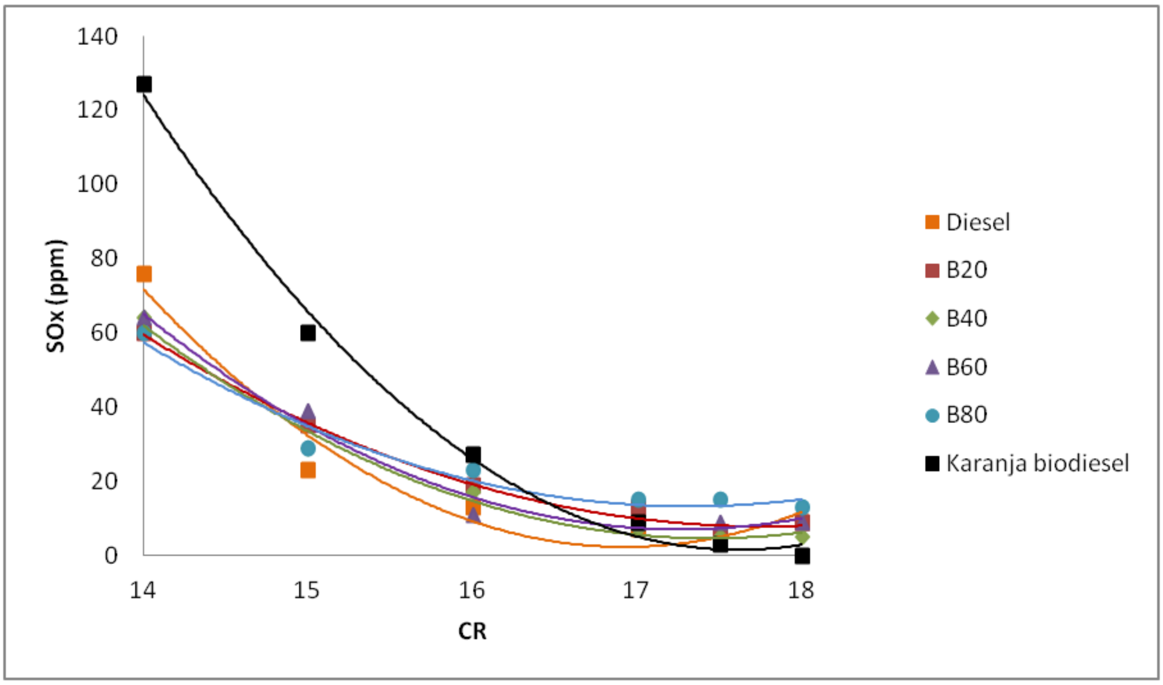


Figure 4.48 Comparison of Variation of SO_X with CR at Load of 12kg and IP of 200bar

The Figure 4.49 illustrates the variation of smoke intensity with CR for all the fuels tested. Smoke intensity is also known as soot or particulate emissions in the form of solid carbon, unburnt hydrocarbons in the exhaust gases. These emissions are a major problem with diesel engine's heterogeneous combustion. Even partial oxidation products are considered under this category. Smoke intensity is measured in Hartridge Smoke Unit (HSU) and expressed in '%'. Higher HSU indicates that either more fuel is burnt or the fuel burning process is hindered by some unfavourable conditions. It is observed from the figure that smoke is less at higher CR for all the fuels. This may be due to a reason that at higher CR the heat of the compressed air is high enough to cause complete combustion of fuel. At lower CR, smoke is high as incomplete combustion of fuel takes place.

Generally smoke intensity is high for Karanja biodiesel & its blends because they have more viscosity than pure Diesel oil and this has a negative effect on the combustion

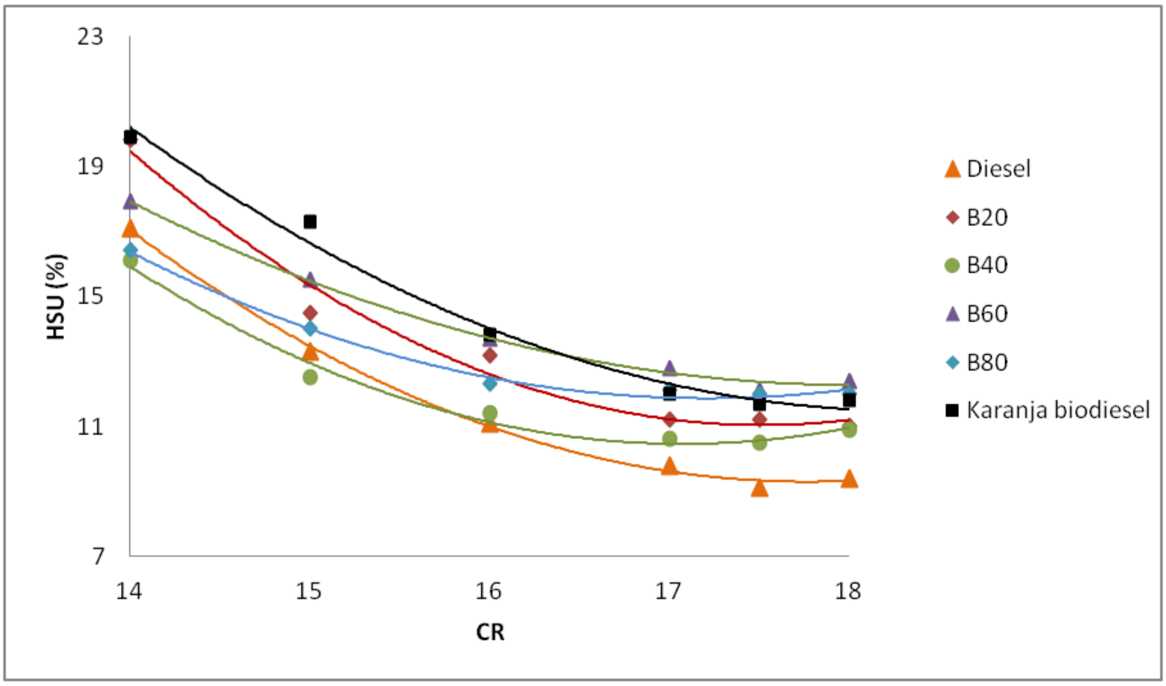


Figure 4.49 Comparison of Variation of HSU with CR for at Load of 12kg and IP of 200bar

process. The HSU decreases by 45% for Diesel oil and 40% for pure Karanja biodiesel as the CR increases from 14 to 18. At a CR of 18, the HSU for pure Karanja biodiesel is 2.4% higher than that for Diesel oil.

The engine may be operated with any higher CR ranging from 16 to 18 so far as the emission constituents such as CO, HC, CO_2, O_2 and SO_x are concerned. The reason is the mentioned emission constituents are least and remain constant in the CR range of 16 to 18. However the NO_x emissions are found to be more at higher CRs with Karanja biodiesel as compared to Diesel oil. Therefore, the selection of CR can be made based on the relative combined effect on thermal performance and emissions. It is preferable to operate the engine at CR 18, as the CR is also selected earlier for thermal performance evaluation.

Figure 4.50 shows the comparison of variation of NO_x emissions and smoke (HSU) of exhaust gas with CR at a load of 12kg and CR of 18 of the present study with that of earlier investigations of Sureshkumar et al. [52], Jindal et al. [66] and Raheman & Phadatare [17]. The biodiesel used by the investigators are Karanja and Jatropha.

Sureshkumar et al. [52] and Raheman & Phadatare [17] studied NO_x at a constant CR of 16.5 and 16 respectively using Karanja biodiesel. Sureshkumar et al. [52] reported 8% higher NO_x while 95% less NO_x is reported by Raheman & Phadatare [17] as compared to that of the present study. Similar increasing trends of NO_x with CR as found in the present study are observed by Jindal et al. [66] using Jatropha biodiesel. It is seen that the NO_x emissions at a CR of 18 is more by 32% for Jatropha as compared to that of Karanja biodiesel used in the present study.

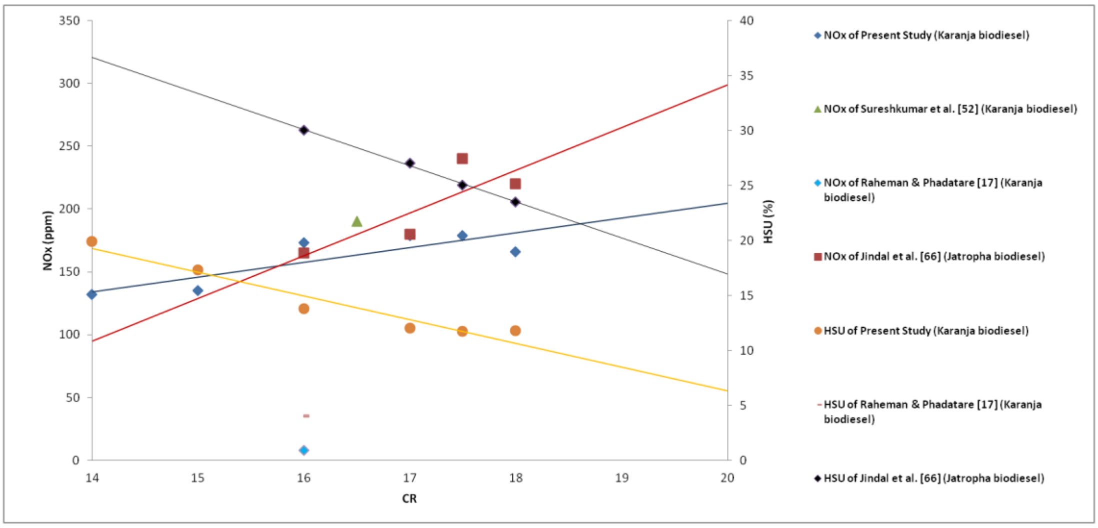


Figure 4.50 Comparison of Variation of NO_x and HSU with CR at a Load of 12kg and IP of 200bar with Earlier Studies

Raheman & Phadatare [17] reported a 71% reduction in HSU using Karanja biodiesel as compared to that of the present study. Similar decreasing trends of HSU with CR as found in the present study are observed by Jindal et al. [66] using Jatropha biodiesel. It is seen that HSU at a CR of 18 is almost double for Jatropha as compared to that of Karanja biodiesel used in the present study.

4.2.3.2 Load

The variations of exhaust emission constituents at different loads from 0kg to 12kg in steps of 3kg at a constant CR of 18 and IP of 200bar are presented in Figures 4.51 to 4.58.

Figure 4.51 gives the comparison of variation of CO emission with Load for Karanja biodiesel and its blends with Diesel oil at load of 12kg and IP of 200bar. It is observed that CO emissions increase with the increase in load. The trend can be attributed to more fuel being consumed at higher loads which means rich running of the engine and there being insufficient oxygen to convert all the carbon in the fuel to carbon dioxide. It is a well known fact that the formation of CO as an emission constituent in the exhaust gases is mainly due to incomplete oxidation of carbon constituent in the fuel.

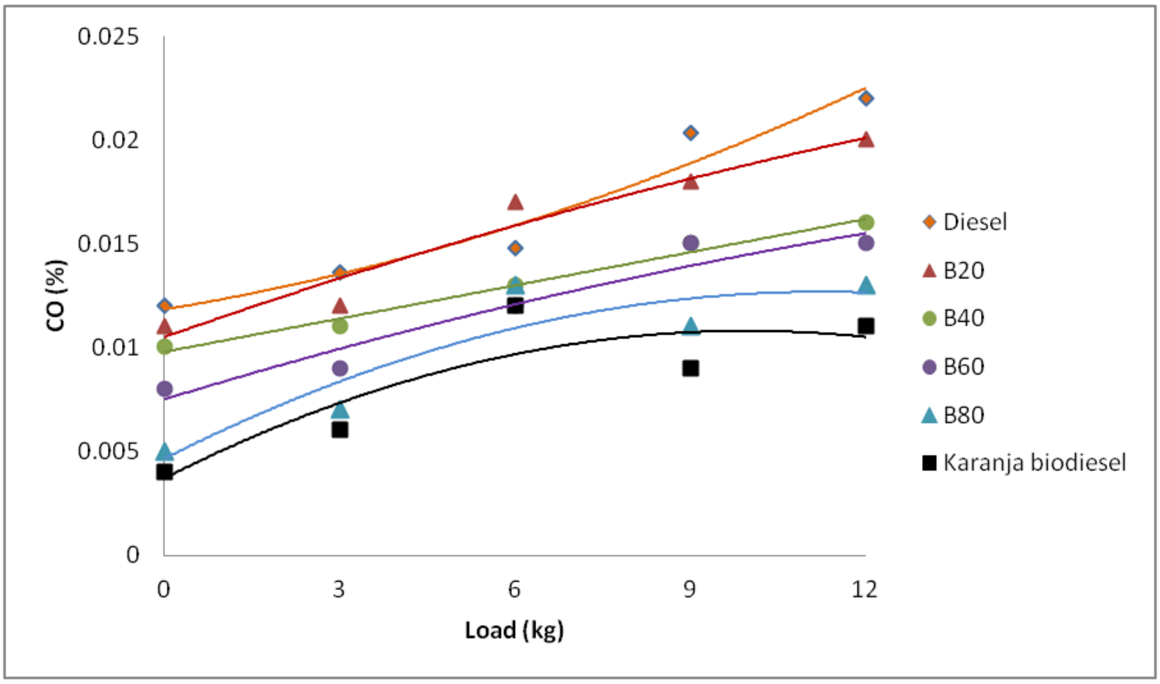


Figure 4.51 Comparison of Variation of CO with Load at CR of 18 and IP of 200bar

As Karanja biodiesel contains more oxygen, the increase in blend proportion reduces CO emissions because oxygen promotes complete combustion. It is found that the CO emissions increase up to about 175% from no load to full load condition for all the fuels tested. At full load of 12kg, it is seen that there is a 66% reduction in CO emissions when Karanja biodiesel is used as compared to Diesel oil.

Figure 4.52 gives the comparison of variation of HC with load for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and IP of 200bar. It can be observed that HC emissions increase with increase in load for all the fuels tested. The increase is steeper at higher loads than at lower loads. The steeper increase at higher loads may be due to the rich fuel air mixture as compared to stoichiometric which leads to improper burning thereby resulting in increase of HC content in the exhaust.

It can be observed that HC emissions decrease with increase in blend proportion at a constant load. The trend can be attributed to the higher oxygen content of Karanja biodiesel due to which complete combustion takes place inside the cylinder. HC emissions increase by 30% to 100% for all blends when load is increased from 0kg to 12 kg. With the increase in blend ratio, up to 66% of reduction is observed in HC emissions for all the loads under consideration. The unburned HC emissions for Diesel oil vary between 6 to 8ppm and for Karanja biodiesel they vary between 2ppm to 4ppm.

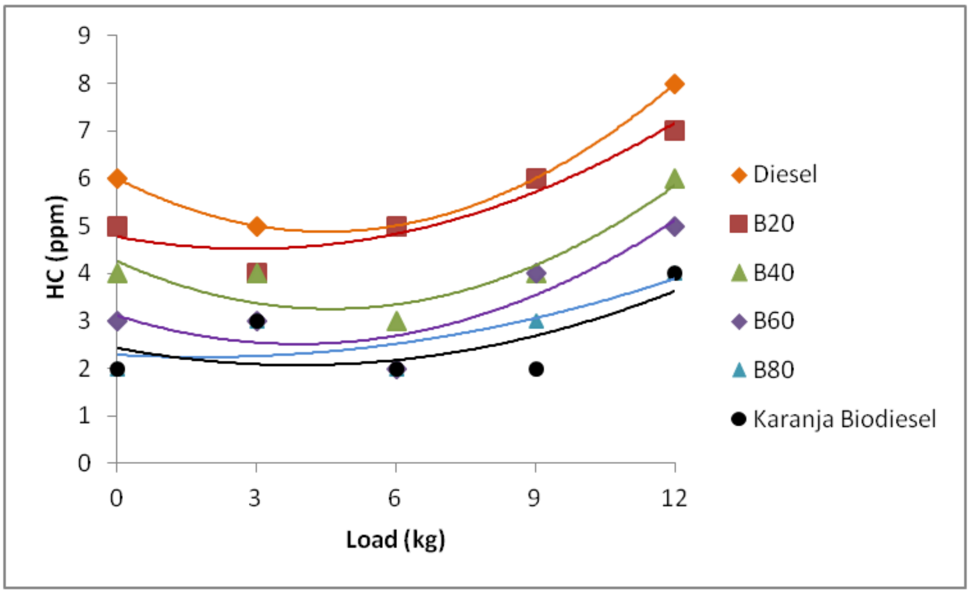


Figure 4.52 Comparison of Variation of HC with Load at CR of 18 and IP of 200bar

Figure 4.53 gives the comparison of variation of NO_x with load for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and IP of 200bar. It can be observed that NO_x emissions increase with increase in load. The trend can be attributed to more temperature with diesel as fuel and higher temperature and inherint higher oxygen content with biodiesel and its blends as fuels.

It can be observed that the NO_x emissions are higher for biodiesel blends. The reason for higher NO_x is higher oxygen content of Karanja biodiesel than Diesel oil. During combustion process of blends, more oxygen is available from fuel and nitrogen from air readily gets combined with oxygen at higher cylinder temperatures and forms compounds like nitrogen dioxide (NO_2) and nitric oxide (NO) which constitute NO_x.

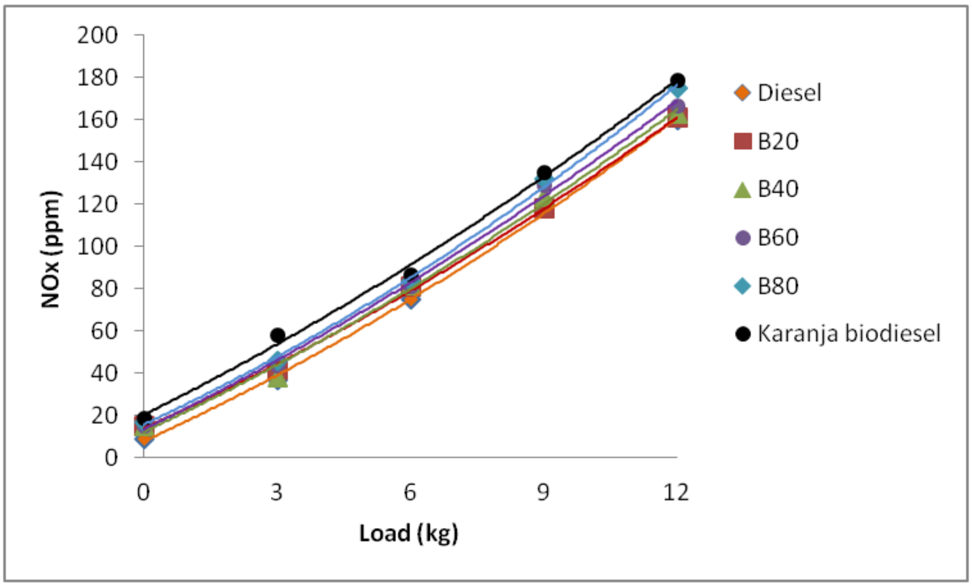


Figure 4.53 Comparison of Variation of NOx with Load at CR of 18 and IP of 200bar

The NO_x emissions for Karanja biodiesel are 11.8% higher than that for Diesel oil at a full load of 12kg. The values of NO_x emissions at the respective tested loads for Diesel oil are 9ppm, 37ppm, 75ppm, 118ppm and 160ppm and for Karanja biodiesel are 19ppm, 58ppm, 87ppm, 135ppm and 179ppm. It is noted that even though NO_x emissions are high for Karanja biodiesel, they are not appreciably high as compared to Diesel oil.

Figure 4.54 presents the comparison of variation of CO_2 with load for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and IP of 200bar. It can be observed that CO_2 increases with the increase in load for all fuels tested. The trend observed may be because of more fuel being burnt at higher loads due to which more carbon is available to form CO_2.

It can also be observed that CO_2 emissions are marginally more for Karanja biodiesel than Diesel oil and it reduces with reduction in blend proportion. The reduction in CO_2 emission is because of high oxygen content in the biodiesel due to which more of the carbon gets oxygenated during combustion inside the cylinder which results in higher CO_2 emission. The percentage increase in CO_2 emissions is about 5% for Karanja biodiesel as compared to Diesel oil at full load of 12kg.

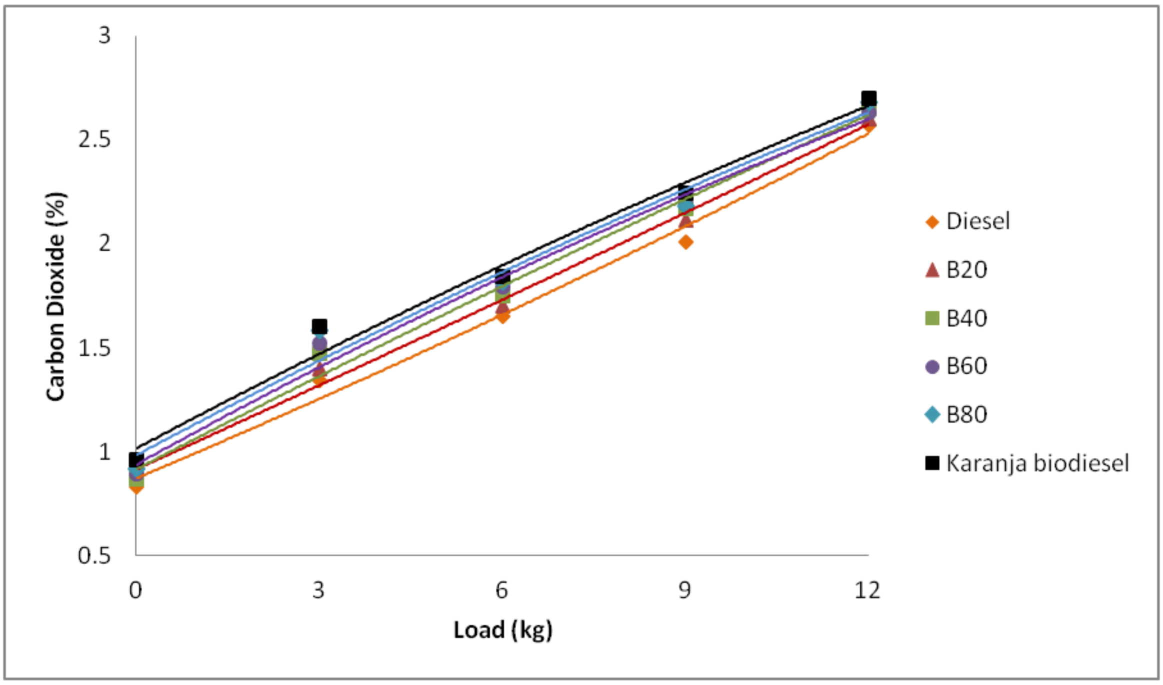


Figure 4.54 Comparison of Variation of CO_2 with Load at CR of 18 and IP of 200bar

Excess air used for combustion results in more of oxygen content in the exhaust. It is also an indicator of the extent of combustion of the fuel and oxygenated nature of the fuel used in the engine. Figure 4.55 presents the comparison of variation of O_2 with load for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and IP of 200bar.

The observed decrease in O_2 content in exhaust with increase in load may be due to richer mixture being burnt in the engine cylinder. This trend also seen in Figures 4.51, 4.53 and 4.54 shows that larger portion of oxygen available in the cylinder additionally due to high temperature reacts with nitrogen and carbon to form CO, NO_x and CO_2 at higher loads. Hence less O_2 is released to the atmosphere. It can also be observed that the O_2 emissions increase with increase in blend proportion. Further, the increase in O_2 emission with increase in blend proportion may be due to the inherent oxygen present in Karanja biodiesel. It can be noted that the percentage of oxygen in the exhaust is maximum for pure Karanja biodiesel and it decreases for other blends in the order B80, B60, B40, B20 and Diesel oil. As the load increases from 0kg and 12kg, the O_2 emissions reduces upto 12% for all the fuels tested. Further, at a load of 12kg, O_2

for Karanja biodiesel is higher by about 0.6% compared to that of Diesel oil. Decrease in oxygen percent in exauahst and converging to a almost a constant value shows a similar trend for the variation of A/F ratio with load.

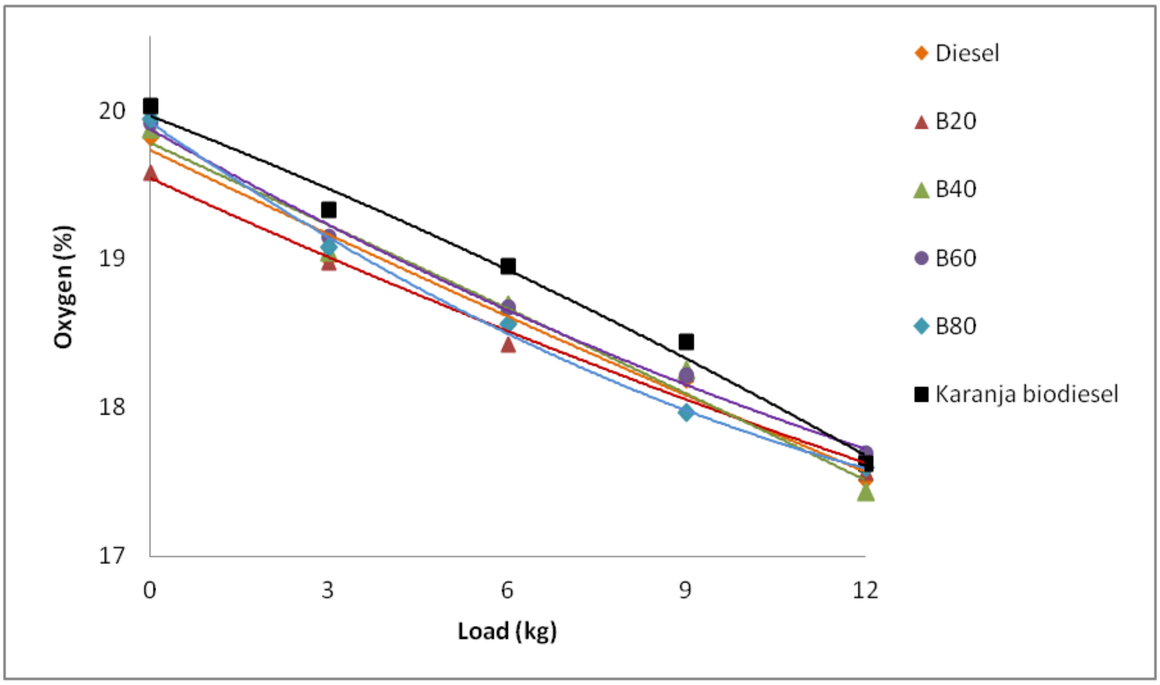


Figure 4.55 Comparison of Variation of O_2 with Load at CR of 18 and IP of 200bar

The comparison of variation of SO_x with load for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and IP of 200bar is given in Figure 4.56. It can be observed that with the increase in load SO_x in the exhaust decreases. The trend may be due to rich fuel air mixture being burnt in the cylinder due to which less oxygen is available to form SO_x.

It is also observed that SO_x emissions decrease with increase in blend proportion. The trend observed is due to the reason that Karanja biodiesel is practically a sulphur free fuel. When engine is fuelled with Karanja biodiesel, traces of sulphur oxide emissions observed at low loads are only due to sulphur present in atmospheric air in vehicular emissions.

With the increase in load from 0kg to 12kg, SO_x emissions reduce by 78%, 84%, 86%, 92%, 100% and 100% for Diesel oil, B20, B40, B60, B80 and Karanja biodiesel respectively.

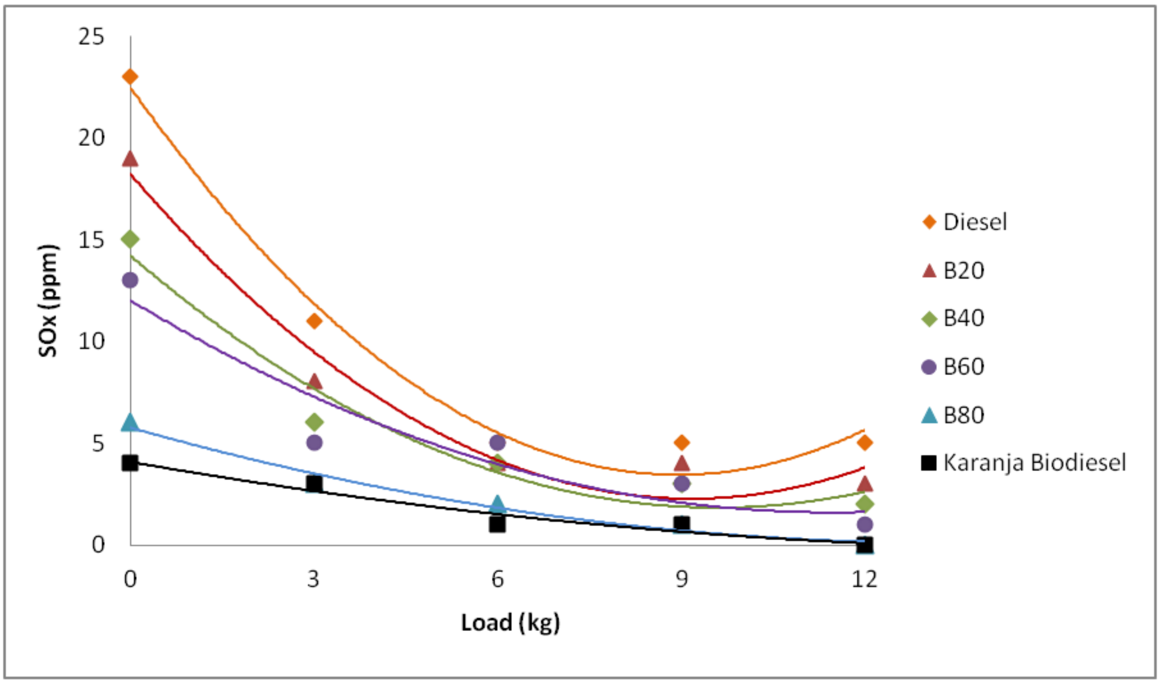


Figure 4.56 Comparison of Variation of SO_x with Load at CR of 18 and IP of 200bar

Figure 4.57 shows the comparison of variation of HSU with load for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and IP of 200bar. The observed increase in smoke intensity (HSU) with increase in load may be due to richer mixture being burnt in the cylinder at higher loads. It is also observed that HSU increases with the increase in blend proportion. The increase in HSU may probably be due to the higher viscosity and lower volatility of Karanja biodiesel due to which it becomes difficult for the fuel to undergo proper atomization and hence incomplete combustion of fuel takes place.

As the load increases from 0kg to 12kg, the HSU for Diesel oil and Karanja biodiesel increase by about 19% and 53% respectively. At a load of 12kg, HSU for Karanja biodiesel is more by about 33% compared to Diesel oil.

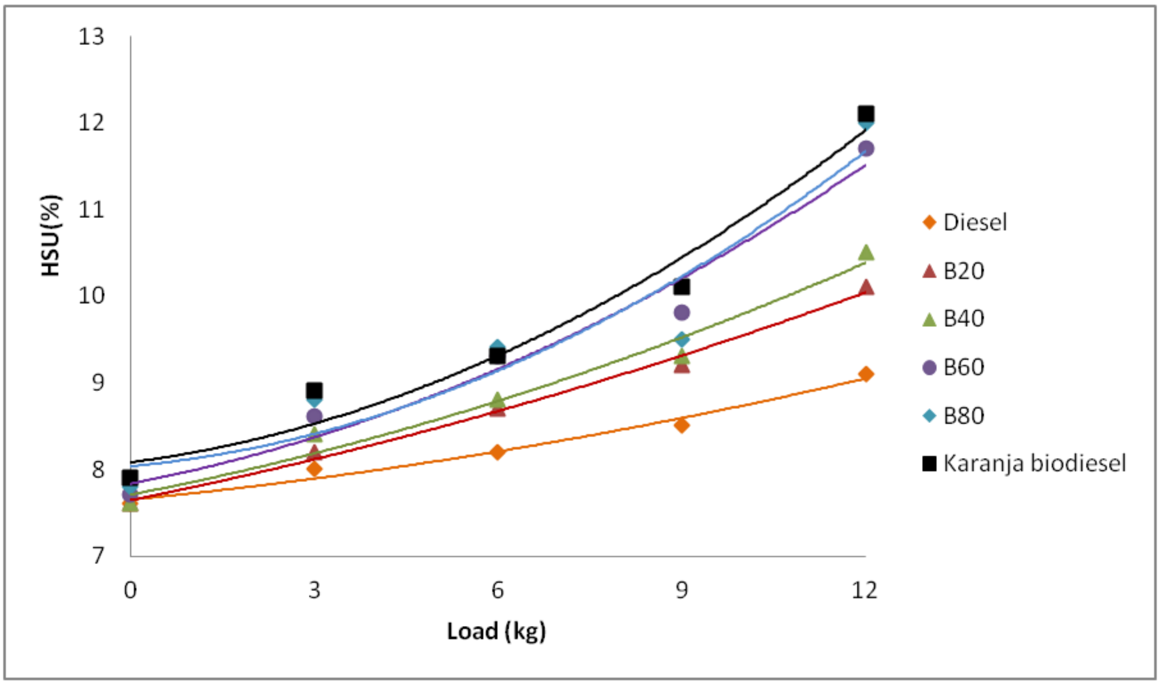


Figure 4.57 Comparison of Variation of HSU with Load at CR of 18 and IP of 200bar

Figure 4.58 shows the comparison of variation of NO_x emissions and smoke (HSU) of exhaust gas with load at a CR of 18 and IP of 200bar of the present study with that of earlier investigations of Sureshkumar et al. [52] and Jindal et al. [66]. The biodiesel used by the investigators are Karanja and Jatropha respectively.

Similar increasing trends of NO_x with load as found in the present study are observed by Sureshkumar et al. [52] using Karanja biodiesel. It is seen that NO_x at a load of 12kg is more by 40% for Sureshkumar et al. [52] as compared to that found in the present study. Jindal et al. [66] studied NO_x emission at full load of 12kg using Jatropha biodiesel. They reported 2.2% less NO_x for Jatropha biodiesel as compared to that of the present study. It is also seen that the HSU at a load of 12kg is almost double for Jatropha as compared to that of Karanja biodiesel used in the present study.

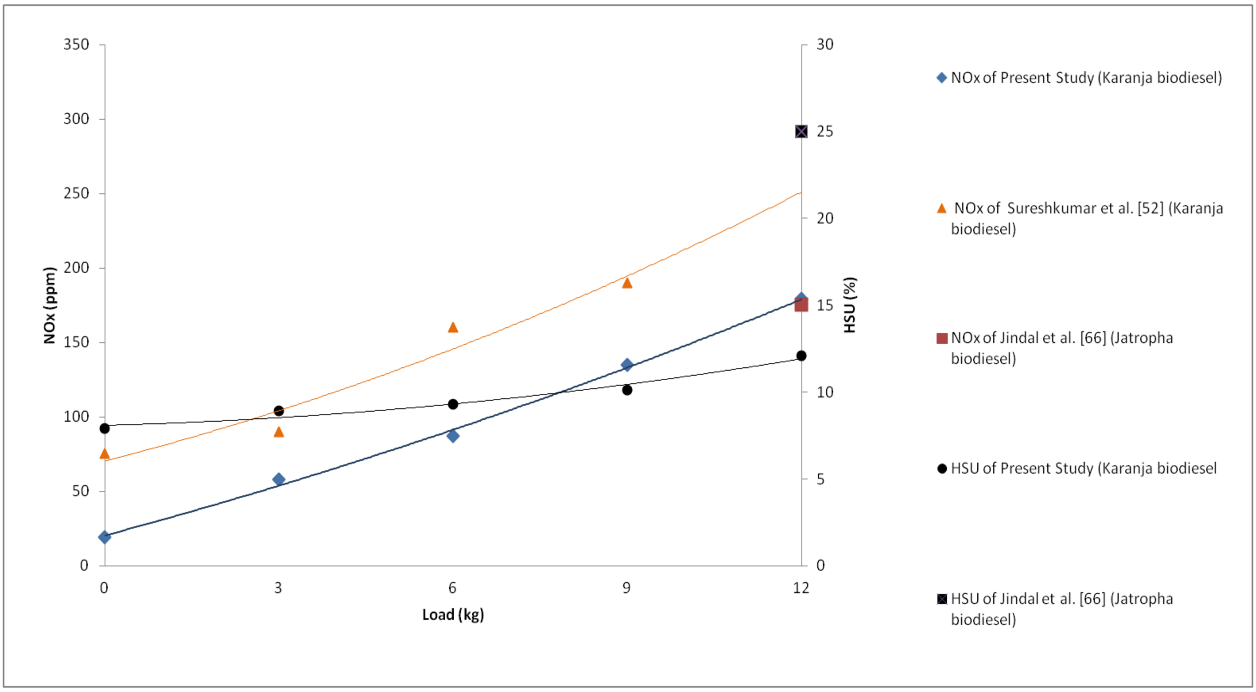


Figure 4.58 Comparison of Variation of NO_x and HSU with Load at a CR of 18 and IP of 200bar with Earlier Studies

4.2.3.3 Injection Pressure

The variations of exhaust emission constituents at different values of IPs from 150bar to 250bar in steps of 50bar at a constant rated load of 12kg and CR of 18 is presented in Figures 4.59 to 4.69.

Figure 4.59 shows the comparison of variation of CO with IP for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and load of 12kg. It is observed that CO emission reduces with increase in IP. The trend observed is due to the reason that as IP increases, the fuel is atomized into very fine droplets and hence more surface area is available for combustion which results in formation of a good quality fuel mixture due to which combustion is complete.

It is also observed that CO emission reduces with increase in blend proportion. The trend observed is because as the concentration of Karanja biodiesel in the blend increases the percentage of oxygen in the blend also increases. Due to higher oxygen content, when the fuel is burnt in the engine cylinder, more of carbon gets oxygenated forming CO_2 resulting in lesser CO emissions.

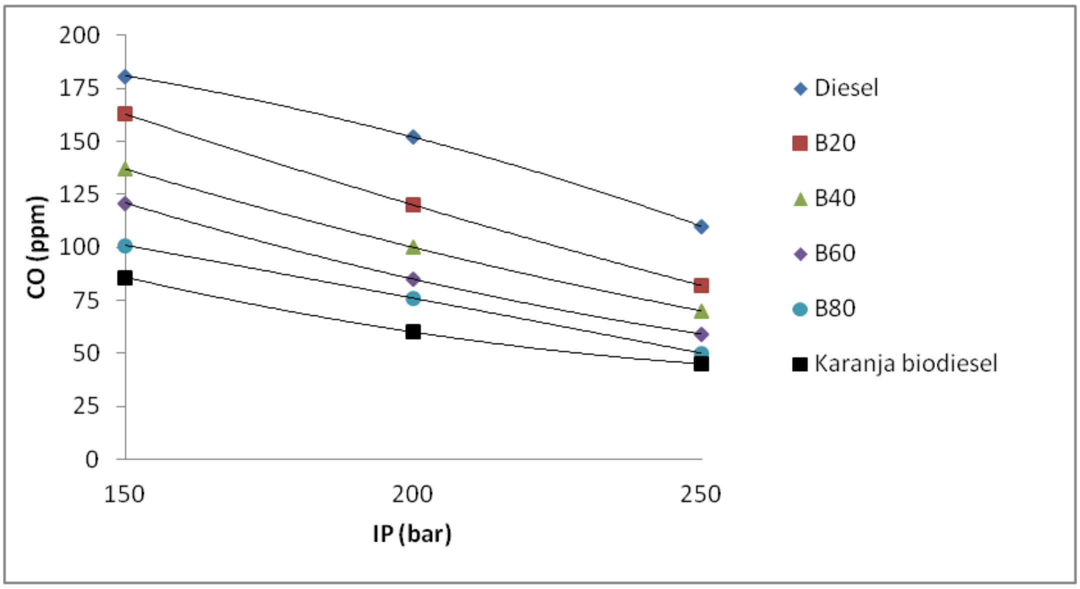


Figure 4.59 Comparison of Variation of CO with IP at CR of 18 and Load of 12kg

The CO emissions reduce up to 39.2%, 49%, 48%, 51.23%, 50% and 47% for Diesel oil, Karanja biodiesel, B80, B60, B40 and B20 respectively when the IP is increased from 150bar to 250bar. It can be noted that at an IP of 250bar, CO emissions for Karanja biodiesel are lesser by about 60% as compared to Diesel oil.

Figure 4.60 shows the comparison of variation of CO emission with IP at a CR of 18 and load of 12kg of the present study with that of Jindal et al. [66] using Jatropha biodiesel. Similar decreasing trends as found in the present study are observed by them.

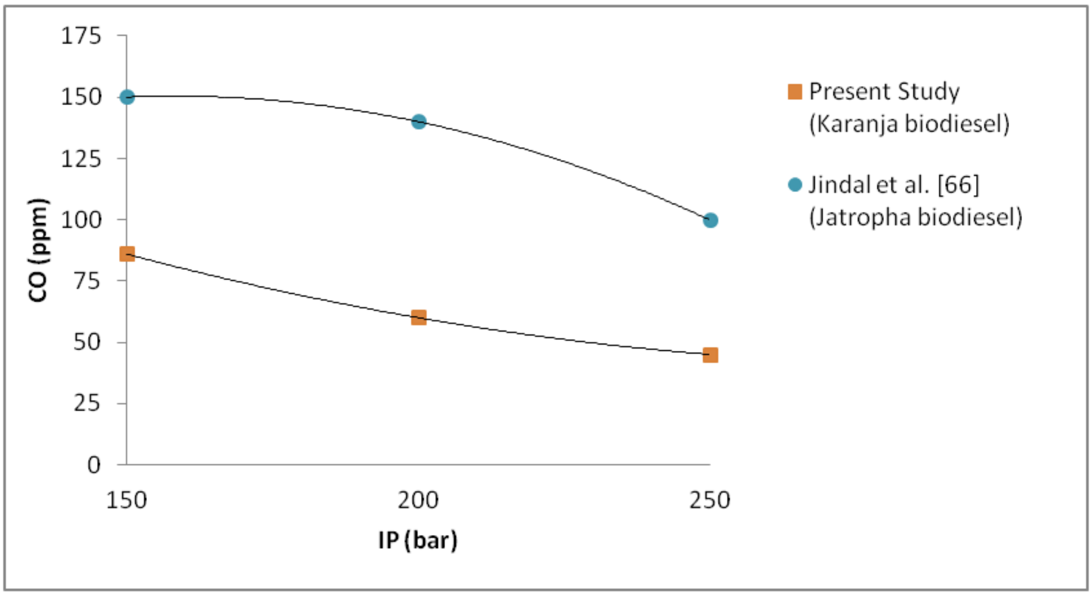


Figure 4.60 Comparison of Variation of CO with IP at a CR of 18 and Load of 12kg with Jindal et al. [66]

The difference between two fuels might be due to better combustion characterstic of Karanja biodiesel at higher CR and IP as compared Jatropha biodiesel of Jindal et.al[66].

It is seen that CO emission at an IP of 250bar is almost double for Jatropha biodiesel as compared to that of Karanja biodiesel used in the present study.

The comparison of variation of HC with IP for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and load of 12kg is shown in Figure 4.61. It can be observed that HC reduces significantly with increase in IP.

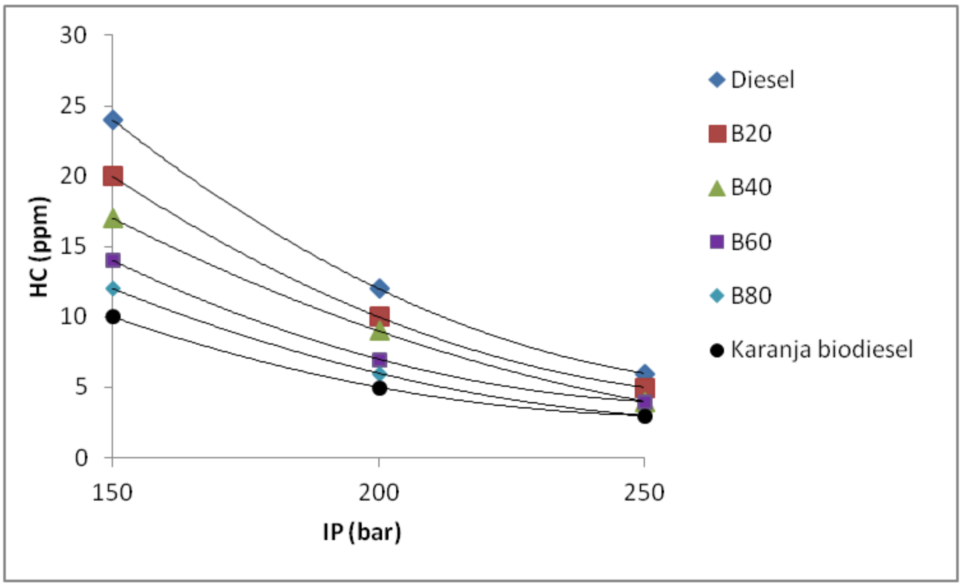


Figure 4.61 Comparison of Variation of HC with IP at CR of 18 and Load of 12kg

The trend observed is because of better combustion of fuel taking place at higher IP. The observed decrease in HC emission with increase in blend proportion may be due to complete combustion of blends as a result of their higher oxygen content. The HC emissions decrease by 75% and 70% for Diesel oil and Karanja biodiesel respectively when the IP increases from 150 bar to 250 bar.

Figure 4.62 shows the comparison of variation of HC emissions with IP at a load of 12kg and CR of 18 of the present study with that of Jindal et al. [66]. Similar decreasing trends of HC with IP as found in the present study using Karanja biodiesel are observed by Jindal et al. [66] using Jatropha biodiesel. It is seen that the HC emissions at an IP of 250bar is higher by 13ppm for Jatropha as compared to that of Karanja biodiesel used in the present study, indicating better combustion of Karanja biodiesel.

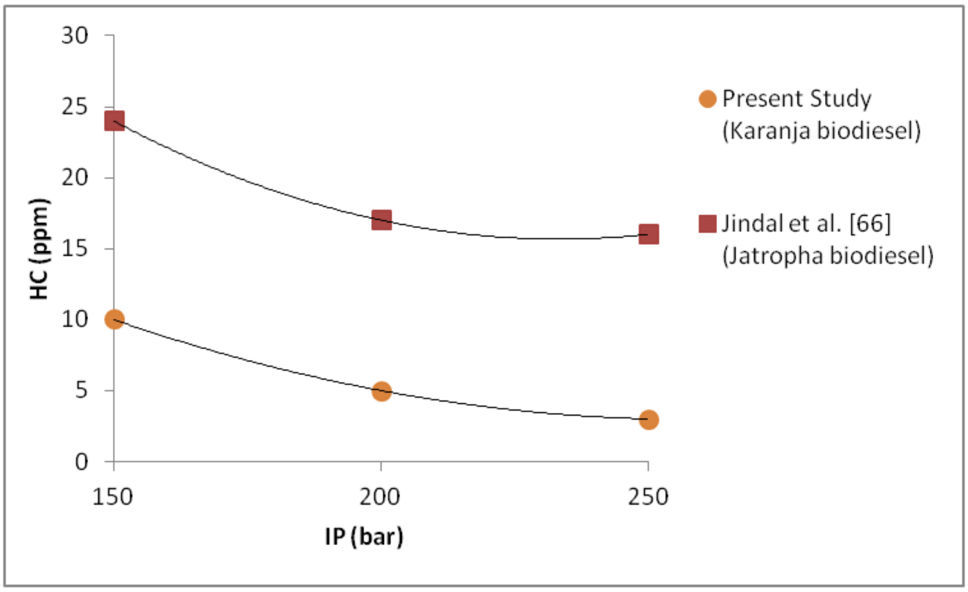


Figure 4.62 Comparison of Variation of HC with IP at CR of 18 and Load of 12kg of Present Study with Jindal et al. [66]

The comparison of variation of NO_x with IP for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and load of 12kg is shown in Figure 4.63. It can be observed that NO_x emissions increase linearly with increase in IP. The trend is due to more fuel being burnt at higher IP. The observed increase in NO_x emissions with blend proportion is mainly due to higher oxygen content in Karanja biodiesel and higher temperature existing in the engine cylinder.

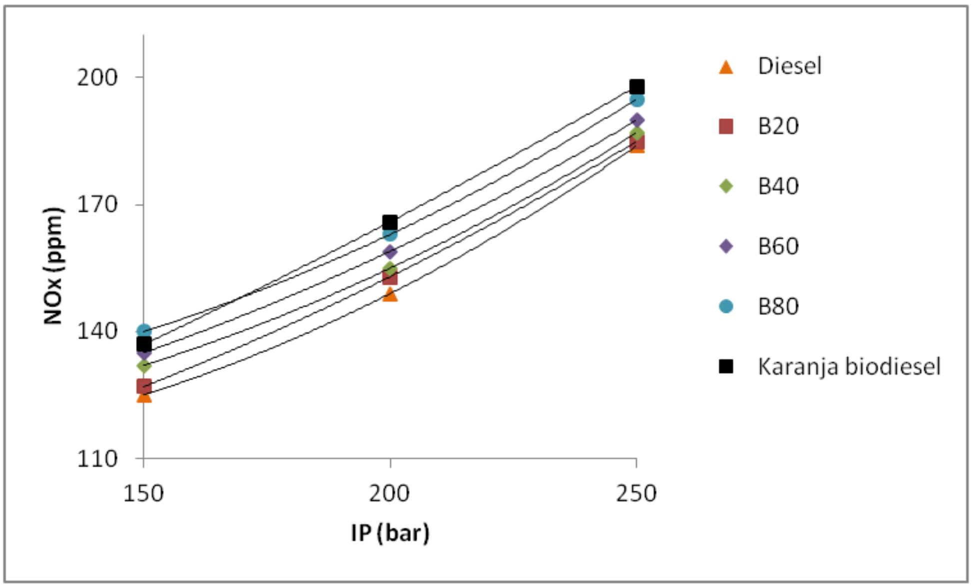


Figure 4.63 Comparison of Variation of NO_x with IP at CR of 18 and Load of 12kg

The NO_x emission for Karanja biodiesel is 7.6% higher than that for Diesel oil at an IP of 250 bar. As the IP increases from 150 to 250 bar, the NO_x emissions increase by about 47% and 44% for Diesel oil and for Karanja biodiesel respectively.

Figure 4.64 gives the comparison of variation of CO_2 with IP for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and load of 12kg. It can be seen that CO_2 content in the exhaust gas increases with the increase in IP. The trend observed is obvious as fuel is burnt to a greater degree at higher IPs due to which more carbon is available to form CO_2.This result is constitent with smaller value of HC for the fuel in present study. (Figures 4.61 and 4.62).

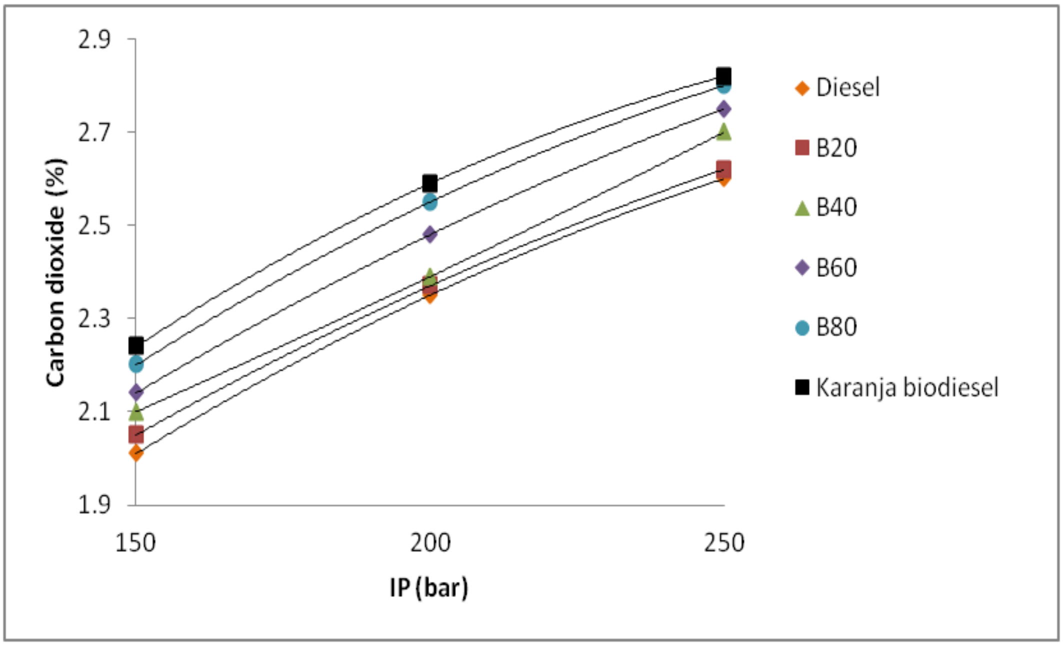


Figure 4.64 Comparison of Variation of CO_2 with IP at CR of 18 and Load of 12kg

It is also observed that CO_2 emissions increase with blend proportion. The trend observed is because of high oxygen content in the Karanja biodiesel due to which more of the carbon gets oxygenated during combustion inside the cylinder which results in higher CO_2 emission. The CO_2 emissions increase by 29% and 25% for Diesel oil and Karanja biodiesel as the IP increases from 150 to 250bar. At an IP 250 bar, CO_2 emission for Karanja biodiesel is found to increase by about 8% as compared to Diesel oil.

Figure 4.65 shows the comparison of variation of CO_2 emissions with IP at a load of 12kg and CR of 18 of the present study with the earlier investigation of Jindal et al. [66]. It is seen that at standard setting of the engine viz. 18CR, 12kg load and 200 bar IP, the

CO_2 emission is more by 82% for Jatropha as compared to that of Karanja biodiesel used in the present study. This is consistent with smaller value of HC for Karanja biodiesel at higher values of IP as compared to Jatropha biodiesel. (Figure 4.62).

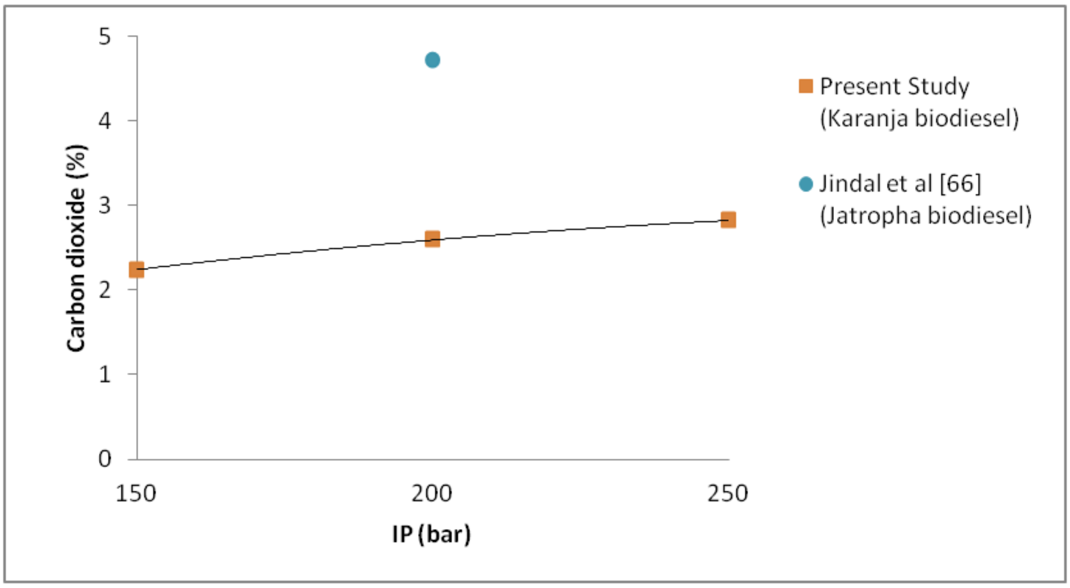


Figure 4.65 Comparison of Variation of CO_2 with IP at a CR of 18 and Load of 12kg of Present Study with Jindal et al. [66]

Figure 4.66 gives the comparison of variation of O_2 with IP for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and load of 12kg. The trend may be due to the reason that fuel is atomized to a higher degree at higher IP, hence combustion of fuel is better which results in more oxygen content in the exhaust. This is also supported by variation of A/F ratio with increase in IP. The percent Oxygen in the exhaust is about 0.25% higher for biodiesel as compared to diesel oil.indicating closer behavior of biodiesel with diesel at higher IPs.

It is also observed that O_2 emissions in the exhaust increase with increase in blend proportion. The observed increase is because of higher oxygen content of Karanja biodiesel. The O_2 emission in the exhaust increases by 12% and 10% for Diesel oil and Karanja biodiesel respectively when the IP is increased from 150 to 250. At an IP of 250bar, O_2 emission for Karanja biodiesel is higher by about 2% as compared to Diesel oil.

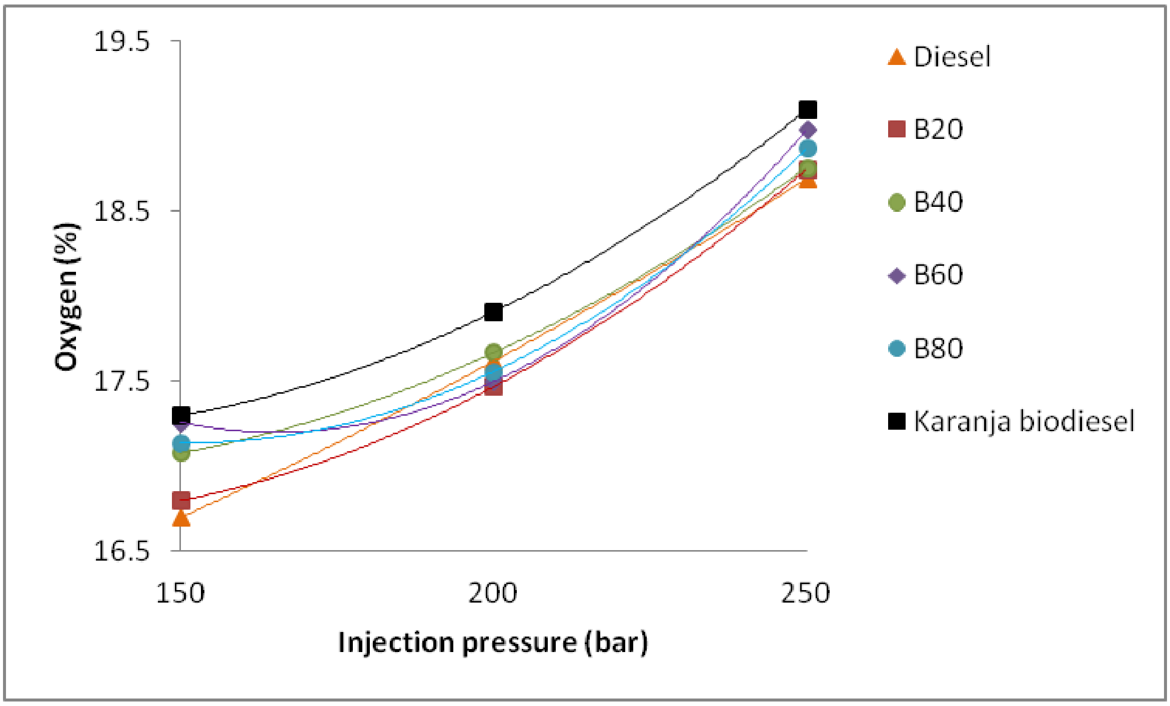


Figure 4.66 Comparison of Variation of O_2 with IP for Tested Fuels at CR of 18 and Load of 12kg

The comparison of variation of SO_x with IP for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and load of 12kg is presented in Figure 4.67. It is observed that SO_x reduce with increase in IP for all the fuels tested. The trend observed is due to complete combustion of fuel which is aided by finer atomization. It is also observed that SO_x emissions decrease with increase in blend proportion. The trend may be due to sulphur free nature of Karanja biodiesel. However, at an IP of 250 bar there is a small increase in SO_x. The reason may be attributed to lean mixture burning associated with higher temperature of combustion. Oxygen present in the exhaust may dissociate at high temperatures and combine with sulphur present in air to produce SO_x.

As the IP increases from 150 to 250 bar the SO_x reduce by 77% and 50% for Diesel oil and Karanja biodiesel respectively. At an IP of 250bar SO_x for Karanja biodiesel is 50% less compared to Diesel oil.

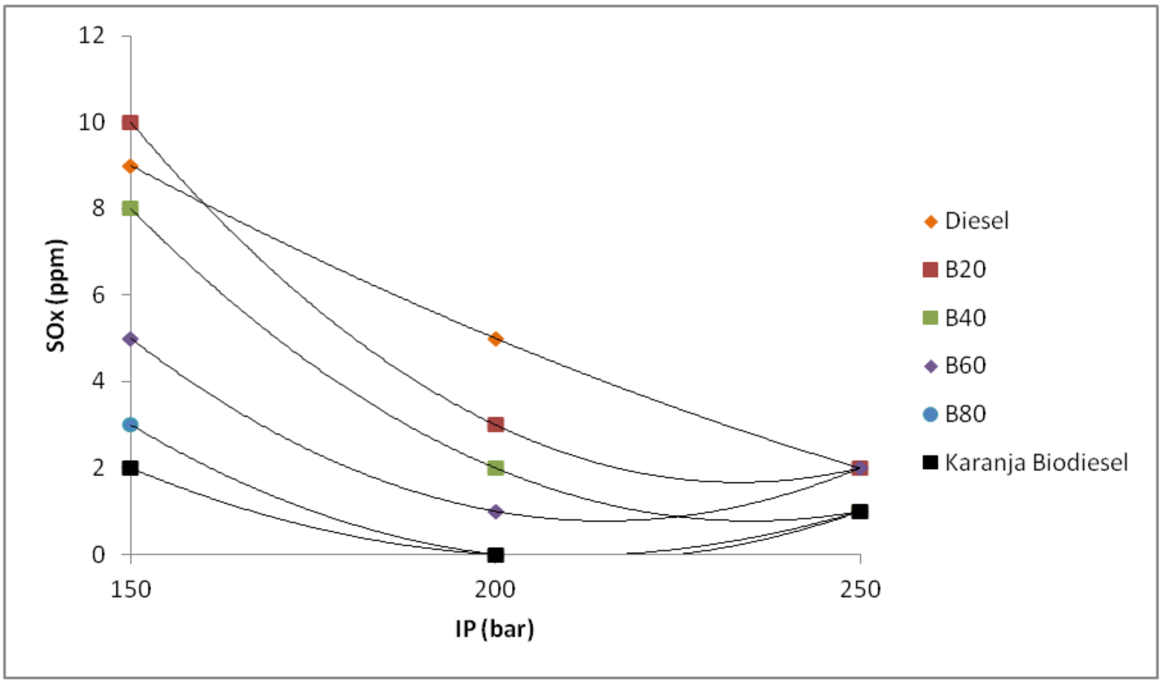


Figure 4.67 Comparison of Variation of SO_x with IP at CR of 18 and Load of 12kg

The comparison of variation of HSU with IP for Karanja biodiesel and its blends with Diesel oil at a CR of 18 and load of 12kg is presented in Figure 4.68. It is observed that HSU decreases with increase in IP for all tested fuels. The trend is due to the fact that at higher IP the degree of the atomization of fuel is high enough to cause complete combustion of fuel.

It is also seen that HSU increases with increase in blend proportion at any constant IP. HSU is higher for Karanja biodiesel by 23% as compared to that of Diesel oil at an IP of 250bar. As the IP increases from 150 to 250 bar the HSU reduces by 58% and 53% for Diesel oil and Karanja biodiesel.

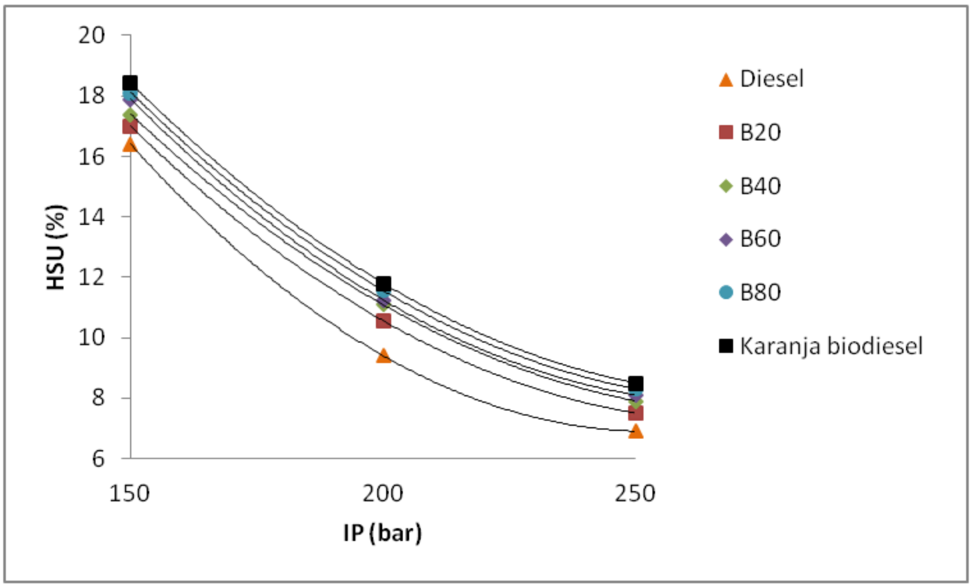


Figure 4.68 Comparison of Variation of HSU with IP at CR of 18 and Load of 12kg

Figure 4.69 shows the comparison of variation of NO_x emissions and HSU of exhaust gas with IP at a load of 12kg and CR of 18 of the present study with that of earlier investigations of Jindal et al. [66].

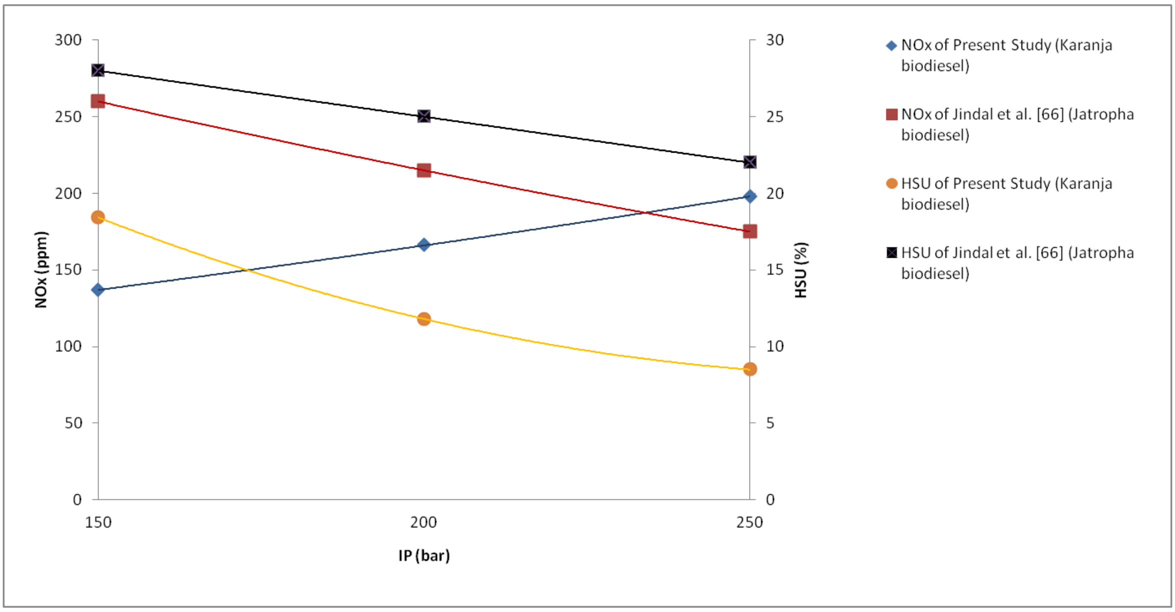


Figure 4.69 Comparison of Variation of NO_x and HSU with IP at a CR of 18 and Load of 12kg with Jindal et al. [66]

Similar decreasing trends of HSU with IP as found in the present study are observed by Jindal et al. [66] using Jatropha biodiesel. It is seen that HSU at an IP of 200bar is almost double for Jatropha as compared to that found in the present study. Jindal et al. [66] also studied the variation of NO_x emission with IP at full load of 12kg using Jatropha biodiesel. The investigators reported an opposite trend as compared to that of the present study. It is also seen that the NO_x at an IP of 250bar is more by about 12% for Jatropha as compared to that of Karanja biodiesel used in the present study. Generally NO_x emissions increase at the expense of HSU with increase in IP. The trend concurs with available literature (Fergusson & Kirkpatrick [98]), but Jindal et al. [66] has observed that both NO_x and HC show a decreasing trend with increase in IP.This is consistent with the value of HC and NO_x at higher IPs. (Figure 4.61, 4.62 and 4.63).

4.2.4 Combustion Analysis

The combustion processes are engine dependent, involving the oxidation of chemical constituents of a fuel resulting in release of thermal energy. A compression ignition engine has separate fuel and air streams that combust as they are mixed together. The chemical reaction which produces a diffusion flame takes place at the interface between the fuel and the air. The heat release begins at a relatively high value and then decreases as the available oxygen is depleted. Combustion chemistry is very complex and depends on the type of fuel used in the combustion process. Oxidation of the hydrocarbon fuel is modelled by earlier investigators using reaction pathways (Ferguson & Kirkpatrick [98]).

The Diesel oil combustion process has been classified into four distinct phases: ignition delay, premixed combustion, mixing controlled combustion and late combustion. Ignition delay is the period during which some fuel has been admitted but not yet ignited. In premixed combustion phase which takes place over a few degrees of crank angle, there is a rapid rise in pressure due to combustion of fuel. In mixing controlled combustion the fuel droplets injected burn almost instantaneously. During late combustion phase some fuel droplets which are not yet burnt, due to their poor distribution, get combusted and heat release continues at a lower rate well into the expansion stroke.

This section deals with the combustion analysis of a VCR diesel engine fuelled with Karanja biodiesel and Diesel oil separately. The combustion parameters considered for analysis in subsequent sections are cylinder pressure, net heat release, rate of pressure rise, mass fraction burnt, mean gas temperature, cylinder pressure-volume (PV) plot, cumulative heat release rate and injection pressure. The study is conducted at full load of 12kg and preset IP of 200bar.

4.2.4.1 Cylinder pressure

Cylinder pressure (CP) with crank angle (CA) over an engine operating cycle gives the quantitative information on the progress of combustion. The variation of CP during a complete cycle with CA for three settings of CRs of 14, 16 and 18 for both Karanja biodiesel and Diesel oil as fuels are shown in Figures 4.70 to 4.75.

The variation of CP with CA at a CR of 14 for Karanja biodiesel and Diesel oil are shown in Figures 4.70 and 4.71 respectively. It is observed that start of combustion (SOC) takes place at 10^0 of CA later for Karanja biodiesel as compared to Diesel oil. The trend observed is because the heat of compressed air is not high enough to initiate combustion at earlier CA in comparison with Diesel oil, which may probably be due to reduced atomization due to higher viscosity and hence less intense premixed combustion phase with Karanja biodiesel as compared to Diesel oil.

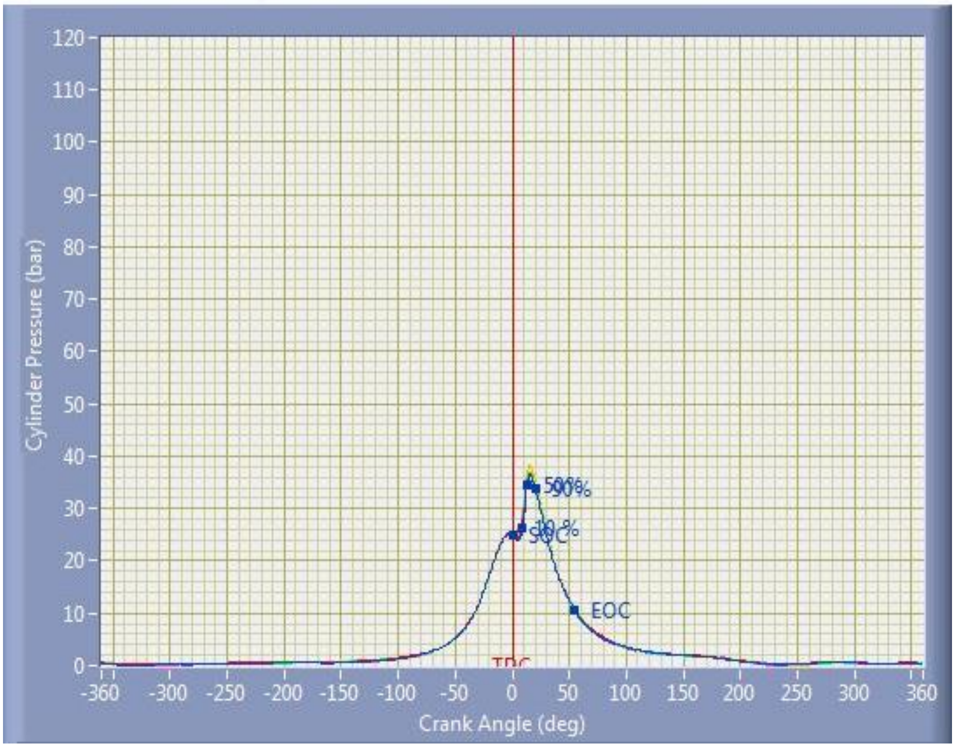


Figure 4.70 Variation of CP With CA at CR of 14 for Karanja Biodiesel

It may also be due to the reason that diffusion burning (during mixing controlled combustion) is more intense with Karanja biodiesel. The peak CP is about 38 bar for Karanja biodiesel as against 40 bar for Diesel oil i.e. CP achieved is lower by 2 bar for Karanja biodiesel. The duration of combustion (CA covered from SOC to end of combustion (EOC)) is 55^0 CA for Karanja biodiesel and 75^0 CA for Diesel oil. It is also observed that EOC is advanced by 10^0 in case of Karanja biodiesel as compared to Diesel oil.

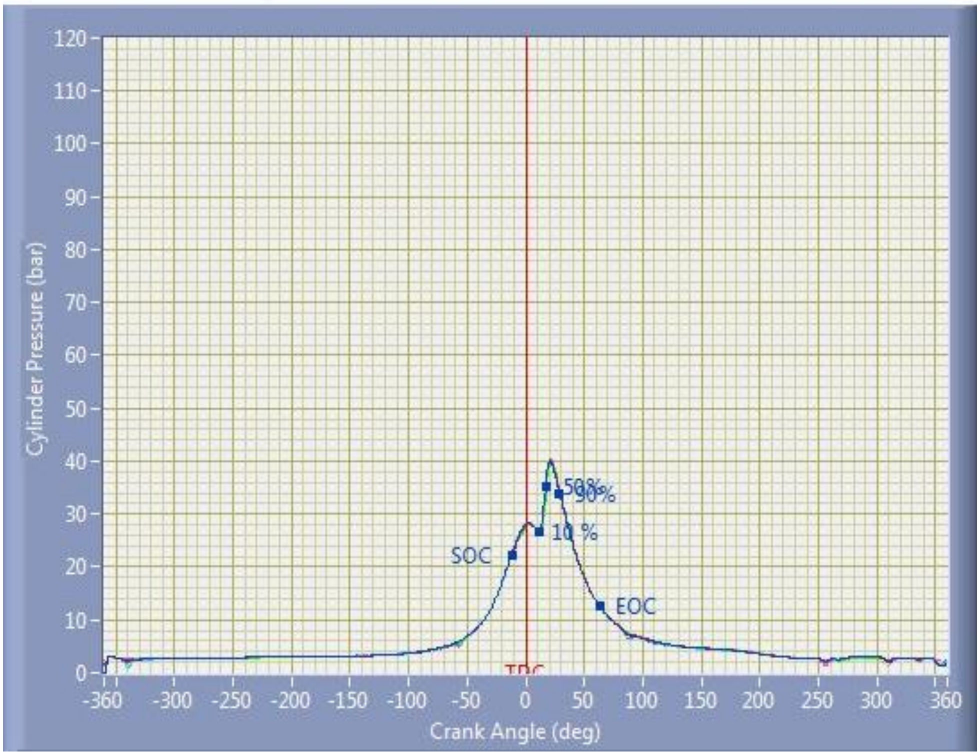


Figure 4.71 Variation of CP With CA at CR of 14 for Diesel Oil

The variation of CP with CA at a CR of 16 for Karanja biodiesel and Diesel oil are shown in Figures 4.72 and 4.73 respectively. It is observed that SOC takes place at TDC for both Karanja biodiesel and Diesel oil. The peak CP is about 42 bar for Karanja biodiesel as against 46 bar for Diesel oil i.e. CP achieved is lower by 4 bar for Karanja biodiesel.

The duration of combustion is 47^0 CA for Karanja biodiesel and 55^0 CA for Diesel oil. It can be noted that the deviation in performance between both the fuels reduces with the increase in CR from 14 to 16. However, engine with Diesel oil as fuel may be marginally performing better due to higher CP achieved.

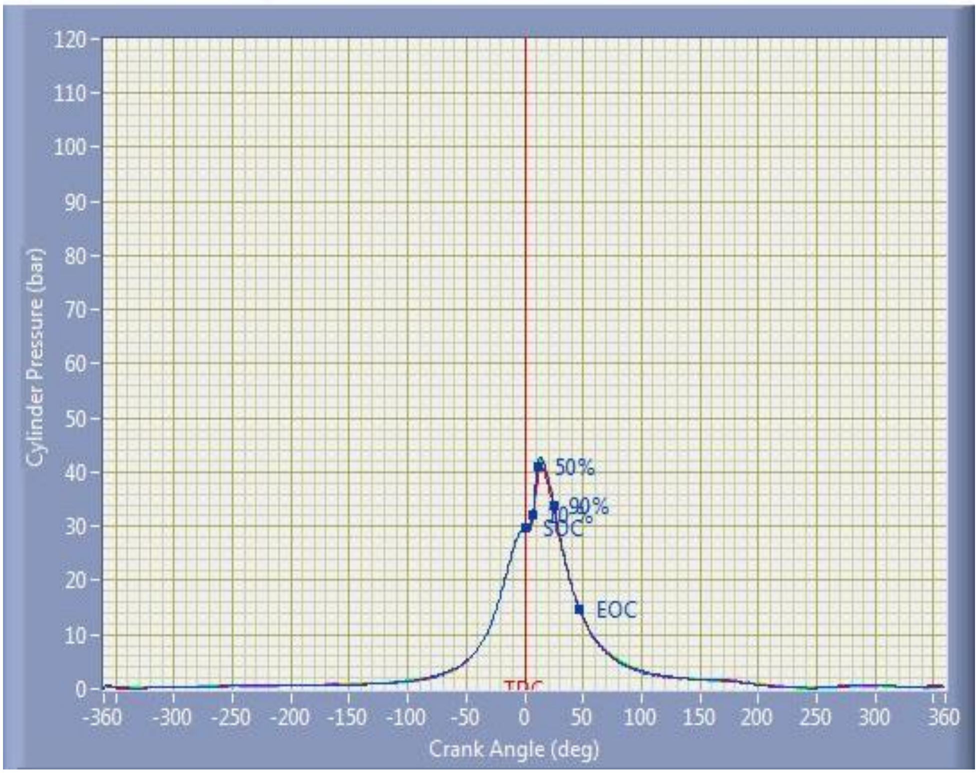


Figure 4.72 Variation of CP With CA at CR of 16 for Karanja Biodiesel

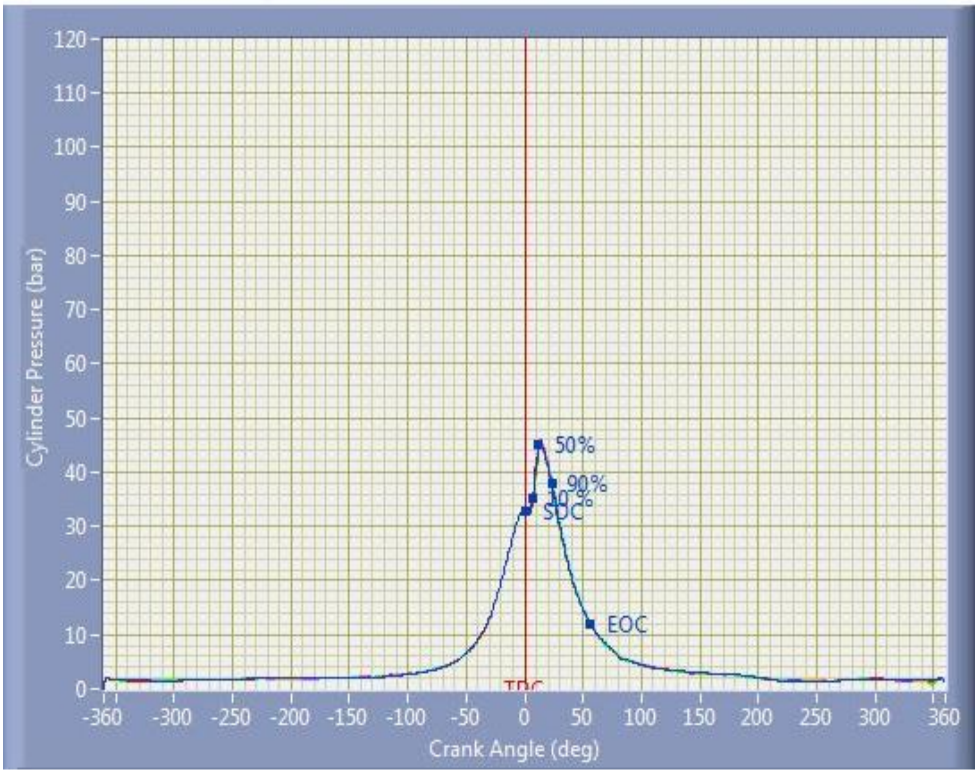


Figure 4.73 Variation of CP With CA at CR of 16 For Diesel Oil

The variation of CP with CA at a CR of 18 for Karanja biodiesel and Diesel oil are shown in Figures 4.74 and 4.75 respectively. It can be observed that the trend of variation of CP with CA is almost same for Karanja biodiesel and Diesel oil. SOC takes place at 25^0 CA before TDC and 20^0 CA before TDC for Karanja biodiesel and Diesel oil respectively indicating about 5^0 advance for Karanja biodiesel. SOC at 25^0 CA before TDC may be attributed to perturbation in the engine. The peak CP attained is same (50 bar) for both the fuels and the duration of combustion is spread over 80^0 of CA for both Karanja biodiesel and Diesel oil. It can be inferred that the engine operated using Karanja biodiesel gives similar performance as that of Diesel oil as fuel at CR of 18.

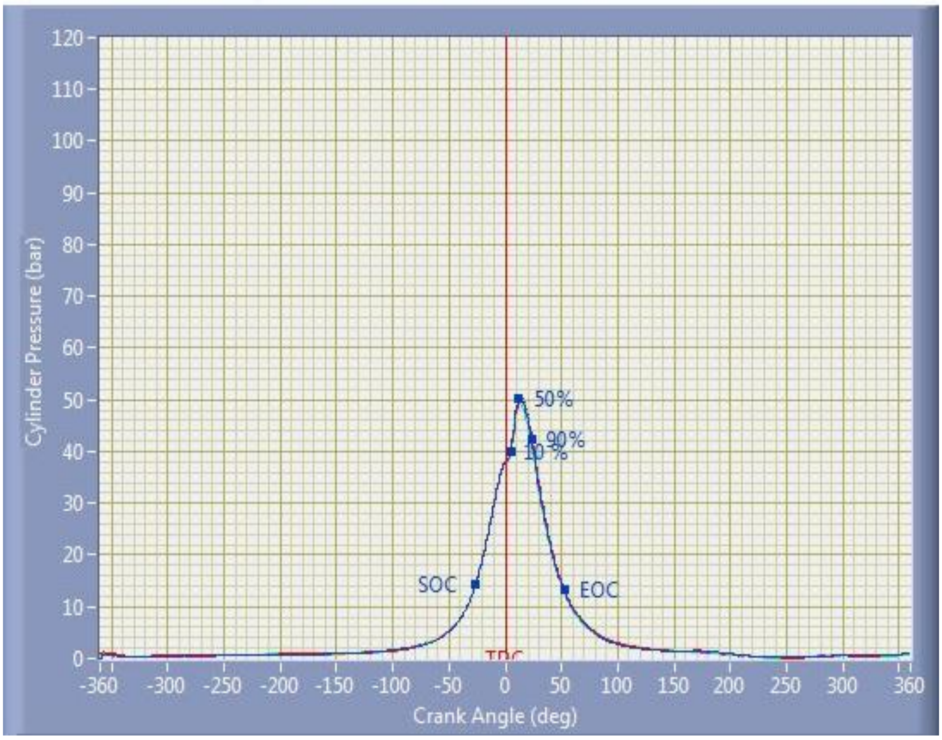


Figure 4.74 Variation of CP With CA at CR of 18 for Karanja Biodiesel

The data of CA positions at different CRs for SOC, Peak CP and EOC is given in Table 4.4. It can be noted that peak pressure development for Karanja biodiesel takes place at earlier CA at 14 CR as compared to Diesel oil. However, the peak pressures at all CRs using both the fuels always takes place after TDC which ensures safe and efficient operation as peak pressure at or before TDC causes knocking which affects engine durability (Ganesan [97]). The peak CP attained for Karanja biodiesel is about 2 to 4 bar less at CR of 14 and 16, whereas at CR of 18 the peak CP is same for both the fuels. As CR increases there is a reduced delay observed for Karanja biodiesel as compared to Diesel by about 5^0 of CA and the peak CP is same for both the fuels.

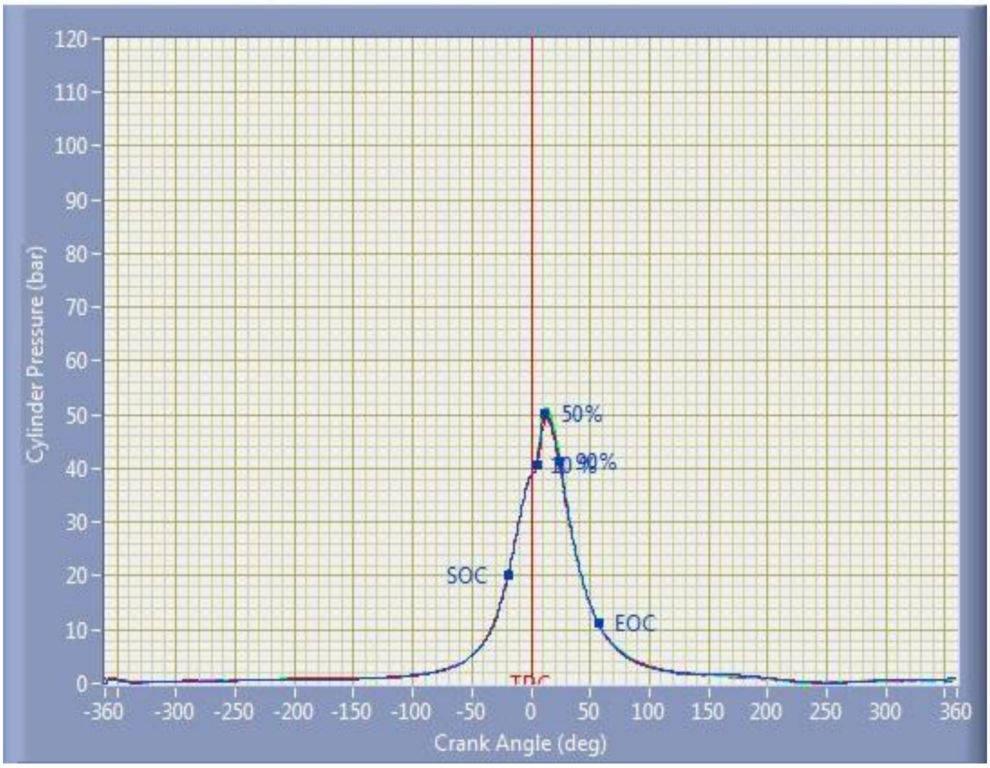


Figure 4.75 Variation of CP With CA at CR of 18 for Diesel Oil

Table 4.4 Crank Angle Data for different CRs for SOC, Peak CP and EOC (Reproduced from Figures 4.70–4.75)

	Karanja biodiesel				Diesel oil			
	SOC	Position of Peak CP (^{0}CA)	EOC	Peak CP (bar)	SOC	Peak CP (^{0}CA)	EOC	Peak CP (bar)
14CR	TDC	15^0 ATDC	55^0 ATDC	38	10^0 BTDC	20^0 ATDC	65^0 ATDC	40
16CR	TDC	10^0 ATDC	47^0 ATDC	42	TDC	10^0 ATDC	55^0 ATDC	46
18CR	25^0 BTDC	15^0 ATDC	55^0 ATDC	50	20^0 BTDC	15^0 ATDC	60^0 ATDC	50

Figure 4.76 shows the comparison of variation of peak CP with CR at full load of 12kg and IP of 200bar of the present study with that of earlier investigations of Sahoo & Das [86], Devan & Mahalaxmi [64], Gattamaneni et al. [85], Buyukkaya [87], Murugan et al. [99], Sahoo & Das [86] used Karanja, Jatropha and Polanga biodiesels, Devan & Mahalaxmi [64] and Gattamaneni et al. [85] used Poon and Rice bran biodiesels respectively. Murugan [99] and Buyukkaya [87] used Distilled tyre pyrolisis oil and Rapeseed oil respectively.

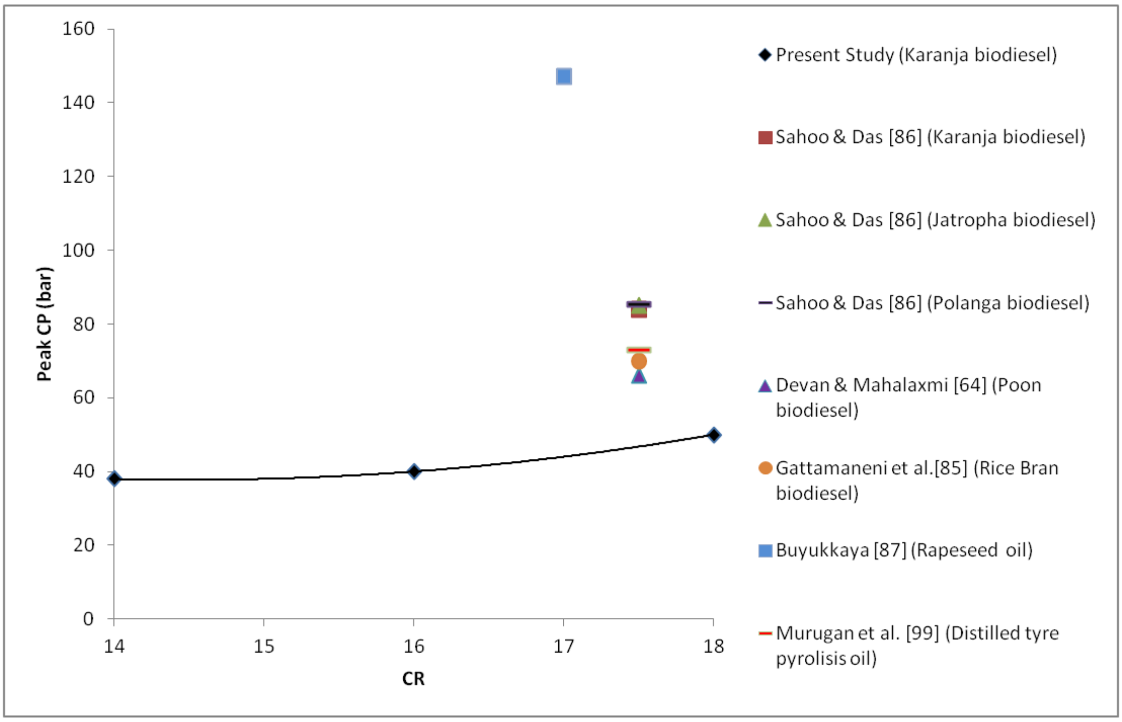


Figure 4.76 Comparison of Variation of Peak CP with CR at a Load of 12kg and an IP of 200bar of the Present Study with that of Earlier Investigations

It is observed that peak CP increases with increase in CR for Karanja biodiesel used in the present study. It can be noted that there is no study reported for variation of Peak CP with CR on a diesel engine. Therefore comparison of trend of the present study with earlier investigations is not possible. However, various studies of peak CP at constant preset CR using different biodiesels are reported.

It is observed that peak CP achieved in investigation by Sahoo & Das [86] using Karanja biodiesel at a constant CR of 17.5 is 75% higher as compared that of present study. They also conducted studies using Jatropha and Polanga biodiesels. The peak CP is higher by about 75% for Jatropha and Polanga biodiesel as compared that of Karanja biodiesel used in the present study. Various investigators also reported higher peak CP at constant CR of 17.5 using Poon biodiesel, Ricebran biodiesel and Distilled tyre pyrolisis oil. It is observed that peak CP is higher by 37.5%, 46% and 52% for Poon biodiesel, Ricebran biodiesel and Distilled tyre pyrolisis oil respectively as compared to Karanja biodiesel used in the present study. At a constant CR of 17, peak CP for Rapeseed oil is more by almost 200% as compared to that of Karanja biodiesel used in the present study. The results of a study on a CI engine indicate that the maximum peak pressure achieved is about 48bar at 18^0 CA after TDC (Ferguson & Kirkpatrick [98]). The values of this study closely match with the present study.

4.2.4.2 Net Heat Release Rate

The variation of net heat release rate during a complete cycle with CA for three settings of CRs of 14, 16 and 18 for both Karanja biodiesel and Diesel oil as fuels are shown in Figures 4.77 to 4.82. The net heat release rate *(dQ_n /dt)* is the difference between the heat released by combustion of fuel (dQ_{ch} /dt) and the heat transfer rate to the walls of the engine cylinder *(dQ_{ht}/dt)*. The net heat release rate also equals to the sum of work done on the piston *[p(dV/dt)]* and change of sensible internal energy of the cylinder contents *(dU_s /dt)*.

The heat transfer rate to the walls is the loss of heat. So, if loss is more the net heat release rate would be less and it can be clearly observed from the equation below that net heat release rate increases with the decrease in the heat transfer rate to the walls keeping heat released by combustion of fuel the same.

$$(dQ_n)/dt = (dQ_{ch})/dt - (dQ_{ht})/dt = p\,dV/dt + (dU_s)/dt \text{ (Heywood [96])}$$

The dual function which is known as Wiebe function can also be used to fit the diesel combustion heat release data.

$$dQ/d\theta = a(Q_p/\theta_p)\,(m_p)\,(\theta/\theta_p)^{m_p-1}\,exp\,[-a(\theta/\theta_p)^{m_p}] + a(Q_d/\theta_d)(m_d)(\theta/\theta_d)^{m_d-1}exp[-a(\theta/\theta_d)^{m_d} \text{ (Ferguson \& Kirkpatrick [98])}$$

The subscripts p and d refer to premixed combustion and mixing controlled combustion portions, respectively, a is the non dimensional constant, θ_p and θ_d are the burning duration for each phase , Q_p and Q_d are the integrated energy for each phase and m_p and m_d are non dimensional shape factors for each phase (Heywood [96], Ferguson & Kirkpatrick [98]).

The variation of net heat release rate with CA over one cycle at a CR of 14 for Karanja biodiesel and Diesel oil are shown in Figures 4.77 and 4.78 respectively. It is observed that the peak net heat release rate occurs at 9^0 CA after TDC for Karanja biodiesel and 16^0 CA after TDC for Diesel oil.

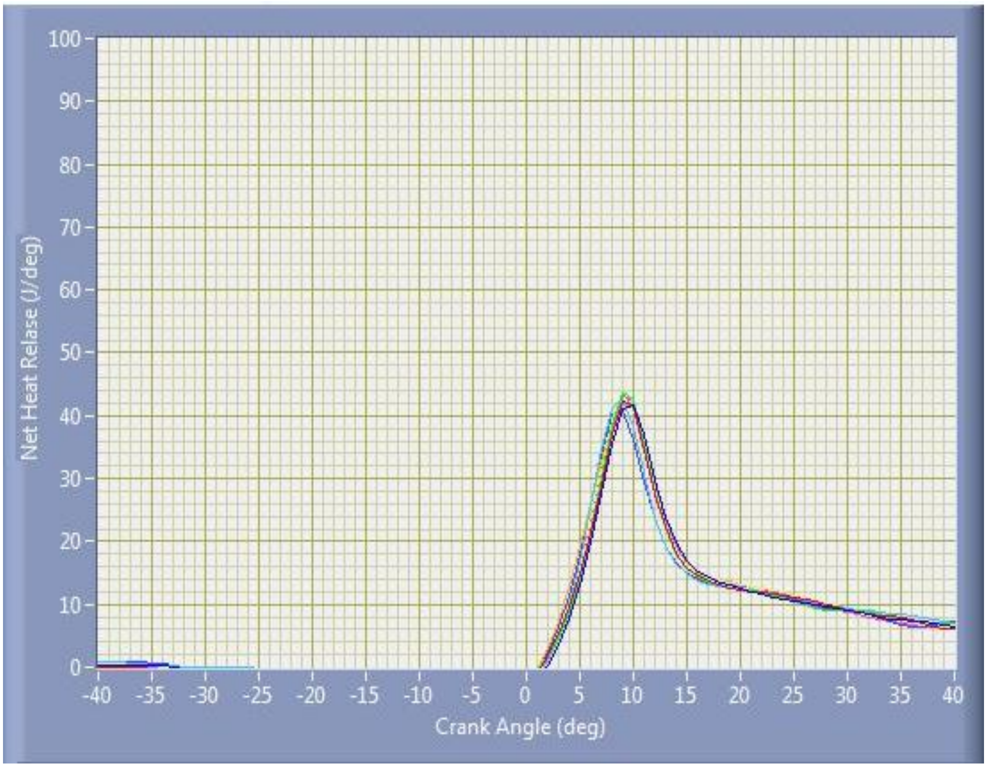


Figure 4.77 Variation of Net Heat Release Rate With CA at CR of 14 for Karanja Biodiesel

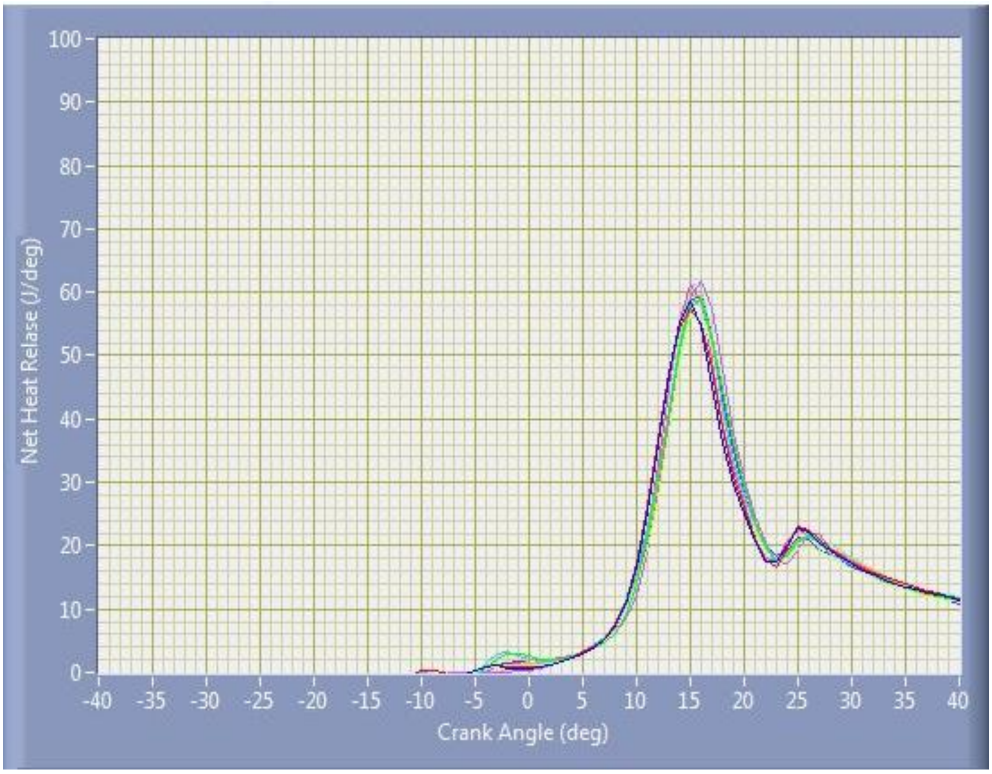


Figure 4.78 Variation of Net Heat Release Rate With CA at CR of 14 for Diesel Oil

The peak value of heat release rate for Karanja biodiesel is 44 J/deg as against 60 J/deg for Diesel oil. The lower net heat release rate for Karanja biodiesel can be attributed to its lower calorific value. It is a well known fact that the double peak shape of the heat release rate curve observed for Diesel oil is the characteristic of diesel combustion (Figure 4.78). The first peak that occurs during the premixed combustion phase results from rapid combustion of the portion of the injected fuel that has vaporized and mixed with air during the ignition delay period. The second peak occurs during the mixing controlled combustion phase. It can be noted that curves of different colours are observed in all the figures of combustion analysis henceforth. It should be noted that these curves are the perturbations of repeated cycles of engine operation.

The variation of net heat release rate with CA for one cycle of engine operation at a CR of 16 for Karanja biodiesel and Diesel oil are shown in Figures 4.79 and 4.80 respectively. It is observed that the peak net heat release rate for Karanja biodiesel is about 39 J/deg corresponding to 8^0 CA after TDC as against 40 J/deg for Diesel oil corresponding to 8^0 CA after TDC. For Karanja biodiesel the peak net heat release rate decreases by about 5 J/deg as compression ratio increases from 14 to 16 and the decrease is about 20 J/deg for Diesel oil.

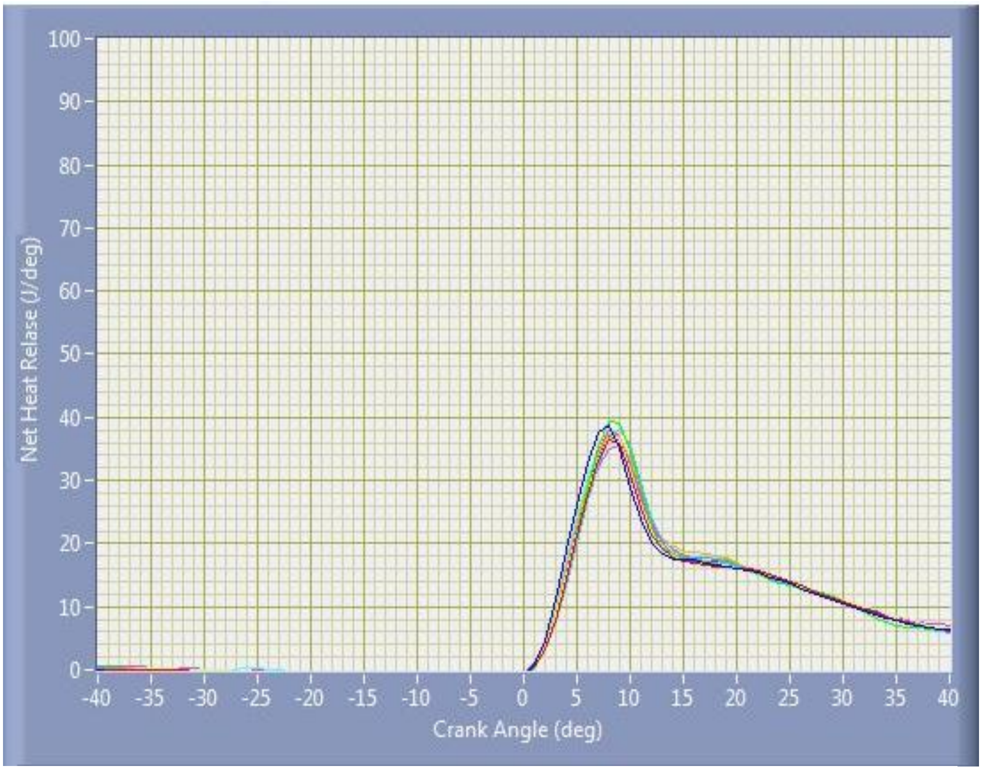


Figure 4.79 Variation of Net Heat Release Rate With CA at CR of 16 for Karanja Biodiesel

Reduced premixed combustion at higher CR for both the fuels can also be attributed to as the reason. The second peak of the net heat release rate curve is observed at the same CA of 18^0 for Karanja biodiesel and Diesel oil corroborates the identical thermal performance of both the fuels at higher CR of 16.

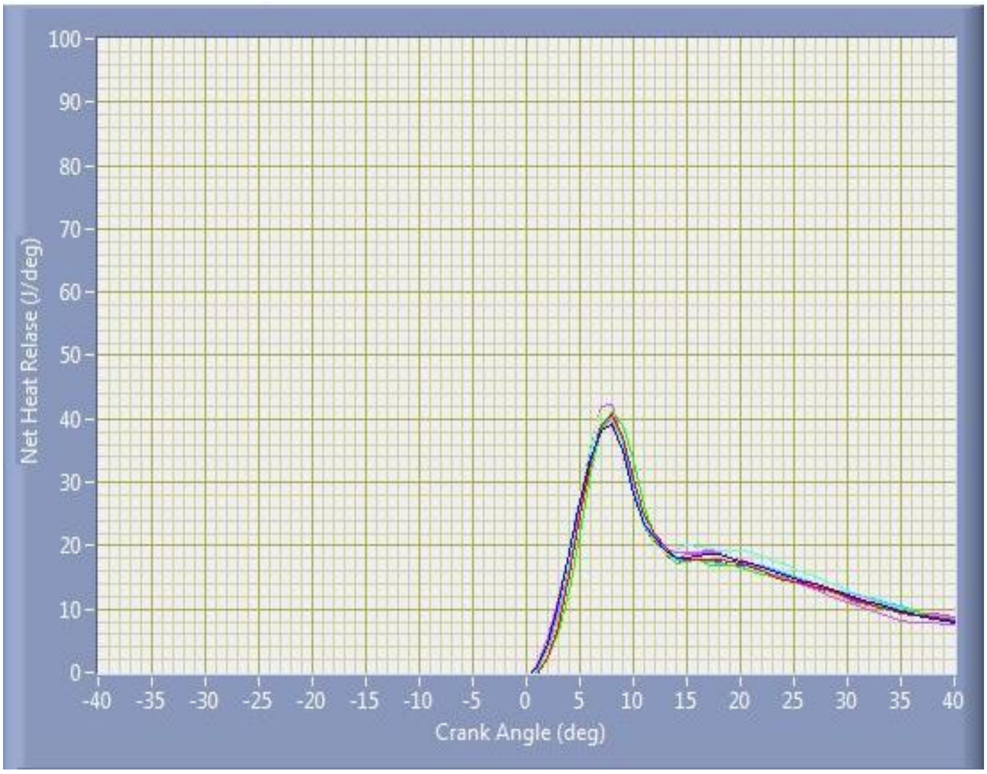


Figure 4.80 Variation of Net Heat Release Rate With CA at CR of 16 for Diesel Oil

The variation of net heat release rate with CA at a CR of 18 over one cycle of engine for Karanja biodiesel and Diesel oil are shown in Figures 4.81 and 4.82 respectively. It can be observed that the peak net heat release rate is same for both the fuels. Also, the second peak observed in case of Karanja biodiesel which is a characteristic of Diesel combustion in a diesel engine is found to occur at CR 18 which corroborates the closeness of thermal performance of Karanja biodiesel as compared to Diesel. The second peak occurs at 18^0 CA for both Karanja biodiesel and Diesel oil. Therefore, it can be inferred that at higher CRs the nature of variation of net heat release rate profile of Karanja biodiesel closely follows that of Diesel oil.

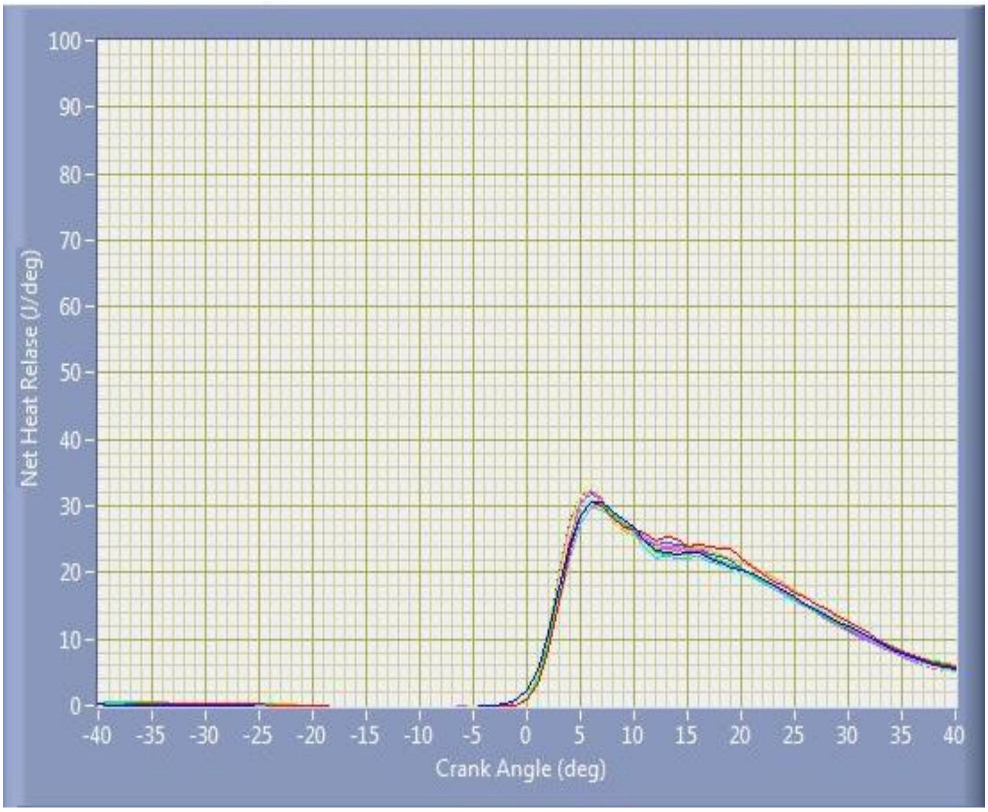


Figure 4.81 Variation of Net Heat Release Rate with CA at CR of 18 for Karanja Biodiesel

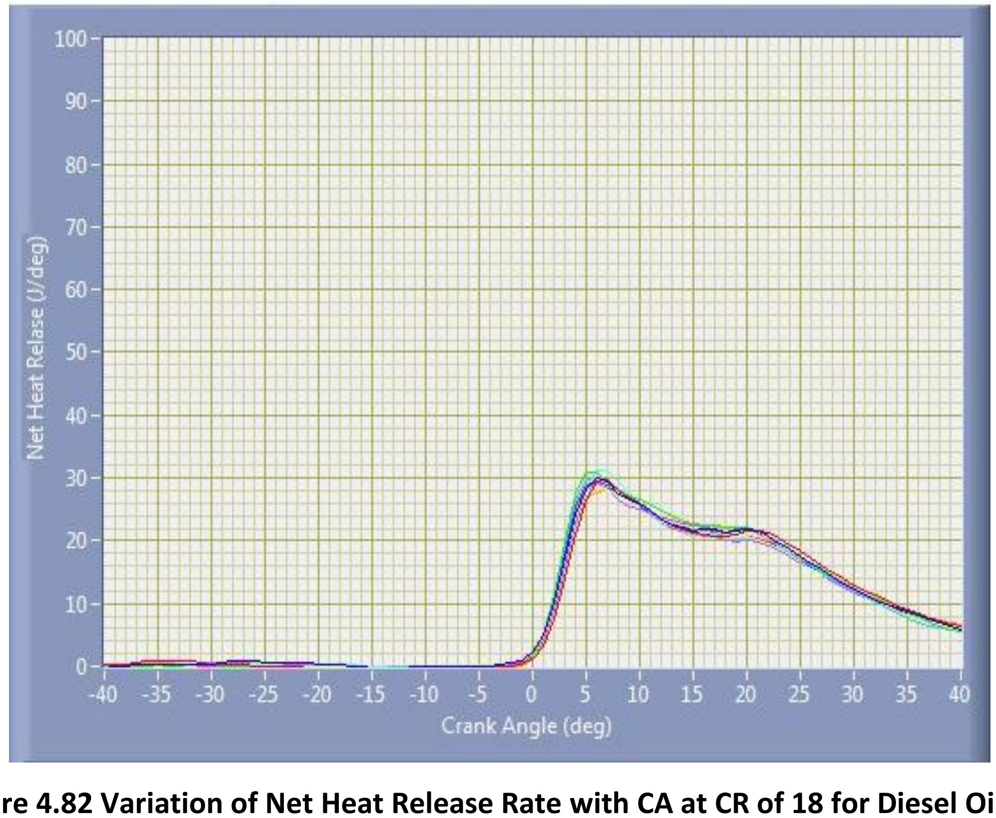


Figure 4.82 Variation of Net Heat Release Rate with CA at CR of 18 for Diesel Oil

Higher CR should be the mode of operation when operating with Karanja biodiesel. It can be noted that the peak net heat release rate for Karanja biodiesel is same as for Diesel oil which may probably be due to equal spread of the duration of combustion & the oxygenated nature of Karanja biodiesel resulting in complete combustion even though Diesel oil has a higher calorific value of about 6% as compared to Karanja biodiesel.

A comparison of peak net heat release rateof Karanja biodiesel with that of Diesel oil is given in Table 4.5. It can be observed that the first peak of heat release rate curve occurs early in case of Karanja biodiesel than Diesel oil. The trend is due to reduced ignition delay resulting in earlier combustion for Karanja biodiesel as its cetane number is more than of Diesel oil (47 for Diesel and 55 for Karanja biodiesel) (H. Aydin, C. Ilkilic [79]). The observed trend of shift of point of occurrence of first peak net heat release rate curve closer to TDC with increase in CR for both the fuels may be attributed to increased intensity of combustion due to higher temperature of compressed air as CR increases. At 14 CR no significant second peak observed for comparison for Karanja biodiesel. At CR of 16 the second peak heat release for Karanja biodiesel is 10% less as compared to Diesel oil. As CR increases from 16 to 18, the second peak net heat release rate increases for Karanja biodiesel and it is same as that of Diesel oil at 18CR.

Table 4.5 Comparison of Peak Net Heat Release Rate of Karanja Biodiesel with that of Diesel Oil (Reproduced From Figures 4.77-4.82)

CR	Fuel	Peak Net Heat Release (J/ ^{0} CA)		Position (^{0}CA)	
		First peak (Premixed Combustion)	Second peak (Mixing Controlled Combustion)	First Peak	Second Peak
14	Karanja biodiesel	44	--	9^0 after TDC	--
	Diesel oil	60	24	16^0 after TDC	25^0 after TDC
16	Karanja biodiesel	39	18	8^0 after TDC	17^0 after TDC
	Diesel oil	40	20	8^0 after TDC	17^0 after TDC
18	Karanja biodiesel	30	24	6^0 after TDC	18^0 after TDC
	Diesel oil	30	24	6^0 after TDC	20^0 after TDC

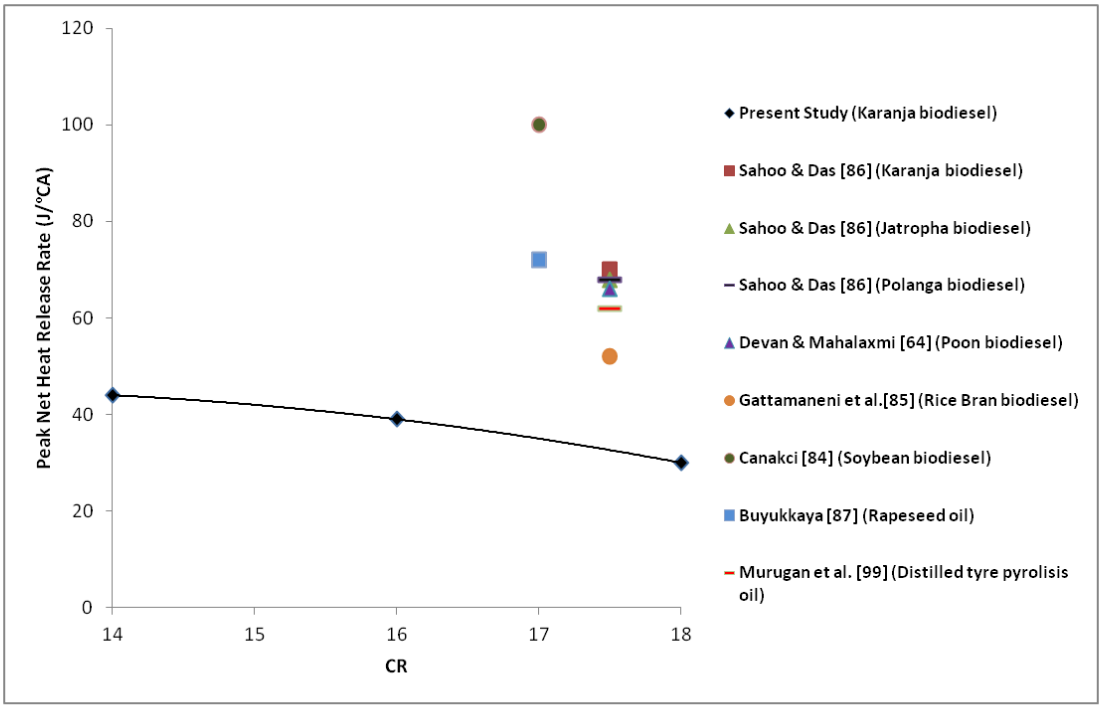


Figure 4.83 Comparison of Variation of Peak Net Heat Release Rate with CR at a Load of 12kg and an IP of 200bar of the Present Study with that of Earlier Investigations

Figure 4.83 shows the comparison of variation of net heat release rate with CR at full load of 12kg and IP of 200bar of the present study with that of earlier investigations of Sahoo & Das [86], Devan & Mahalaxmi [64], Gattamaneni et al. [85], Buyukkaya [87], Canakci [84], Murugan [99]. Sahoo & Das [86] used Karanja, Jatropha and Polanga biodiesels, Devan & Mahalaxmi [64], Gattamaneni et al. [85] and Canakci [84] used Poon, Rice bran and Soybean biodiesels respectively. Murugan [99] and Buyukkaya [87] used Distilled tyre pyrolisis oil and Rapeseed oil respectively.

It is observed that net heat release rate decreases with increase in CR for Karanja biodiesel used in the present study. It can be noted that there is no study reported for variation of net heat release rate with CR. Therefore comparison of trend of the present study with earlier investigations is not possible. However, various studies of net heat release rate at constant preset CR using different biodiesels are reported.

It is observed that net heat release rate achieved in investigation by Sahoo & Das [86] using Karanja biodiesel at a constant CR of 17.5 is 112% higher as compared that of present study. They also conducted studies using Jatropha and Polanga biodiesels. The net heat release rate is higher by about 106% for Jatropha and Polanga biodiesel as compared that of Karanja biodiesel used in the present study. Various investigators also reported higher net heat release rate at constant CR of 17.5 using Poon biodiesel, Ricebran biodiesel and Distilled tyre pyrolisis oil. It is observed that net heat release rate is higher by 100%, 57% and 87% for Poon biodiesel, Ricebran biodiesel and Distilled tyre pyrolisis oil respectively as compared to Karanja biodiesel used in the present study. At a constant CR of 17, net heat release rate for Rapeseed oil and Soybean biodiesel is 94% and 170% higher respectively as compared to that of Karanja biodiesel used in the present study. The observed deviation could not be attributed to any specific reason existing in the available literature. However, an earnest effort has been made in this direction and the results are presented for further exploration and analysis in this area.

4.2.4.3 Rate of Pressure Rise

Variations of rate of pressure rise (also known as pressure derivative, $dP/d\Theta$) with CA as obtained by the plots directly indicated by the software for Karanja biodiesel and Diesel oil at CRs of 14, 16 and 18 are presented in the Figures 4.84-4.89. $dP/d\Theta$ curves indicate the slope of PΘ diagram. The maximum rate of pressure rise $(dP/d\Theta)_{max}$ and CA at which it occurs are clearly visible from these figures. The rate of change of pressure is substantially affected by the rate of burning. The occurrence of maximum rate of pressure rise should be close to TDC for proper burning and knock free combustion to take place (Heywood [96], V. Ganesan [97]). The shapes of the curves depend on whether the combustion process is fast and robust. Under robust conditions the distributions are closer to normal distributions i.e. attainment of peak pressure and rate of pressure rise taking place after TDC but not too far from the same. The point of attainment of peak pressure indicates that major part of the heat release has taken place. This phase is followed by a sharp fall in the rate of pressure rise (Heywood [96]) and a steady value of $dp/d\Theta$.

The variation of rate of pressure rise with CA at a CR of 14 for Karanja biodiesel and Diesel oil are shown in Figures 4.84 and 4.85 respectively. It is observed that for Karanja biodiesel as fuel, the maximum rate of pressure rise occurs at 10^0 of CA after TDC and the value is 2.5 bar/^{0}CA whereas it is 2.6 bar/^{0}CA at 15^0 after TDC for Diesel oil.

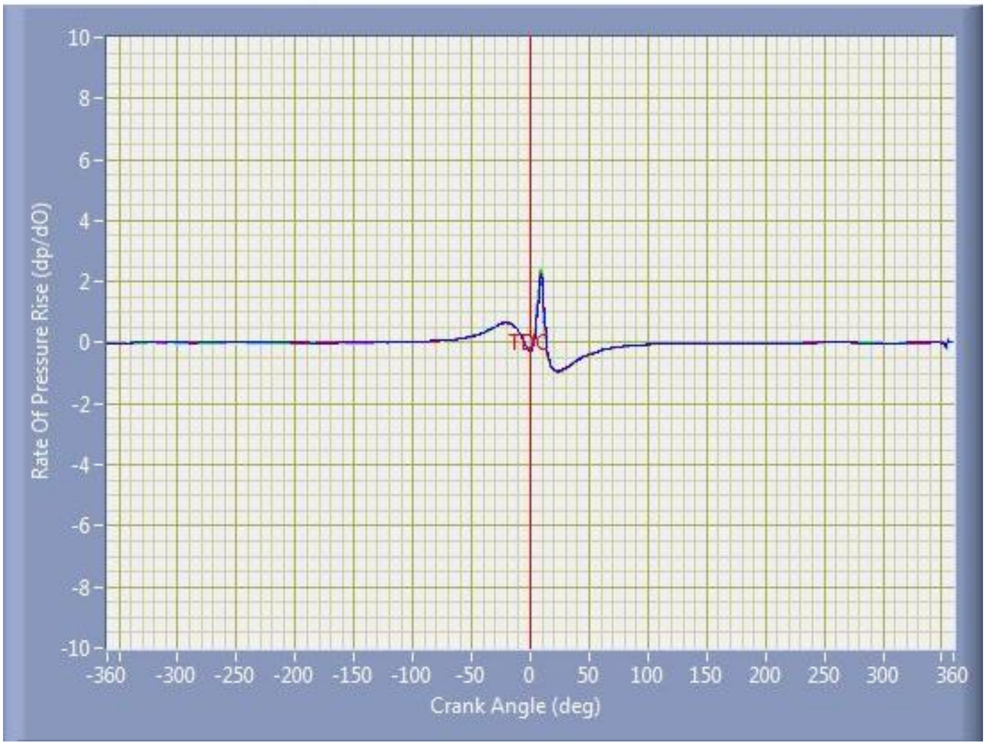


Figure 4.84 Variation of Rate of Pressure Rise with CA at CR of 14 for Karanja Biodiesel

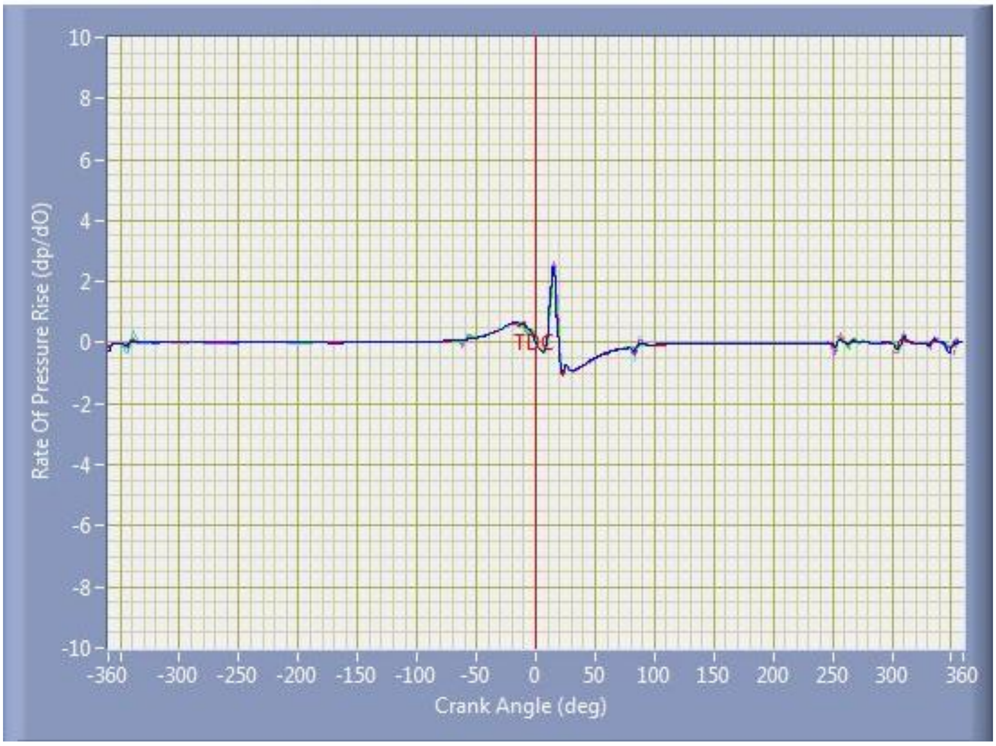


Figure 4.85 Variation of Rate of Pressure Rise With CA at CR of 14 for Diesel Oil

It can also be observed that the maximum rate of pressure rise takes place closer to TDC is case of Karanja biodiesel as fuel than Diesel oil. This observation can be supported by the fact that Karanja biodiesel has higher cetane number due to which combustion begins earlier and is fairly smooth. The spikes appearing with respect to Diesel oil combustion is due to experimental perturbations.

The variation of rate of pressure rise with CA at a CR of 16 for Karanja biodiesel and Diesel oil are shown in Figures 4.86 and 4.87 respectively. It is observed that the maximum rate of pressure rise is 2.5 bar/^{0}CA and it occurs at 8^0 after TDC for both Karanja biodiesel and Diesel oil. It can be noted here is that the maximum rate of pressure rise occurs 2^0 for Karanja biodiesel and 7^0 for Diesel oil closer to TDC as compared to that at 14 CR which results in knock free combustion.

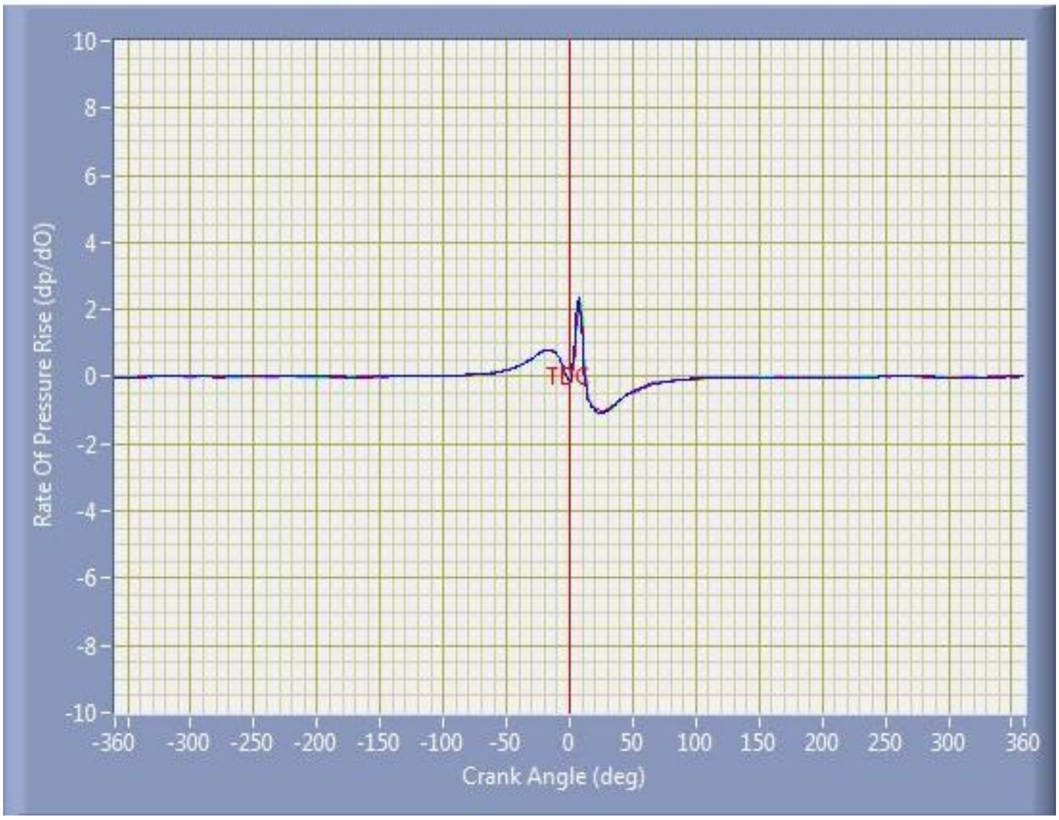


Figure 4.86 Variation of Rate of Pressure Rise With CA at CR of 16 for Karanja Biodiesel

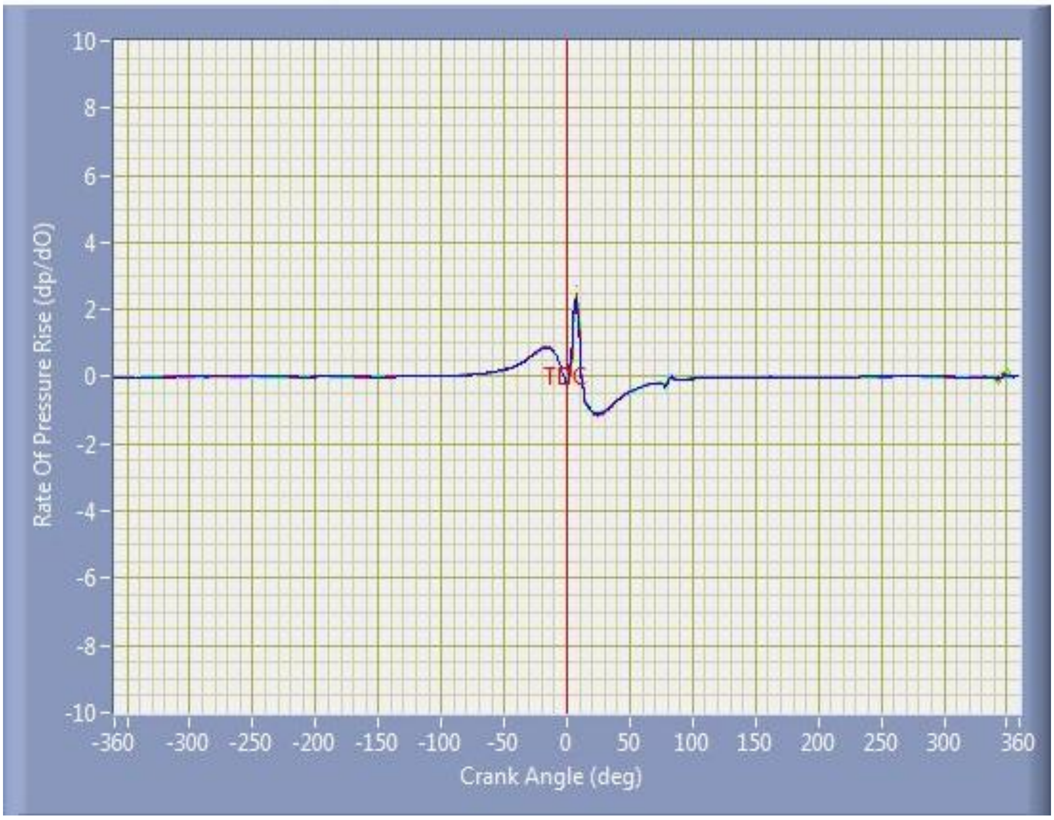


Figure 4.87 Variation of Rate of Pressure Rise with CA at CR of 16 for Diesel Oil

The variation of rate of pressure rise with CA at a CR of 18 for Karanja biodiesel and Diesel oil are shown in Figures 4.88 and 4.89 respectively. The trend of variation observed is similar to that observed at CR of 16 but maximum rate of pressure rise is 2 bar/deg of crank angle and takes place at 5^0 CA after TDC for both Karanja biodiesel and Diesel oil. All the discussions indicate that an engine can perform better with reduced chances of abnormal combustion at a CR of 18 as the maximum rate of pressure rise occurs closer to but after TDC. The performance with Karanja biodiesel is similar that of Diesel oil.

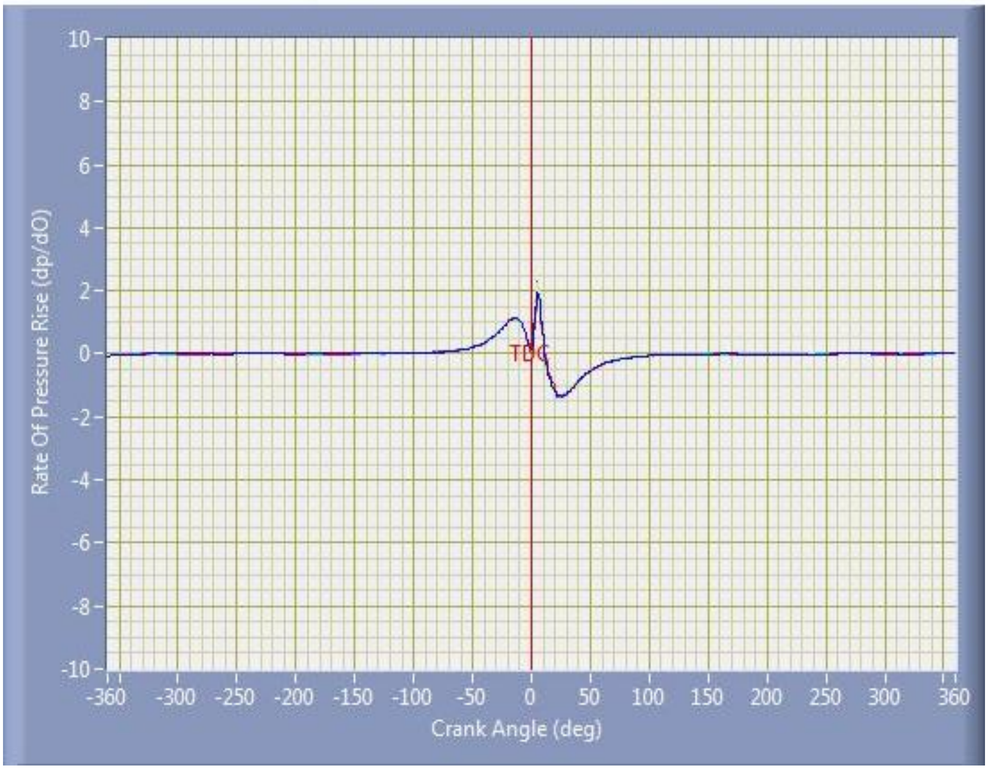


Figure 4.88 Variation of Rate of Pressure Rise With CA at CR of 18 for Karanja Biodiesel

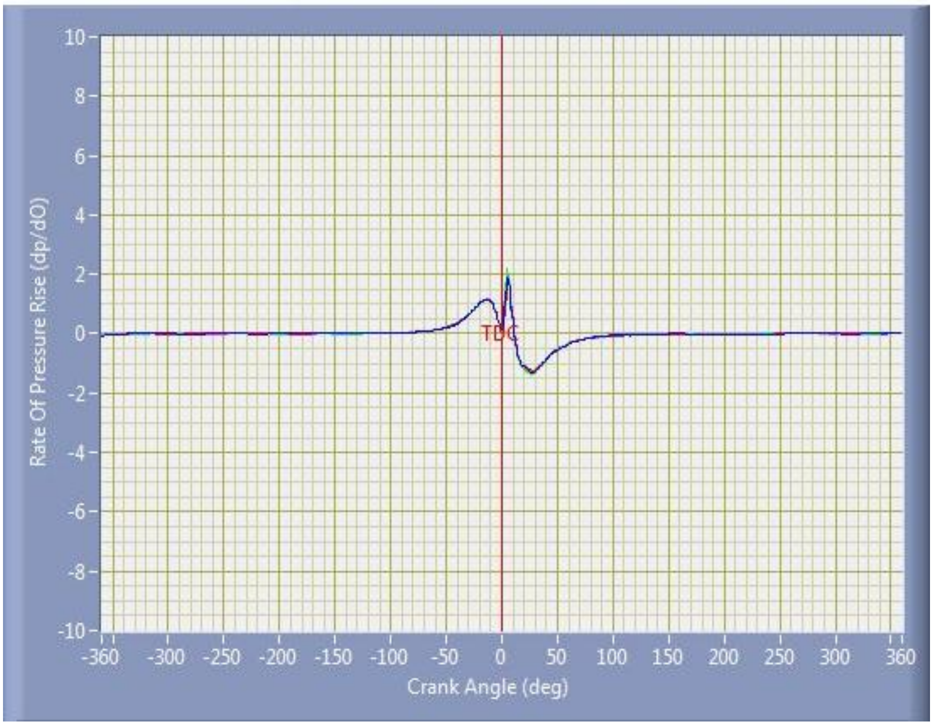


Figure 4.89 Variation of Rate of Pressure Rise with CA at CR of 18 for Diesel Oil

From Figures 4.84-4.89 it can be observed that each curve consists of two peaks and one trough for the fuels tested. They correspond to the phases of premixed combustion, mixed controlled combustion and late combustion respectively. Table 4.6 gives a comparison of rate of pressure rise of Karanja biodiesel with that of Diesel oil. It can be noted that the deviation of performance between Karanja biodiesel and Diesel oil reduces with increasing CR. It is observed that the first peak increases with increase in CR from 0.7 to 1.2 bar/^{0}CA for both the fuels due to higher rate of premixed combustion as CR increases.

Table 4.6 Comparison of Rate of Pressure Rise of Karanja Biodiesel With that of Diesel Oil (Reproduced From Figures 4.84-4.89)

CR	Fuel	Rate of Pressure Rise (bar/^{0}CA)		Position (^{0}CA)	
		First Peak	Second Peak	First Peak	Second Peak
14	Karanja biodiesel	0.7	2.5	20^0 before TDC	10^0 after TDC
	Diesel oil	0.7	2.6	20^0 before TDC	15^0 after TDC
16	Karanja biodiesel	0.8	2.5	17^0 before TDC	8^0 after TDC
	Diesel oil	1	2.5	18^0 before TDC	8^0 after TDC
18	Karanja biodiesel	1.2	2	15^0 before TDC	5^0 after TDC
	Diesel oil	1.2	2	13^0 before TDC	5^0 after TDC

Figure 4.90 shows the comparison of variation of maximum rate of pressure rise with CR at full load of 12kg and IP of 200bar of the present study with that of earlier investigations of Gattamaneni et al. [85] and Murugan [99]. The investigators used Rice bran biodiesel and Distilled tyre pyrolisis oil respectively.

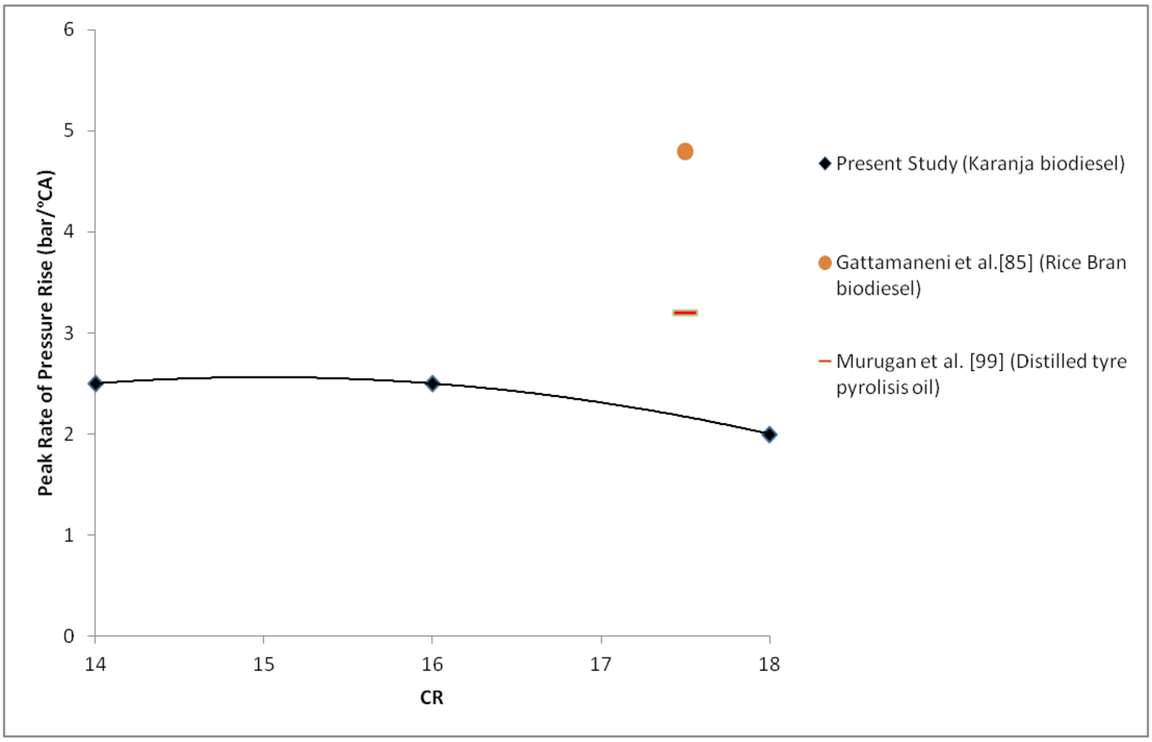


Figure 4.90 Comparison of Variation of Peak Rate of Pressure Rise with CR at a Load of 12kg and an IP of 200bar of the Present Study with that of Earlier Investigations

It is observed that maximum rate of pressure rise is uniform but marginally decreases with increase in CR for Karanja biodiesel used in the present study. The trend may be due to more uniform pressure rise as a result of even spread of combustion duration for different phases. It can be noted that there is no study reported for variation of maximum rate of pressure rise with CR. Therefore comparison of trend of the present study with earlier investigations is not possible. However, various studies of maximum rate of pressure rise at constant preset CR using different biodiesels are presented in Figure 4.90.

It is observed that at a constant CR of 17.5, maximum rate of pressure rise for Ricebran biodiesel and Distilled tyre pyrolisis oil is higher by almost 118% and 45% respectively as compared to that of Karanja biodiesel used in the present study. The reasons for the trend could not be ascertained from the earlier literature.

4.2.4.4 Mass Fraction Burnt

Variation of mass fraction burnt with CA for the engine operated with Karanja biodiesel and Diesel oil at CRs of 14, 16 and 18 are given in Figures 4.91 to 4.96. When combustion of the charge in a CI engine begins, a single flame front travels through the charge until it gets fully burnt. During the burning of the charge, the combustion zone can be divided into burnt and unburned zone. As the combustion progresses, the burnt zone part increases and unburned zone part reduces. Mass fraction burnt is the fraction of charge burnt for each cycle and it is expressed in percentage. In other words, the percent mass fraction burnt is an indication of the percentage of fuel burnt in the combustion chamber during the combustion process (Ferguson & Kirkpatrick [98]).

The variation of mass fraction burnt with CA at a CR of 14 for Karanja biodiesel and Diesel oil are shown in Figures 4.91 and 4.92 respectively. It is seen that 90 % of the fuel mass is burnt at 21^0 CA after TDC (over a total of 21^0CA) when Karanja biodiesel is the fuel, whereas, the same amount of fuel mass is burnt at 27^0 CA after TDC (over a total of 40^0 CA) using Diesel oil as fuel. A higher rate of fuel burning (same mass fraction is burnt in less CA) is observed for Karanja biodiesel as compared to Diesel oil (Table 4.7). The trend may be due to higher oxygen content and higher value of cetane number of Karanja biodiesel which initiates early combustion, complete burning of fuel over lesser ^{0}CA as compared to Diesel. 80% of mass is burned over 14^0 CA for Karanja biodiesel and over 17^0 CA for Diesel oil.

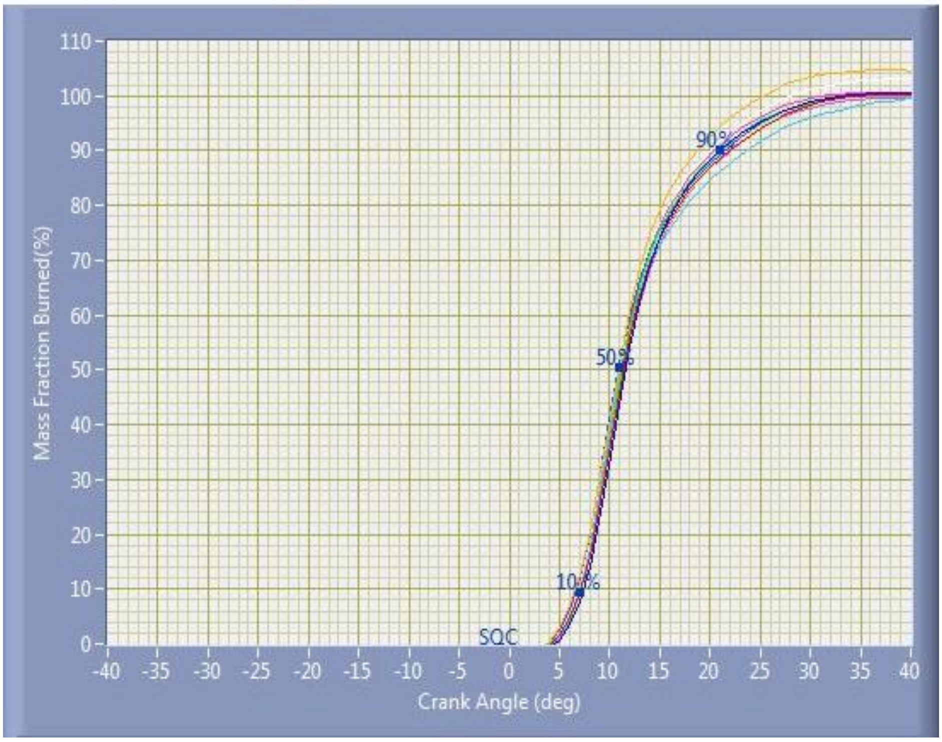


Figure 4.91 Variation of Mass Fraction Burnt With CA at CR of 14 for Karanja Biodiesel

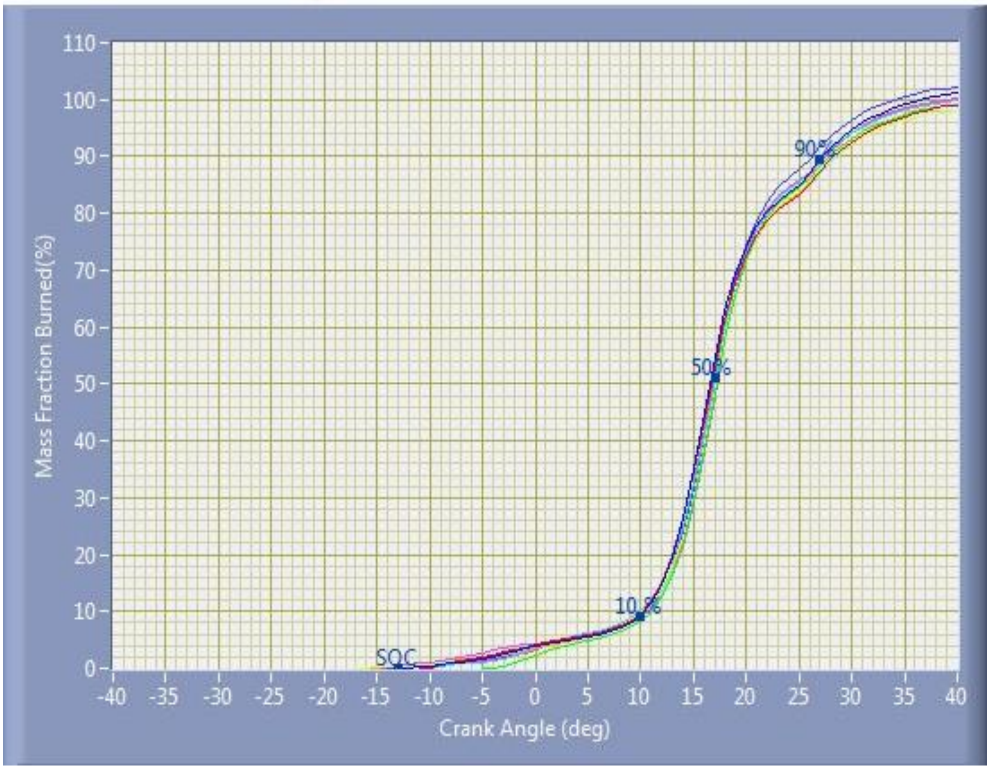


Figure 4.92 Variation of Mass Fraction Burnt With CA at CR of 14 for Diesel Oil

The variation of mass fraction burnt with CA at a CR of 16 for Karanja biodiesel and Diesel oil are shown in Figures 4.93 and 4.94 respectively. 90% fuel mass is burnt at 23^0 CA after TDC for Karanja biodiesel and 22^0 CA after TDC for Diesel oil as observed. It can be noted that as the CR is increased from 14 to 16 the deviation of performance between the two fuels used reduces. The same observation can also be made from Figures 4.70 to 4.75, 4.77 to 4.82 and 4.84 to 4.89. 80% mass fraction is burnt nearly over the same value of 17^0 CA for Karanja biodiesel and Diesel oil indicating that the performance is similar for both fuels.

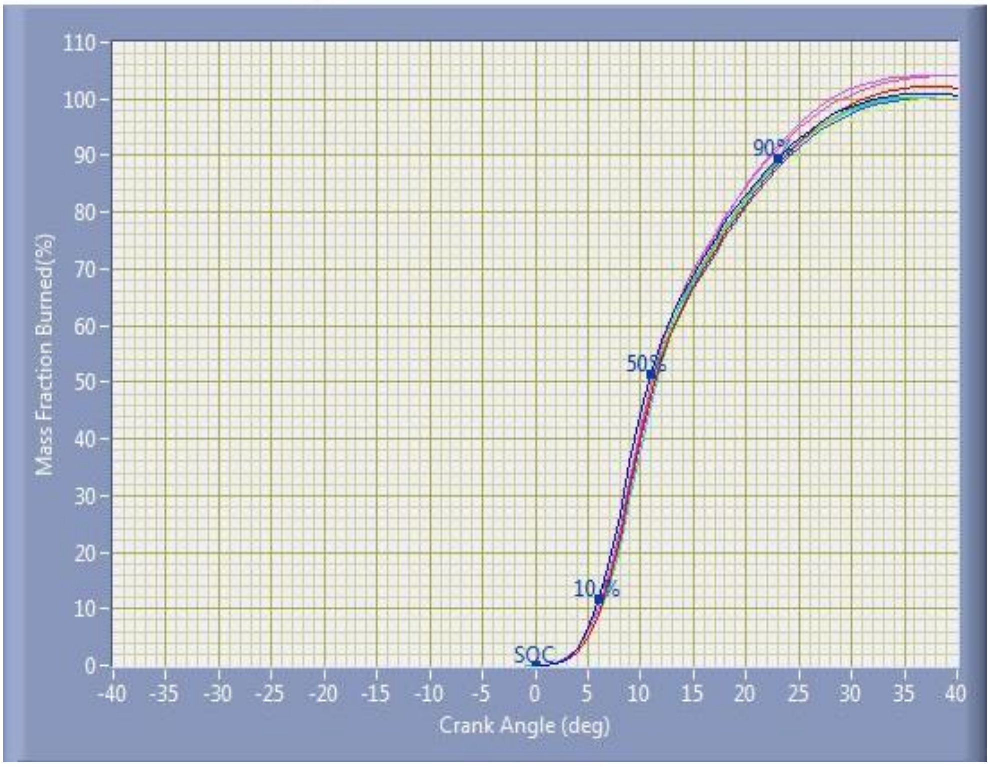


Figure 4.93 Variation of Mass Fraction Burnt With CA at CR of 16 for Karanja Biodiesel

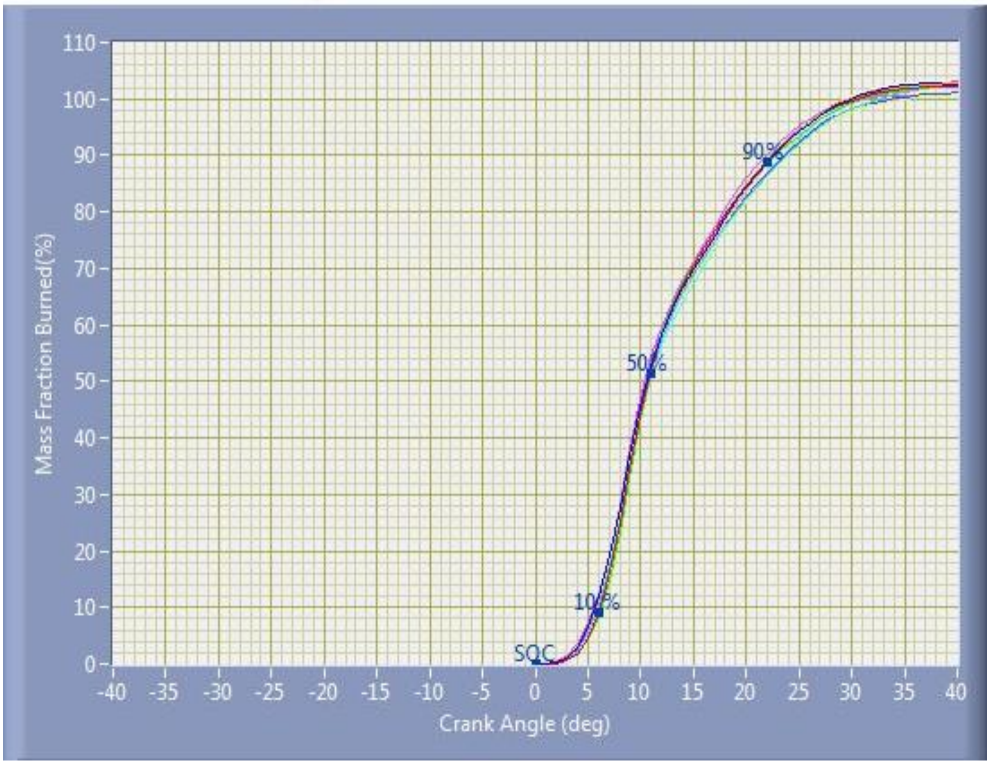


Figure 4.94 Variation of Mass Fraction burnt with CA at CR of 16 for Diesel Oil

The variation of mass fraction burnt with CA at a CR of 18 for Karanja biodiesel and Diesel oil are shown in Figures 4.95 and 4.96 respectively. It can be observed that there is no difference in mass fractions burnt with CA for both the fuels at higher CR of 18. Hence, it is better to substitute Diesel oil with Karanja biodiesel completely in a diesel engine at higher CRs.

Figure 4.95 Variation of Mass Fraction Burnt with CA at CR of 18 for Karanja Biodiesel

It should be noted that curves of different colours are observed in all the figures of combustion analysis. The curves are the perturbations of repeated cycles of engine operation as mentioned earlier. Table 4.7 shows the comparison of fuel mass fraction burnt with the engine operated using Karanja biodiesel and Diesel oil as fuels. It can be observed that similar behaviour is found at higher CR of 16 and 18 between Karanja biodiesel and Diesel oil at higher CRs of 16 and 18 with respect to mass fraction burnt. Although, at CR of 14 there is an earlier combustion for all mass fractions burnt by about 4^0 to 5^0 CA after TDC with Karanja biodiesel as compared to Diesel oil.

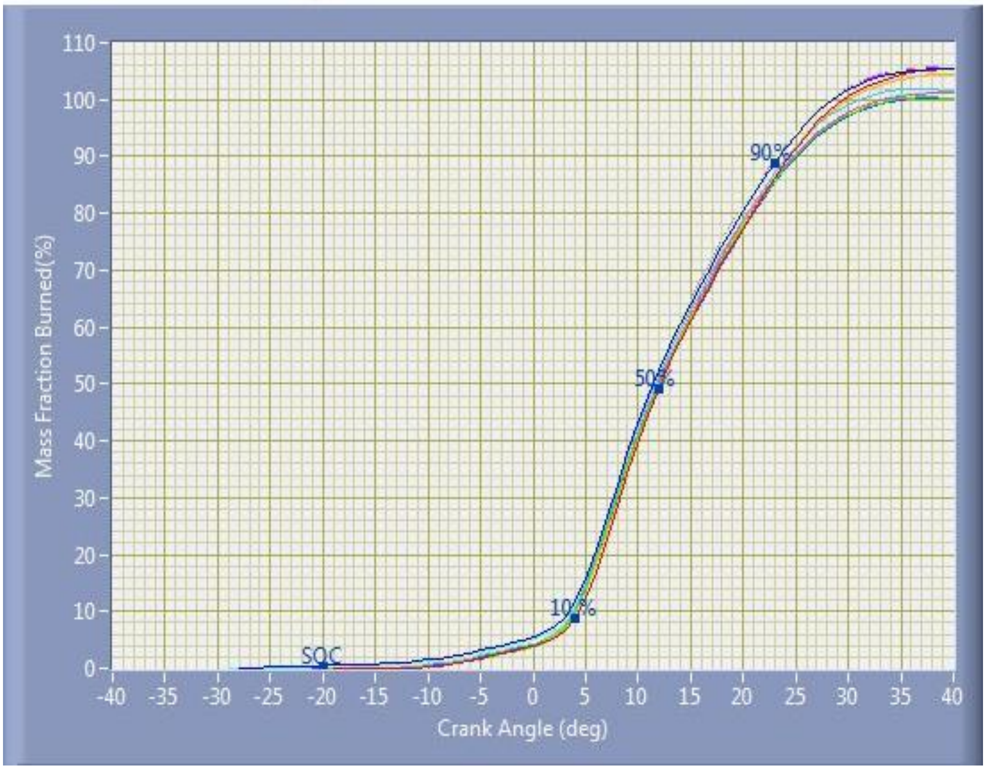


Figure 4.96 Variation of Mass Fraction Burnt With CA at CR of 18 for Diesel Oil

Table 4.7 Comparison of Mass Fraction of Fuel Burnt Between Diesel Oil and Karanja Biodiesel (Reproduced From Figures 4.91-4.96)

CR	Mass fraction burned (%)	Position (^{0}CA)	Fuel
14	10	7^0 after TDC	Karanja Biodiesel
	10	10^0 after TDC	Diesel oil
	50	11^0 after TDC	Karanja Biodiesel
	50	17^0 after TDC	Diesel oil
	90	21^0 after TDC	Karanja Biodiesel
	90	27^0 after TDC	Diesel oil
16	10	6^0 after TDC	Karanja Biodiesel
	10	6^0 after TDC	Diesel oil
	50	11^0 after TDC	Karanja Biodiesel
	50	11^0 after TDC	Diesel oil
	90	23^0 after TDC	Karanja Biodiesel

	90	22^0 after TDC	Diesel oil
18	10	4^0 after TDC	Karanja Biodiesel
	10	4^0 after TDC	Diesel oil
	50	11^0 after TDC	Karanja Biodiesel
	50	12^0 after TDC	Diesel oil
	90	23^0 after TDC	Karanja Biodiesel
	90	23^0 after TDC	Diesel oil

4.2.4.5 Mean Gas Temperature

Variation of mean gas temperature with CA for the engine operated with Karanja biodiesel and Diesel oil at CRs of 14, 16 and 18 are given in Figures 4.97 to 4.102. The average value of burned & unburned gas temperatures existing in the combustion chamber during a cycle is called mean gas temperature. The gas in the cylinder is the mixture of burnt and unburnt fuel air mixture. Mean gas temperature decides the rate of reaction during burning of fuel and it is desirable that the value be closer to the adiabatic flame temperature. Adiabatic flame temperature is the temperature achieved for the products of combustion if the reaction is adiabatic and no loss of thermal energy takes place in any other manner, ensuring that all the heat evolved helps in increasing the products of combustion to a maximum possible value.

The variation of mean gas temperature with CA at CR of 14 for Karanja biodiesel and Diesel oil are shown in Figures 4.97 and 4.98 respectively. It is observed that the minimum value of mean gas temperature is 40 ^{0}C less for Karanja biodiesel compared to Diesel oil at 40^0 CA before TDC. It remains almost constant and then rises to 1060^0C at 25^0 CA after TDC in case of Karanja biodiesel whereas the same is around 1430^0C at 35^0 CA after TDC for Diesel oil. It can be noted that the maximum mean gas temperature for Karanja biodiesel is about 25% less as compared to that of Diesel oil. The lower mean gas temperature for Karanja biodiesel may be due to the fact that heat release is less as calorific value of Karanja biodiesel is about 6% less than that of Diesel oil. It is observed that for Karanja biodiesel, the sudden rise in the mean gas temperature takes place at 5^0 after TDC and at 10^0 after TDC for Diesel oil. The earlier rise for Karanja biodiesel may be due to its lesser ignition delay.

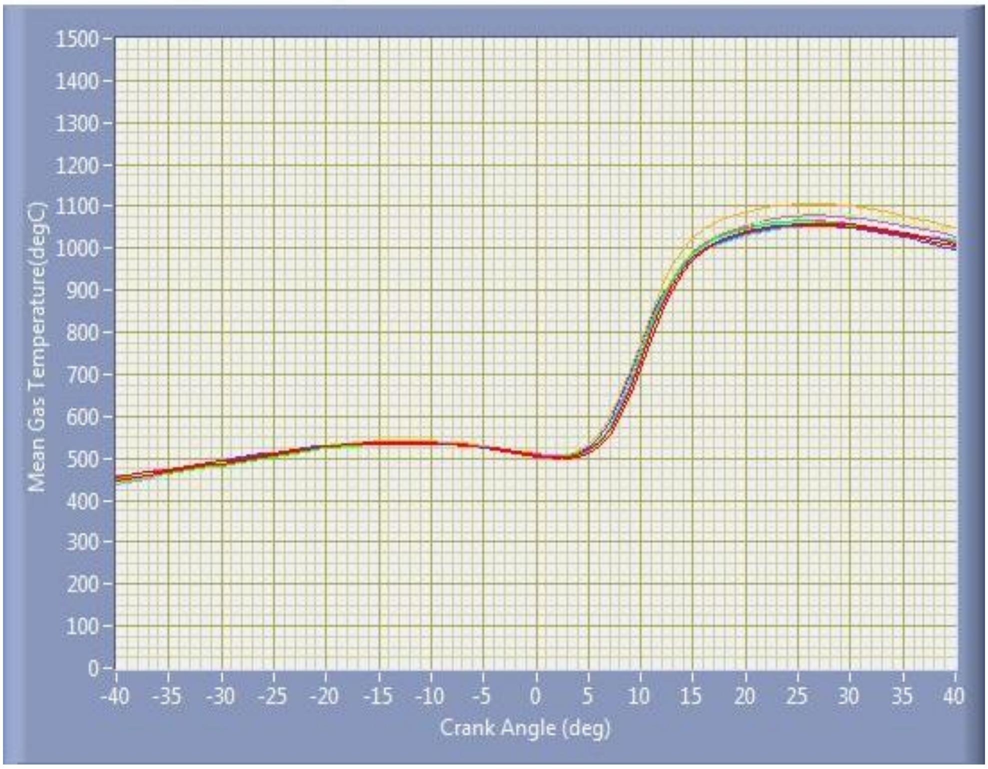


Figure 4.97 Variation of Mean Gas Temperature With CA at CR of 14 for Karanja Biodiesel

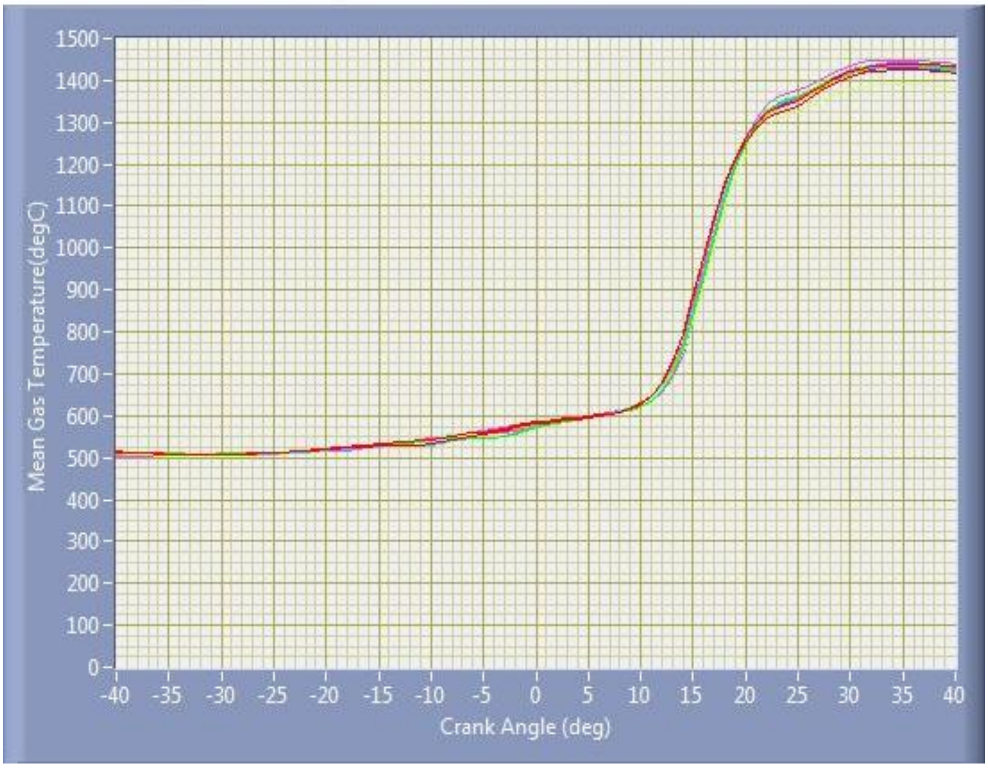


Figure 4.98 Variation of Mean Gas Temperature With CA at CR of 14 for Diesel Oil

The variation of mean gas temperature with CA at a CR of 16 for Karanja biodiesel and Diesel oil are shown in Figures 4.99 and 4.100 respectively. The minimum mean gas temperature is 421^0C and 500^0C for Karanja biodiesel and Diesel oil respectively at 40^0 before TDC. Mean gas temperature starts to increase at 5^0 CA after TDC for both the fuels tested reaching a value of 1130^0C and 1200^0C for Karanja biodiesel and Diesel oil respectively. The gap between the maximum value of mean gas temperature between the two fuels reduces as CR increases from 14 to 16.

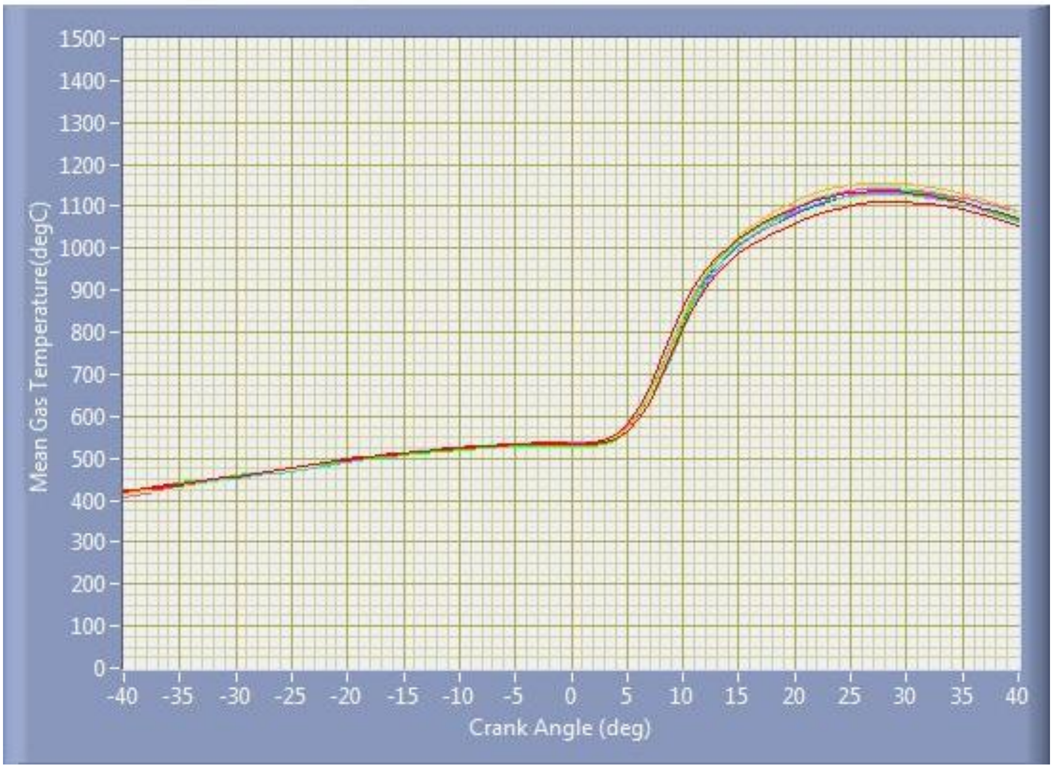


Figure 4.99 Variation of Mean Gas Temperature with CA at CR of 16 for Karanja Biodiesel

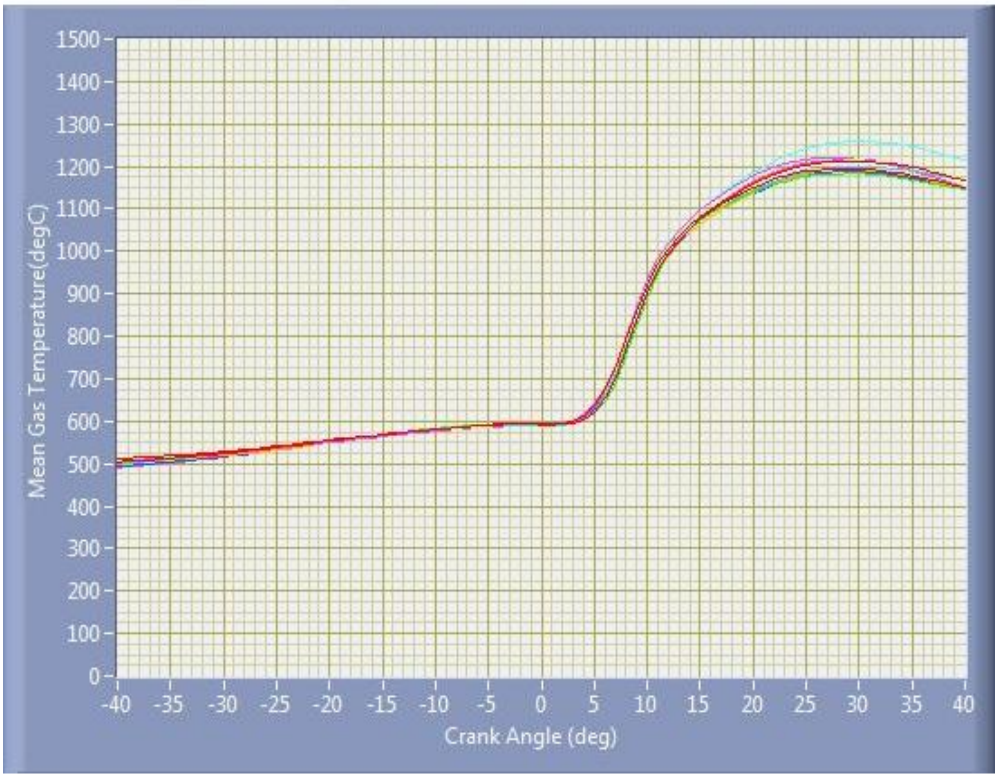


Figure 4.100 Variation of Mean Gas Temperature With CA at CR of 16 for Diesel Oil

The variation of mean gas temperature with CA at a CR of 18 for Karanja biodiesel and Diesel oil are shown in Figures 4.101 and 4.102 respectively. It is observed that the maximum mean gas temperature for Karanja biodiesel and Diesel oil are 1280^0C and 1275^0C respectively. The minimum temperatures achieved are identical and the crank angle positions at which it is observed are same for the fuels. Therefore, it can be inferred that Karanja biodiesel performs similar to Diesel oil at higher CR of 18 even though its calorific value is less and has a relatively higher viscosity.

Table 4.8 gives a comparison of mean gas temperatures for Karanja biodiesel with that of Diesel oil at CRs of 14, 16 & 18, full load of 12kg and IP of 200bar. It is observed that the mean gas temperature increases with increase in CR for Karanja biodiesel whereas the trend is opposite for Diesel oil. It can be attributed to the fact that more heat is released in diffusion burning (mixing controlled combustion) phase for Karanja biodiesel as compared to Diesel oil (Refer Table 4.5). Greater mass fraction is burnt at earlier CA for Karanja biodiesel as CR increases also results in increase of mean gas temperature (Refer Table 4.7). Delayed combustion and reduce in the peak net heat release rate with increase in CR account for the decrease in mean gas temperature with

Diesel oil. It can be noted that the behavior of Karanja biodiesel compared to Diesel oil are identical qualitatively at higher CR of 18.

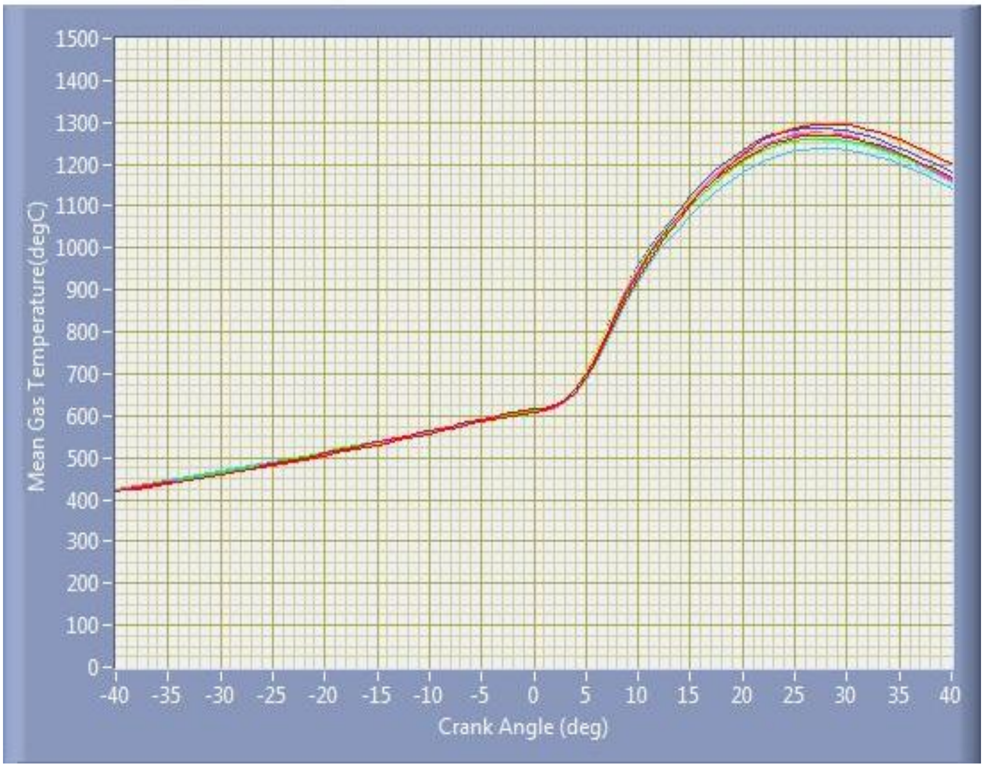


Figure 4.101 Variation of Mean Gas Temperature With CA at CR of 18 for Karanja Biodiesel

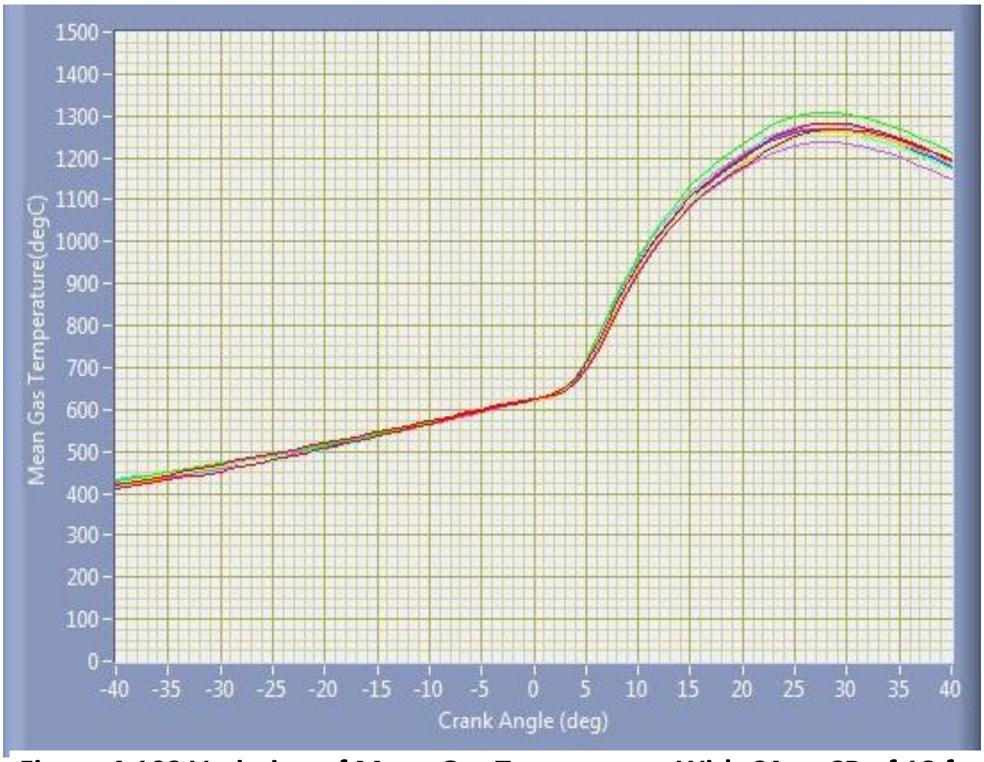


Figure 4.102 Variation of Mean Gas Temperature With CA at CR of 18 for Diesel Oil

A comparison between Table 4.7 and Table 4.8 shows that the mean gas temperature is highest when more than 90% of mass fraction is burnt for both fuels. It is obvious as mean gas temperature, is the result of thermal energy released during combustion of fuel. So more fuel burnt results in higher mean gas temperature.

Table 4.8 Comparison of Mean Gas Temperature of Karanja Biodiesel With that of Diesel Oil (Reproduced From Figures 4.97-4.102)

CR	Fuel	Mean Gas Temperature (^{0}C)		Position (^{0}CA)	
		Max	Min	Max	Min
14	Karanja biodiesel	1060	460	25^0 after TDC	40^0 before TDC
	Diesel oil	1430	500	35^0 after TDC	40^0 before TDC
16	Karanja biodiesel	1130	421	27^0 after TDC	40^0 before TDC
	Diesel oil	1200	500	28^0 after TDC	40^0 before TDC
18	Karanja biodiesel	1280	425	28^0 after TDC	40^0 before TDC
	Diesel oil	1275	425	27^0 after TDC	40^0 before TDC

4.2.4.6 PV characteristics

Variation of CP with cylinder volume for the engine operated with Karanja biodiesel and Diesel oil at CRs of 14, 16 and 18 are given in Figures 4.103 to 4.108. Cylinder pressure changes with crank angle as a result of cylinder volume change during combustion of fuel.. Since both the compression of the unburned mixture prior to combustion and the expansion of burned gases following the end of combustion are close to isentropic processes, the observed behaviour is as expected (Heywood [96]).

It can be inferred from the figures that peak value of cylinder pressure at lower compression ratio (CR 14) is lower for Karanja biodiesel (38 bar) by about 2bar as compared to that of Diesel oil (40 bar). As CR increases to 16, CP increases to 44bar and 46bar for Karanja biodiesel and Diesel oil respectively. However, as CR increases to 18, cylinder pressure increases to 50bar for both the fuel runs on the engine implying that identical performance exists at higher compression ratio of 18 exists for both fuels. For CR of 14, the areas of the PV plots for Karanja biodiesel are found to be less by about 30% as compared to Diesel oil which means work done during the cycle is comparatively less for Karanja biodiesel than Diesel oil. At CR of 16 the area is only 3% less for Karanja biodiesel. This is observed by area measurements of PV plots. As the CR increases to 18, the areas are almost identical , indicating same performance for both fuels. Between CR of 16 to 18 peak CP increases by 31% (38 to 50 bar) and 25% (40 to 50 bar) for Karanja biodiesel and Diesel oil respectively. The results obtained are interpreted earlier from the figures of CP with CA.

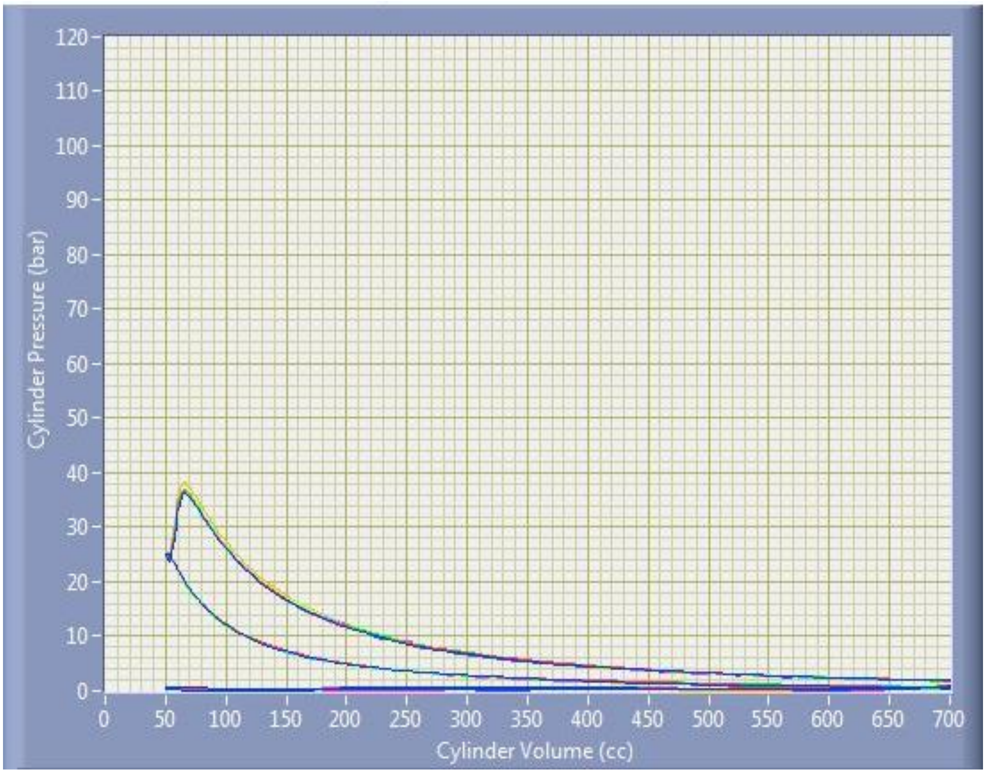


Figure 4.103 Variation of CP With Cylinder Volume at CR of 14 for Karanja Biodiesel

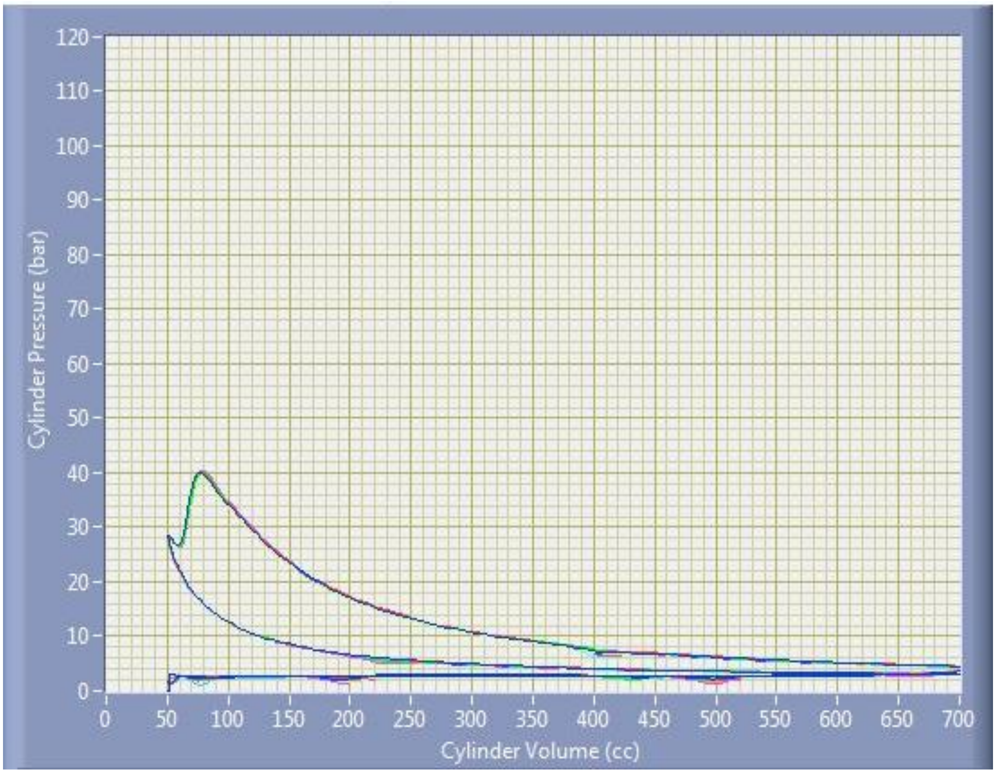


Figure 4.104 Variation of CP With Cylinder Volume at CR of 14 for Diesel Oil

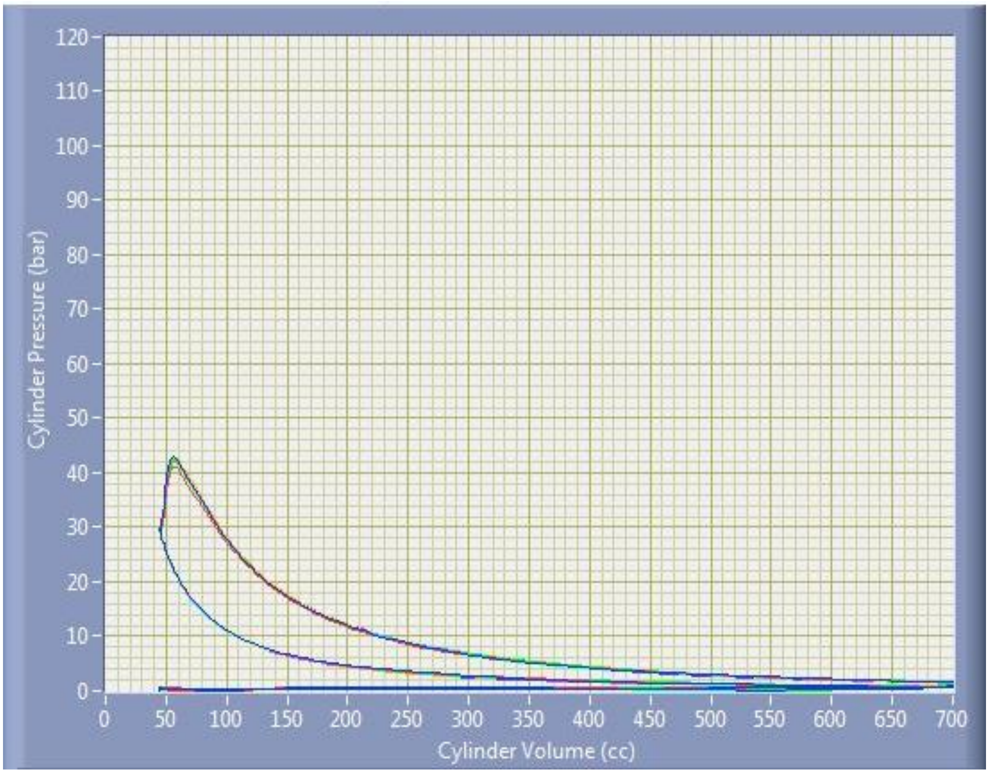


Figure 4.105 Variation of CP With Cylinder Volume at CR of 16 for Karanja Biodiesel

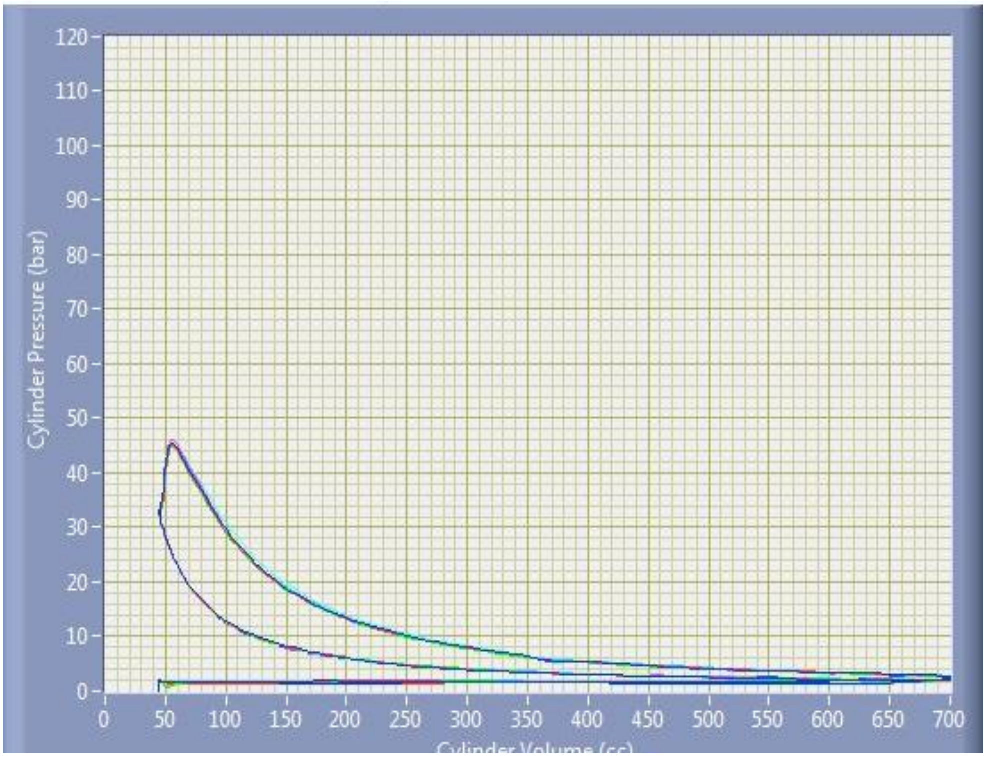


Figure 4.106 Variation of CP With Cylinder Volume at CR of 16 for Diesel Oil

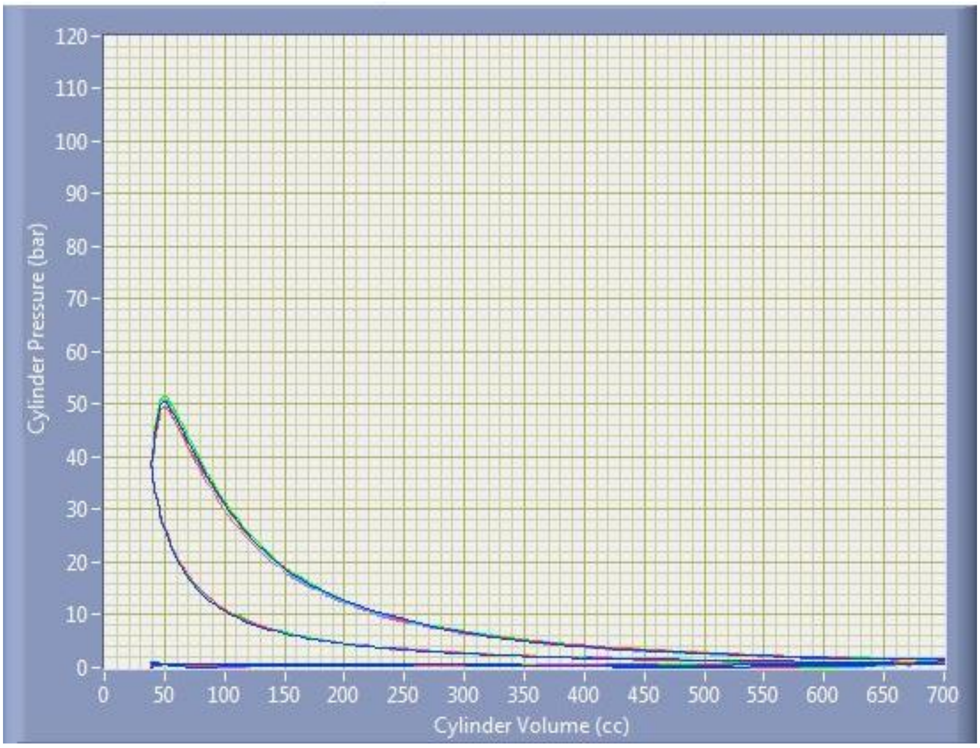


Figure 4.107 Variation of CP With Cylinder Volume at CR of 18 for Karanja Biodiesel

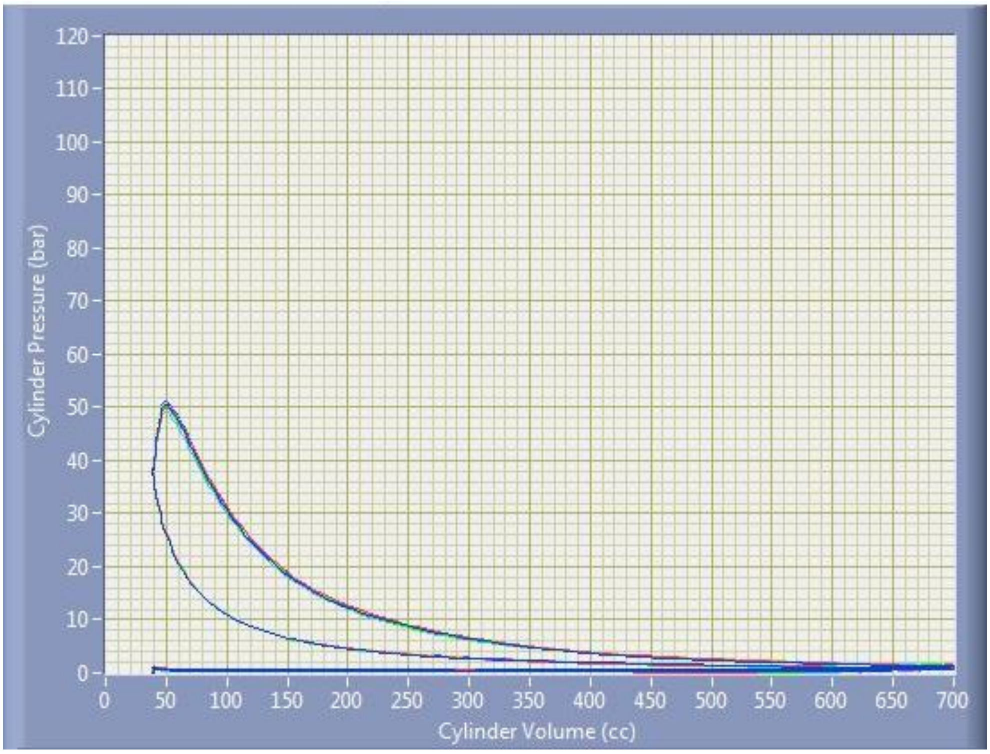


Figure 4.108 Variation of CP With Cylinder Volume at CR of 18 for Diesel Oil

4.2.4.7 Cumulative Heat Release

Variation of cumulative heat release with CA for the engine operated with Karanja biodiesel and Diesel oil at CRs of 14, 16 and 18 are given in Figures 4.109 to 4.114. Cumulative heat is the total chemical energy of the fuel released by combustion process. This is indicated by Q_{ch} in the equation.

$$(dQ_n)/dt = (dQ_{ch})/dt - (dQ_{ht})/dt = p\,dV/dt + (dU_s)/dt$$

The cumulative heat release is the sum of net heat release and heat transfer to the walls of the combustion chamber. It is also the sum of the sensible internal energy of the cylinder contents, the heat transfer rate to the walls and the rate at which the work is done on the piston (Heywood [96]).

The variation of cumulative het release with CA at a CR of 14 for Karanja biodiesel and Diesel oil are shown in Figures 4.109 and 4.110 respectively. It is observed that the cumulative heat release is about 0.02 kJ less at 40^0 before TDC for Karanja biodiesel as compared to Diesel oil.

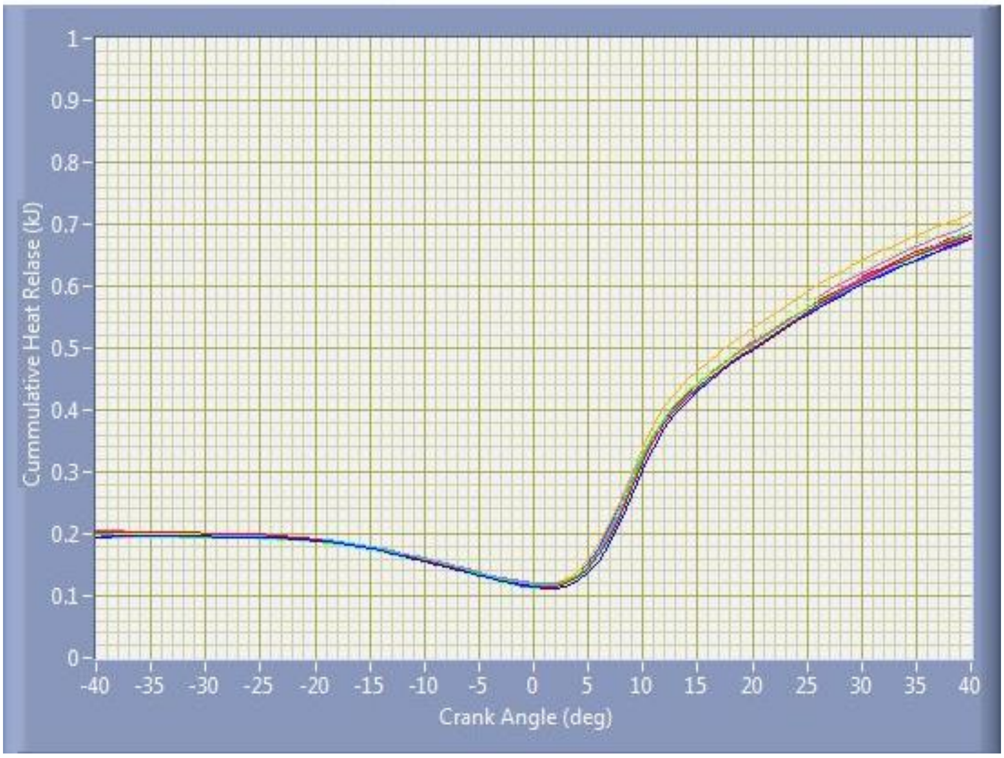


Figure 4.109 Variation of Cumulative Heat Release With CA at CR of 14 for Karanja Biodiesel

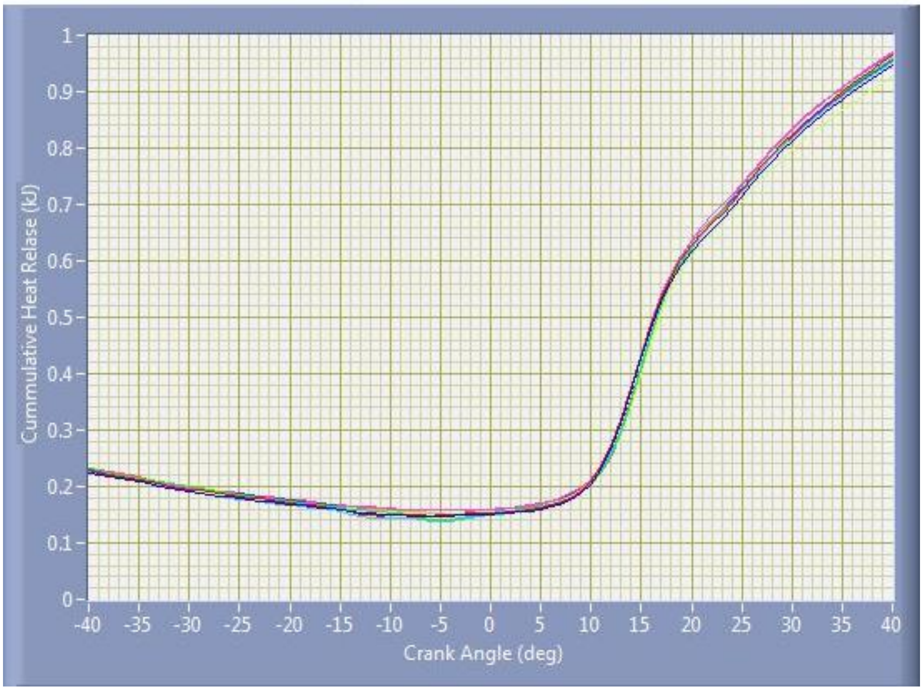


Figure 4.110 Variation of Cumulative Heat Release With CA at CR of 14 for Diesel Oil

Heat release drops to a minimum of 0.12 kJ at 1.5^0 CA after TDC for Karanja biodiesel and 0.14 kJ at 5^0 CA before TDC for Diesel oil. The drop in the curve is more noticeable for Karanja biodiesel than Diesel oil. The highest value of cumulative heat release achieved is 0.7 kJ for Karanja biodiesel and 0.97 kJ for Diesel oil which occurs at 40^0 after TDC for both fuels tested.

The variation of cumulative heat release with CA at a CR of 16 for Karanja biodiesel and Diesel oil are shown in Figures 4.111 and 4.112 respectively. The lowest value of cumulative heat release at a CR of 16 is about 0.13 kJ for Karanja biodiesel and 0.14 kJ for Diesel oil. The cumulative heat release raises to a maximum of 0.75 kJ at 40^0 CA after TDC for Karanja biodiesel whereas it is about 0.82 kJ at 40^0 CA after TDC for Diesel oil. The lower value of heat release for Karanja biodiesel may be due to its lower calorific value as compared to Diesel oil.

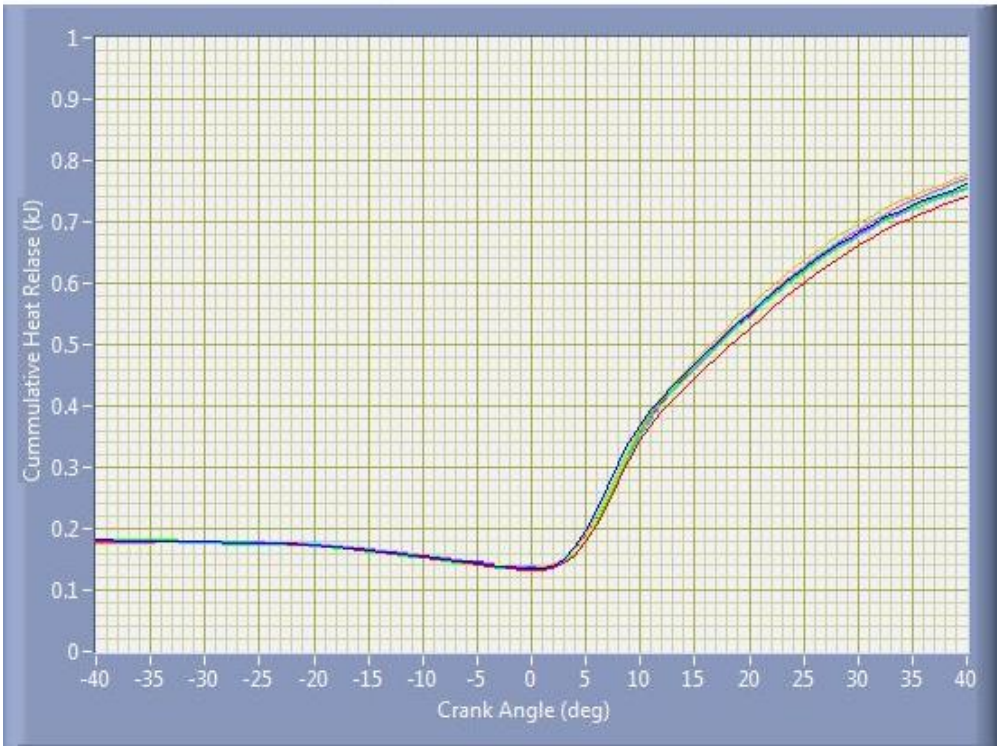


Figure 4.111 Variation of Cumulative Heat Release With CA at CR of 16 for Karanja Biodiesel

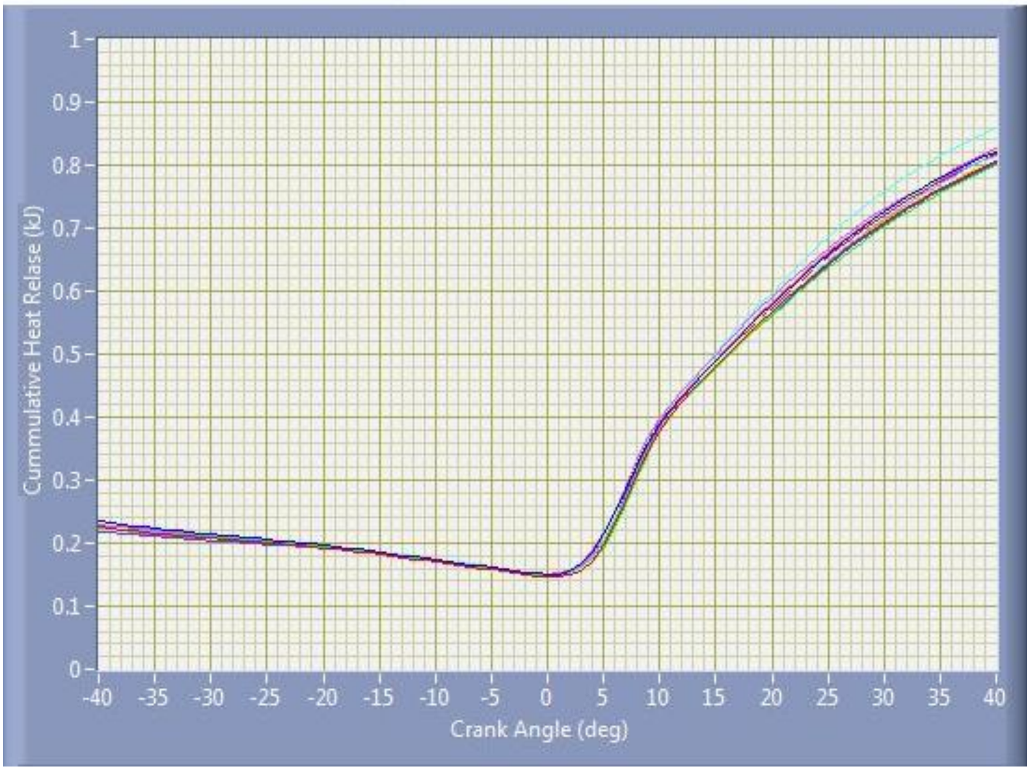


Figure 4.112 Variation of Cumulative Heat Release With CA at CR of 16 for Diesel Oil

The variation of cumulative heat release with CA at a CR of 18 for Karanja biodiesel and Diesel oil are shown in Figures 4.113 and 4.114 respectively. It can be observed that the trends for Karanja biodiesel and Diesel oil are the same with respect to cumulative heat release. The maximum value achieved is around 0.86kJ for both Karanja biodiesel and Diesel oil and it occurs at 40^0 CA after TDC. The minimum cumulative heat release value of 0.18kJ are identical for both the fuels. Hence, it can be noted that the performance of Karanja biodiesel is comparable to that of Diesel oil at higher CRs.

Table 4.9 gives the comparison of cumulative heat release of Karanja biodiesel with that of Diesel oil. It can be noted that cumulative heat release increases with increase in CR for Karanja biodiesel (22%) while an opposite trend is observed for Diesel oil (11%). Cumulative heat release is the integrated or ensemble average of the total number of cycles over the duration of combustion period.

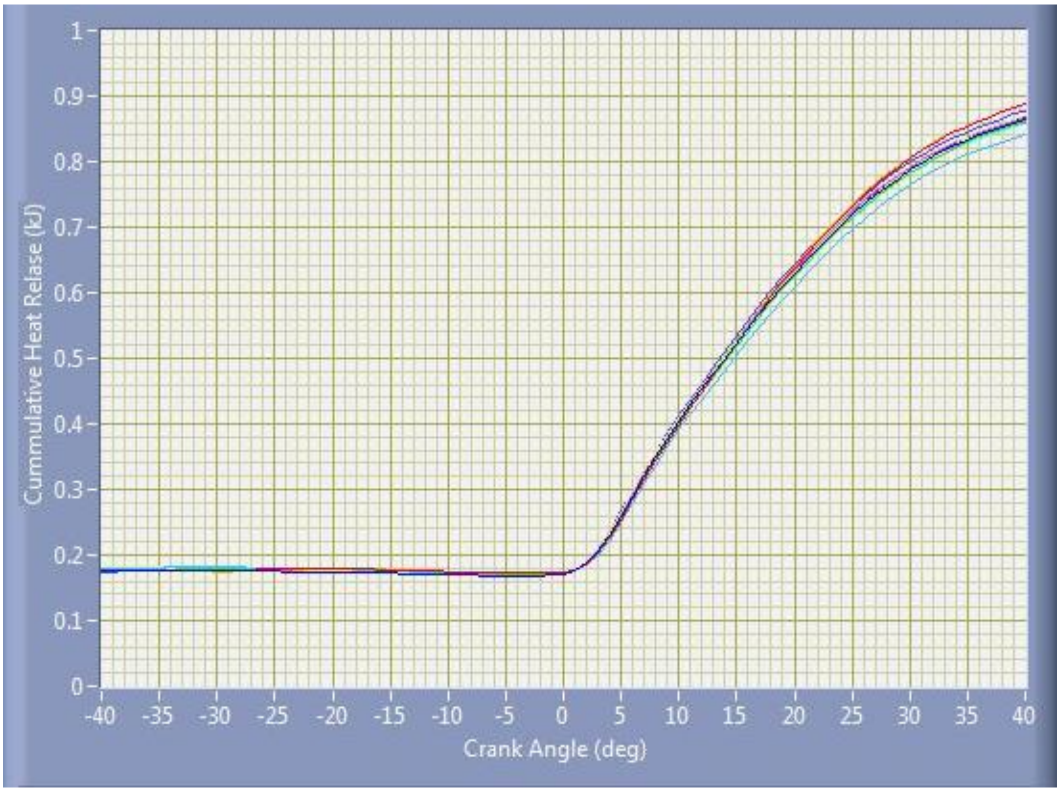


Figure 4.113 Variation of Cumulative Heat Release With CA at CR of 18 for Karanja biodiesel

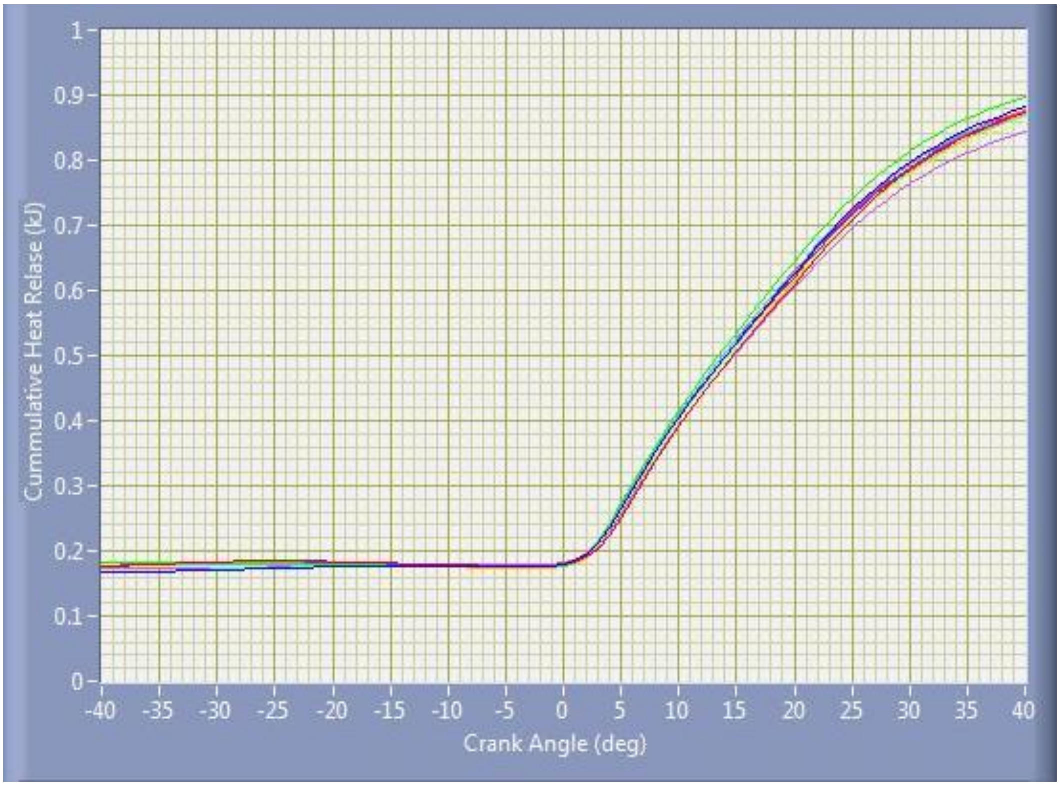


Figure 4.114 Variation of Cumulative Heat Release With CA at CR of 18 for Diesel oil

Table 4.9 Comparison of Cumulative Heat Release of Karanja Biodiesel With that of Diesel Oil (Reproduced From Figures 4.106-4.111)

CR	Fuel	Cumulative Heat Release (kJ)		Crank Angle (^{0} CA)	
		Max	Min	Max	Min
14	Karanja biodiesel	0.7	0.12	40^0 after TDC	1.5^0 after TDC
	Diesel oil	0.97	0.14	40^0 after TDC	5^0 before TDC
16	Karanja biodiesel	0.75	0.13	40^0 after TDC	TDC
	Diesel oil	0.82	0.14	40^0 after TDC	TDC
18	Karanja biodiesel	0.86	0.18	40^0 after TDC	TDC
	Diesel oil	0.86	0.18	40^0 after TDC	TDC

4.2.4.8 Injection Pressure

Variation of injection pressure and cylinder pressure data for the engine in one cycle of operation are shown superposed at full load of 12kg and CR of 14 in Figures 4.115, 4.116 and 4.117.

A typical data for cylinder pressure and fuel line pressure (IP) with crank angle is as shown in Figures 4.115, 4.116 and 4.117 for injection pressure settings of 150, 200 and 250bar at full load and CR of 14. The injection pressure rise was observed over a crank angle of 25^0CA and cylinder pressure rise over 175^0CA for one cycle of operation. The fuel line pressure plot is an indicator of the rate of fuel injection into the cylinder. The cylinder pressure varies between 0 to 40 bar and injection pressure varies from 0 to 150, 200 and 250 bar. The same trend is observed at all different values of injection pressures tried. The duration of fuel injection remains same in all cases at around 25^0CA. The value concurs with the duration of injection as mentioned in the available literature for diesel running on a CI engine (Ferguson & Kirkpatrick, Chapter Thermodynamic Analysis). The injection line pressure can be selected above or below the standard value 210 bar based on the type of fuel and its properties such as compressibility and viscosity. Generally, the injection pressure value selected for biodiesel is more than that for Diesel oil because of higher rate of fuel injection required to achieve the same pressure rise and hence the power output.. It should be noted that due to software limitation the injection pressure is indicated as diesel pressure on the ordinate of the plot indicates here (Figures 4.115, 4.116 and 4.117).

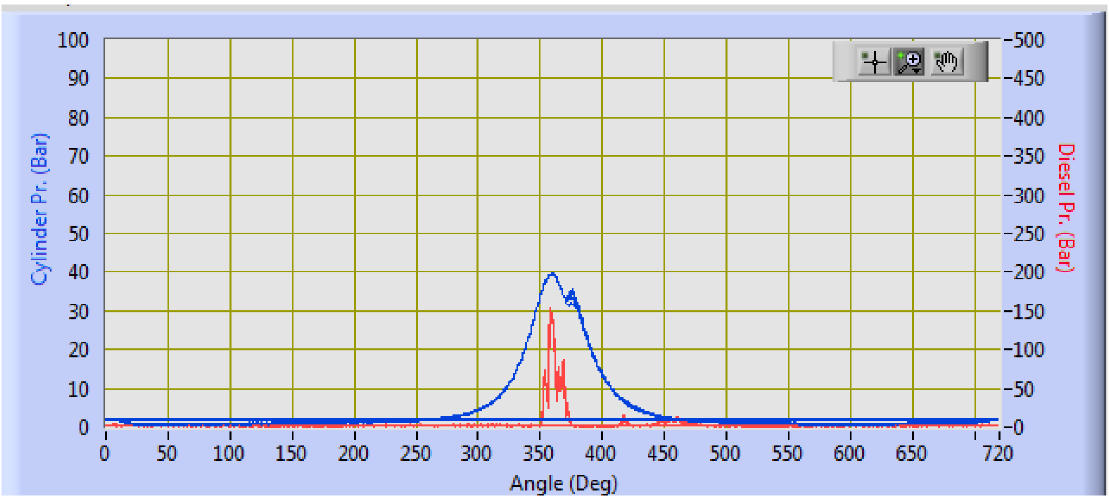


Figure 4.115 Variation of Cylinder and Injection Pressures with CA (150bar)

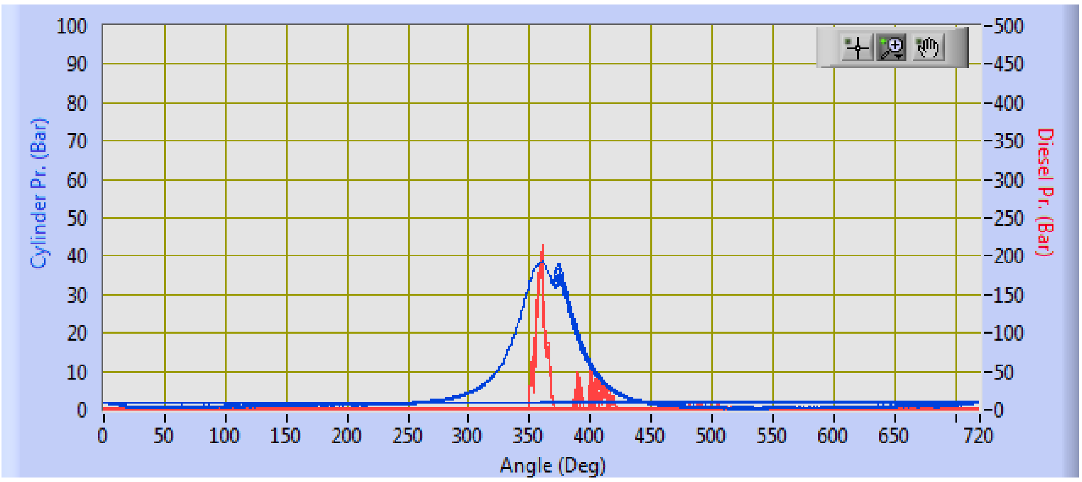


Figure 4.116 Variation of Cylinder and Injection Pressures with CA (200bar)

The cylinder peak pressure is almost same in all cases (around 40 bar) as seen from Figures 4.115 to 4.117. The attainment of the desired injection pressure takes place over crank advancement of 22-25^0 of CA in all cases. Higher IP is desirable for biodiesels as reported from thermal performance and emissions. Some experimental perturbation is noticed in the form of a small second peak cannot be substantiated.

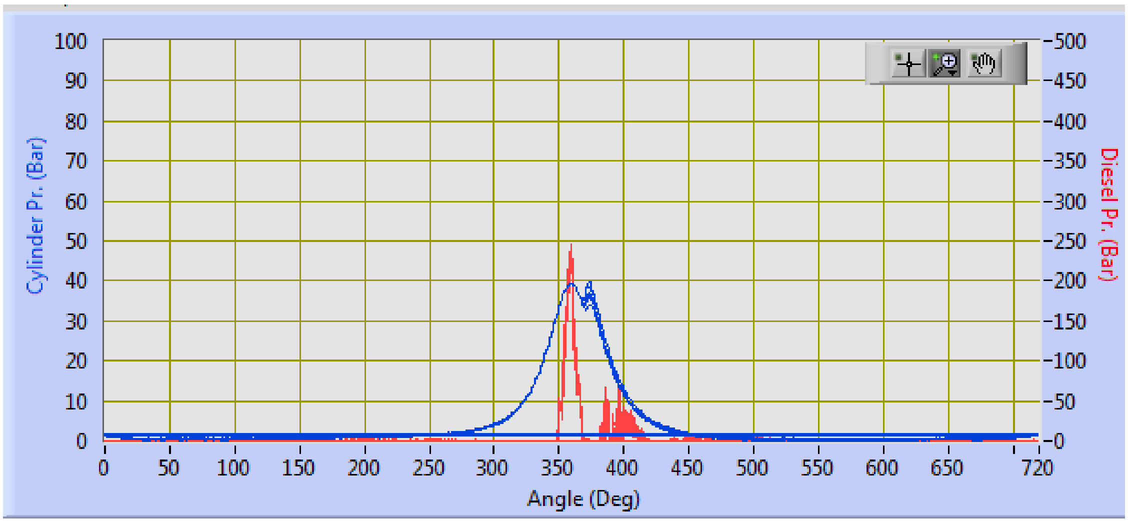


Figure 4.117 Variation of Cylinder and Injection Pressures with CA (250 bar)

4.3 Summary

In the earlier sections the thermal performance and emissions characterictics evaluated by operating a four stroke, variable compression ratio diesel engine using Karanja biodiesel and its blends with diesel are given. The experiments are performed using combinations of different preset compression ratios of 14, 15, 16, 17, 17.5 and 18, different injection pressures of 150 bar, 200 bar and 250 bar, and varying loads from 0kg to 12 kg in steps of 3kg. The thermal performance parameters evaluated are BTHE, BSFC, BMEP, Volumetric Efficiency, HBP and HGas, EGT. The emission constituents measured are CO, HC, NO_x, CO_2, O_2 and SO_x. Along with emission constituents, the smoke intensity is also measured in terms of HSU (%) & K (m^{-1}). A detailed combustion analysis of the engine running on pure diesel and Karanja biodiesel at different preset compression ratios of 14, 16 and 18 and full load is also conducted. The combustion parameters considered for analysis are cylinder pressure, net heat release, rate of pressure rise, mass fraction burnt, mean gas temperature, cylinder pressure-volume (PV) plot and cumulative heat release rate (follow same order). Prior to carrying out the exhaustive experimental study, a validation test is conducted on the engine using Diesel oil alone as fuel to check/ascertain the readiness of the experimental setup at standard rated conditions of 12kg, IP of 200 bar, CR of 17.5. The data from the validation test are also used as a base line for further comparison of the results with Karanja biodiesel & its blends.

4.3.1 Thermal Performance

Even though the rated CR of the engine used for experimentation is 17.5, the variation of thermal performance parameters is studied at different preset CRs maintaining a constant load of 12kg and IP of 200bar in order to determine the optimum CR for Karanja biodiesel and its blends. It is observed that with the increase in CR the performance with Karanja biodiesel and its blends approach to that of Diesel oil. At a CR of 18, BTHE, BSFC and EGT for Karanja biodiesel are about 8% less, 18% more and 2% less respectively as compared to Diesel oil which indicates that Karanja biodiesel performs close to Diesel oil at CR of 18. Due to this reason CR of 18 is selected for comparison of variation of other thermal performance parameters with load and IP for further analysis.

The difference of performance of blends from Diesel oil increases with increase in blend proportion. It is observed that the variation in CR and blend proportion does not have an effect on volumetric efficiency and BMEP of the engine. Volumetric efficiency is marginally (2%) higher for Karanja biodiesel as compared to Diesel oil. However HBP increases with increase in CR and decrease in blend proportion. It is noted that the increase in HBP with CR is (linear) continuous for Karanja biodiesel while it increases upto CR of 16 then remains almost constant upto CR of 17 and subsequently decreases for other blends. Therefore, it can be inferred that combustion is better at CR of 16 for the blends and at CR of 18 for Karanja biodiesel. The change in CR does not have much effect on the HGas of the engine. However, at most of the CRs it is less for Karanja biodiesel than Diesel oil.

The thermal performance evaluation indicates that the blend B20 operates closest to Diesel oil with respect to thermal performance. However, there is not much deviation between the performance of Karanja biodiesel and Diesel oil at higher CRs. Hence higher CRs particularly CR of 18 should be the mode of operation when engine is fuelled with Karanja biodiesel. Also, higher IP of 250bar is preferable for Karanja biodiesel due to its higher viscosity.

As load on engine is increased from 0kg to 12kg, BMEP, BTHE, HBP, EGT are found to increase while BSFC decreases. The variation in load did not have any effect on HGas and Volumetric efficiency of the engine. Higher volumetric efficiency is desirable since it is an indication of greater fuel economy for engine and higher volumetric efficiency is noted with both fuels (more than 90%). As the injection pressure is increased, BMEP, volumetric efficiency and HGas remained unaffected, whereas, BSFC decreases. The BTHE, EGT and HBP are found to be higher at higher injection pressure. It is observed that for Karanja biodiesel as fuel, load of 12kg and IP of 250 bar result in highest BTHE of 31% and lowest BSFC of 0.33 kg/kWh which are 6% lower and 17% higher as compared to that of Diesel oil respectively at same operating conditions.

The variation of BTHE and BSFC with CR of the present study are compared with that found in the literature. There are no reported studies available on variation of BTHE and BSFC with CR except Jindal et al. [66] and Raheman & Ghadge [40]. It should be noted that Jatropha and Mahua biodiesels are using by the investigators while Karanja biodiesel is used in the present study. An attempt also is made to compare the results reported by Banapurmath et al. [44] and Raheman and Phadatare [17] who conducted studies at constant CRs of 17.5 and 16 respectively. Banapurmath et al. [44] has reported a 10% higher value of BTHE at a constant CR of 17.5 while 14% lesser BTHE has been reported by Raheman and Phadatare [17] at a constant CR of 16 using Karanja biodiesel as compared to the present study. Further, comparison is made for BTHE at a constant CR of 18 for different fuels i.e., Jatropha and Mahua biodiesels are lesser by 12% for Jatropha biodiesel (Jindal et al. [66]) and 17% for Mahua biodiesel (Raheman & Ghadge [40]) with that of Karanja biodiesel used in the present study. Similar comparison is also made for BSFC at constant CR. About 3% higher value of BSFC at a constant CR of 16.5 is reported by Sureshkumar et al. [52] while 21% higher BSFC has been reported by Raheman and Phadatare [17] at a constant CR of 16 using Karanja biodiesel as compared to the present study. It is seen that the value of BSFC at 18 CR is 9% more for Jatropha biodiesel (Jindal et al. [66]) and 56% more for Mahua biodiesel (Raheman & Ghadge [40]) as compared to that of Karanja biodiesel used in the present study.

4.3.2 Emission Constituents

At higher CRs of 16 to 18, it is found that the fuel gets combusted in a much efficient way due to high temperature of compressed air. Therefore, most of the exhaust emissions are found to decrease at higher CRs. Also at higher CRs, smoke is less due to complete combustion of fuel. The engine may be operated with any higher CR ranging from 16 to 18 as far as the emission constituents of CO, HC, CO_2, O_2 and SO_x are concerned. The reason is that the mentioned emission constituents are least and remain constant in the CR range of 16 to 18. However, the NO_x emissions are found to be more at higher CRs with Karanja biodiesel as compared to Diesel oil. There is a trade off between NO_x and smoke emission as NO_x emissions are found to decrease with increase in smoke as lesser flame temperature decreases the amount of soot oxidised with the result higher smoke & relatively lesser NO_x levels. Therefore, the selection of CR can be made based on the relative combined effect on thermal performance and emission characteristics. It is preferable to operate the engine at CR of 18, as selected earlier for optimum thermal performance evaluation. If NO_x is considered then we have to operate at CR 16 but it is found that the decrease in BTHE of about 13% and increase of BSFC of about 13% are not affordable just for the sake of NOx emissions.

Higher oxygen content of Karanja biodiesel influences the exhaust emissions considerably. At a CR of 18, load of 12kg and IP of 200bar, it is observed that CO_2 emissions are more by 24% for Karanja biodiesel than that of diesel. It may be noted that due to higher oxygen content in the biodiesel more of the carbon gets oxygenated during combustion inside the cylinder which results in higher CO_2 emission. Higher CO_2 emissions lead to lower CO emissions as more of carbon is already oxygenated. CO emission is less by 67% for Karanja biodiesel as compared to that of Diesel oil. Also it can be noticed that the NO_x emissions are higher by 33% for Karanja biodiesel as compared to that of Diesel oil. The percentage reduction in HC emissions for Karanja biodiesel is 60% as compared to that of Diesel oil. The content of O_2 with exhaust increased with increase in blend proportion as it is an oxygenated fuel. Karanja biodiesel is sulphur free fuel and hence, fine traces of SO_x found in the exhaust may be from atmospheric air. In the context of NO_x emissions selection of suitable blends instead of pure Karanja biodiesel can be thought of, striking a balance between NO_x emissions on one end and all other emissions along with thermal performance on the other hand at the same point of time.

Among all the emission constituents measured, CO, CO_2, NO_x and HC are found to increase with increase in load, whereas O_2 and SO_x indicated a decrease. At higher loads, more fuel is burnt due to which more carbon is available from fuel to form CO_2 and CO. The unburnt HC are formed due to rich burning. At higher loads, the content of O_2 in the exhaust is less as a richer mixture is burnt inside the cylinder. At higher loads more fuel is burnt in rich zone and hence higher smoke level is observed. Smoke consists of solid carbon, organic fraction consisting of hydrocarbons and their partial oxidation products.

As the IP is increased, CO_2, O_2 and NO_x increase while, HC, CO and smoke opacity are decreased. A higher IP causes better atomisation which results in exposure of more surface area of fuel droplets to hot compressed air causing complete combustion of fuel as a result of which most of the harmful exhaust emissions reduce.

The emission characteristics evaluation indicates that the Karanja biodiesel (B100) gives minimum harmful emissions as compared to all other blends. Further, at a higher CR of 18 and IP of 250bar the fairly reduced exhaust emissions are observed irrespective of the fuel. Therefore, operating the diesel engine with Karanja biodiesel at a CR of 18 and IP of 250bar results in minimum emissions but for NO_x emissions.

The results of NO_x and smoke opacity (HSU) of the present study are compared with that found in the literature. Sureshkumar et al. [52] reported a 15% higher NO_x at a CR of 16.5 while 95% less NO_x at a CR of 16 is reported by Raheman & Phadatare [17] using Karanja biodiesel as compared to that of the present study. It is seen that NO_x emission at CR of 18 are higher by 32% for Jatropha biodiesel (Jindal et al. [66]) as compared to that of Karanja biodiesel used in the present study. Raheman & Phadatare [17] reported a 71% reduction in HSU at a constant CR of 16 respectively using Karanja biodiesel as compared to that of the present study. It is seen that HSU at a CR of 18 is almost double for Jatropha biodiesel (Jindal et al. [66]) as compared to that of Karanja biodiesel used in the present study.

4.3.3 Combustion Analysis

The peak CP development for Karanja biodiesel takes place at an earlier CA at 14 CR as compared to Diesel oil. However, the peak pressures at all CRs using both the fuels always takes place after TDC which ensures safe and efficient operation as peak pressure at or before TDC causes knocking which affects engine durability (Ganesan [97]). The peak CP attained for Karanja biodiesel is about 2 and 4 bar less at CR of 14 and 16 respectively, whereas at CR of 18 the peak CP is same for both the fuels. Peak cylinder pressure is an indicator of the intense premixed combustion. From the cylinder pressure data the location of peak pressure, the instantaneous heat release & the burn fraction can be identified. It has no direct bearing on the performance of the engine. Generally, with low cetane number fuels such as Diesel oil the peak CP is highest. It is the mean effective pressure which decides the performance parameters like the thermal efficiency and BSFC of the engine.

The double peak shape of the heat release rate curve which is the characteristic of Diesel oil combustion is observed only for Diesel oil at CR of 14. It is observed that the first peak of heat release curve occurs early in case of Karanja biodiesel than Diesel oil. The point of occurrence of first peak net heat release shifts closer to TDC with increase in CR for both the fuels. At 14 CR no significant second peak observed for Karanja biodiesel. At CR of 16 the second peak heat release for Karanja biodiesel is 10% less as compared to Diesel oil. As CR increases from 16 to18, the second peak net heat release rate increases for Karanja biodiesel and it is same as that of Diesel oil at 18CR. The trend observed corroborates the closeness of thermal performance of Karanja biodiesel as compared to Diesel.

At CR of 14, the maximum rate of pressure rise takes place closer to TDC in case of Karanja biodiesel as fuel than Diesel oil. With the increase in CR from 14 to 16 the point of occurrence of peak rate of pressure rise shifts by about 2^0 towards TDC for both the fuels. The peak rate of pressure rise at CR of 14 and 16 is about 2.5bar /^{0}CA for both the fuels. However, at CR of 18 it occurs at same CA and has same magnitude of 2 bar/^{0}CA for both the fuels.

The behaviour of Karanja biodiesel in terms of mass fraction burnt is similar to that of Diesel oil at CRs of 16 and 18. At CR of 14, however, there is an earlier combustion by about 4^0 to 5^0 CA after TDC with Karanja biodiesel as compared to Diesel oil. It is noted that the mean gas temperature (average of burned and unburned gas temperatures in cylinder) is highest when more than 90% of the fuel mass fraction is burnt for both fuels. It is obvious as mean gas temperature is the result of thermal energy released during combustion of fuel. So more fuel burnt results in higher mean gas temperature. The mean gas temperature increases with increase in CR for Karanja biodiesel and its behaviour is identical compared to Diesel qualitatively at higher CR of 18. The analysis of PV characteristics and cumulative heat release also indicate that Karanja performs identical to Diesel oil at a CR of 18.

It is inferred from the combustion analysis that as the CR increases from 14 to 18 at full load and IP of 200bar, the difference in performance between Karanja biodiesel and diesel reduces i.e. the operation of the engine fuelled with Karanja biodiesel tends towards that fuelled with diesel. The combustion characteristics of diesel engine using Karanja biodiesel is similar to that using pure diesel at higher CR of 18 which is very promising as far as Karanja biodiesel as an alternative fuel in diesel engines is concerned.

From the thermal performance, emission characteristics and combustion analysis, it is observed that Karanja biodiesel operates closer to Diesel oil at a CR of 18. Proper combustion ensures reduced emission levels and optimum thermal performance. Although B20 blend gives better thermal performance compared higher blends of Karanja, it is not recommended as it poses problems of higher levels of exhaust emissions. It is observed that harmful emissions such as CO, HC are 81% and 75% higher for B20 blend as compared that of Karanja biodiesel. Hence Karanja biodiesel can be successfully used without any hardware modifications on the engine.

4.3.4 Limitations of the Experimental Study

The experimental study conducted provides large number of results. But, it is impossible to select the input parameters such as CR, IP and blend for obtaining optimum thermal performance and emissions from the engine. Hence, a suitable computational study is required to be carried out in order to meet the objective of finding the optimum combination of the input parameters under different preset priorities. The preset priorities is a combination of weightages in proportions of performance and emissions like 50-50, 80-20, 20-80, etc. A balance can be struck as to what could be the suitable combination of optimum input parameters which will result in maximising the thermal performance while minimising the emission constituents.

Multi-objective optimization using genetic algorithm technique is one of the suitable tools available that can be used to optimize the thermal performance and exhaust emissions. Genetic algorithms are computerized search and optimization algorithms based on the mechanics of natural genetics and natural selection. GAs are radically different from most of the traditional optimization methods. GAs work with a string-coding of variables instead of the variables. The advantage of working with the coding of variables is that the coding discretizes the search space, even though the function may be continuous. On the other hand, since GAs require only function values at various discrete points, a discrete or discontinuous function can be handled with no extra cost. This allows GAs to be applied to a wide variety of problems. Another advantage is that the GA operators exploit the similarities in string-structures to make an effective search.

Further, ANN can be used to obtain the output parameters using the optimized input parameters by modeling. ANN is a theoretical simulation of actual engine behaviour. ANNs don't rely upon a pre-defined mathematical equation to relate system input/output. A proper ANN structure is developed for each system to capture the system behavior of a complex system. With high complexity of combustion relations and emission phenomena it is suitable to model by ANN. ANNs have been used for two main tasks: 1) function approximation and 2) classification problems. Neural networks offer a general framework for representing non-linear mappings. The application of neural networks to predict thermal performance and exhaust gas constituents belongs to the class of function approximation applications. Suitable software is to be used for modeling an ANN model. The software that can be selected for neural network modeling is EasyNN purely because of its simplicity in developing and training models involving feed forward multilayered neural networks with back-propagation training algorithm.

Chapter 5

Computational Study

The computational study conducted consists of optimization of thermal performance and emissions constituents and ANN modeling of the variable compression ratio diesel engine. The optimization conducted using genetic algorithm technique is explained in Section 5.1 and ANN in Section 5.2.

Data obtained from experiments needs to be treated in a number of different ways to get meaningful insight into the system being studied. Numerous modeling techniques and multiple models may be developed for engines system. It is important to select a suitable modeling technique to capture the relationship between input and output of the system accurately and efficiently. For systems involving mutually conflicting out comes effected by a number of input variables, it is essential to determine the optimum state of the system to achieve desired output by setting appropriate levels of inputs. For this purpose, use of suitable optimization technique is essential.

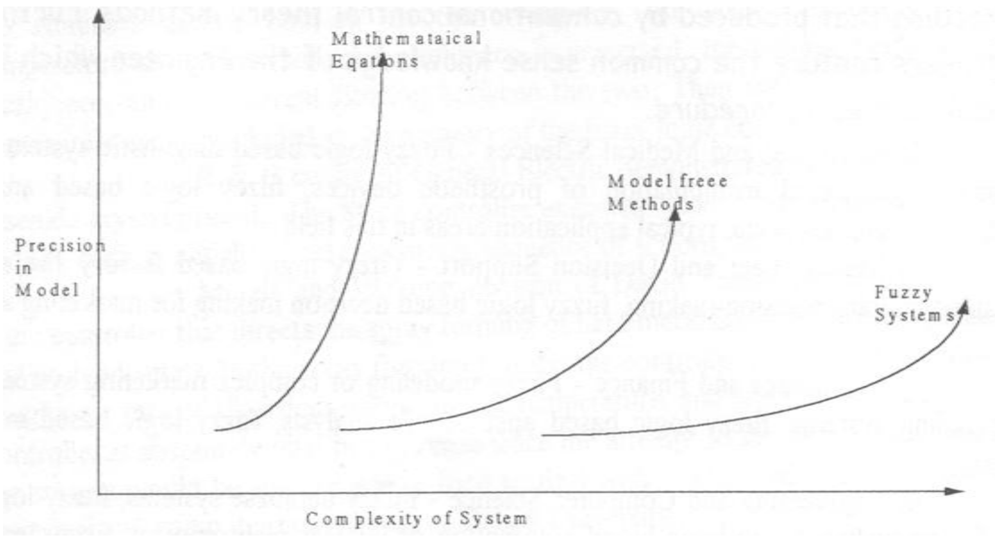


Figure 5.1 Complexity of System & Precision Level of Different Models

With increase in knowledge about a system or process, its complexity decreases and understanding increases. Decrease in complexity leads to increase in precision afforded by computational methods useful in modeling of the system or process. As seen in Figure 5.1, for systems that are little complex and hence little uncertain, closed form of mathematical models provide precise description of system. For systems little more complex but for which significant data is available, model free methods like ANNs provide powerful and robust means to reduce uncertainty using pattern based learning. For most complex systems where little numerical data exists and where only ambiguous or imprecise information may be available, fuzzy models provide a method to understand and represent system behaviour by interpolation between observed inputs and outputs.

In the recent years, many models and simulations have been tried to give a clear view about the diesel engine performance, fuel characteristics, emission etc. under varied conditions of speed, load and other operating parameters. One of these techniques is the ANN modeling which encompasses very sophisticated techniques capable of modeling complex functions and processes. The true power and advantage of neural networks lies in their ability to represent both linear and nonlinear relationships as well as having the capability of learning by example. For processes that have non-linear characteristics such as those found in diesel engine performance modeling, traditional linear models are simply inadequate. In comparison to traditional computing methods, neural networks offer a different way to analyze data and to recognize patterns within that data by being generic non-linear approximators. Artificial Intelligence (AI) techniques seem to be best solution for predicting engine emissions since they do not demand any additional sensor installation.

When mathematical models fail to capture the input/output relationship within the limits of permissible error and sufficient data regarding the system available, ANN is a pertinent tool to model successfully the system behaviour. ANNs are called model free models since they don’t rely upon a pre-defined mathematical equation to relate system input/output. A proper ANN structure is developed for each system to capture the system behavior of a complex system. With high complexity of combustion relations and emission phenomena it is suitable to model by ANN. ANNs have been used for two main tasks: 1) function approximation and 2) classification problems. Neural networks offer a general framework for representing non-linear mappings. The application of neural networks to predict thermal performance and exhaust gas constituents belongs to the class of function approximation applications.

In order to use optimization algorithms in engineering design activities, the first task is to formulate the optimization problem. The formulation process begins with identifying the important design variables that can be changed in a design. The other design parameters are usually kept fixed. Thereafter, constraints associated with the design are formulated. The constraints may arise due to resource limitations such as deflection limitations, strength limitations, frequency limitations, and others. Constraints may also arise due to codal restrictions that govern the design. The next task is to formulate the objective function which the designer is interested in minimizing or maximizing. The final task of the formulation phase is to identify some bounding limits for the design variables.

The formulation of an optimization problem can be more difficult than solving the optimization problem. Every optimization problem requires different considerations for formulating objectives, constraints and variable bounds.

5.1 Multi-objective Optimization

Optimization is the task of finding one or more solutions which correspond to minimizing (or maximizing) one or more specified objectives and which satisfy all constraints (if any). A single-objective optimization problem involves a single objective function and usually results in a single solution, called an optimal solution. On the other hand, a multi-objective optimization task considers several conflicting objectives simultaneously. In such a case, there is usually no single optimal solution, but a set of alternatives with different trade-offs. Despite the existence of multiple optimal solutions, in practice, usually only one of these solutions is to be chosen. Thus, compared to single objective optimization problems, in multi-objective optimization, there are at least two equally important tasks: an optimization task for finding an optimal solution (involving a computer based procedure) and a decision making task for choosing a single most preferred solution. The latter typically necessitates preference information from a decision maker.

IC engines are the lightest power generating units known and therefore are of greatest applications in transportation. An engine is expected to give highest possible efficiency with least possible emissions in order to meet the environmental standards. Generally, it is observed that engine gives maximum efficiency and also maximum emissions at high loads. This creates a scenario wherein, to get maximum efficiency we must compromise on emissions or to get minimum emissions we must compromise on efficiency. This is a case of Optimization to strike an optimal combination of input parameters to achieve maximization of thermal performance and minimization of emission constituents.

The results obtained in the experimental study conducted at different preset compression ratios, different loads and different injection pressures show that as thermal efficiency of the engine increases, the harmful emissions constituents also increase. Hence, it is difficult to get the accurate minimum possible values of harmful emissions and maximum possible value of thermal efficiency. Hence an optimization is performed to obtain optimum values of load, compression ratio, injection pressure and blend which give minimum possible emissions and maximum possible thermal efficiency i.e. optimise thermal performance and emission constituents.

There are several algorithms which can be used to solve an optimization problem. The usage of these algorithms depends on the type of the problem, environment and the results required in real life situation. For engine applications, use of genetic algorithm (Refer Appendix VIII) is considered one of the best ways to solve the problem because of its inherent uniqueness. The optimization using genetic algorithm toolbox in MATLAB software is carried out to optimize the chosen engine input parameters to achieve specified priorities.

The genetic algorithm toolbox makes use of the correlations obtained by performing nonlinear regression analysis on the experimental data for all Karanja biodiesel and diesel blends at all loads, compression ratios and injection pressures. DATAFIT software which is developed by Oakdale engineering, USA is used to obtain the correlations.

The correlations evaluated are,

Square: $Y = A(x_1) + B(x_2) + C(x_3) + D(x_1)^2 + E(x_2)^2 + F(x_3)^2 + G(x_1)(x_2) + H(x_2)(x_3) + I(x_3)(x_1)$

Cubic: $Y = A(x_1) + B(x_2) + C(x_3) + D(x_4) + E(x_1)^2 + F(x_2)^2 + G(x_3)^2 + H(x_4)^2 + I(x_1)(x_2) + J(x_2)(x_3) + K(x_3)(x_4) + L(x_3)(x_4) + M(x_1)^3 + N(x_2)^3 + O(x_3)^3 + P(x_4)^3$

Exponential: $Y = e^{[A(X_1) + B(X_2) + C(X_3) + D]}$

Linear with Constant: $Y = A(x_1) + B(x_2) + C(x_3) + D$

The optimization process is carried out after fixing the upper and lower bounds for constraints, defining the fitness function and number of variables. The constraints are load, blend, compression ratio and injection pressure. The optimization is carried out considering full load condition and hence, both upper and lower bounds for loads are 12kg. The upper and lower bounds for the constraints are given in Table 5.1. The correlations obtained by nonlinear regression analysis of the experimental data are used as fitness functions and the number of variables are given as '4' viz. load, compression ratio, injection pressure, blend proportionand are varied during experimentation.

Table 5.1 Upper and Lower Bounds for Constraints

Constraints	Upper bound	Lower bound
Load (kg)	12	12
Compression Ratio	14	18
Injection Pressure (bar)	150	250
Blend	20	100

The Optimization is conducted considering three cases, namely

1. Thermal performance
2. Emission constituents
3. Both thermal performance and emission constituents together with equal weightages to each

The optimization of thermal performance, emission constituents and both thermal performance & emission constituents together giving equal weightages to each are, respectively, explained in sections 5.1.1, 5.1.2 and 5.1.3

5.1.1 Thermal Performance

The optimization of thermal performance is conducted by considering three important parameters viz. BTHE, BSFC and EGT. BTHE is always preferred to be maximum, BSFC and EGT to be minimum. Hence, this creates a multimodal scenario of optimization.

The solution procedure for optimization of thermal performance is as follows:

1. Three types of mathematical equations viz. Polynomials, exponentials and power functions are developed from the captured experimental data (Refer Appendix V) to form a relation between the input (Load, CR, IP, Blend) and output (BTHE, BSFC and EGT) parameters using the DATAFIT software which is developed by Oakdale engineering, USA.
2. The mathematical equation with best fit is selected based on best value of coefficient of determination (R^2) for all parameters. Cubic polynomial is found to be the best fit on both criterion and for all output parameters. Table 5.2 shows the cubic polynomials for thermal performance parameters, with their respective R^2 values.

Table 5.2 Polynomials for Thermal Performance Parameters with their Respective R^2 Values

Parameter	Polynomial	R^2
BTHE	BTHE = -541317.7 -3.34E-02 (x_1) - 0.413 (x_2)+ 8480.6 (x_3) - 5.03E-05 (x_4) + 2.616E-03 $(x_1)^2$ + 2.45E-02 $(x_2)^2$ - 43.3 $(x_3)^2$+ 2.27E-06 $(x_4)^2$ + 3.39E-04 $(x_1)(x_2)$ + 4.14E-06 $(x_2)(x_3)$ - 2.26E-02 $(x_3)(x_4)$+ 2.26E-02 $(x_4)(x_1)$ -8.46E-05 $(x_1)^3$ - 4.89E-04 $(x_2)^3$+ 7.21E-02 $(x_3)^3$ -1.026E-08 $(x_4)^3$	0.9408
BSFC	BSFC = -541296.66 -0.29(x_1) -3.99(x_2) + 8480.68(x_3) -8.84E-04(x_4) + 0.02$(x_1)^2$ + 0.23$(x_2)^2$ -43.3$(x_3)^2$ + 2.77E-05$(x_4)^2$ + 3.02E-03$(x_1)(x_2)$ -3.4E-08$(x_2)(x_3)$+ 0.03$(x_3)(x_4)$ -3.13E-02$(x_3)(x_4)$ -7.51E-04$(x_1)^3$ -4.71E-03$(x_2)^3$+ 7.21E-02$(x_3)^3$ -1.44E-07$(x_4)^3$	0.9245
EGT	EGT= -531591.25+ 13.7(x_1) -1831.77(x_2) + 8487.49(x_3) + 0.24(x_4) -6.27E-03$(x_1)^2$ + 111.28$(x_2)^2$ -43.33$(x_3)^2$ -7.98E-03$(x_4)^2$ -0.16$(x_1)(x_2)$ -2.43E-06$(x_2)(x_3)$ -5.34E-03$(x_3)(x_4)$+ 5.34E-03$(x_4)(x_1)$ + 3.29E-02$(x_1)^3$ -2.25$(x_2)^3$ + 7.22E-02$(x_3)^3$+ 6.45E-05$(x_4)^3$	0.9723
x_1= Load, x_2= CR, x_3= IP, x_4= Blend		

3. The equations thus obtained are defined as a function in MATLAB and saved as a dot m (.m) file. This function is called as the fitness function for optimization in MATLAB. The GENETIC ALGORITHM tool box of MATLAB is used for defining and solving the problem. Figure 5.2 shows optimization problem definition screen.
4. Record the value of the optimum and the result after a few repeated trials to eliminate the effects of specific initialisation.

The problem definition screen is shown in Figure 5.2, while Figure 5.3 shows display screen of MATLAB program after optimization of thermal performance is carried out. The problem definition screen is where the solver, fitness function and bounds are defined. The optimization is carried out considering full load condition and hence, both upper and lower bounds for loads are 12kg. The set bounds for the constraints are given in Table 5.1. The same are entered in the order of 'Load, CR, IP, Blend' in the software. The options column on the left hand side is used to select pareto front display. The optimization is carried out by clicking the START button. After the optimization is done, number of iterations is displayed in the current iteration box.The the function values are also displayed suitably.

Figure 5.4 shows pareto front for optimization of thermal performance parameters to find maximum BTHE with minimum BSFC and EGT for the diesel engine operating on different Karanja biodiesel and diesel blends at different CRs, IPs and full load. Pareto front shows the trade-off between the two objectives. The objectives are the functions which the software forms inherently based on the maximising and minimising parameters given. Objective 1 is minimisation of BSFC and EGT and objective 2 is the maximisation of BTHE. The pareto front is plotted in objective function space. It gives a set of points corresponding to different values of objective 1 and objective 2. The software uses pareto front to strike a balance between objective 1 and objective 2 and picks up a point to suggest the optimum values of input parameters such as CR, IP and blend. The points appear to lie in the range -3472.3 to -3472.25 and -3800 to -3850 for objective 1 and objective 2 respectively. The values of objective 1 and objective 2 are used to determine the optimum input parameters by the software by following a set of complex calculations. The optimum values of CR, IP and blend suggested by the software after carrying the optimization run are 17, 228 bar, B70 respectively.

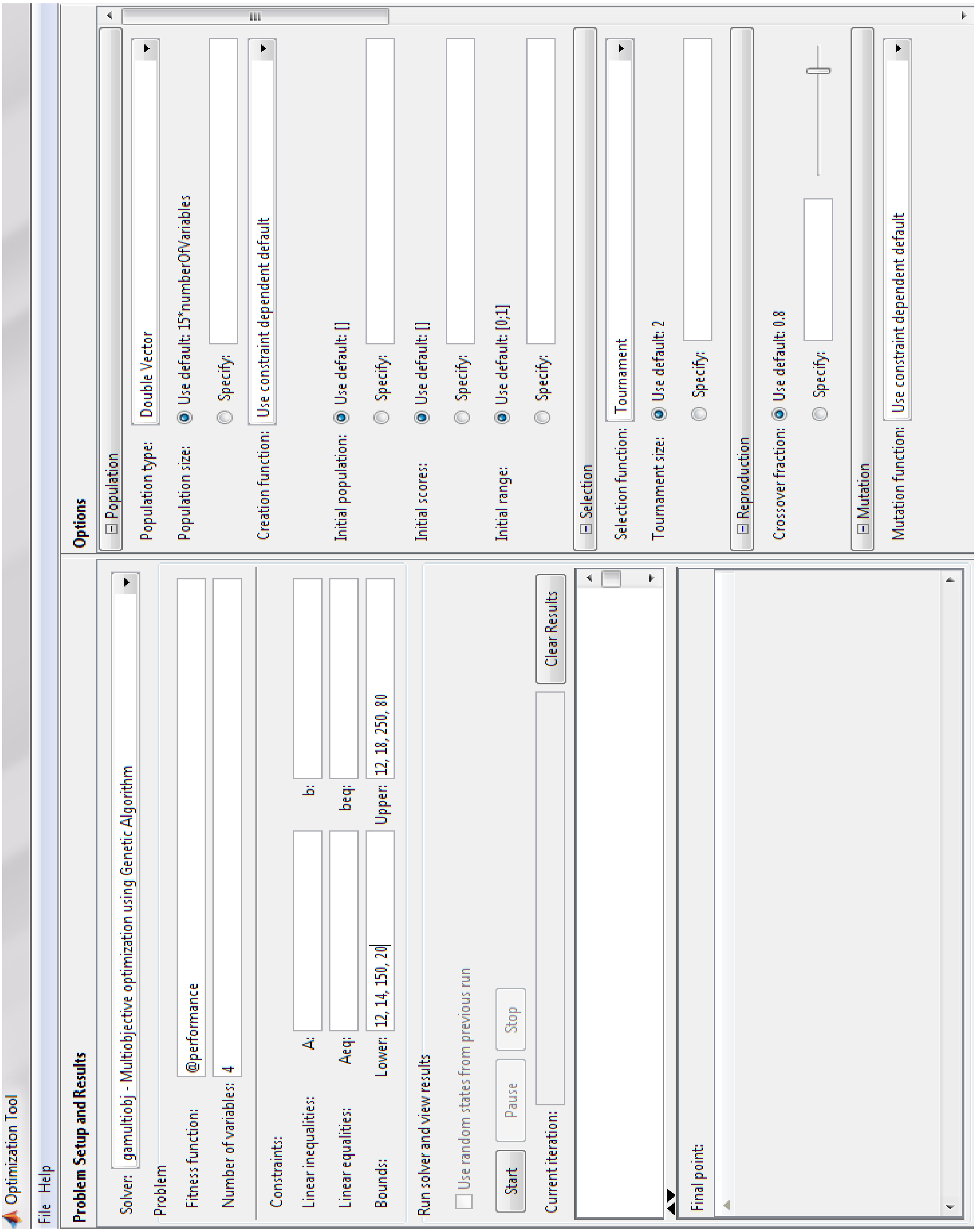


Figure 5.2 Problem Definition Screen

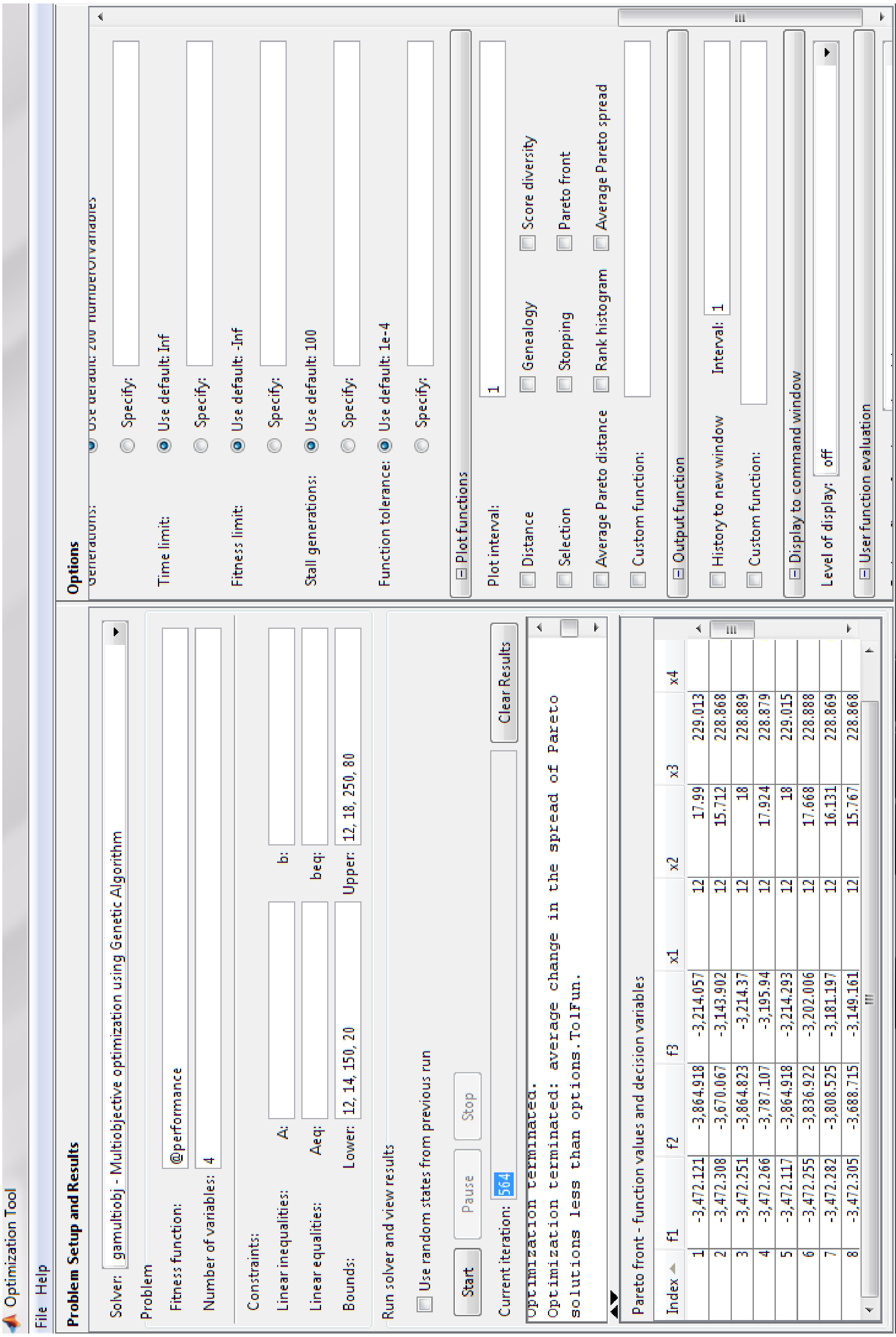


Figure 5.3 Display Screen of MATLAB Program for Optimization of Thermal Performance

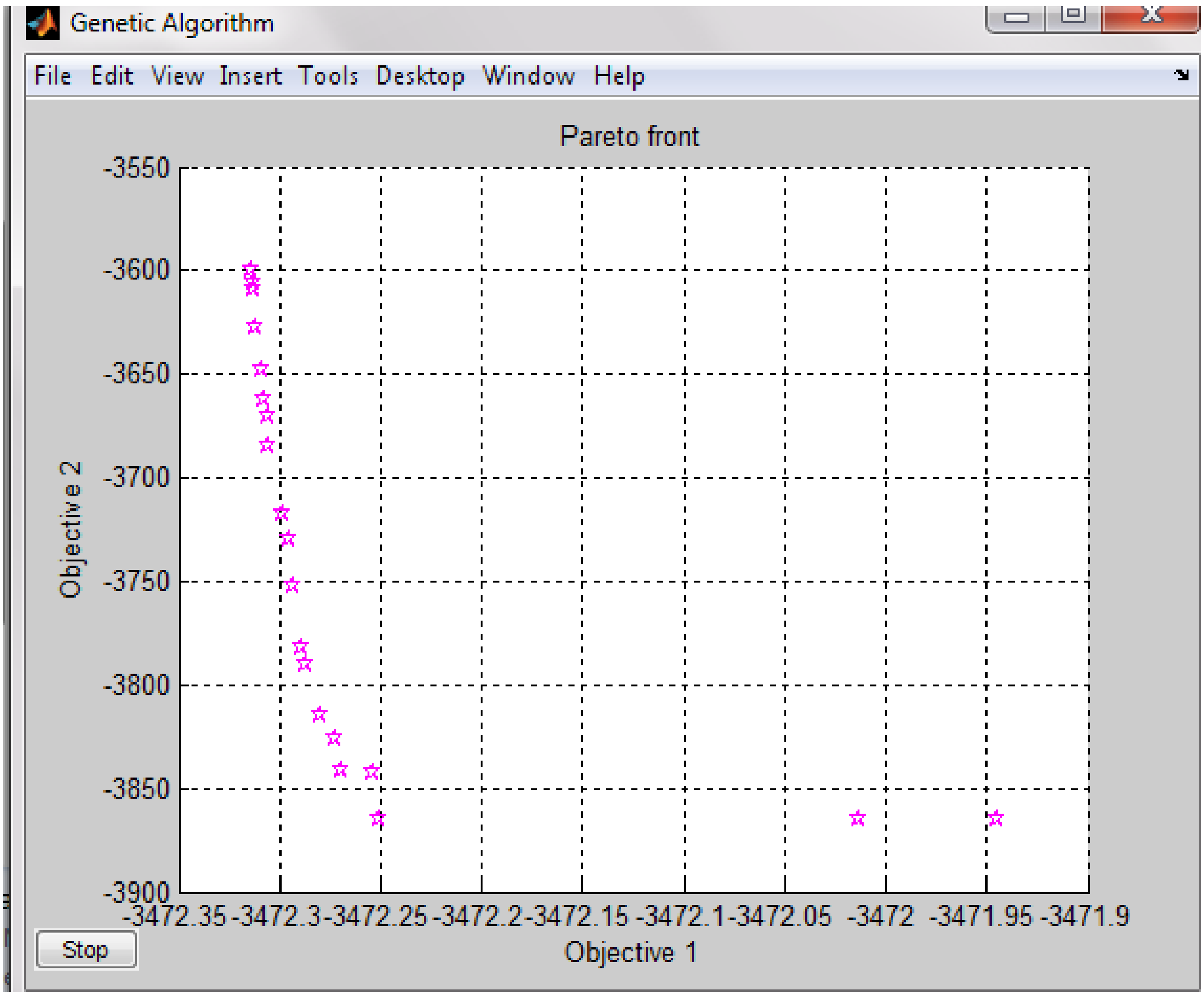


Figure 5.4 Pareto Front for Optimisation of Thermal Performance Parameters

5.1.2 Emission Constituents

The optimization of emission constituents is conducted by considering all the constituents of the exhaust gas measured in the experimental study. Among the emissions considered, oxygen is preferred to be maximum as it is not harmful to the environment and remaining emission constituents such as HC, CO, NO_x, SO_x and CO_2 to be minimum. This creates a multimodal scenario as oxygen is to be maximised at the expense of other constituents. The same steps adopted in solution procedure in section 5.1.1 of thermal performance are followed here also.

The polynomials for emission constituents are given in Table 5.3 with their respective R^2 values. Figure 5.5 shows the screen indicating process of optimization of emission constituents.

Table 5.3 Polynomials for Emission Constituents with their Respective R^2 values

Parameter	Polynomial	R^2
O_2	O_2 = -541388.21 -0.37 (x_1) + 15.41 (x_2) + 8480.75*(x_3) + 8.92E-03 (x_4) + 2.52E-02 $(x_1)^2$ -0.95 $(x_2)^2$ -43.30 $(x_3)^2$ -1.18E-04 $(x_4)^2$+ 5.63E-03 (x_1) (x_2)- 1.36E-06 (x_2) (x_3) -5.56E-02 (x_3) (x_4)+ 5.56E-02 (x_3) (x_4) -1.62E-03$(x_1)^3$+ 1.95E-02 $(x_2)^3$+ 7.21E-02 $(x_3)^3$+ 4.16E-07 $(x_4)^3$	0.9209
HC	HC = exp[-9.6E-02 (x_1) - 0.53 (x_2) - 1.38E-02 (x_3) - 1.29E-02 (x_4) + 14.016]	0.6638
CO	CO= -542389.53 - 121.82 (x_1) + 925.92 (x_2) + 8479.54 (x_3) - 5.63 (x_4) - 10.67 $(x_1)^2$ - 84.99 $(x_2)^2$ - 43.3 $(x_3)^2$ + 0.104 $(x_4)^2$ + 8.58 (x_1) (x_2) - 9.211E-06 (x_2) (x_3) - 0.27 (x_3) (x_4) + 0.27 (x_3) (x_4) + 0.66 $(x_1)^3$ + 2.1 $(x_2)^3$ + 0.072 $(x_3)^3$ - 5.57E-04 $(x_4)^3$	0.8668
NO_X	NO_X = -593.41 + 14.28 (x_1) + 70.25 (x_2) - 0.605 (x_3) - 0.311 (x_4) + 0.72 $(x_1)^2$ - 2.15 $(x_2)^2$ + 1.87E-03 $(x_3)^2$ + 4.13E-03 $(x_4)^2$ - 0.603 (x_1) (x_2) + 2.66E-02 (x_2) (x_3) + 1.53E-04 (x_3) (x_4) - 2.423E-02 (x_4) (x_1)	0.9495
CO_2	CO_2 = -541263.79 + 0.34 (x_1) -12.4 (x_2) + 8480.83 (x_3) -6.51E-03 (x_4) -2.12E-02 $(x_1)^2$+ 0.77 $(x_2)^2$ -43.3 $(x_3)^2$ + 9.16E-05 $(x_4)^2$ - 6.87E-03 (x_1) (x_2) -1.44E-05 (x_2) (x_3) + 6.82E-03 (x_3) (x_4) -6.82E-03 (x_3) (x_4)+ 1.26E-03 $(x_1)^3$ -1.6E-02 $(x_2)^3$ + 7.21E-02 $(x_3)^3$ -3.25E-07 $(x_4)^3$	0.9258
x_1= Load, x_2= CR, x_3= IP, x_4= Blend		

Figure 5.6 shows pareto front for optimization of emission constituents after the application of multi-objective optimization of emission constituents for maximisation of O_2 emissions with minimisation of other emission constituents (HC, CO, NO_x, SO_x and CO_2) for the diesel engine operating on different Karanja biodiesel and diesel blends at varying compression ratios, injection pressures and full load. Objective 1 is maximisation of O_2 and objective 2 is the minimisation of other emissions viz. HC, CO, NO_x, SO_x and CO_2. The pareto front gives set of points corresponding to different values of objective 1 and objective 2. It can be observed that most of the points are close to objective 2 axis. The reason is that, objective 1 is only maximisation of O_2 whereas objective 2 is minimisation of HC, CO, NO_x, SO_x and CO_2. Hence, more preference is given to objective 2 due to more number of parameters involved. The software uses pareto front to strike a balance between objective 1 and objective 2 and picks up a point to suggest the optimum values of input parameters such as CR, IP and blend for optimum emissions. The point picked up by the software may lie between a value of 0 to 5 and -

4000 to -3000 for objective 1 and objective 2 respectively. The values of objective 1 and objective 2 are used to determine the optimum input parameters by the software by following a complex set of calculations. The optimum values of CR, IP and blend suggested by the software after carrying the optimization run are 18, 220 bar, B70 respectively.

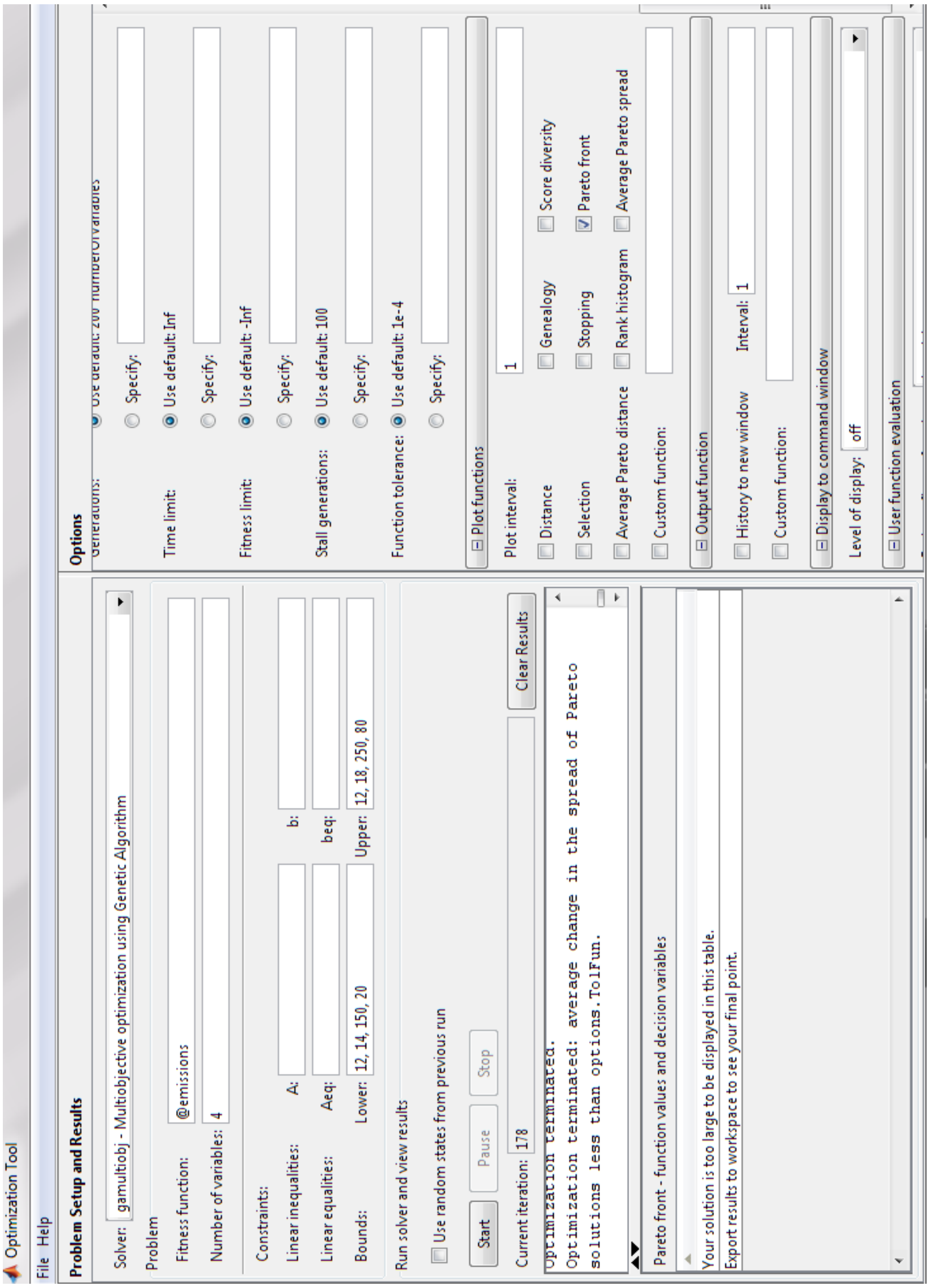


Figure 5.5 Display Screen of MATLAB Program for Optimization of Emission Constituents

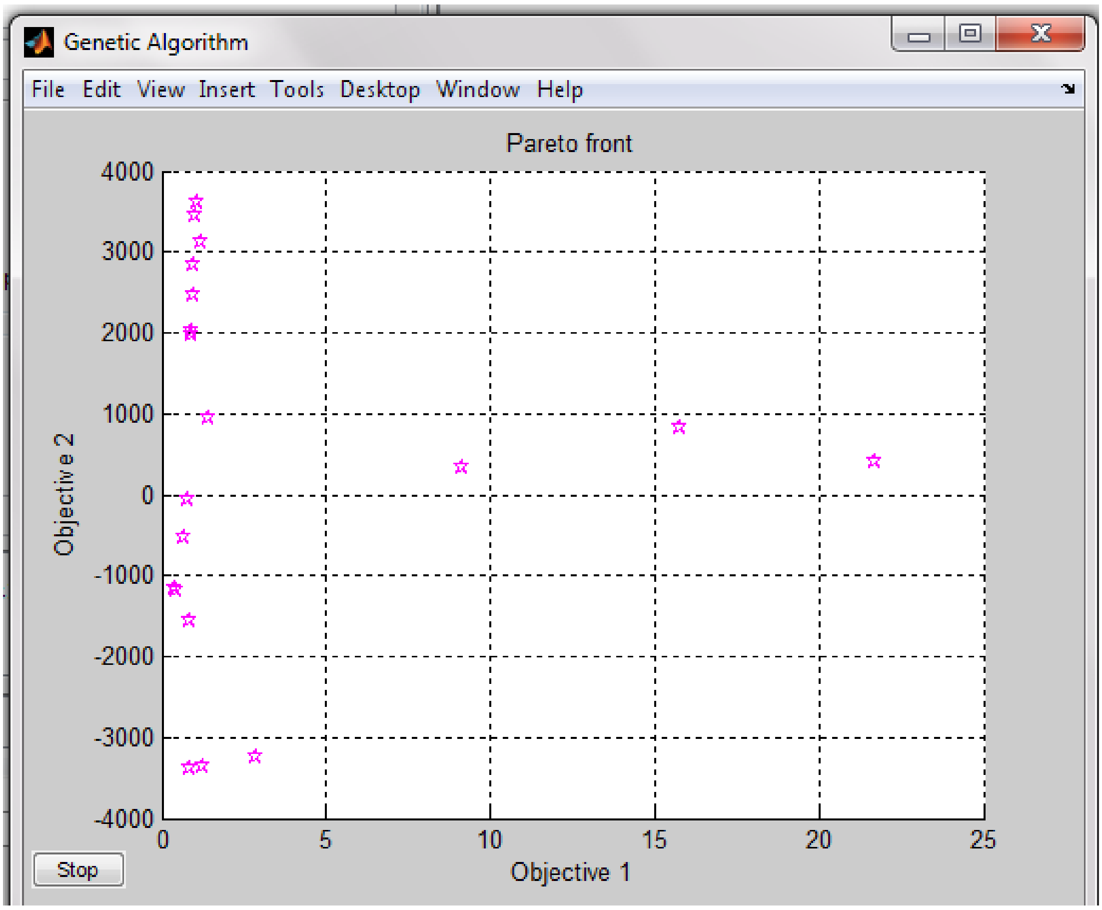


Figure 5.6 Pareto Front for Optimisation of Emission Constituents

5.1.3 Both Thermal Performance and Emission Constituents

The optimization of both thermal performance and emission constituents is conducted by considering all the performance parameters and emission constituents considered in sections 5.1.1 and 5.1.2. The solution procedure is also the same as followed for thermal performance optimization and emission constituents optimization.

Figure 5.7 shows the screen showing optimization of thermal performance and emission constituents.

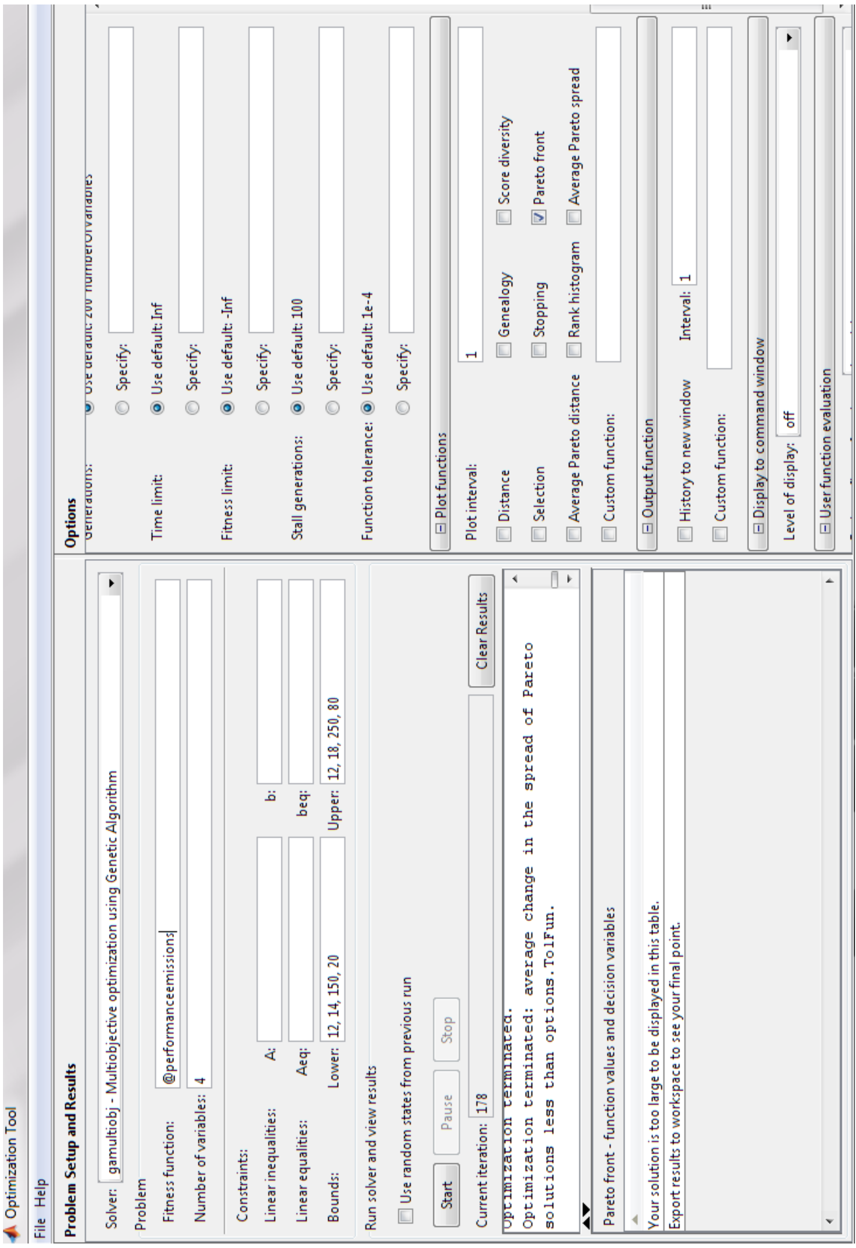


Figure 5.7 Display Screen of MATLAB Program for Optimization of Thermal Performance and Emission Constituents

Figure 5.8 shows pareto front for optimization of both thermal performance parameters and emission constituents giving equal weightages. Here, objective 1 is maximisation of thermal performance and objective 2 is the minimisation of exhaust emissions. The software uses a pareto front to strike a balance between objective1 and objective 2 and picks up a point to suggest the optimum values of input parameters such as CR, IP and blend. The point picked up by the software found to lie in a range of 0 to 5 and -4000 to -3000 for objective 1 and objective 2 respectively. The values of objective 1 and objective 2 are used to determine the optimum input parameters by the software by following a complex set of calculations. The optimum values of CR, IP and blend suggested by the software after carrying the optimization run are 18, 228 bar, B60 respectively.

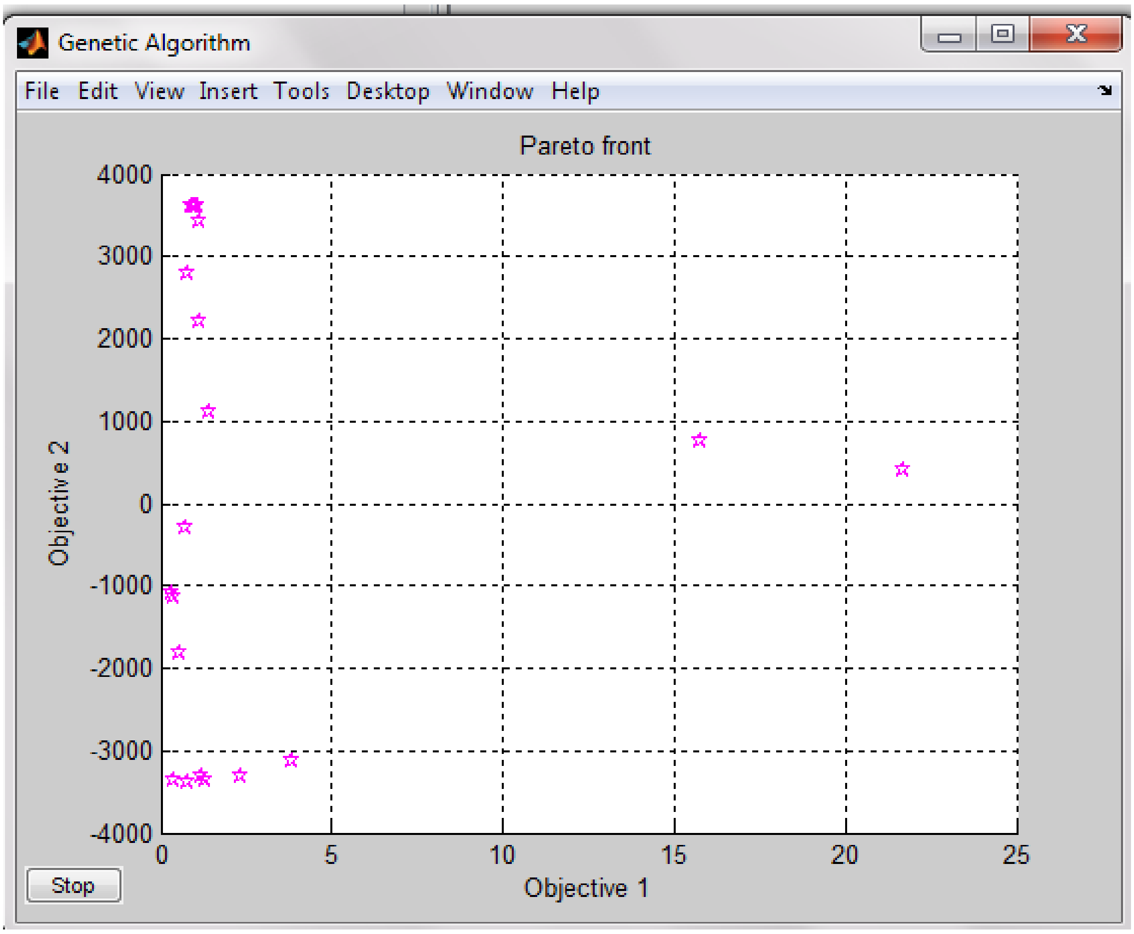


Figure 5.8 Pareto Front for Optimisation of Both Thermal Performance Parameters and Emission Constituents Giving Equal Weightages

5.1.4 Weighted Multi-objective Optimization

The multi-objective optimization carried out in the preceding section is conventional method. In such optimization equal preference is given to each individual parameter of both thermal performance and exhaust emissions. But, usually, the purpose of an IC engine is to operate with maximum thermal efficiency and at considerably lesser emissions. Rarely there may be cases where emissions are of more importance than efficiency. Under all such cases, weighted optimization** is to be carried out wherein suitable weights (preferences) are given to the thermal performance parameters and emission constituents.

The weighted optimization is carried out using genetic algorithm solver and the procedure is similar to that for conventional optimization. The only difference is that weights are defined for all the equations of output parameters which are obtained by nonlinear regression analysis (refer Table 5.2 and 5.3). Figure 5.9 shows the MATLAB editor which is used to define weights w (1) to w (8) for the equations f (1) to f (8).

Table 5.4 gives a few combinations of the results of weighted optimization of thermal performance and exhaust emissions. It is seen that the weightage given to thermal performance and exhaust emissions does not affect the values of CRs chosen. In all the cases, the optimum CR and IP are found to be 18 and 228bar respectively. Therefore, it can be inferred that the results obtained using optimization are in close agreement with those found in the experimental study. However the blend to be used appears to vary between B67 to B74 in most of the cases as observed from the table, based on the allotted weightages.

Further, it is also observed that the optimum values of CR and IP for equal weightages of 0.5 (50%) each to thermal performance and exhaust emissions match but blend do not match with those obtained from conventional multi-objective optimization[*]. The reason is that, in conventional optimization equal weightages are alloted to all parameters irrespective of whether they are related to thermal performance or exhaust emissions. But, in weighted optimization, 0.5 (50%) weightage given to thermal performance evaluation gets divided equally among BTHE, BSFC and EGT and the weightage given to exhaust emissions say gets divided equally among exhaust emissions constituents of HC, CO, NO_x, HSU, O_2, CO_2 and SO_x.

[*]Conventional multi-objective optimization: GA toolbox of MATLAB by default allots equal weightages to each individual parameter.

[**]Weighted optimization: Any selected value of weightage can be allotted collectively to a group of parameters or each individual parameter specifically

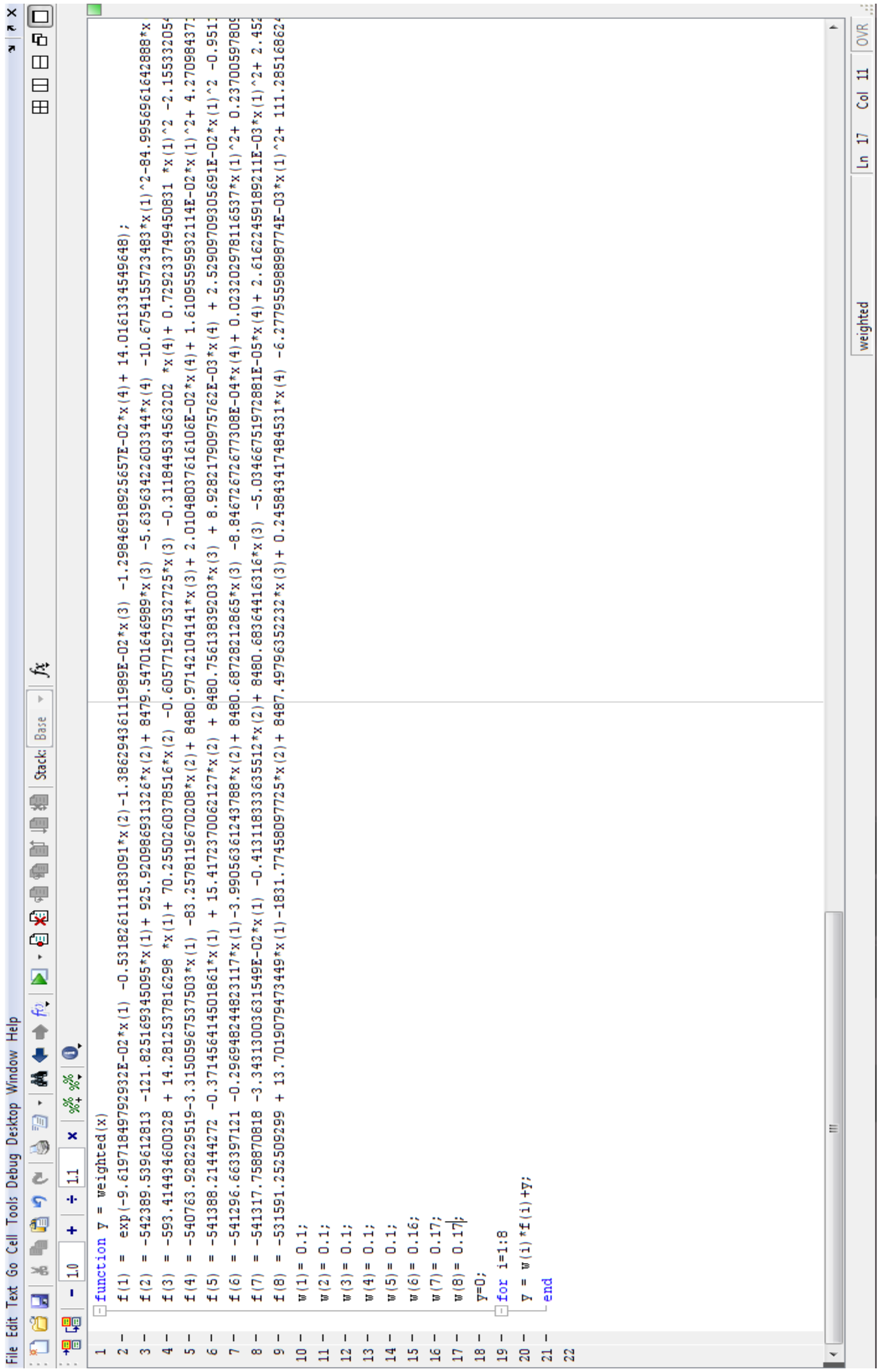


Figure 5.9 MATLAB Editor

It can also be noted that the optimum blend for 100% weightage to thermal performance is B72 which is in line with that obtained using conventional optimization (B70). But, when 100% weightage is given to exhaust emissions, the optimum blend through weighted optimization method is found to be B41. The value does not agree with that obtained by using conventional optimization. Here, in weighted optimization, highest weightage given to emissions indicate that B41 gives least emissions as compared to other higher blends.

Table 5.4 Weightages For Thermal Performance and Emissions In Multi objective Optimization

Weightage to Thermal Performance	Weightage to Emissions	CR	IP	Blend
1	0	18	228	72
0.8	0.2	18	228	74
0.7	0.3	18	228	72
0.6	0.4	18	228	75
0.5	0.5	18	228	71
0.4	0.6	18	228	72
0.3	0.7	18	228	70
0.2	0.8	18	228	67
0	1	18	228	41

Different optimization runs carried out considering thermal performance, emission constituents, both thermal performance and emission constituents giving equal and different weightages, show that the values of CR, IP and blend for optimum performance and emissions of the engine fuelled with Karanja biodiesel and diesel blends are found to be 18, 228bar and B70 respectively.

The experimental results presented in chapter 4 do not give the output parameters for the specified optimum values of CR, IP and blend as engine tests are not conducted for this configuration. Hence, there is a need to obtain the output parameters corresponding to the optimum values of inputs. An ANN can be used in order to obtain the corresponding output parameters, which is discussed in the following section.

5.2 ANN Modeling

An ANN is a system based on the operation of biological neural networks, in other words, is an emulation of biological neural system. ANNs are among the newest signal processing technologies nowadays. For further details Appendix IX can be referred.

In the present study, an ANN model is built to simulate the actual engine behaviour in order to obtain the output parameters of thermal performance and emission constituents corresponding to the optimum input parameters obtained. The model is built by using the data obtained by conducting a series of experiments on the four stroke, single cylinder, variable compression ratio, direct injection, diesel engine using blends of Karanja biodiesel and diesel as fuel. As discussed in chapter 4, the experiments are conducted by varying load, compression ratio, blend proportion and injection pressure. The experimental data is divided according to 80-20 rule, with 80% of the available data being used for training of the network and remaining 20% for validation.

Tables 5.5 and 5.6 represent a sample of input and output data that is used in the ANN modeling of engine performance and exhaust gas emission constituents respectively. The engine load is varied from 0kg to 12kg in steps of 3kg, compression ratios considered for conducting the experiments are 14, 15, 16, 17, 17.5 and 18, while the injection pressure is varied from 150bar to 250bar in steps of 50bar. The experiments are carried out for the blends of B20, B40, B60, B80, pure Karanja biodiesel and pure diesel. The data obtained by conducting these experiments is used to model the neural network for predicting thermal performance and exhaust gas emissions of the engine.

The software selected for neural network modeling in the present study is Easy NN. It is selected purely because of its simplicity in developing and training models involving feed forward multilayer neural networks with back-propagation training algorithm. Easy NN grows multilayer neural networks from the data in a *Grid*. The neural network input and output layers are created to match the grid input and output columns.Hidden layers connecting the input and output layers can then be grown to hold the optimum number of nodes. Each node contains a neuron and its connection address. The whole process is automatic. The data grid containing input/output matched pairs for training and validation of the neural network is produced by importing data from spreadsheet files, tab separated plain text files, and comma separated files, bitmap files or binary files. The grid can also be produced manually using the Easy NN grid editing facilities. Numeric, text, image or combinations of the data types in the grid can be used to grow the neural networks. The neural networks learn the training data in the grid and they can use the validation data available in the grid for self validation at the same time. When training finishes the neural networks can be tested using the querying data in the grid, the interactive query facilities or querying data in separate files. The steps that are required to produce neural networks are automated in Easy NN. The learning can be terminated by specifying maximum number of cycles or the targeted maximum error.

Table 5.5 Neural Network Input Output Sample Data for Engine Thermal Performance

Input Parameters				Output Parameters		
Load (kg)	Fuel	Compression Ratio	Injection Pressure	BSFC (kg/kWh)	BTHE (%)	EGT (^{0}C)
0	B0	18	200	27.80	0.45	175.67
3	B0	18	200	0.6	15.72	196.27
6	B0	18	200	0.42	23.15	237.55
9	B0	18	200	0.35	26.17	288
12	B0	18	200	0.3	31.25	350.75
0	B20	18	200	21.25	0.43	185.56
3	B20	18	200	0.61	15.26	210.96
6	B20	18	200	0.45	22.35	254.67
9	B20	18	200	0.3	25.86	289.53
12	B20	18	200	0.31	30.5	348.56
0	B40	18	200	20.13	0.4	180.65
3	B40	18	200	0.63	15.04	201.85
6	B40	18	200	0.4	22.7	239.68
9	B40	18	200	0.3	25.42	286.53
12	B40	18	200	0.32	30.05	347.96
0	B60	18	200	34.36	0.38	182.89
3	B60	18	200	0.65	14.82	208.23
6	B60	18	200	0.44	20.38	247.9
9	B60	18	200	0.35	25.08	299.65
12	B60	18	200	0.32	29.32	356.04
0	B80	18	200	25.40	0.36	187.53
3	B80	18	200	0.67	13.63	209
6	B80	18	200	0.39	20.35	242.69
9	B80	18	200	0.36	25.46	290.62
12	B80	18	200	0.33	28.75	346.99
0	B100	18	200	22.05	0.32	179.35
3	B100	18	200	0.7	12.58	198.69
6	B100	18	200	0.45	20.03	240.38
9	B100	18	200	0.37	24.39	281.85

12	B100	18	200	0.34	28.65	343.77

Table 5.6 Input Output Sample Data of Emission Constituents For Neural Network

Input Parameters				Output Parameters				
Load (kg)	Fuel	Compression Ratio	Injection Pressure	CO_2 (%)	CO (%)	NO_x (ppm)	HC (ppm)	O_2 (%)
0	B0	18	200	0.83	0.012	9	6	19.81
3	B0	18	200	1.34	0.013	37	5	18.99
6	B0	18	200	1.65	0.014	75	5	18.67
9	B0	18	200	2.01	0.020	118	6	18.18
12	B0	18	200	2.57	0.022	160	8	17.51
0	B20	18	200	0.88	0.011	15	5	19.58
3	B20	18	200	1.4	0.012	42	4	18.97
6	B20	18	200	1.7	0.017	81	5	18.42
9	B20	18	200	2.11	0.018	118	6	18.22
12	B20	18	200	2.6	0.02	161	7	17.56
0	B40	18	200	0.87	0.01	15	4	19.87
3	B40	18	200	1.47	0.011	38	4	19.04
6	B40	18	200	1.75	0.013	82	3	18.69
9	B40	18	200	2.17	0.015	124	4	18.25
12	B40	18	200	2.65	0.016	163	6	17.43
0	B60	18	200	0.89	0.008	15	3	19.91
3	B60	18	200	1.52	0.009	44	3	19.15
6	B60	18	200	1.79	0.012	81	2	18.67
9	B60	18	200	2.19	0.015	129	4	18.21
12	B60	18	200	2.63	0.015	167	5	17.69
0	B80	18	200	0.92	0.005	17	2	19.94
3	B80	18	200	1.58	0.007	46	3	19.08
6	B80	18	200	1.82	0.013	84	2	18.56
9	B80	18	200	2.18	0.011	132	3	17.96
12	B80	18	200	2.68	0.013	175	4	17.59
0	B100	18	200	0.96	0.004	19	2	20.03
3	B100	18	200	1.6	0.006	58	3	19.33
6	B100	18	200	1.84	0.012	87	2	18.95
9	B100	18	200	2.24	0.009	135	2	18.44
12	B100	18	200	2.7	0.011	179	4	17.62

5.2.1 Thermal Performance

The neural network for predicting thermal performance is developed by considering the operating parameters like load, compression ratio, injection pressure and blend as input parameters. The output parameters considered are brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature. Figure 5.10 shows the layout of simulation of actual engine using ANN model for engine performance. The ANN uses the same input parameters as received by the engine and gives corresponding outputs as that fron the engine. Therefore, the ANN model effectively replaces the engine.

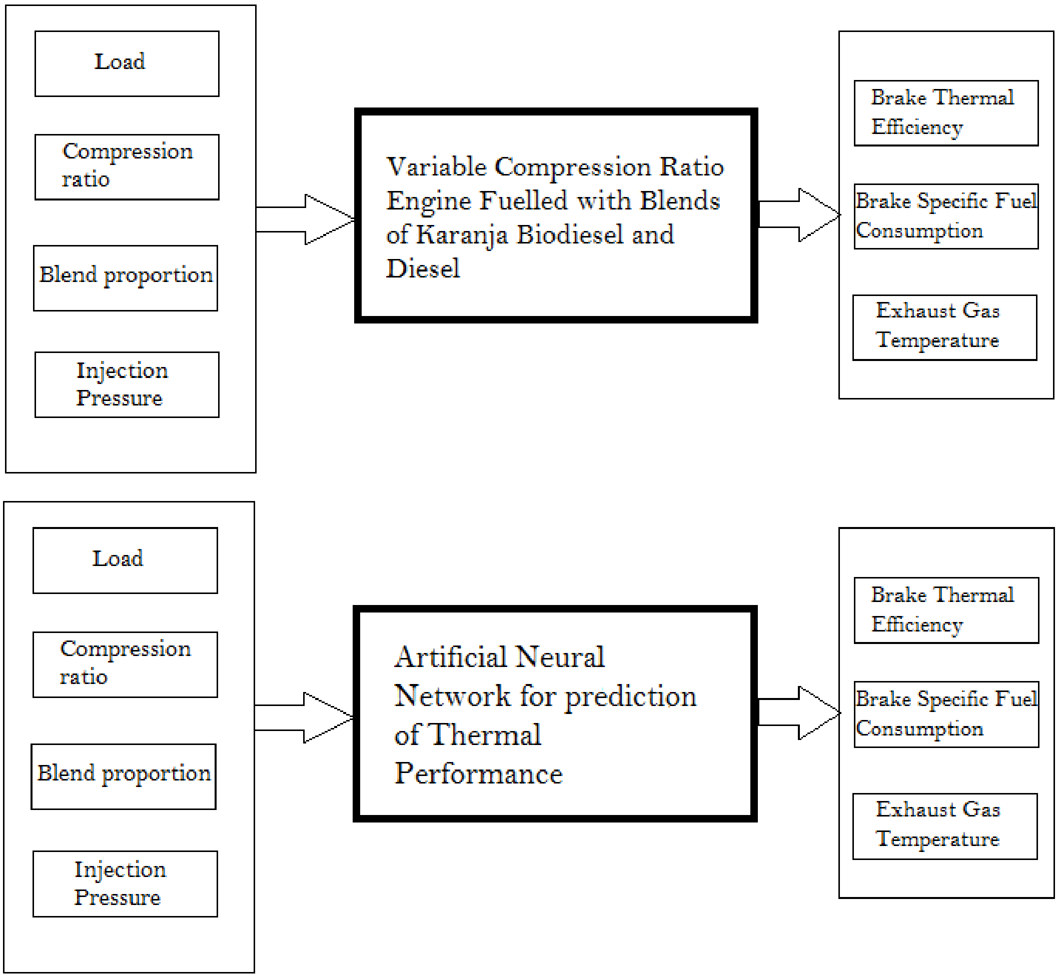


Figure 5.10 Layout of Simulation of Actual Engine using ANN Model for Engine Performance

The sample of results obtained experimentally by carrying out series of experiments on the variable compression ratio engine are listed in Table 5.5, which are used for developing the neural network model. Various architectures of neural networks are tried and an appropriate architecture is chosen such that the average error of the network for all outputs is within 5%. Table 5.7 gives the details used for modeling this neural network. According to the table, beginning from 2 hidden layered neural network model, having initially 8 cells in the hidden layer. The number of cells in the hidden layer is increased up to 30 while monitoring the error resulting at the end of training.

Table 5.7 Details Used to Model the Neural Network

Network Type	Feed Forward
Inputs for the neural network model	Load, Compression ratio, Blend, Injection pressure
Number of cells in input layer	4
Outputs from the neural network model	Brake thermal efficiency, Brake Specific Fuel Consumption, Exhaust Gas Temperature
Number of cells in output layer	3
Number of Hidden Layers	2
Initial Number of Cells in a Hidden Layer	8
Maximum Number of Cells in a Hidden Layer	30
Propagation Rule	Weighted Sum Rule
Activation Function	Logistic Function
Output Function	Identity Function
Learning Rule	Back Propagation

The criteria for the termination of training selected are a) permissible error and b) maximum number of cycles in training and validation. The limiting value for all the errors over the entire data is selected as 0.05 (5%) while the permissible error for validation sets is specified as 3% of the target value. The maximum number of training cycles is limited to 1000000 for each learning set. The training stops when any one of the above criteria, namely, all errors being less than 0.05, all validation points within 3% of target values or 1000000 training cycles being completed. The learning rate is kept as 0.6 and momentum as 0.8 for the stable learning and convergence of weights. The number of learning cycles before any validation cycle is executed is set to 3000. The number of validation cycles in one instance of validation is set to 100. These values are set in the control window of the software as shown in Figure 5.11.

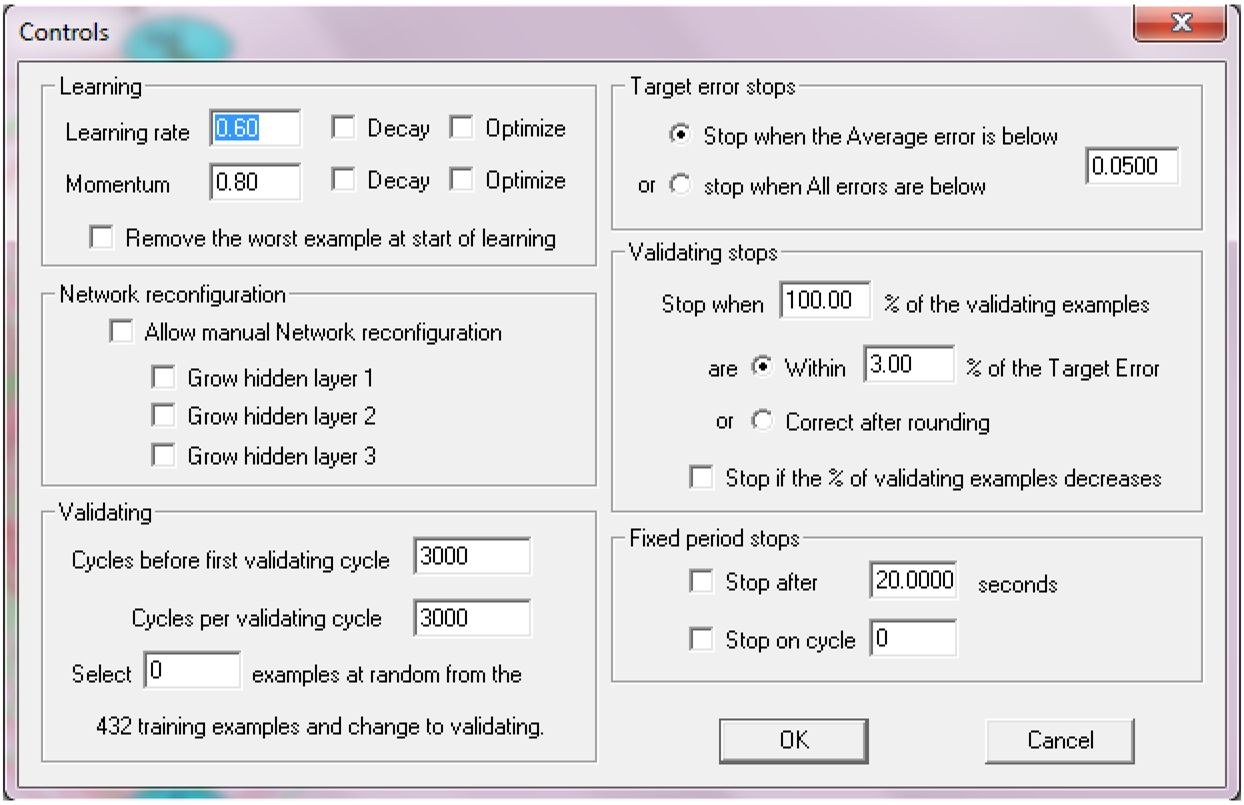


Figure 5.11 Setting Learning Control for Training of ANN Model

The neural network generated has 4 cells in the input layer, 8 cells in the first hidden layer, 10 cells in the second hidden layer and 3 cells in the output layer as shown in Figure 5.12. The architecture of such a neural network is denoted as 4.8.10.3 where the numbers denote the number of cells in the input layer, first hidden layer, second hidden layer and output layer respectively.

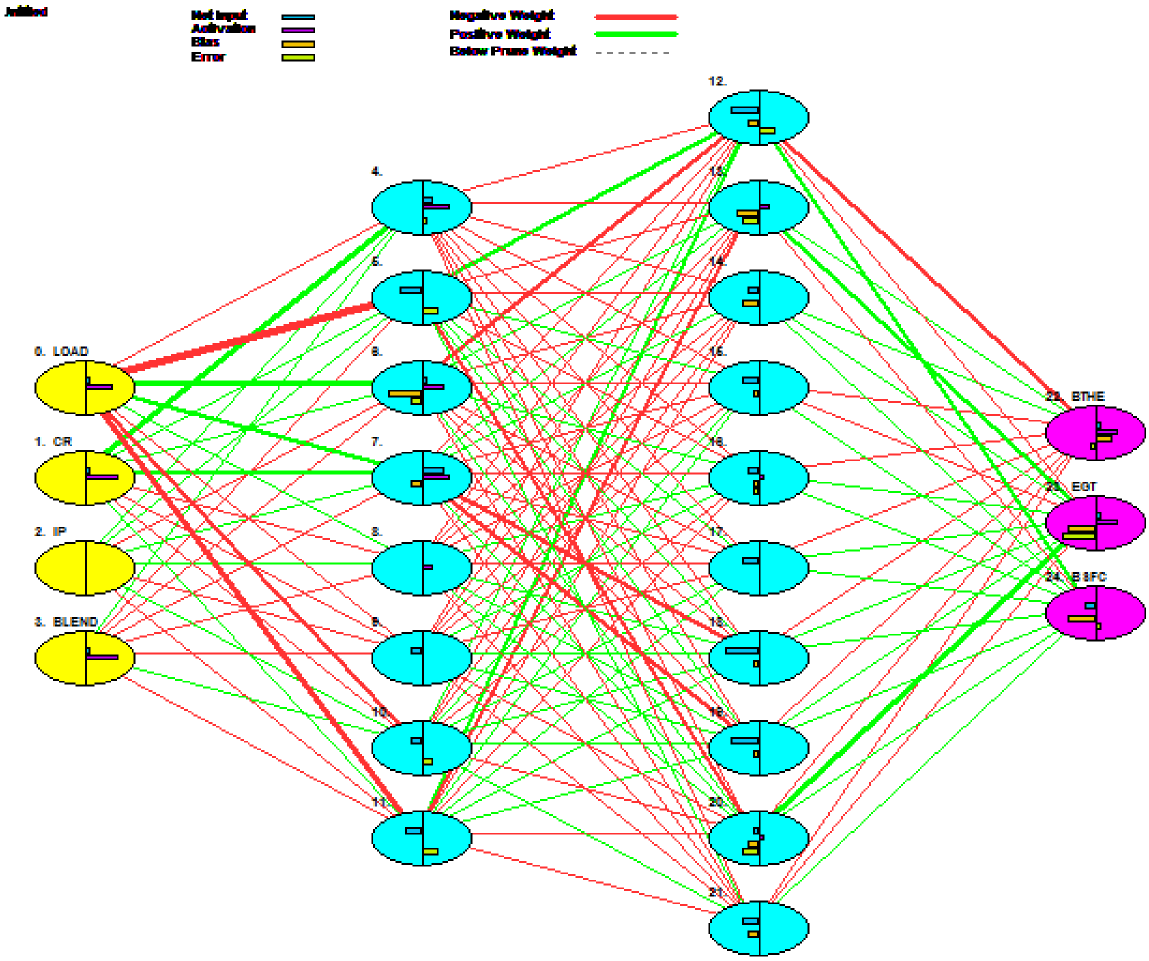


Figure 5.12 ANN Model for Predicting Thermal Performance with Architecture 4.8.10.3

On training of the network the error propagation graph for each training cycle is obtained as indicated in Figure 5.13. The maximum, average and minimum error are all seen to reduce at a fast pace in the early training period. But, towards the last training cycles, there is very small change in these error values. Further, the training ends with average error value less than the value of 5% permitted as the target error value. The training does not achieve the limiting error value of 3% of target value selected for the validation set. The training stops after number of training cycles without the error limit set for validation points being achieved because no further reduction in error is seen for a number of consecutive cycles.

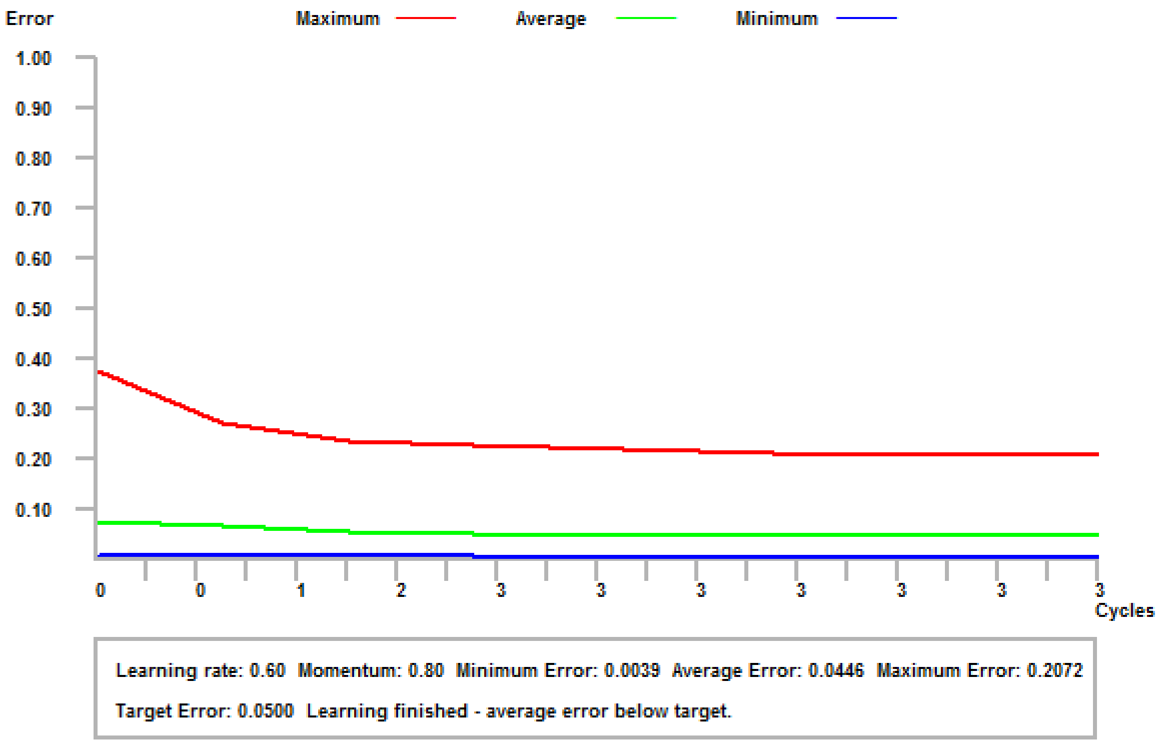


Figure 5.13 ANN Model Training & Error Propogation with Increasing Number of Training Cycles for the 4.8.10.3

Table 5.8 shows the average, minimum and maximum error for different architectures of the neural network. It can be observed that the average errors of all the architectures are almost identical and hence it becomes difficult to select the best architecture for the model. In such a case, the R test can be used in order to determine the best architecture. The values of rms error and R are evaluated by using Equations 5.1 to 5.6

Table 5.8 Errors for Different Architectures of Neural Network

Sr. No	Architecture of the Model	Average Error (%)	Minimum Error (%)	Maximum Error (%)	Validation stops within limiting Error (%)	Remarks
1	4.8.10.3	4.983	0.4482	22.6534	3	Training stopped with all errors within 5%
2	4.8.9.3	4.5806	0.3713	21.9288	3	
3	4.7.9.3	4.7719	0.2836	21.4245	3	
4	4.7.8.3	4.7719	0.2836	21.4245	3	
5	4.5.6.3	4.8108	0.3058	21.6008	3	

6	4.7.7.3	4.7889	0.3729	22.257	3	
7	4.22.3	4.7053	0.513	21.8532	3	
8	4.21.3	4.7962	0.6696	23.4381	3	
9	4.20.3	4.9679	0.2332	24.2883	3	
10	4.19.3	4.7524	0.4905	22.6710	3	

For error calculation, Equations 5.1 to 5.6 are used.

Error for each case is defined as

$$\%Error = \frac{|A_e - A_p|}{A_e} \qquad (5.1)$$

where, A_e = The output value as obtained from theoretical analysis

A_p = The output value predicted by the neural network model

The average error for entire epoch (complete set of input-output pairs) is defined as

$$\%Error_{av} = \frac{1}{N}\sum_{i=1}^{N}\frac{|A_{ei} - A_{pi}|}{A_{ei}} \qquad (5.2)$$

The maximum error is defined as

$$\%Error_{max} = max\left(\sum_{i=1}^{N}\frac{|A_{ei} - A_{pi}|}{A_{ei}}\right) \qquad (5.3)$$

and the minimum error is defined as

$$\%Error_{min} = min\left(\sum_{i=1}^{N}\frac{|A_{ei} - A_{pi}|}{A_{ei}}\right) \qquad (5.4)$$

For each architecture of neural network model the root mean square value of error is

$$Error_{rms} = \sqrt{\frac{1}{N}\sum_{i=1}^{N}\left(\frac{|A_{ei} - A_{pi}|}{A_{ei}}\right)^2} \tag{5.5}$$

The confidence R (coefficient of determination) can be used to decide upon the best architecture. The R values can be determined as

$$R = \frac{1}{N}\sum_{i=1}^{N} R_i = \frac{1}{N}\sum_{i=1}^{N}\frac{A_e}{A_p} \tag{5.6}$$

The training and test errors for the networks are listed in Table 5.9. It can be observed that the values of errors are well within specified limits for all the neural network model architectures evaluated. On the basis of R test, it is found that the model having architecture 4.22.3 is a good model for which value of R is closest to unity among other models. Hence this model is selected as the best representative model for the prediction of thermal performance constituents.

Table 5.9 Training and Test Errors for Neural Network Architectures

Architecture of the Model	Average Error (%)	R
4.8.10.3	4.983	0.91517
4.8.9.3	4.5806	0.91637
4.7.9.3	4.7719	0.92456
4.7.8.3	4.7719	0.92092
4.5.6.3	4.8108	0.92916
4.7.7.3	4.7889	0.92428
4.22.3	4.7053	0.94355
4.21.3	4.7962	0.928815
4.20.3	4.9679	0.94024
4.19.3	4.7524	0.93864
4.23.3	4.5668	0.95002

5.2.2 Emission Constituents

The constituents of exhaust gas measured during the experiments are CO, CO_2, HC, O_2, NO_X, and SO_2. The exhaust gas emissions are directly dependent on input parameters like engine load, compression ratio, injection pressure, blend proportion. Thus, in order to develop a neural network which predicts the exhaust gas emission constituents for diesel engine working with Karanja biodiesel and its blends with diesel, the operating conditions such as engine load, CR, IP and blend proportion are given to the model as the input. The output parameters from the model are the exhaust gas constituents. Figure 5.14 shows the layout of simulation of actual engine using ANN model for exhaust emissions. The ANN uses the same input parameters as received by the engine and gives similar outputs as that of the engine. Therefore, the ANN model effectively replaces the actual engine.

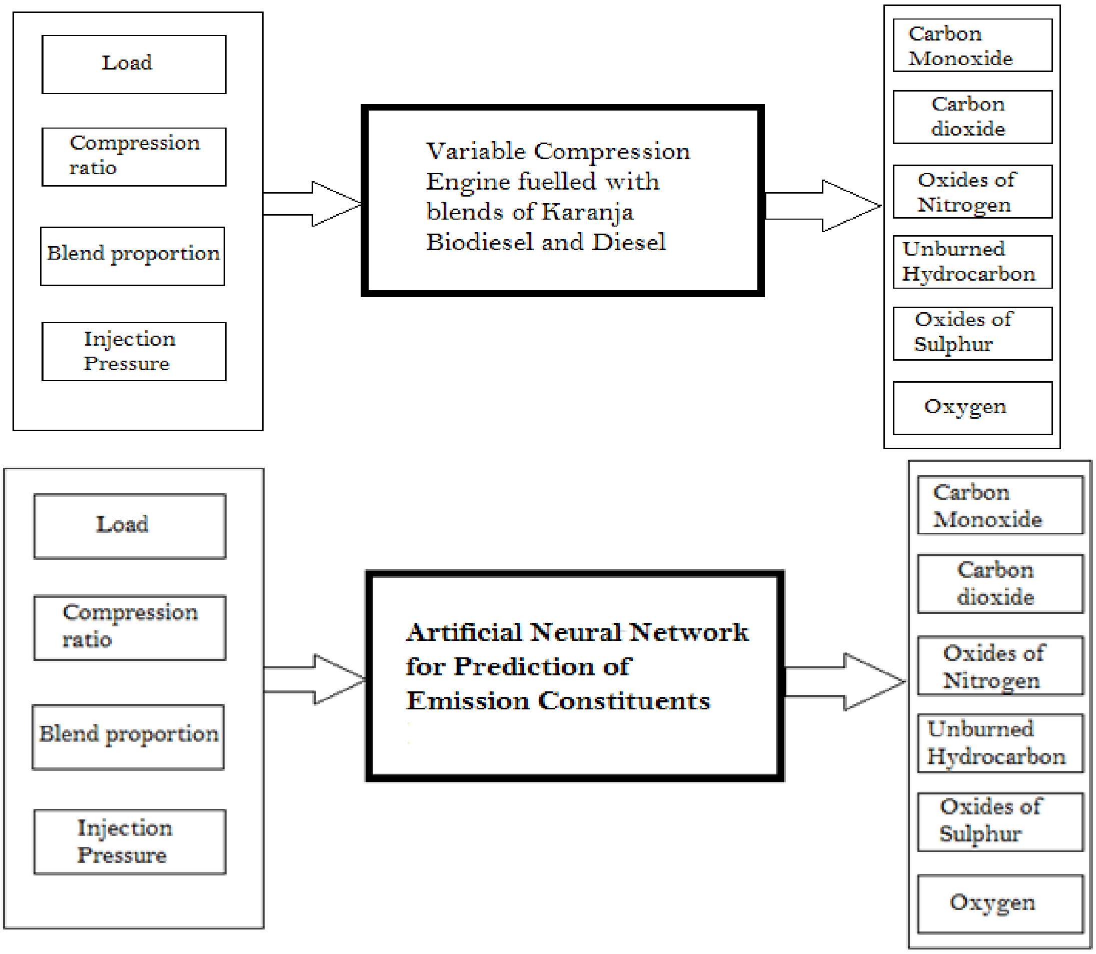


Figure 5.14 Simulation of Actual Engine Using ANN Model for Exhaust Emissions

The sample of results obtained experimentally by carrying out series of experiments on the variable compression ratio engine are listed in Table 5.6, which are used for developing the neural network model. Various architectures of neural networks are tried and an appropriate architecture is determined such that the average error of the network for all outputs is within 5%. Table 5.10 gives the details used for modeling this neural network. As per the table, beginning from 2 hidden layered neural network model, having initially 8 cells in the hidden layer, the number of cells in the hidden layer is increased up to 30 while monitoring the error, resulting at the end of training.

Table 5.10 Neural Network Modeling for Emission Constituents

Network Type	Feed Forward
Inputs for the neural network model	Load, Compression ratio, Blend, Injection pressure
Number of cells in input layer	4
Outputs from the neural network model	Brake thermal efficiency, Brake Specific Fuel Consumption, Exhaust Gas Temperature
Number of cells in output layer	3
Number of Hidden Layers	2
Initial Number of Cells in a Hidden Layer	8
Maximum Number of Cells in a Hidden Layer	30
Propagation Rule	Weighted Sum Rule
Activation Function	Logistic Function
Output Function	Identity Function
Learning Rule	Back Propagation

The limiting value for all the errors over the entire data is selected as 0.05 (5%) while the permissible error for validation sets is specified as 3% of the target value. The maximum number of training cycles is limited to 1000000 for each learning set. The training stops when any one of the above criteria, namely, all errors being less than 0.05, all validation points within 3% of target values or 1000000 training cycles being completed. The learning rate is kept as 0.6 and momentum as 0.8 for the stable learning and convergence of weights. The number of learning cycles before any validation cycle is executed is set to 3000. The number of validation cycles in one instance of validation is set to 100. These values are set in the controls window of the software as shown in Figure 5.15.

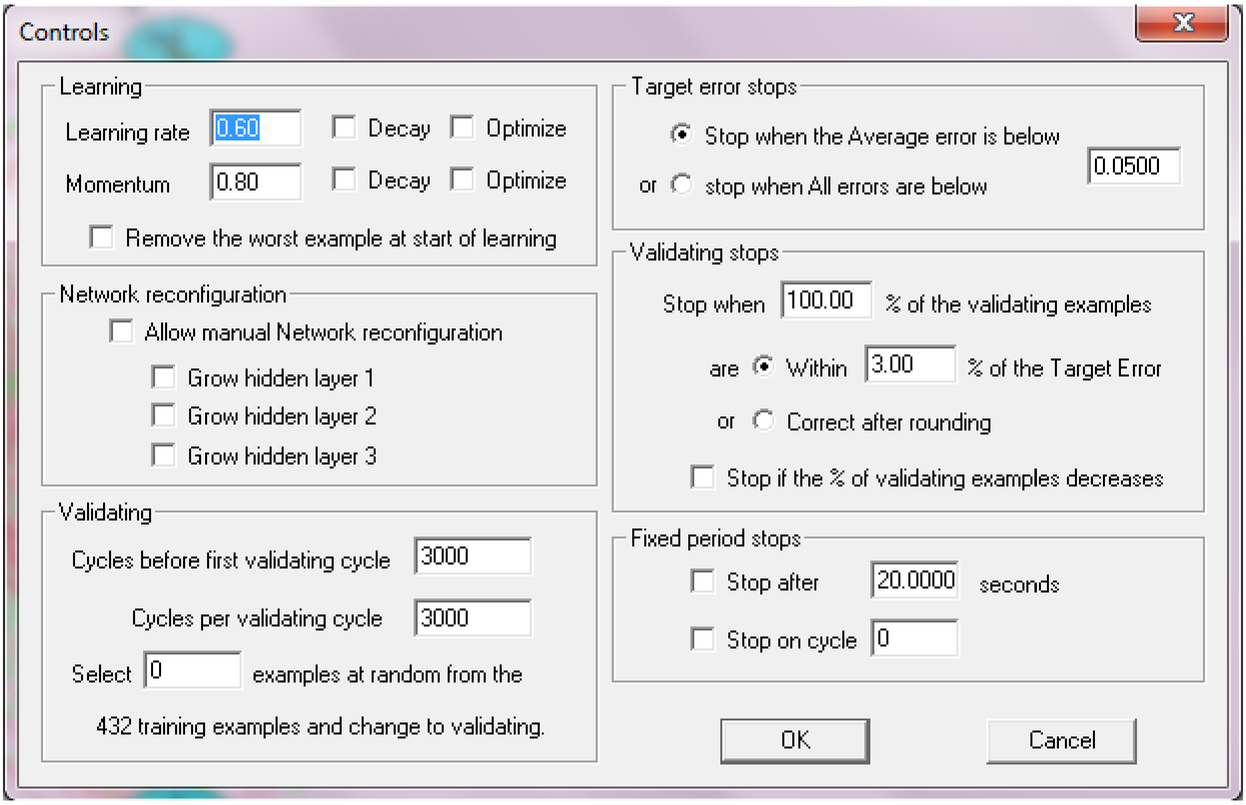


Figure 5.15 Setting Learning Controls for Training of ANN Model

The neural network generated has 4 cells in the input layer, 8 cells in the first hidden layer, 10 cells in the second hidden layer and 6 cells in the output layer as shown in Figure 5.16. The architecture of such a neural network is denoted as 4.8.9.5 where the numbers denote the number of cells in the input layer, first hidden layer, second hidden layer and output layer respectively.

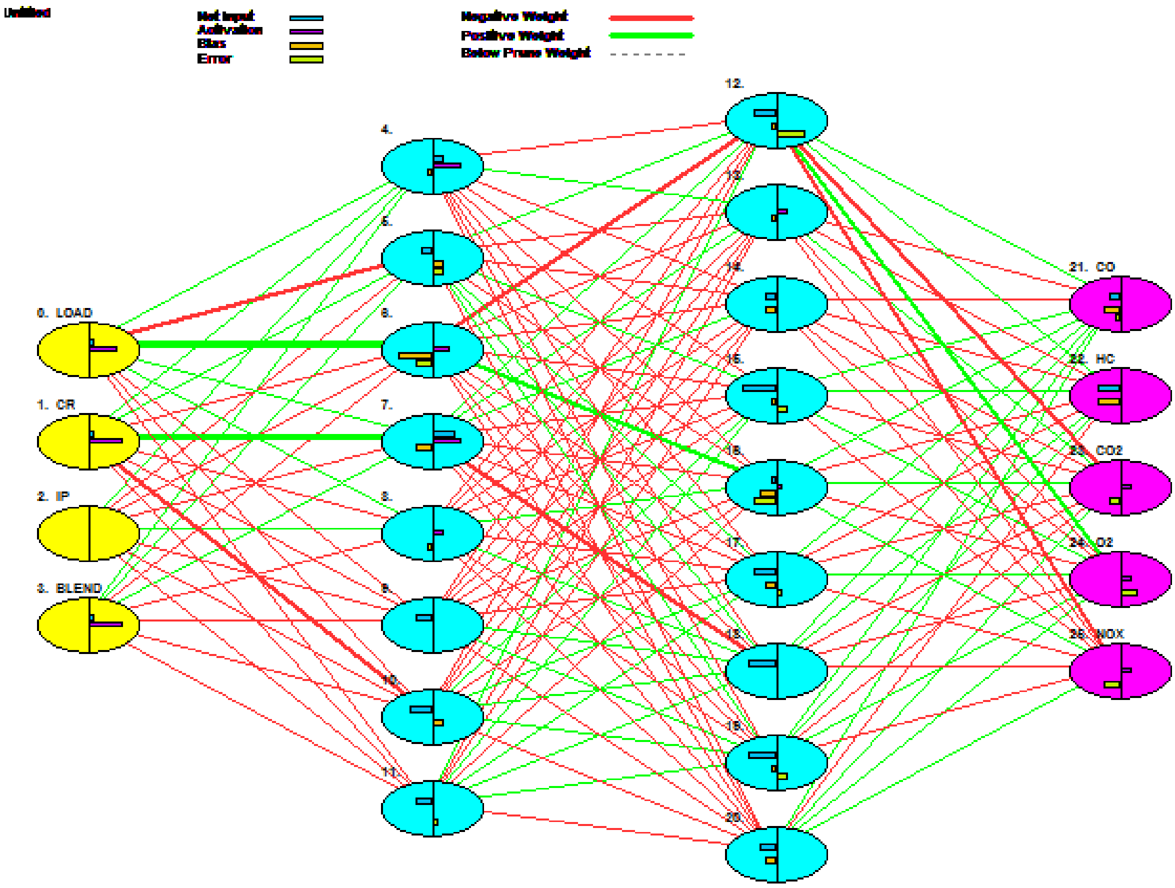


Figure 5.16 ANN Model of Exhaust Gas Constituents with Architecture 4.8.9.5

On training of the network the error propagation graph for each training cycle is obtained as indicated in Figure 5.17. The maximum, average and minimum error are all seen to reduce at a fast pace in the early training period. But, towards the last training cycles, there is very small change in these error values. Further, the training ends with average error value less than the value of 5% permitted as the target error value. The training does not achieve the limiting error value of 3% of target value selected for the validation set. The training stops after number of training cycles without the error limit set for validation points being achieved because no further reduction in error is seen for a number of consecutive cycles.

Table 5.11 gives the average, minimum and maximum errors for different architectures tested. It can be observed that the average errors of all the architectures are almost identical and hence it becomes difficult to select the best architecture for the model. In such a case, the R test can be used in order to determine the best architecture.

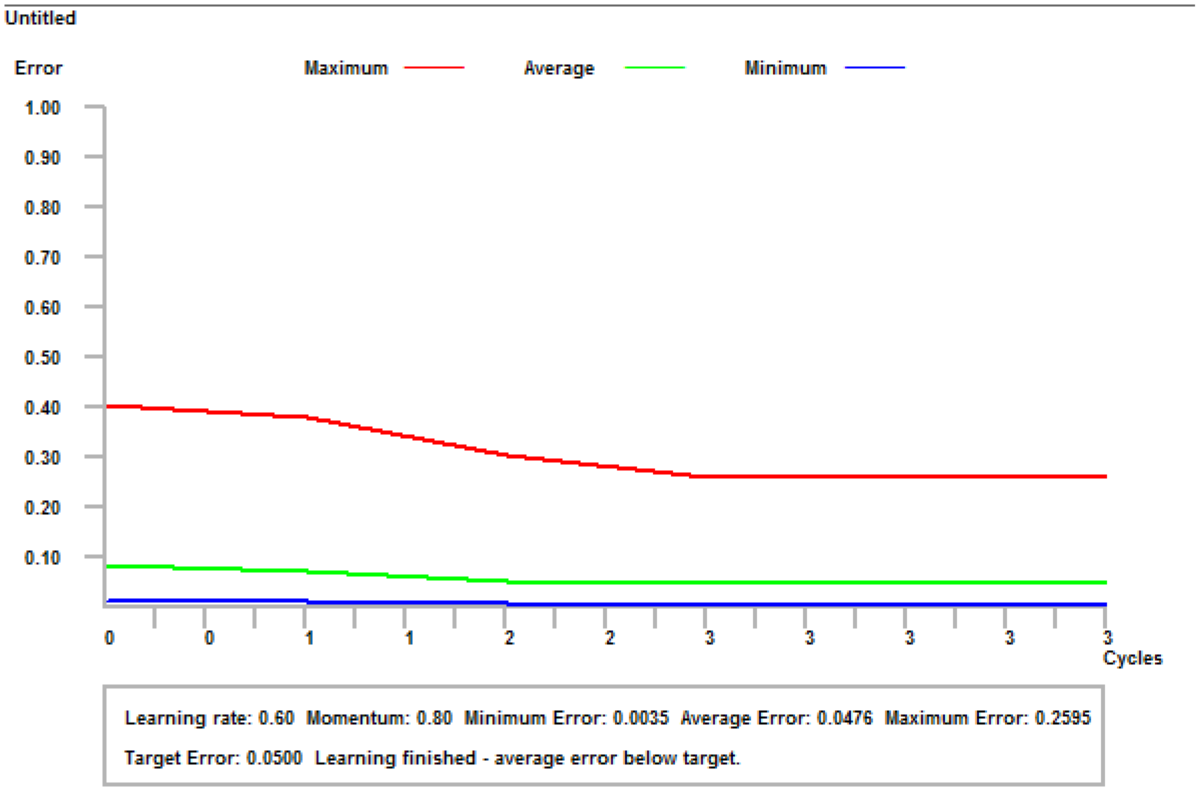


Figure 5.17 ANN Model Training & Error Propogation Graph With Increasing Number of Training Cycles for 4.8.9.5

Table 5.11 Neural Network Architecture & Corresponding Training Results for Gas Emissions

Sr. No	Architecture of the Model	Average Error (%)	Minimum Error (%)	Maximum Error (%)	Validation stops within limiting Error (%)	Remarks
1	4.8.10.5	4.6410	0.5076	22.3102	3	Training stopped with all errors within 5%
2	4.8.8.8.5	4.9514	0.7642	22.8478	3	
3	4.6.6.4.5	4.9928	0.648	26.9356	3	
4	4.6.6.5	4.9968	0.4639	22.9947	3	
5	4.6.7.5	4.9726	0.5657	23.3737	3	
6	4.20.20.5	4.8227	0.6691	22.9378	3	
7	4.6.7.5	4.7852	0.564	22.675	3	
8	4.6.6.6.5	4.6724	0.763	23.7865	3	
9	4.8.7.5	4.7895	0.7236	24.9720	3	
10	4.8.8.5	4.5463	0.7342	23.8765	3	

The training and test errors for the networks are listed in Table 5.12. It can be observed that the values of error are well within specified limits for all the neural network model architectures evaluated. On the basis of R test, it is found that the model having architecture 4.8.10.5 is a good model for which value of R is closest to unity among other models. Hence this model is selected as the best representative model for the prediction of thermal performance constituents.

Table 5.12 Training and Test errors for Different Architectures

Architecture of the Model	Average Error (%)	R
4.8.10.5	4.641	1.00288
4.8.8.8.5	4.9514	0.87712
4.6.6.4.5	4.9928	0.95073
4.6.6.5	4.9968	0.78219
4.6.7.5	4.9726	0.8043
4.20.20.5	4.8227	0.97175
4.6.6.6.5	4.7852	0.92968
4.8.9.5	4.6724	1.0028
4.8.7.5	4.7895	0.82335
4.8.8.5	4.5463	0.8976

It is seen from ANN modeling for prediction of exhaust gas constituents and thermal performance that ANN models can successfully capture the complex input-output relationships and still provide small prediction errors.

In order to determine the best ANN model a number of architectures are tried and tested for error, spread and coefficient of determination. The representative model selected for thermal performance has architecture of 4.22.3. It has an average error of nearly 4.79%, minimum error of 0.66% and maximum error of 23%. The coefficient of determination is very close to unity being 0.92 and the spread is 0.1576. This ensures that in the test range if this model is subject to any condition for which experiment is not conducted, the error will not exceed 5% on an average and 23% maximum. This is shown by applying the model to a selected set of data. The results predicted by 4.22.3

ANN model are compared with experimental results and the error is evaluated (Refer Table 5.13)

On similar lines, the representative model selected for emission constituents has an architecture of 4.8.10.5. It has an average error of nearly 5%, minimum error of 0.5% and maximum error of 22%. The coefficient of determination is very close to unity being 1.0028 and the spread is 0.2876. This ensures that in the test range if this model is subject to any condition for which experiment is not conducted, the error will not exceed 5% on an average and 22% maximum. This is shown by applying the model to a selected set of data. The results predicted by 4.8.10.5 ANN model are compared with experimental results and the error is evaluated (Refer Table 5.14)

For the ANN models selected for both thermal performance and emission constituents, the average prediction error is hence close to 5% which is in line with the model errors reported in most literature reviewed. Further, the ANN model for thermal performance modeling required one hidden layer while that for exhaust gas constituents required two. The number of cells in these hidden layers is larger for exhaust gas constituents. This indicates a much more complex relationship between the exhaust gas constituents and the input parameters as compared to the thermal performance parameters.

Tables 5.13 and Table 5.14 show that the ANN models developed for prediction of thermal performance and exhaust emission constituents have an acceptable error and hence can be used for obtaining the output parameters corresponding to optimized input parameters. The optimum values of CR, IP and blend obtained through optimization using genetic algorithm tool are 18, 228bar and B70 respectively. The output parameters corresponding to CR, IP and blend of 18, 228bar and B70 are given in Table 5.15.

Table 5.13 Comparison of Results of ANN Model and Experimental Data for Thermal Performance

Load (kg)	Fuel	CR	IP (bar)	BSFC (kg/kWh)			BTHE (%)			EGT (^{0}C)		
				Exp	ANN	Error (%)	Exp	ANN	Error (%)	Exp	ANN	Error (%)
3	Diesel	18	200	0.60	0.54	10.00	15.72	14.76	6.10	196.27	190.85	2.76
6	Diesel	18	200	0.42	0.39	7.14	23.15	24.65	-6.47	237.55	220.89	7.01
9	Diesel	18	200	0.35	0.36	-2.85	26.17	27.12	-3.63	288	267.43	7.14
12	Diesel	18	200	0.30	0.29	3.33	31.25	29.87	4.41	350.75	376	-7.19
3	B20	18	200	0.61	0.58	4.91	15.26	16.89	-10.68	210.96	195.46	7.34
6	B20	18	200	0.45	0.48	-6.66	22.35	23.06	-3.17	254.67	265.37	-4.20
9	B20	18	200	0.30	0.33	-10.00	25.86	23.87	7.69	289.53	300.78	-3.88
12	B20	18	200	0.31	0.35	-12.90	30.5	28.21	7.50	348.56	360.56	-3.44
3	B40	18	200	0.63	0.55	12.69	15.04	16.87	-12.16	201.85	225.46	-11.69
6	B40	18	200	0.40	0.42	-5.00	22.7	23.65	-4.18	239.68	245.67	-2.49
9	B40	18	200	0.30	0.28	6.66	25.42	25.12	1.18	286.53	295.43	-3.10
12	B40	18	200	0.32	0.36	-12.5	30.05	28.9	3.82	347.96	330.98	4.87

Table 5.14 Comparison of Results of ANN and Experimental Data for Emission Constituents

Load (kg)	Fuel	CR	IP (bar)	CO (%)			HC (ppm)			CO_2 (%)			O_2 (ppm)			NO_x (ppm)		
				Exp	ANN	Error (%)	Exp	ANN	Error (%)	Exp	ANN	Error (%)	Exp	ANN	Error (%)	Exp	ANN	Error (%)
3	Diesel	18	200	0.013	0.0134	-3.07	5	4.8	4.00	1.34	1.26	5.97	18.99	19.78	-4.16	37	35	5.40
6	Diesel	18	200	0.014	0.0142	-1.42	5	4.7	6.00	1.65	1.73	-4.84	18.67	18.28	2.08	75	69	8.00
9	Diesel	18	200	0.020	0.0213	-6.50	6	5.5	8.33	2.01	2.14	-6.46	18.18	17.38	4.40	118	109	7.62
12	Diesel	18	200	0.022	0.0224	-1.81	8	7.5	6.25	2.57	2.68	-4.28	17.51	16.32	6.79	160	171	-6.87
3	B20	18	200	0.012	0.013	-8.33	4	4.2	-5.00	1.4	1.54	-10.00	18.97	18.45	2.74	42	45	-7.14
6	B20	18	200	0.017	0.016	5.88	5	5.2	-4.00	1.7	1.54	9.41	18.42	18.67	-1.35	81	78	3.70
9	B20	18	200	0.018	0.0185	-2.77	6	6.5	-8.33	2.11	1.98	6.16	18.22	18.43	-1.15	118	113	4.23
12	B20	18	200	0.02	0.0211	-5.50	7	7.4	-5.71	2.6	2.89	-11.15	17.56	18.45	-5.06	161	169	-4.96
3	B40	18	200	0.011	0.0113	-2.72	4	3.8	5.00	1.47	1.34	8.84	19.04	18.67	1.94	38	39	-2.63
6	B40	18	200	0.013	0.0128	1.53	3	2.8	6.66	1.75	1.5	14.28	18.69	19.89	-6.42	82	75	8.53
9	B40	18	200	0.015	0.0146	2.66	4	4.3	-7.50	2.17	2.34	-7.83	18.25	17.3	5.20	124	118	4.83
12	B40	18	200	0.016	0.0159	0.62	6	6.2	-3.33	2.65	2.54	4.15	17.43	17.9	-2.69	163	169	-3.68

Table 5.15 Output Parameters Corresponding to CR, IP and Blend of 18, 228bar and B70

THERMAL PERFORMANCE	Output Parameters	Value
	BTHE (%)	27.49
	BSFC (kg/kWh)	0.33
	EGT (^{0}C)	345.56
EMISSION CONSTITUENTS	CO (%)	0.006
	HC (ppm)	3
	CO_2 (%)	3.08
	O_2 (%)	17.80
	NO_X (ppm)	160

Table 5.16 presents a comparison between the values of output parameters obtained through the ANN corresponding to the optimised input parameter for and those obtained through experimentation for Karanja biodiesel and Diesel oil. The experimental results given in the table correspond to a full load of 12kg, CR of 18 and IP of 200bar.

Table 5.16 Comparison of Thermal Performance and Emission Constituents for Diesel Oil, B70 and Karanja Biodiesel

THERMAL PERFORMANCE	Output Parameters	Diesel	B70	Karanja biodiesel
	BTHE (%)	29.20	27.49	26.64
	BSFC (kg/kWh)	0.31	0.33	0.34
	EGT (^{0}C)	350.75	345.56	343.77
EMISSION CONSTITUENTS	CO (%)	0.02	0.006	0.005
	HC (ppm)	5.00	3.00	2.00
	CO_2 (%)	2.62	3.08	3.26
	O_2 (%)	17.62	17.80	17.91
	NO_x (ppm)	125	160	166

It can be observed that thermal performance is best for Diesel oil and emission constituents are least for Karanja biodiesel. It can be also noted that the thermal performance and emission constituents for B70 are in between those for Diesel oil and Karanja biodiesel. The optimum values for blend B70 are obtained by striking a compromise or balance between Diesel oil and Karanja biodiesel. If the engine is operated with B70 blend, the values of BTHE, BSFC and EGT are found lesser by about 6%, 6% and 1.4% respectively as compared to Diesel oil and the emission constituents of CO, HC, CO_2, O_2 and NO_x are found more by about 20%, 50%, 5%, 0.6% and 4% respectively as compared to Karanja biodiesel.

Chapter 6

Concluding Remarks

The experimental study is conducted on a four stroke, variable compression ratio diesel engine using Karanja biodiesel and its blends with diesel. The thermal performance and emissions characteristics are evaluated by running the engine at different combinations of preset CRs of 14, 15, 16, 17, 17.5 and 18, IPs of 150 bar, 200 bar and 250 bar, and varying loads from 0kg to 12 kg in steps of 3kg. The thermal performance parameters evaluated are BTHE, BSFC, BMEP, Volumetric Efficiency, HBP, HGas and EGT while the emission constituents measured are CO, HC, NO_x, CO_2, O_2 and SO_x. Along with emission constituents, the smoke intensity is also measured in terms of HSU. A detailed combustion analysis of the engine running on pure diesel and Karanja biodiesel at different preset compression ratios of 14, 16 and 18 and full load is also conducted. The combustion parameters analysed are cylinder pressure, net heat release rate, mass fraction burnt, mean gas temperature, rate of pressure rise, pressure volume plot, cumulative rate of heat release and injection pressure. Thus, the output parameters are BTHE, BSFC, BMEP, Volumetric Efficiency, HBP, HGas and EGT for thermal performance and CO, HC, NO_x, CO_2, O_2 and SO_x for emission constituents. Further, for optimization genetic algorithm (GA) analysis is carried out selecting only BTHE, BSFC and EGT as thermal performance parameters since other parameters directly influence the already selected once and CO, HC, NO_x, CO_2 and O_2 as emissions constituents. SO_x is not included as one of the output parameters in emission constituents as it is negligible in case of engine operated with Karanja biodiesel.

It should be noted that the three different studies conducted on thermal performance, emission constituents and combustion analysis leads to prediction of each of them in isolation, although a unification of thermal performance and emission constituents with combustion analysis is a possibility. However, it is difficult to strike an optimum combination of the possible maximum thermal performance and minimum emission constituents with respect to a combination of CR, IP, bio-diesel blend when the engine operates at a preset load (Say at full load) using only the experimental data. Hence, a suitable technique of optimization along with a modeling is to be chosen to strike an optimum balance between the chosen four input parameters to predict the output parameters.

Thus, a computational study consisting of multi-objective optimization of thermal performance and emission characteristics using GA technique and modeling using ANN for the engine is carried out. The combination of optimization and modeling is intended to find the optimum combination of the four input parameters, viz, CR, IP, load and blend and subsequently to predict the output parameters, viz. BTHE, BSFC and EGT for thermal performance and CO, HC, NO_x, CO_2 and O_2 for emission constituents for the optimum combination of input parameters. The GA toolbox of MATLAB is used for the purpose of optimization. The ANN is modeled by using the selected results of the experimental study using EASYNN-PLUS software. The accuracy with which this neural network works is judged by comparing the outputs from the network with the experimental data. The output parameters can be determined from the optimized input parameters.

Based on the experimental and computational studies, following are the important observations made and the conclusions drawn thereon.

1. A single cylinder, four stroke, variable compression ratio (VCR) CI engine originally designed to operate on diesel as fuel may be operated on pure Karanja biodiesel without any system hardware modifications.
2. Based on the experimental study, it can be concluded that with the increase in CR, the performance of diesel engine operated using Karanja biodiesel and its blends approach to that operated using Diesel oil. And at a higher CR and IP of 18 and 250 bar respectively, the thermal performance of Karanja biodiesel is closest to that of Diesel oil compared to that operated at other CRs of 14, 15, 16, 17 and 17.5. Higher IP of 250bar is preferable for Karanja biodiesel due to its higher viscosity. Thus, the best combination of input parameters in terms CR and IP will be 18 and 250 bar respectively when the engine is operated at full load. The experiments, however, can not specify any optimum Karanja biodiesel blend that will simultaneously maximize thermal performance and minimize emission constituents.
3. The thermal performance evaluation, in isolation, indicates that the blend B20 operates closest to that of Diesel oil whereas, the engine operated with any higher CR ranging from 16 to 18, the emission constituents of CO, HC, CO_2, O_2 and SO_x are the least and remains constant in the CR range of 16 to 18. Both the

above observations are made when the engine is operated with IP at 200 bar and full load at 12 kg.

4. On the basis of comparison made with earlier investigations with similar input parameters of CR, IP and load of 18, 200 bar and 12 kg (full load) respectively, the thermal performance of the engine operated with Karanja biodiesel is found to be superior in comparison with that operated using Jatropha and Mahua biodiesels. BTHE of the engine operated using Jatropha and Mahua biodiesels are found to be lesser by 12% and 17% respectively than that operated using Karanja biodiesel. BSFC is found to be 9% and 56% more respectively for Jatropha and Mahua biodiesels operated engines as compared to that of Karanja biodiesel used in the present study.
5. Karanja biodiesel (B100) gives minimum harmful emissions as compared to all other blends. Further, at a higher CR of 18 and IP of 250bar, the fairly reduced exhaust emissions are observed irrespective of the fuel blend used. Therefore, operating the diesel engine with Karanja biodiesel at a CR of 18 and IP of 250 bar results in minimum emissions but for NO_x emissions. If NO_x is also to be minimised, then the engine should be operated at CR of 16, which will result in a decrease in BTHE of about 13% and an increase in BSFC of 14% which are not affordable just for the sake of reduced NO_x emissions. A comparison of present study made with earlier investigations revealed that NO_x emission at a CR of 18 are found to be higher by 32% for Jatropha biodiesel as compared to that of Karanja biodiesel used in the present study
6. Although, B20 blend gives better thermal performance compared higher blends of Karanja, it is not recommended as it poses problems of higher levels of exhaust emissions.
7. It is inferred from the combustion analysis that as CR increases from 14 to 18 at full load and IP of 200bar, the difference in performance between Karanja biodiesel and Diesel oil reduces which results in almost the same level of thermal performance for the engine when fuelled with Karanja biodiesel as compared to that fuelled with Diesel oil. The combustion characteristics of diesel engine using Karanja biodiesel is similar to that using pure Diesel oil at higher CR of 18 which is very promising as far as Karanja biodiesel as an alternative fuel in diesel engines is concerned.

8. From the large number of experimental data for thermal performance and emission constituents obtained for various input parameters such as load, CR, IP and blend, picking up an optimum combination of the input parameters manually is not possible. The effects of blend proportion, load, compression ratio (CI) and injection pressure (IP) on thermal performance and emission constituents create a multi-objective scenario. Therefore, there is a need to find optimum conditions considering both thermal performance and emission constituents. GA is one optimizing technique that can help tackle the multi-objective scenario. Using GA as the optimization tool, it is concluded that the optimum values of CR, IP, and blend proportion are 18, 228 bar, and B70 respectively when the engine is operated at full load which results in maximizing the thermal performance by maximizing BTHE and minimizing BSFC and EGT and minimizing all other emission constituents including NO_x except O_2.
9. As the optimum values of input parameters, viz., CR, IP, and blend proportion are found to be 18, 228 bar, and B70 while operating the engine at full load using GA and the combination of these input parameters are not readily available through the hardware experiments conducted, it is necessary to use a modeling tool to predict the output parameters of thermal performance and emission constituents with an acceptable error limit. For this purpose, a multi layer feed forward ANN model with 80-20 rule for training the network is selected. From the comparison of a selected set of readings obtained from experimental study with that from ANN, it can be concluded that models developed for prediction of thermal performance and exhaust emission constituents have an acceptable error and hence can be used for obtaining the output parameters correspond to optimized input parameters. Further, if the engine is operated using B70 blend, the BTHE, BSFC and EGT have to be compromised by about 6%, 6% and 1.4% respectively compared to Diesel oil and the emission constituents namely CO, HC, CO_2, O_2 and NO_x by 20%, 50%, 5%, 0.6% and 3.6% respectively compared to Karanja biodiesel.

Appendix I

Karanja

Karanja is a medium sized tree that generally attains a height of about 8 meters and a trunk diameter of more than 50 cm. The trunk is generally short with thick branches spreading into a dense hemispherical crown of dark green leaves. The bark is thin gray to grayish-brown, and yellow on the inside. The tap root is thick and long, lateral roots are numerous and well developed.

Karanja tree grows mainly in Western India and near Mumbai. The tree is occasionally seen on roadsides in Peninsular India. It is indigenous species and available throughout India from the foothills of the Himalayas down to the south of the Peninsula, especially not far from the seacoasts. Native to the Asian subcontinent, this species has been introduced to humid tropical lowlands in Malaysia, Australia, the Seychelles, the United States and Indonesia. Karanja trees are green during summer and they add to natural beauty. They also provide shelter and cool air. Karanja trees are normally planted along the highways, roads and canals to stop soil erosion. If the seeds fallen along road side are collected, and oil is extracted at village level expellers, tons of oil will be available for lighting the lamps in rural area. It is the best oil for lighting.

Karanja is called as Koroch in Bangladesh. In Bangladesh, it is a fresh water flooded tree. The seedlings of Koroch can survive in 1.5 meters deep water submergence for five to six months duration at a stretch. There are nearly 30,000 square km of water reservoirs in India. This tree can be cultivated in such water storage reservoirs up to 1.5 meters depth and reap additional economic value from unused reservoir lands.

The leaves of Karanja tree are alternate, compound pinnate type consisting of 5 or 7 leaflets which are arranged in 2 or 3 pairs, and a single terminal leaflet (Refer Plate I.1). Leaflets are 5-10 cm long, 4-6 cm wide, and pointed at the tip. The flowers borne by these trees are pink, light purple, or white. Pods are elliptical in shape. They are 3-6 cm long and 2-3 cm wide, thick walled, and usually contains a single seed. Seeds are 10-20 cm long, oblong, and light brown in color.

Plate I.1 Leaves and Seeds of Karanja Tree

Karanja is one of the few nitrogen fixing trees to produce seeds containing 30-32% oil. Native to humid and subtropical environments, Karanja grows in areas having an annual rainfall ranging from 500 to 2500 mm. In its natural habitat, the maximum temperature ranges from 27 ^{o}C to 38 ^{o}C and the minimum 1 ^{o}C to16 ^{o}C. Mature trees can withstand water logging and slight frost.

Karanja trees can grow on most of the soil types ranging from stony to sandy to clay. It does not grow well on dry sands. It is highly tolerant of salinity and drought. Hence, it is commonly grown along waterways or seashores, with its roots in fresh or salt water. The growth rates are found to be fastest on well drained soils with assured moisture.

There are various uses of Karanja tree. Some of them are listed below

Wood: Karanja is commonly used as fuel wood. Its wood is medium or sometimes coarse textured. However, it is not durable and is susceptible to insect attack. Thus the wood is not considered a quality timber. The wood is used for cabinet making, cart wheels, agricultural implements, tool handles and combs.

Oil: A thick yellow-orange to brown oil is extracted from seeds. The oil is extracted from the seeds using mechanical expellers and crushers. The oil has a bitter taste and a disagreeable aroma, thus it is not considered edible. In India, the oil is used as a fuel for cooking and lighting lamps. The oil is also used as a lubricant, water-paint binder, pesticide etc. The oil is known to have value in folk medicine for the treatment of rheumatism, as well as human and animal skin diseases. It is effective in enhancing the pigmentation of skin affected by leucoderma or scabies.

Fodder and feed: The leaves of karanja trees are eaten by cattle and readily consumed by goats. However, in many areas it is not commonly eaten by farm animals. Its fodder value is greatest in arid regions. The oil cake, remaining when oil is extracted from the seeds, is used as poultry feed.

Other uses: Dried leaves are used as an insect repellent in stored grains. The oil cake, when applied to the soil adds pesticidal value to the soil, particularly against nematodes and also improves soil fertility.

The Karanja biodiesel used to conduct the experiments is commercially procured from MINT BIOFUELS, Pune. The process used by MINT BIOFUELS to produce Karanja biodiesel is shown in Figure I.1. The seeds of Karanja are crushed through an oil expeller, which separates the oil and cake. The oil is filtered and passed through a chemical reactor for mixing it with methanol in the presence of a catalyst. Here fatty acids are separated, and esterification process takes place. The end product is bio-diesel. By-product glycerin is separated and excess methanol is recovered. The by-products of esterification have got commercial utility.

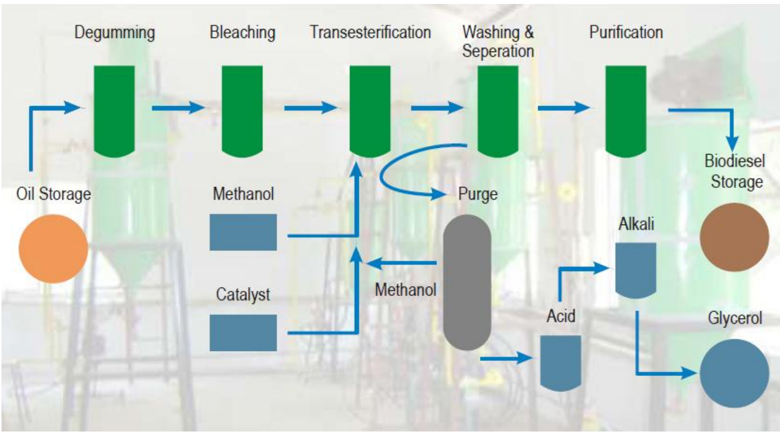


Figure I.1 Biodiesel Production Process

Figure I.2 shows the reaction of esterification process. During the esterification process, the vegetable oil is reacted with methyl alcohol in the presence of a catalyst, usually a strong alkaline like sodium hydroxide. The alcohol reacts with the fatty acids to form biodiesel (methyl ester) and crude glycerol.

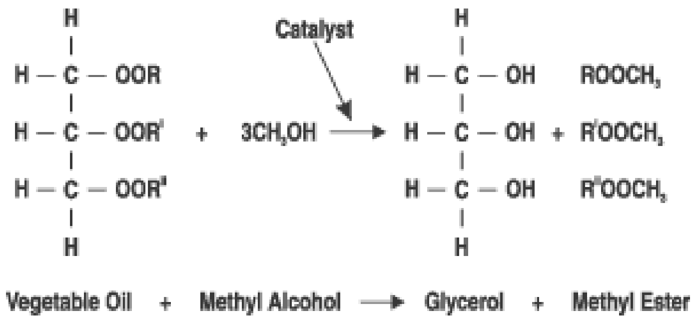


Figure I.2 Esterification of Vegetable Oil

Appendix II

Review Related Studies

Some noted investigators have authored various literature reviews which comprise of a collection of works conducted by various investigators around the globe. Such literature reviews are briefly explained in this appendix.

A.S. Ramadhas et al. [100] reviewed the studies conducted by many investigators from various countries using vegetable oils as I.C. engine fuel substitutes. The review includes papers from 1981 to 2002. The review indicates that vegetable oils posed some problems when subjected to prolonged usage in compression ignition engines because of their high viscosity and low volatility. The common problems were poor atomization, carbon deposits, ring sticking, fuel pump failure, etc. hence it was suggested to convert them into their blends with diesel or esters(known as biodiesels) to minimize these problems. The results showed that thermal efficiency was comparable to that of diesel engines with small amount of power loss while using vegetable oils. The particulate emissions of vegetable oils are higher than that of diesel fuel with a reduction in NO_X. Vegetable oil methyl esters gave performance and emission characteristics comparable to that of Diesel oil. Hence, they may be considered as diesel fuel substitutes. Raw vegetable oil can be used as fuel in diesel engines with some minor modifications. The use of vegetable oils as I.C. engine fuels can play a vital role in helping the developed world to reduce the environmental impact of fossil fuel.

E. Shahid and Y. Jamal [101] reviewed the use of biodiesels for compression ignition engines based on the study of reports of about 50 investigators who published their results between 1900 and 2005. These investigators used different types of raw and refined oils. It was reported that the use of raw vegetable oil as fuel did not show satisfactory results and caused problems of injector coking and piston ring sticking. However, transesterified biodiesel fuels brought down the properties like density, viscosity and flash point as compared to raw biofuels and comparable to that of Diesel oils. The transesterified fuels were blended with Diesel oils in different proportions. Blends above 20% showed maintenance problems & resulted in engine wear when used for long term applications. Some authors reported success in using vegetable oils as diesel fuel extenders in blends of more than 20% even in long term studies. It was reported that there was a slight decrease in brake power and a slight increase in fuel consumption with biodiesel applications. Lubricating properties of biodiesels are better than diesel which can help to increase the engine life and were found to be environment friendly with much less produce of NO_x and HC emissions, absolutely no SO_x and with CO_2 level maintained almost a constant. It was concluded in the review that vegetable oils in refined, degummed, dewaxed & transestrified form can replace Diesel oil. Particularly rapeseed and palm oil are most suitable as diesel fuel extenders. Unmodified vegetable oils can be used only for small engines for a short term period. For long term use for heavy engines either biodiesels or their blends with diesels are recommended. They observed that Indirect fuel injection systems are more successful with vegetable oil applications as compared to direct injection systems. They report that the engine is to be started with Diesel oil alone and after warming up could be shifted to biodiesel-diesel blends. More care, regularity in maintenance and periodic services of the engine is required. Biodiesel generates much less pollutant gases as compared to diesel. Hence it would be better to use B100 in the urban area applications in particular. Studies on cottonseed, rapeseed, soybean, palm, peanut oils and other miscellaneous studies of vegetable oils in dual fuel mode conducted by various investigators are taken for comparison.

Y.C. Sharma et al. [102] concluded from his review that biodiesel studies are mainly from edible and non edible oils. The review consists of studies from 1978 to 2008. The main advantage in the usage of biodiesels is attributed to lesser exhaust emissions in terms of carbon monoxide, hydrocarbons, particulate matter, polycyclic aromatic hydrocarbon compounds and Nitrited polycyclic aromatic hydrocarbon compounds. Transesterification is the process successfully employed at present to reduce the viscosity of biodiesel and improve other characteristics. Methanol being cheaper is the commonly used alcohol during transesterification reaction. Among the catalysts, homogeneous catalysts such as sulphuric acid, sodium hydroxide, potassium hydroxide are commonly used at industrial level production of biodiesel. Heterogeneous catalysts such as calcium oxide, magnesium oxide and others are also being tried to decrease the catalyst amount and production cost of biodiesel. Transesterification reaction can be completed even without catalyst by using supercritical methanol but it will increase the production cost of biodiesel as it is energy intensive. The molar ratio of alcohol to oil required is 3:1 by stoichiometry, but excess molar ratio has been used for biodiesel production & for better yield in lesser time. The molar ratio employed during acid esterification is between 6:1 and 18:1 whereas the molar ratio used in alkaline transesterification ranges between 5:1 and 12:1 after reducing the acid value to less than 2.0% approximately. The temperature ranges between 318 and 338 K as the boiling point of methanol is 337.7 K and heating beyond this temperature would burn methanol. However, higher temperature is employed while using supercritical methanol (473–573 K). Depending on the feedstock taken; amount & type of alcohol and catalyst; temperature employed; mode and rate of stirring; there is difference in the yield of biodiesel which varies from 80 to 100%.

Another added advantage of biodiesel is that it is biodegradable in nature. When used as blend along with diesel fuel, it shows positive synergic effect of biodegradation by means of co- metabolism. Major disadvantage of biodiesel is the inverse relationship of oxidation stability of biodiesel with its low temperature properties which includes cloud point and pour point. Higher composition of saturated fatty acids in feedstock will increase the oxidation stability of biodiesel but will lower its cloud and pour points. Whereas, higher composition of unsaturated fatty acids will enhance the cloud point and pour point of biodiesel but will have poor oxidation stability. Hence, a balance has to be maintained between the ratio of saturates and unsaturates, for the oil to be used as a feedstock for biodiesel production.

From Indian perspective, they observed that as India is rich in plant biodiversity, there are many plant species whose seeds remain unutilized and underutilized. They are being tried for biodiesel production. These species have shown promises and fulfil the requirements & meet various biodiesel standards. However, they also state that there still is paucity in terms of all the standards which should be fulfilled for their large commercial application and acceptance from public and governing bodies.

A.Murugesan et al. [103] presented a review of the prospects and opportunities of introducing vegetable oils and their derivatives i.e., bio-diesel as fuels in diesel engines. The review is done for researches conducted from 1981 to 2007. It was highlighted in this review that vegetable oils are very promising as alternative fuels in diesels engines because their properties are comparable with those required to operate diesel engines. Optimization of alkali based transesterification of Pongamia Pinnata oil for the production of bio-diesel was discussed. They observed that bio-diesel could bring down the emissions to a safer level. The suitability of injection timing indicated that it should be retarded. A review of performance of bio-diesel blends indicated that the

BSFC is about 8% to 10% higher than that of Diesel oil because of a decrease in the calorific value of fuel with an increase in bio-diesel percentage in the blends. From the review, they concluded that bio-diesels blended with diesel is a suitable strategy for saving of fossil fuel consumption and B20 is best alternative fuel for diesel in conformity with performance and emission with base line data of diesel.

S. Basha and K. RajaGopal [104] authored a literature review which was based on biodiesel production, combustion, performance and emissions. This study was based on the reports of about 130 scientists who published their results between 1980 and 2008. It was concluded in the review that biodiesel is one of the best available sources to fulfil the energy demand of the world. Though more than 350 oil bearing crops were identified ,but among them only few like sunflower, rapeseed, palm and jatropha were considered as potential alternative fuels for diesel engines according to this review. Many of the investigators mentioned that short term engine test using vegetable oils are promising but long term test showed long carbon build up, lubricating oil contamination resulting in engine failure. Hence blending with diesel or chemically altering the oil into esters (bio-diesel) was carried out. Running in this manner on a CI engine provides engine power output comparable to that of base line data which is obtained using diesel as fuel. It was reported that the combustion characteristics of biodiesel are similar as diesel and blends were found to have shorter ignition delay, higher ignition temperature, higher injection pressure and peak heat release. The engine power output was found to be equivalent to that of diesel fuel. In addition, it was observed that the base catalysts are more effective than acid catalysts and enzymes. The tests with refined oil blends indicated considerable improvement in performance. The emission of unburnt hydrocarbon from the engine was found to be more on the all the fuel blends as compared to diesel. The emission of oxides of nitrogen from the engine were found to be higher on all fuel blends as compared to Diesel oil.

H. C. Ong et al. [105] reviewed and compared the performance and emissions of Jatropha, Palm and Penaga Laut biodiesels. The review is conducted for studies carried out between the years 1954-2010. Penaga Laut is also known as Callophyllum Inophyllum and is found in Malaysia, India, Myanmar etc. In India it is popularly known as Polanga. From the review, it was observed that most of the research work conducted on biodiesels by various investigators showed combustion characteristics of biodiesels are same as diesel and biodiesel blends can reduce hydrocarbons, smoke opacity, particulate matters, carbon dioxide and carbon monoxide emissions but slightly higher NO_x emissions. The exhaust emissions of NO_x can be controlled by adopting techniques such as changing the composition of biodiesels in the blend, improve cetane number of biodiesel, retardation of injection timing and exhaust gas recirculation. The high viscosity and low volatility of vegetable oils cause poor combustion in diesel engines. Several ways to reduce the viscosity of vegetable oils have been discovered by various investigators. They are preheating of oil, micro emulsion with solvents, dilution, blending with diesel, thermal cracking and transesterification. Thermal cracking is also called as pyrolysis. Palm is widely produced in Malaysia hence it is economically viable to use palm oil in that part of the world. Palm is one of the most oil bearing crops in terms of land utilisation, efficiency and productivity. But, when compared with non edible oils, edible oils cannot be preferred as they are the sources of food. This shifts the attention to nonedible oils like Jatropha and Polanga which are grown in tropical and subtropical climatic region countries. Among the non edible oils the most preferred one is Jatropha oil as it is a drought resistant plant and can be cultivated success fully in wastelands. Polanga can also be considered as a as a potential biodiesel fuel and could be transesterified. However, compared to Palm and Jatropha biodiesel industry, production of polanga biodiesel is still at a nascent stage.

The different review related studies conducted on biodiesels are shown in a chronological order in Table II.1. Almost all studies reveal that biodiesels blended with diesel in a definite proportion can be used in existing diesel engines. But the disadvantage of using biodiesel is that it gives higher percentage of NO_x emissions compared to diesel, however the other harmful emissions are less with biodiesels. Even though NO_x emissions are higher with biodiesels they are not appreciably higher compared to diesel. The reviews also validate the fact that transesterification is the most accepted and widely used method for production of biodiesels as it yields biodiesel whose properties are closer to that of conventional diesel. The yield percent of biodiesel is more using transesterification method.

Table II.1 Summary of Review Related Studies Carried Out On Biodiesels

Sr. No	Investigators	Year	Observation made from the review
1	A.S. Ramadhas et al.	2004	The review showed that vegetable oils posed some problems when subjected to prolonged usage in compression ignition engines because of their high viscosity and low volatility.
2	E. Shahid and Y. Jamal	2007	The review showed that indirect fuel injection systems are more successful with vegetable oil applications as compared to direct injection systems. It was also reported that the engine is to be started with diesel alone and after warming up could be shifted to biodiesel-diesel blends.
3	Y.C. Sharma et al.	2008	Transesterification is the process successfully employed at present to reduce the viscosity of biodiesel and improve other characteristics.
4	A.Murugesan et al.	2009	A review of performance of bio-diesel blends indicated that the BSFC is about 8% to 10% higher than that of diesel because of a decrease in the calorific value of fuel with an increase in bio-diesel percentage in the blends.
5	S. Basha and K. Raja Gopal	2009	The engine power output was found to be equivalent to that of diesel fuel. In addition, it was observed that the base catalysts are more effective than acid catalysts and enzymes
6	H. C. Ong et al.	2011	The exhaust emissions of NO_x can be controlled by adopting techniques such as changing the composition of biodiesels in the blend, improve cetane number of biodiesel, retardation of injection timing and exhaust gas recirculation.

Appendix III

Sample Experimental Data

This appendix gives experimental data for few selected test runs conducted to study thermal performance and emission constituents for selected compression ratios, injection pressures and loads. B100 indicates pure Karanja biodiesel.

Table III.1 Thermal performance data of the VCR diesel engine running on Diesel oil (Diesel, diesel) and different blends of diesel and biodiesel

1. **Compression ratio : 18**
2. **Injection pressure : 200 bar**

Load (kg)	Fuel	BMEP (bar)	BSFC (kg/kWh)	BTHE (%)	Vol. Eff (%)	HBP (%)	HGas (%)	EGT (^{0}C)
0	Diesel	0.01	27.80	0.45	89.39	0.33	22.2	175.67
3	Diesel	0.95	0.6	15.72	89.59	15.04	23.51	196.27
6	Diesel	1.73	0.42	23.15	89.29	21.46	22.78	237.55
9	Diesel	2.55	0.35	26.17	89.14	26.17	23.13	288
12	Diesel	3.36	0.3	31.25	87.6	29.29	24.08	350.75
0	B20	0.02	21.25	0.43	93.76	0.43	45.86	185.56
3	B20	1.08	0.61	15.26	93.93	15.72	32.93	210.96
6	B20	2.1	0.45	22.35	93.57	22.59	31.57	254.67
9	B20	3.11	0.3	25.86	93.36	29.68	30.9	289.53
12	B20	4.11	0.31	30.5	92.59	30.5	29.03	348.56
0	B40	0.02	20.13	0.4	94.44	0.45	36.82	180.65
3	B40	1.15	0.63	15.04	94.44	15.26	28.71	201.85
6	B40	2.05	0.4	22.7	94.07	22.7	29.2	239.68
9	B40	3.12	0.3	25.42	93.59	29.64	30.56	286.53
12	B40	4.16	0.32	30.05	93.19	32.63	31.18	347.96
0	B60	0.01	34.36	0.38	90.61	0.26	29.69	182.89

3	B60	1.16	0.65	14.82	90.62	12.97	21.67	208.23
6	B60	2.12	0.44	20.38	90.42	20.38	22.72	247.9
9	B60	3.08	0.35	25.08	90.25	25.85	24.49	299.65
12	B60	4.15	0.32	29.32	89.95	29.32	24.92	356.04
0	B80	0.02	25.4	0.36	92.98	0.36	39.99	187.53
3	B80	1.12	0.67	13.63	92.32	14.82	29.28	209
6	B80	2.1	0.39	20.35	92.09	23.21	29	242.69
9	B80	3.11	0.36	25.46	92.08	34.18	35.14	290.62
12	B80	4.12	0.33	28.75	91.17	32.25	30.35	346.99
0	B100	0.01	22.05	0.32	89.65	0.41	27.3	179.35
3	B100	1.04	0.7	12.58	89.72	13.63	21.97	198.69
6	B100	2.08	0.45	20.03	89.47	20.03	21.7	240.38
9	B100	3.07	0.37	24.39	89.45	24.39	21.46	281.85
12	B100	4.14	0.34	28.65	89.19	26.64	21.71	343.77

Table III.2 Thermal performance data of the VCR diesel engine running on Diesel oil and different blends of diesel and biodiesel

1. **Load : 12kg**
2. **Injection pressure : 200 bar**

Compression ratio	Fuel	BMEP (bar)	BSFC (kg/kWh)	BTHE (%)	Vol. Eff (%)	HBP (%)	HGas (%)	EGT (^{0}C)
14	Diesel	4.08	0.31	27.78	88.63	27.76	41.57	423.71
15	Diesel	4.06	0.29	30.69	89.27	30.69	27.8	407.05
16	Diesel	4.09	0.31	29.01	88.89	29.01	24.92	398.1
17	Diesel	4.13	0.3	30.56	89.01	30.56	25.92	383.77
17.5	Diesel	4.11	0.29	30.66	88.52	30.66	24.79	366.92
18	Diesel	4.06	0.31	29.2	87.6	29.29	24.08	350.75
14	B20	4.12	0.32	25.89	92.2	26.49	32.2	420.87

15	B20	4.08	0.31	28.96	92.3	29.62	31.57	406.25
16	B20	4.09	0.32	27.57	91.65	28.89	29.81	396.76
17	B20	4.1	0.31	29.65	92.98	28.87	34.96	377.98
17.5	B20	4.12	0.29	29.16	93.39	29.16	30.02	360.67
18	B20	4.11	0.32	28.54	92.59	28.84	29.03	348.56
14	B40	4.1	0.34	24.87	93.69	25.48	31.71	419.6
15	B40	4.14	0.32	26.34	93.75	28.54	31.53	400.45
16	B40	4.16	0.33	26.98	93.61	28.67	32.38	390.67
17	B40	4.1	0.32	27.45	93.4	28.51	30.44	372.79
17.5	B40	4.14	0.31	28.12	93.34	28.34	31.58	358.35
18	B40	4.16	0.32	27.95	93.19	28.01	31.18	347.96
14	B60	4.11	0.35	23.79	89.88	24.05	24.22	418.46
15	B60	4.12	0.34	24.89	89.87	27.59	24.45	398.67
16	B60	4.14	0.33	25.87	87.15	28.04	24.61	386.98
17	B60	4.13	0.31	26.43	87.05	28.97	23.51	367.89
17.5	B60	4.14	0.32	26.46	88.56	27.67	22.72	356.78
18	B60	4.15	0.33	27.03	89.95	27.45	24.92	347.3
14	B80	4.12	0.37	22.65	92.34	23.59	31.48	417.39
15	B80	4.08	0.35	23.78	92.22	26.41	32.02	395.8
16	B80	4.14	0.34	24.65	93.76	27.54	29.55	383.98
17	B80	4.1	0.33	25.76	90.9	27.79	29.71	364.01
17.5	B80	4.11	0.31	26.01	90.95	27.56	29.26	354.25
18	B80	4.12	0.34	26.98	91.17	26.98	30.35	346.99
14	B100	4.11	0.39	21.33	89.23	23.04	23.39	415.87
15	B100	4.13	0.36	22.67	89.41	25.13	24.31	390.61
16	B100	4.25	0.35	23.98	89.2	25.57	24.32	380.15
17	B100	4.09	0.33	24.61	92.34	26.48	31.48	360.75
17.5	B100	4.07	0.32	25.61	89.76	26.46	23.87	350.78

18	B100	4.14	0.34	26.64	89.19	26.64	21.71	343.77

Table III.3 Thermal performance data of the VCR diesel engine running on Diesel oil and different blends of diesel and biodiesel

1. **Load : 12kg**
2. **Compression ratio : 18**

Injection Pressure (bar)	Fuel	BMEP (bar)	BSFC (kg/kWh)	BTHE (%)	Vol. Eff (%)	HBP (%)	HGas (%)	EGT (^{0}C)
150	Diesel	4.09	0.32	31.34	88.68	23.56	25.87	326.24
200	Diesel	4.06	0.3	31.9	88.51	29.29	24.08	350.75
250	Diesel	4.11	0.28	32.7	89.29	39.41	23.98	370.26
150	B20	4.09	0.33	29.85	90.67	23.45	28.9	324.7
200	B20	4.11	0.31	30.5	92.59	28.84	29.03	348.92
250	B20	4.12	0.29	31.25	91.76	37.41	27.46	368.9
150	B40	4.15	0.33	29.64	94.44	22.15	32.9	323.56
200	B40	4.16	0.31	30.25	93.19	28.01	31.18	346.89
250	B40	4.18	0.29	31.09	94.07	37.1	33.87	366.9
150	B60	4.13	0.34	28.64	89.85	21.09	23.89	321.9
200	B60	4.15	0.32	29.89	89.95	27.45	24.92	345.78
250	B60	4.17	0.3	29.87	89.97	36.23	25.67	366.09
150	B80	4.1	0.35	28.03	92.98	21.15	29.89	320.76
200	B80	4.12	0.33	29.65	91.17	26.98	30.35	344.87
250	B80	4.15	0.32	29.02	92.09	34.96	31.2	365.23
150	B100	4.15	0.35	27.46	89.65	20.98	21.23	319.56
200	B100	4.14	0.34	28.09	89.19	26.64	21.71	343.77
250	B100	4.18	0.32	28.83	89.47	32.78	23.98	364.59

Table III.4 Thermal performance data of the VCR diesel engine running on Diesel oil and different blends of diesel and biodiesel

1. **Load : 12 kg**
2. **Compression ratio : 18**
3. **Injection pressure 200 bar**

Fuel	BMEP (bar)	BSFC (kg/kWh)	BTHE (%)	Vol. Eff (%)	HBP (%)	HGas (%)	EGT (^{0}C)
Diesel	3.36	0.3	31.25	88.51	29.29	24.08	350.75
B20	4.11	0.31	30.5	92.59	30.5	29.03	348.56
B40	4.16	0.32	30.05	93.19	32.63	31.18	347.96
B60	4.15	0.32	29.32	89.95	29.32	24.92	356.04
B80	4.12	0.33	28.75	91.17	32.25	30.35	346.99
B100	4.14	0.34	28.65	89.19	26.64	21.17	343.77

Table III.5 Exhaust emission data of the VCR diesel engine running on Diesel oil and different blends of diesel and biodiesel

1. **Compression ratio : 14**
2. **Injection pressure : 200 bar**

Load (kg)	Fuel	CO_2 (%)	CO (%)	NO_x (ppm)	HC (ppm)	O_2 (%)	SO_x (ppm)	HSU (%)
0	Diesel	0.83	0.012	9	6	19.81	23	23
3	Diesel	1.34	0.0136	37	5	18.99	11	11
6	Diesel	1.65	0.0148	75	5	18.67	5	5
9	Diesel	2.01	0.0203	118	6	18.18	5	5
12	Diesel	2.57	0.022	160	8	17.51	5	5
0	B20	0.88	0.011	15	5	19.58	19	19
3	B20	1.4	0.012	42	4	18.97	8	8
6	B20	1.7	0.017	81	5	18.42	4	4
9	B20	2.11	0.018	118	6	18.22	4	4

12	B20	2.6	0.02	161	7	17.56	3	3
0	B40	0.87	0.01	15	4	19.87	15	15
3	B40	1.47	0.011	38	4	19.04	6	6
6	B40	1.75	0.013	82	3	18.69	4	4
9	B40	2.17	0.015	124	4	18.25	3	3
12	B40	2.65	0.016	163	6	17.43	2	2
0	B60	0.89	0.008	15	3	19.91	13	13
3	B60	1.52	0.009	44	3	19.15	5	5
6	B60	1.79	0.012	81	2	18.67	5	5
9	B60	2.19	0.015	129	4	18.21	3	3
12	B60	2.63	0.015	167	5	17.69	1	1
0	B80	0.92	0.005	17	2	19.94	6	6
3	B80	1.58	0.007	46	3	19.08	3	3
6	B80	1.82	0.013	84	2	18.56	2	2
9	B80	2.18	0.011	132	3	17.96	1	1
12	B80	2.68	0.013	175	4	17.59	0	0
0	B100	0.96	0.004	19	2	20.03	4	4
3	B100	1.6	0.006	58	3	19.33	3	3
6	B100	1.84	0.012	87	2	18.95	1	1
9	B100	2.24	0.009	135	2	18.44	1	1
12	B100	2.7	0.011	179	4	17.62	0	0

Table III.6 Exhaust emission data of the VCR diesel engine running on Diesel oil and different blends of diesel and biodiesel

1. **Load : 12 kg**
2. **Injection pressure : 200 bar**

Compression ratio	Fuel	CO_2 (%)	CO (%)	NO_x (ppm)	HC (ppm)	O_2 (%)	SO_x (ppm)	HSU (%)
14	Diesel	2.72	0.043	225	18	15.79	76	17.1
15	Diesel	2.48	0.0293	217	16	16.67	23	13.3
16	Diesel	2.4	0.0205	186	12	17.42	13	11.1
17	Diesel	2.45	0.0166	188	7	17.63	7	9.8
17.5	Diesel	2.57	0.015	156	6	17.62	5	9.1
18	Diesel	2.62	0.0152	125	5	17.62	9	9.4
14	B20	2.75	0.049	177	17	17	60	19.8
15	B20	2.65	0.037	170	14	17.23	35	14.5
16	B20	2.54	0.018	151	8	17.41	19	13.2
17	B20	2.55	0.009	149	7	17.5	13	11.2
17.5	B20	2.52	0.004	146	6	17.59	5	11.2
18	B20	2.78	0.012	130	6	17.47	9	11
14	B40	2.82	0.053	162	16	17.26	64	16.1
15	B40	2.7	0.022	150	13	17.49	29	12.5
16	B40	2.68	0.018	168	8	17.31	17	11.4
17	B40	2.64	0.009	158	6	17.54	7	10.6
17.5	B40	2.75	0.007	138	4	17.69	5	10.5
18	B40	2.7	0.007	140	5	17.67	5	10.9
14	B60	2.88	0.068	149	13	17.34	64	17.9
15	B60	2.71	0.027	143	11	17.47	39	15.5
16	B60	2.75	0.008	165	6	17.4	11	13.7
17	B60	2.65	0.009	166	5	17.32	9	12.8
17.5	B60	2.84	0.008	167	4	17.43	9	12.1

18	B60	2.8	0.007	146	5	17.5	9	12.4
14	B80	2.96	0.079	140	14	17.37	60	16.4
15	B80	2.75	0.019	139	10	17.41	29	14
16	B80	2.75	0.017	145	5	17.64	23	12.3
17	B80	2.71	0.011	146	4	17.21	15	12.1
17.5	B80	3.05	0.01	136	3	17.56	15	12
18	B80	3.1	0.006	160	3	17.56	13	12
14	B100	3.26	0.084	132	12	16.81	127	19.9
15	B100	2.71	0.049	135	8	17.09	60	17.3
16	B100	2.86	0.018	173	5	17.29	27	13.8
17	B100	2.85	0.011	179	3	17.45	9	12
17.5	B100	3.15	0.007	179	3	17.51	3	11.7
18	B100	3.26	0.005	166	2	17.91	3	11.8

Table III.7 Exhaust emission data of the VCR diesel engine running on Diesel oil and different blends of diesel and biodiesel

1. **Load : 12 kg**
2. **Compression ratio : 18**

Injection Pressure (bar)	Fuel	CO_2 (%)	CO (ppm)	NO_x (ppm)	HC (ppm)	O_2 (%)	SO_x (ppm)	HSU (%)
150	Diesel	2.01	181	125	24	16.7	9	16.4
200	Diesel	2.35	152	149	12	17.62	5	9.4
250	Diesel	2.6	110	184	6	18.69	2	6.9
150	B20	2.05	163	127	20	16.8	10	17
200	B20	2.37	120	153	10	17.47	3	10.56
250	B20	2.62	82	185	5	18.75	2	7.5
150	B40	2.1	137	132	17	17.08	8	17.4

200	B40	2.39	100	155	9	17.67	2	11.09
250	B40	2.7	70	187	4	18.76	1	7.9
150	B60	2.14	121	135	14	17.26	5	17.9
200	B60	2.48	85	159	6	17.5	1	11.25
250	B60	2.75	59	190	3	18.98	2	8.1
150	B80	2.2	101	140	12	17.14	3	18.12
200	B80	2.55	76	163	6	17.56	0	11.56
250	B80	2.8	50	195	3	18.87	1	8.3
150	B100	2.24	86	137	10	17.3	2	18.43
200	B100	2.59	60	166	5	17.91	0	11.8
250	B100	2.82	45	198	3	19.1	1	8.5

Table III.8 Exhaust emission data of the VCR diesel engine running on Diesel oil and different blends of diesel and biodiesel

1. **Load : 12 kg**
2. **Compression ratio : 18**
3. **Injection pressure : 200 bar**

Fuel	CO_2 (%)	CO (ppm)	NO_x (ppm)	HC (ppm)	O_2 (%)	SO_x (ppm)	HSU (%)
Diesel	2.62	0.0152	125	5	17.62	9	9.4
B20	2.78	0.012	130	6	17.47	9	11
B40	2.7	0.007	140	5	17.67	5	10.9
B60	2.8	0.007	146	5	17.5	9	12.4
B80	3.1	0.006	160	3	17.56	13	12
B100	3.26	0.005	166	2	17.91	3	11.8

Table III.9.1: Airfuel ratio data of the VCR Diesel Engine running on Diesel Oil and different Blends of Diesel and Biodiesel

1. Load : 12kg
2. IP : 200bar

CR	HSD	B20	B40	B60	B80	B100
14	36.94	28.87	29.22	21.86	28.99	20.78
15	27.51	30.41	30.88	23.85	30.36	22.72
16	26.04	28.72	32.55	25.13	28.73	23.69
17	27.46	35.96	32.57	25.12	29.95	28.99
17.5	27.32	32.22	34.24	25.17	30.06	25.06
18	25.94	32.13	34.18	26.43	33.35	23.81

Table III.9.2: Airfuel ratio data of the VCR Diesel Engine running on Diesel Oil and different Blends of Diesel and Biodiesel

1. CR : 18
2. IP : 200 bar

Load	Diesel	B20	B40	B60	B80	Karanja biodiesel
0	68.64	108.16	108.52	70.14	91.57	79.11
3	49.3	63.83	58.46	42.31	57.26	45.34
6	38.18	41.62	48.72	36.24	47.69	35.83
9	31.09	32.13	41.64	31.53	47.27	29.5
12	25.94	27.86	34.18	26.43	33.35	23.81

Table III.9.3 : Airfuel ratio data of the VCR Diesel Engine running on Diesel Oil and different Blends of Diesel and Biodiesel

1. Load : 12kg
2. CR : 18

IP	Diesel	B20	B40	B60	B80	Karanja biodiesel
150	25.2	27.1	33.5	25.9	32.6	23.2
200	25.94	27.86	34.18	26.43	33.35	23.81
250	26.8	28.5	34.9	27.1	33.56	24.4

VARIATION OF AIRFUEL RATIO WITH COMPRESSION RATIO, LOAD AND INJECTION PRESSURE.

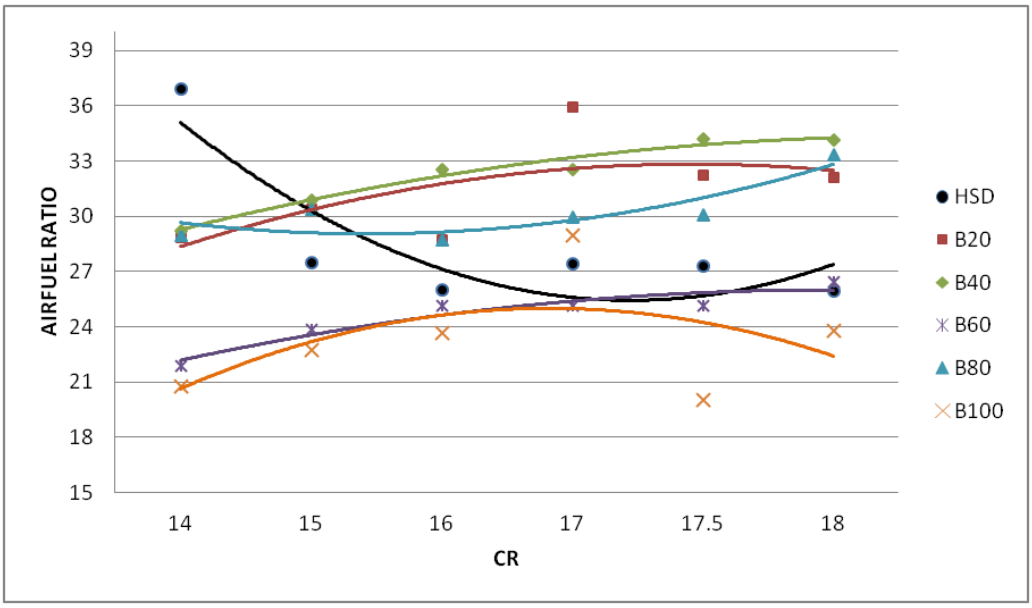


Figure 1: Variation of Airfuel ratio with CR at a load of 12kg and IP of 200bar

It is seen that(Figure 1) as CR increases (Figure 1) from 14 to 18 A/F ratio decreases for diesel from 35 to 27 initially upto 15CR and is maintained the almost same at about 26 at all higher CRs. Similar trend of constant Airfuel ratio of about 24 is maintained at higher CRs for B100. A lower Airfuel for biodiesel infers slightly higher rate of fuel consumption for biodiesel as compared to diesel. B20 behaves closer to diesel at higher CRs. Relatively higher A/Fs for fuel blends may be due to dissimilar combustion characteristics of diesel and biodiesel. A/Fs show an increase in trend with CR for biodiesel and other blends due to improved combustion characteristics because of higher temperature of compressed air. Lower the blend proportion higher the A/F ratio since the behaviour is closer to diesel.

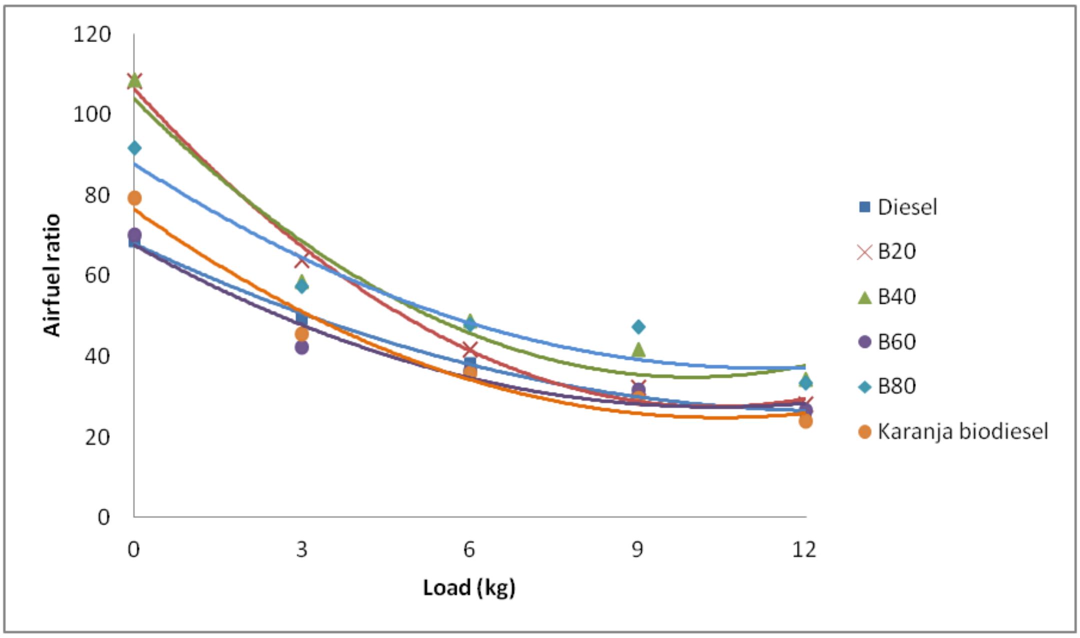


Figure 2: Variation of Airfuel ratio with load at CR of 18 and IP of 200bar

The plot below (Figure 2) shows the variations of the Air-fuel ratio with Load,. With increase in load it is observed that the Air-fuel ratio show a decreasing a trend for all tested fuels, indicating that a richer mixture is required at higher loads. At low loads the Air-fuel ratio is about 10% higher for biodiesel as compared. But as load increases the A/F requirement is the same for both the fuels, indicating comparable performance at higher loads. B20 behaves closer to diesel at higher loads as expected.

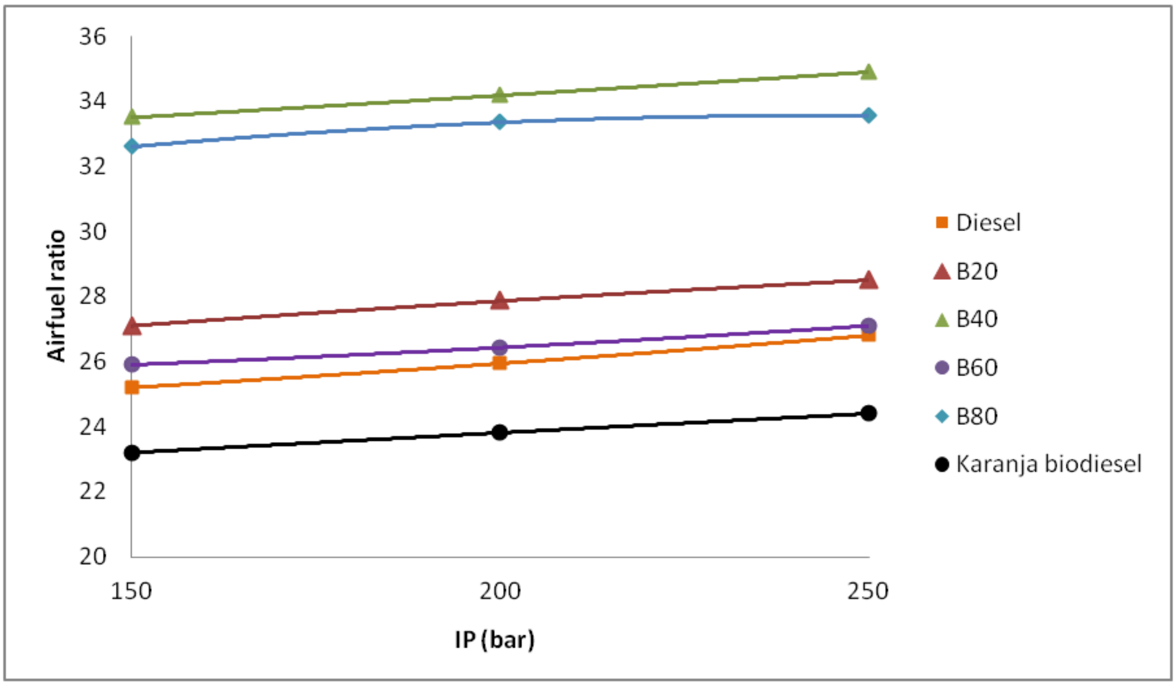


Figure 3: Variation of Air-fuel ratio with IP for full load of 12kg and CR of 18

It is observed that A/F increases (Figure 3) marginally with increase in injection pressure for all the fuels tested at a load of 12kg and CR of 18. This indicates that higher injection pressure results in better atomization of fuel and relatively higher the A/F ratio required which indicates that the engine runs on relatively lean mixture to take up the same load at higher CRs.

Appendix IV

Specimen Calculations

The specimen calculations of BTHE, BSFC, BMEP and Volumetric efficiency are given in the appendix. The calculations shown are for the reading corresponding to CR of 18 and fuel as Diesel oil given in Table III.2 in Appendix III.

IV.1 Data

IC Engine set up under Test: Single cylinder, VCR Diesel having power 3.50 kW, Four stroke , Constant Speed, Water Cooled, Diesel Engine,

Speed: 1500 rpm,

Cylinder Bore Diameter: 87.50 mm,

Stroke Length: 110.00 mm,

Connecting Rod Length: 234.00 mm,

Compression Ratio: 18.00,

Swept Volume: 661.45 cc

Brake Power: 3.36 KW

Fuel Flow Rate: 1.04 kg/hr

Air Flow Rate: 28.63 kg/hr

Calorific Value of fuel: 42000 kJ/kg

Density of Air (ρ_a): 1.127 kg/cu m

IV. 2 Brake Mean Effective Pressure

$$BMEP = \frac{BP\ (KW) \times 60}{L \times A \times \left(\frac{N}{n}\right) \times No\ of\ cylinders \times 100}$$

$$= \frac{3.36 \times 60}{0.11 \times \left(\frac{\pi}{4}\right) \times 0.0875^2 \times \left(\frac{1472}{2}\right) \times 100}$$

$$= 4.14\ bar$$

IV.3 Brake Specific Fuel Consumption

$$BSFC = \frac{Fuel\ flow\ in\ \frac{kg}{hr}}{BP}$$

$$= \frac{1.04}{3.36}$$

$$= 0.31\ \frac{kg}{kWh}$$

IV.4 Brake Thermal Efficiency

$$\eta_{bth} = \frac{BP \times 3600 \times 100}{Fuel\ flow\ in\ \frac{kg}{hr} \times Calorific\ Value}$$

$$= \frac{3.36 \times 3600 \times 100}{1.04 \times 42000}$$

$$= 28\ \%$$

IV.5 Volumetric efficiency

$$\eta_{vol} = \frac{Airflow\ in\ kg/hr \times 100}{\left(\frac{\pi}{4}\right) \times D^2 \times L \times \left(\frac{N}{n}\right) \times 60 \times No\ of\ cylinders \times \rho_a}$$

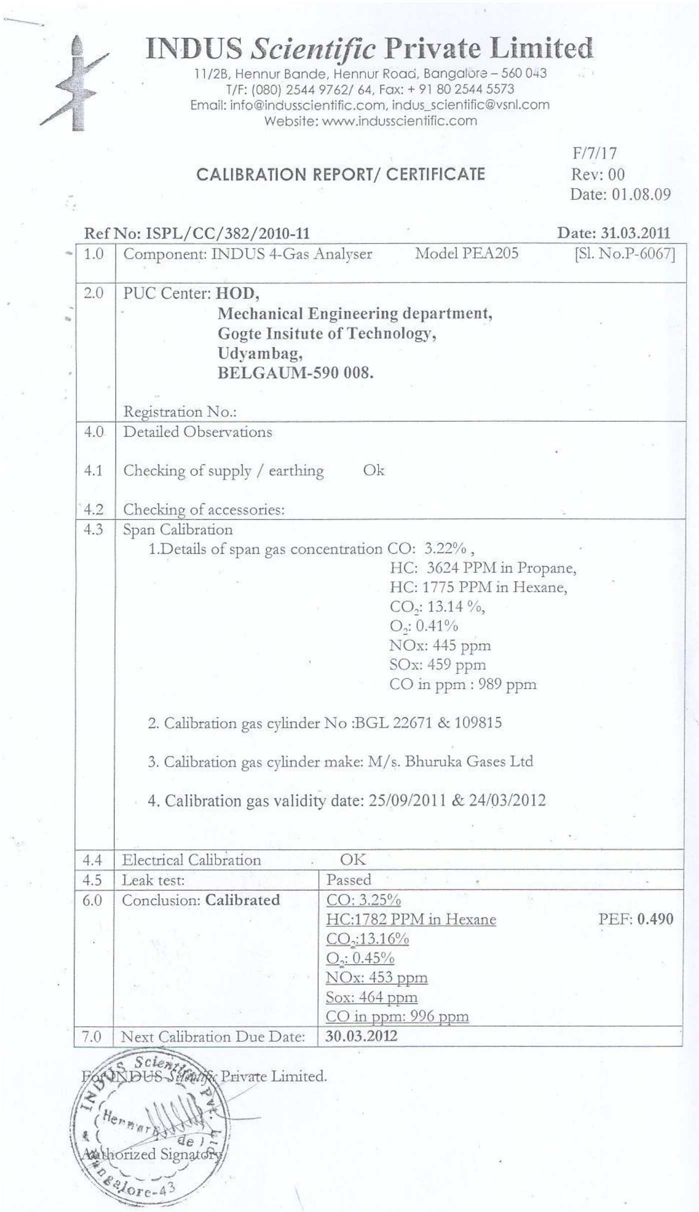
$$= \frac{28.63 \times 100}{\left(\frac{\pi}{4}\right) \times 0.0875^2 \times 0.11 \times \left(\frac{1472}{2}\right) \times 60 \times 1 \times 1.127}$$

$$= 87\%$$

Appendix V

Calibration Certificates

The appendix consists of calibration certificates for the multi gas analyser, smoke meter, pressure sensor & certificate of properties of Karanja biodiesel. The multi gas analyser and smoke meter, are of make INDUS Scientific Pvt. Ltd, Bengaluru. The pressure sensor is of make PCB Peizotronics and the biodiesel is commercially procured from MINT Biofuels, Pune.



INDUS *Scientific* Private Limited
11/2B, Hennur Bande, Hennur Road, Bangalore – 560 043
T/F: (080) 2544 9762/ 64, Fax: + 91 80 2544 5573
Email: info@indusscientific.com, indus_scientific@vsnl.com
Website: www.indusscientific.com

F/7/17
Rev: 00
Date: 01.08.09

CALIBRATION REPORT/ CERTIFICATE

Ref No: ISPL/CC/382/2010-11 — Date: 31.03.2011

1.0	Component: INDUS 4-Gas Analyser Model PEA205 [Sl. No.P-6067]	
2.0	PUC Center: **HOD, Mechanical Engineering department, Gogte Insitute of Technology, Udyambag, BELGAUM-590 008.** Registration No.:	
4.0	Detailed Observations	
4.1	Checking of supply / earthing Ok	
4.2	Checking of accessories:	
4.3	Span Calibration 1.Details of span gas concentration CO: 3.22% , HC: 3624 PPM in Propane, HC: 1775 PPM in Hexane, CO_2: 13.14 %, O_2: 0.41%, NOx: 445 ppm, SOx: 459 ppm, CO in ppm : 989 ppm 2. Calibration gas cylinder No :BGL 22671 & 109815 3. Calibration gas cylinder make: M/s. Bhuruka Gases Ltd 4. Calibration gas validity date: 25/09/2011 & 24/03/2012	
4.4	Electrical Calibration	OK
4.5	Leak test:	Passed
6.0	Conclusion: **Calibrated**	CO: 3.25% HC:1782 PPM in Hexane PEF: **0.490** CO_2:13.16% O_2: 0.45% NOx: 453 ppm Sox: 464 ppm CO in ppm: 996 ppm
7.0	Next Calibration Due Date:	**30.03.2012**

For INDUS Scientific Private Limited.

Authorized Signatory

Figure V.1 Calibration Certificate of Multigas Analyser

INDUS *Scientific* Private Limited

11/2B, Hennur Bande, Hennur Road, Bangalore – 560 043
T/F: (080) 2544 9762/ 64, Fax: + 91 80 2543 0914
Email: info@indusscientific.com, indus_scientific@vsnl.com
Website: www.indusscientific.com

ISO 9001 DAC

CALIBRATION REPORT/ CERTIFICATE

F/7/17
Rev: 00
Date: 01.08.09

Ref No: ISPL/CC/383/2010-11 **Date: 31/03/2011**

1.0	Component: INDUS Make Smoke Meter Model No.OMS 103 [Sr. No. O-3277] [Upgraded as per revised Test Procedure MORTH/CMVR/TAP/115/116 Issue No. 2 Part VIII]	
2.0	PUC Center: **HOD,** **Mechanical Engineering department,** **Gogte Insitute of Technology,** **Udyambag,** **BELGAUM-590 008.** Registration No.:	
3.0	Objective of the test: To carry out Physical and calibration of Smoke Meters as per the test procedure specified in Annexure 1 of CMVR / TAP 115-116 Part-8.	
4.0	Detailed Observations	
4.1	Checking of supply / earthing. **OK**	
4.2	Checking of accessories:	
4.3	Span Calibration Details of Natural Density filters used for mid point calibration filter value [**K=1.60m^{-1}**]	
4.4	Electrical Calibration **OK**	
4.5	RPM & Oil Temperature	**OK**
5.0	One no. of Diesel vehicle checked for idling Emission / Free acceleration. Measurement	
6.0	Conclusion: Calibrated	Filter Value: **K: 1.60m^{-1}**
7.0	Next Calibration Due Date:	**30/07/2011**

For **INDUS** ***Scientific*** **Private Limited**

Authorised Signatory

Figure V.2 Calibration Certificate of Smoke Meter

CALIBRATION CERTIFICATE

Model: SM111A22
Serial #: 15345
Description: Pressure Sensor
Type: ICP

Date: 8/9/2010
By: David Kimmel, Cal. Tech.
Station: Dead Weight #5 (Test Procedure AT601-2)

Sensitivity*: 1.001 mV/PSI
145.2 mV/MPa

Temp: 70 deg F [21deg C]
Humidity: 56 %

Linearity*: 0.4% FS
Uncertainty**: +/- 1 %

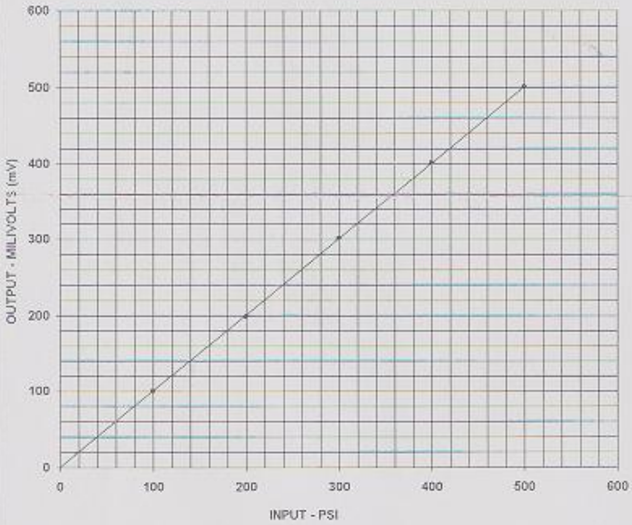
Cert #: 367859

Bias: 10.67 VDC

* Zero based, least-squares straight line

** Measurement uncertainty represented using a coverage factor of k=2 which provides a level of confidence of approximately 95 %.

Condition of Unit:
As Found: Not applicable
As Left: In tolerance, new unit



TEST DATA

INPUT (PSI)	OUTPUT (mV)
100	99.8
200	198.4
300	301.6
400	401.1
500	500.4

Notes:
1 Station #19
2 Calibration is traceable to NIST and is accredited to ISO 17025 and ANSI/NCSL Z540-1-1994.
3 NIST traceability through PCB control # CA727.
4 This certificate may not be reproduced, except in full, without written approval from PCB Piezotronics, Inc.

ACCREDITED
CALIBRATION CERT #1862.01

PCB PIEZOTRONICS

Tel: 716-684-0001 Fax: 716-684-0987 Email: sales@pcb.com
3425 Walden Avenue, Depew NY 14043

ISO 9001 CERTIFIED

Page 1 of 1

Figure V.3 Calibration Certification of the Piezosensor

Mint Biofuels Ltd. mint

CERTIFICATE OF ANALYSIS

BATCH NO : K 10604
Material : Biodiesel
FEED : Karanja
Quantity : 70 Lit (2 Plastic cones)

NO.	PROPERTY	UNIT	VALUE	ASTM D 6751 -09 STANDARDS
1.	Kinematic Viscosity @ 40 ° C	Cst	5.21	1.9 - 6.0
2.	Flash Point (Closed cup)	° C	160	130 min.
3.	Density @15 °C	Kg /cu m	874	Not mentioned
4	Water and Sediments	% volume	< 0.05	0.05 max
5.	Free Glycerin	% mass	0.012	0.020 max
6.	Total Glycerin	% mass	0.180	0.240 max
7.	Copper strip corrosion		Class 1	No 3 max
8	Carbon Residue (100%) Sample	% mass	0.028	0.05 max
9	Acid Number	mgKOH/gm	<0.50	0.50 max.
10	Calorific Value	Kcal/Kg	9300 to 9500	Not mentioned

Customer : To Ruturaj Patil
Report Date : 17/09/2010

Prepared by	Checked by
17/9/10	17/09/2010
prakash patil (Chemist)	sunil wadekar (Sr.Chemist)

S. No. 1073/1,2,3, Mutha Road, Pirangoot, Pune - 412111(India) Tel. : 020-22922350-57 Fax : 020-22922180/134 email : mint@mintbiofuels.com Website : www.mintbiofuels.com

Figure V.4 Properties of Karanja Biodiesel

Appendix VI

Studies Related To Production of Biodiesel

The appendix reviews the different techniques utilised for production of biodiesels by various investigators around the globe. The most preferred technique is transesterification as it yields the biodiesel whose properties are very close to diesel. Transesterification can be carried out in various ways by maintaining different equilibrium conditions. A new technique called peroxidation process in being highlighted in this review. The appendix also consists of a study wherein tools of biotechnology are employed in order to change the chemical structure of oils.

Ramadhas et al [21] developed a two-step esterification method to produce biodiesel from crude rubber seed oil. The esterification process consists of the following two stages, viz. (i) Acid-esterification: The acid-catalyzed pre-treatment process reduced the high FFA content of the crude oil to about 2% FFA, (ii) Alkaline-esterification: The products of first step were transesterified using alkaline catalyst. Sulphuric acid is used as catalyst. Alcohol to vegetable oil molar ratio is one of the important factors that affects the conversion efficiency of the process. For the stoichiometric transesterification, 3 mol of alcohol are required for each mole of the oil. However, in practice, the molar ratio should be higher than this theoretical ratio in order to drive the reaction towards early completion.

Usta et al. [26] investigated the effects of the methyl ester produced from a hazelnut soapstock/waste sunflower oil mixture using methanol, sulphuric acid and sodium hydroxide in a two stage process on the performance and emissions of a Ford XLD 418T four stroke, four cylinder, turbocharged indirect injection (IDI) Diesel engine at both full and partial loads. The CR of the engine was 21.5:1. During the production of biodiesel, Hazelnut soapstock and waste sunflower oil were mixed in approximately equal volume proportions. It was found that when a base catalysed transesterification process was directly applied to the mixture, the high free fatty acid content in the soap stock causes fairly high soap formation, which diminished the ester yield. Therefore, it was necessary to reduce the free fatty acid content of the mixture using an acid catalysed pre-treatment at 35 ^{0}C to esterify the free acids before transesterifying the triglycerides with an alkaline catalyst to complete the reaction at 55 ^{0}C.

Reyes and Sepulveda [29] Biodiesel production was carried out in a 20 litre non pressurized reactor, with control of stirring speed and temperature. Waste non-refined salmon oil with high free fatty acid content from food industry was used as oil source to make biodiesel. Two types of biodiesel fuels were produced. One of the fuels, called crude biodiesel was produced by sulphuric acid esterification at 60^0C for 65 minutes followed by a caustic soda transesterification at 55^0C for 60 minutes and then a gravitational separation of glycerine. In both reactions, a molar ratio of 6:1 methanol to triglycerides was used. The final biodiesel was carefully washed to eliminate residues of Glycerine and acid-base compound formed in the process. After a final washing and heating of the crude biodiesel at 70^0C for 30 minutes, the resulting viscosity and melting point of the fuel were 4.96 Cs and -11.0 ^{0}C respectively. This fuel held red colour impurities which were due to the artificial colorants added to the pellets. They were fed to the salmons and therefore were transferred to the fish oil. The final percentage conversion of glycerides to methyl ester obtained in crude biodiesel was 90%. On the other hand, the refined biodiesel fuel was produced from distillation of the crude biodiesel, which permits to achieve elimination of colour residues coming from salmon feeding and unconverted glycerides. Viscosity and melting point of refined biodiesel were 3.46Cs and -10.5 ^{0}C respectively.

Lin and Lin [30] produced biodiesel using peroxidation process. In this study, soybean oil (used as raw oil) and methanol in a molar proportion of 1:6 were reacted to produce biodiesel. Then, 1wt% of NaOH was added as a catalyst in the transesterification reaction. Hydrogen peroxide agent, which has a strong chemical activity, was used to react with sample of biodiesel in the peroxidation process. The oxygen content was significantly increased from 9.09 to 9.94 wt%, and the saturated carbon bonds increased significantly by 1.6 wt%, from 16.6 to 18.2 wt%, for the biodiesel produced with the additional peroxidation process.

Sahoo et al. [36] produced high acid value Polanga (Calophyllum Inophyllum L.) oil based mono esters by triple stage transesterification process and blended with high speed diesel on a small-size water-cooled direct injection diesel engine. The density and viscosity of the polanga oil methyl ester formed after triple stage transesterification were found to be close to those of petroleum Diesel oil. The three stages used for transesterification were.

(i) Zero catalyzed transesterification: In the first stage, the organic matters and other impurities present in the unrefined filtered polanga oil using reagent were removed.

(ii) Acid catalyzed transesterification: The intermediate stage, the acid value of the oil about 4 mg KOH/gm corresponding to a FFA level of 2% was reduced.

(iii) Alkaline catalyzed transesterification: The product of the intermediate stage (pure triglycerides) was transesterified to mono-esters of fatty acids (biodiesel) using alkali catalyst.

The formation of methyl ester by three stage transesterification stochiometrically required six moles of alcohol for every mole of triglyceride. However, transesterification is an equilibrium reaction in which an excess of the alcohol is required to drive the reaction close to completion. The optimum ratio was found to be 6:1 molar ratio of methanol to oil (triglyceride) which is sufficient to give 85% yield of ester.

Meng et al. [61] studied the production of biodiesel from Waste cooking oil (WCO) through experimental investigation of reaction conditions such as methanol/oil molar ratio, alkaline catalyst amount, reaction time and reaction temperature. With the methanol/oil molar ratio increasing, WCO conversion efficiency was found to be correspondingly increasing. The maximum conversion efficiency (88.9%) was achieved at 7:1 methanol/oil molar ratio. With further increase in molar ratio, the conversion efficiency more or less remained the same. The WCO conversion efficiency also was found to be increasing proportionally with increasing amount of NaOH. The maximum WCO conversion efficiency (85.0%) was observed at 1.0 wt % NaOH. Addition of excess amount of catalyst, gave rise to the formation of an emulsion, which increased the viscosity and led to the formation of gels. The WCO conversion efficiency rapidly increased with the reaction time ranging between 30 minutes and 60 minutes, thereafter, the conversion efficiency kept rising very slowly and then practically constant above 86% at 90 minutes. The maximum WCO conversion efficiency was obtained at 50 °C temperature. Based on the results of preliminary experiments, a four-factor (A. methanol/oil molar ratio, B. NaOH amount, C. Reaction temperature and D. Reaction time) and three-level (for A 3:1, 6:1, 9:1; for B 0.7 wt.%, 1.0 wt.%, 1.3 wt.%; for C 35 °C, 50 °C, 60 °C and for D 30 minutes, 60 minutes, 90 minutes) orthogonal test was designed to determine optimum conditions for the transesterification process of WCO.

The optimum experimental conditions, which were obtained from the orthogonal test, were methanol/oil molar ratio 9:1, with 1.0 wt. % sodium hydroxide, temperature of 50 °C and time of 90 minutes. Verified experiments showed methanol/oil molar ratio 6:1 was more suitable in the process, and under that condition WCO conversion efficiency increased to 89.8% and the physical and chemical properties of biodiesel sample satisfied the requirement of relevant international standards.

Kinney and Clemente [106] implemented the tools of biotechnology to modify the fatty acid profile of soybean for performance enhancement. The content of linoleic and linolenic acids was reduced in order to reduce oxidative reactivity of the oil. It was also found that addition of ricinoleic acid increase the lubricity of soyabeen oil. Finally, it was concluded that designing soybean oil with 10–20% ricinoleic acid coupled with 70–80% oleic acid would enhance the performance of biodiesel fuel. They quoted also an economic analysis conducted by the Minnesota Department of Agriculture which revealed that if soybean oil-derived biodiesel was used at either 2% or 5% blend with petroleum diesel, this in turn would generate a significant economic stimulus, including job creation in rural areas.

Lertsathapornsuk et al. [107] modified a household microwave (800W) into a biodiesel reactor (Refer Figure VI.1) for continuous transethylation of waste frying palm oil. The waste palm oil biodiesel was then tested in a 100 kW diesel generator as a neat fuel (B100) and 50% blend with diesel No. 2 fuel (B50). It was reported that transmethylation under microwave radiation was very rapid. The biodiesel obtained was found to be having cetane number higher than Diesel oil. HC emissions increased with the increase in load of the engine in all types of fuels. The reason was reported to be a consequence of the increase in fuel consumption at higher loads. It was also reported that as the biodiesel contained about 11% oxygen and its cetane index is much higher than diesel more complete combustion took place which resulted in reduction of CO emissions.

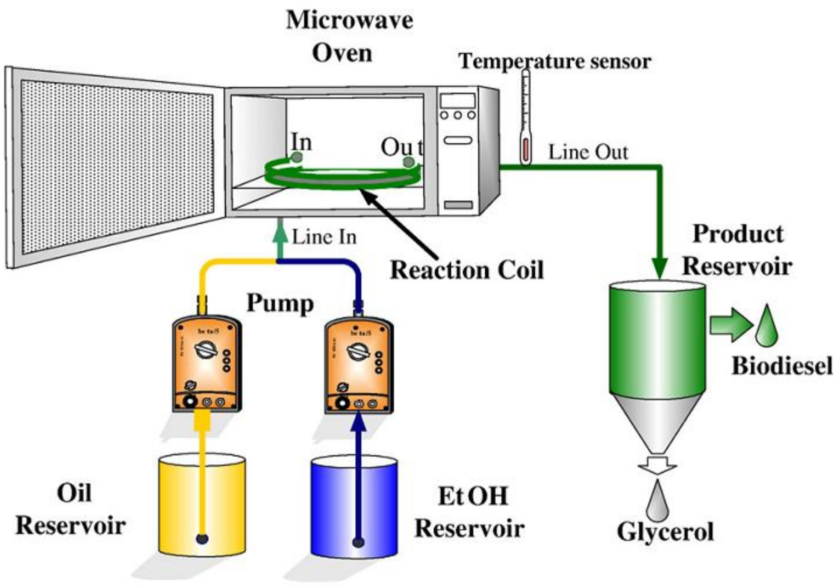


Figure VI.1 Schematic of Continuous Microwave Reactor [107]

Ali Keskin et al. [108] studied the usability of cotton oil soapstock biodiesel–diesel fuel blends as an alternative fuel for diesel engines. The biodiesel was produced using the following process as given in Figure VI.2

It was found that high calorific value and cetane number, low sulphur and low aromatic content, and similar characteristics are advantageous of cotton oil soap stock biodiesel–diesel fuel blends. The investigators by studies reported that the properties of blend fuels, such as oxygen content, higher cetane number and low levels of sulphur content improved combustion process.

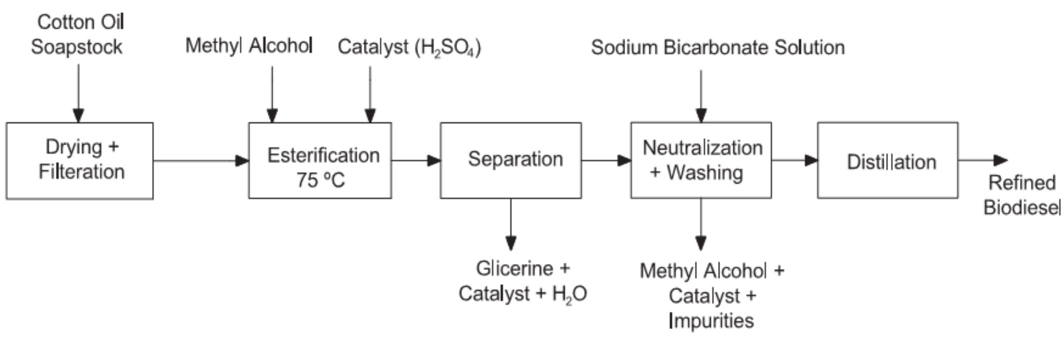


Figure VI.2 Flow Diagram of Cotton Oil Soap stock Biodiesel Production Process [108]

S. Murugan et al. [99] studied the use of tyre pyrolysis oil (TPO) in diesel engines. In order to produce oil, pyrolysis of rubber pieces obtained from an automobile tyre was carried out. After determining its properties, it was found that the viscosity of TPO was higher than diesel by about 1.5 times and the flash point and the fire point of TPO were closer to those of diesel. It was reported that the TPO-diesel fuel blend had longer ignition delay, higher NO_x and higher HC emissions. The ignition delay was longer due to the higher viscosity of TPO–diesel fuel blends that results in poor atomization. Fuels with longer ignition delays exhibit higher maximum rates of heat release, resulting in higher cylinder temperatures. Thus, NO_x is produced in the local high temperature region. Higher HC emissions are probably due to higher viscosity, density, poor volatility, and rich fuel mixtures at higher loads.

Ayhan Demirbas [109] conducted an experiment to demonstrate the production of bio-diesel from linseed oil fuels in non catalytic super critical method using fuels such as methanol and ethanol. Methanol was commonly used due to its lower cost. Methyl esters are renewable and are new clean engine fuel alternatives to petro diesel. It was stated that the most important variables affecting the yield of methyl esters during the transesterification reaction are the reaction temperature and the molar ratio of alcohol to vegetable oil. The yield conversion rate rises from 50% to 95% for first 10 minutes. The reaction temperature was in the range of 503K to 523K. The viscosity of linseed oil ethyl esters was slightly higher than that methyl ester. As seen in Figure II.3, the most volatile fuel was diesel fuel. The volatility of linseed oil methyl ester was higher than that of ethyl ester at all temperatures. The volatility of linseed oil was lower than those of corresponding methyl and ethyl esters at all temperatures.

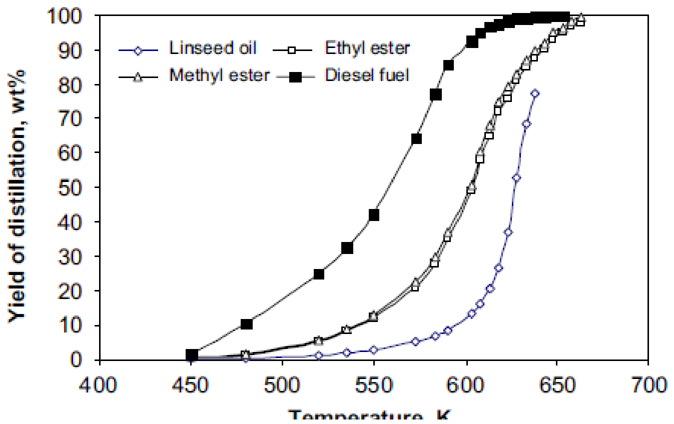


Figure VI.3 Distillation Curves of Diesel Fuel, Linseed Oil, and Methyl and Ethyl Esters of Linseed Oil [109]

TableVI.1 Summary of Studies Carried out on Production of Biodiesels

Sr. No	Investigators	Year	Fuel	Remarks
1	Ramadhas et al.	2005	Biodiesel from rubber seed oil	Developed a two-step esterification method to produce biodiesel from crude rubber seed oil.
2	Usta et al.	2005	Biodiesel from a hazelnut soap stock	It was found that when a base catalysed transesterification process was directly applied to the mixture, the high free fatty acid content in the soap stock causes fairly high soap formation, which diminished the ester yield.
3	Kinney and Clemente	2005	Biodiesel from soybean oil	Implemented the tools of biotechnology to modify the fatty acid profile of soybean for performance enhancement.
4	Reyes and Sepulveda	2006	Biodiesel from refined salmon oil	Biodiesel production was carried out in a 20 litre non pressurized reactor, with control of stirring speed and temperature.
5	Lin and Lin	2006	Biodiesel from soybean oil	Biodiesel is produced using peroxidation process. A hydrogen peroxide agent, which has a strong chemical activity, was used to react with sample of biodiesel in the peroxidation process
6	Sahoo et al.	2007	Biodiesel from polanga seed oil	Biodiesel was produced by triple stage transesterification process. The density and viscosity of the polanga oil methyl ester formed after triple stage transesterification were found to be close to those of petroleum Diesel oil.
7	Meng et al.	2008	Biodiesel from Waste cooking oil	Verified experiments showed methanol/oil molar ratio 6:1 was more suitable in the process, and under that condition WCO conversion efficiency led to 89.8% and the physical and chemical properties of biodiesel sample satisfied the requirement of relevant international standards.
8	Sinha and Agarwal	2008	Biodiesel from rice bran oil	Carried out the transesterification process for production of biodiesel. Its characteristic properties were found suitable to be used as fuel in diesel engines as they were comparable to that of diesel. However, the density, flash point temperature was slightly higher.
9	Lertsathapornsuk et al.	2008	Biodiesel from waste frying palm oil	Modified a household microwave into a biodiesel reactor for continuous transethylation of waste frying palm oil.
10	Keskin et al.	2008	Biodiesel from cottonseed oil	Before the transesterification, cotton oil soap stock was filtered and dried for removing water and impurities at 100 ^{0}C. After transestrification Crude biodiesel was distillated for obtaining pure methyl ester by means of batch distillation apparatus.
11	Murugan et al.	2008	Biodiesel from Tyre pyrolysis oil	Pyrolysis of rubber pieces obtained from an automobile tyre was carried out.
12	Krishnakumar et al.	2008	Biodiesel from rice bran oil and Jatropha oil	Catalytic transesterification method was used for converting oils into methyl esters. Transesterification reaction was carried out in a batch reactor.
13	Demirbas	2009	Biodiesel from linseed oil	Biodiesel was produced using Non catalytic super critical

				method using fuels such as methanol and ethanol.

The studies carried out on production of biodiesel from the year 2005 to year 2009 are listed in Table VI.1. From the table it can be observed that transesterification is the best and optimum method for production of biodiesel. However various investigators have developed different methods of transesterification, most of the methods yields almost same results. The various transesterification methods developed are two-step transestrification, base catalysed transesterification, triple stage transesterification, transethylation etc. The properties of biodiesel produced by all the transesterification processes are comparable to that of conventional Diesel oil and hence this throws some light on a point that biodiesel can be successfully used in existing diesel engines.

Appendix VII

Uncertainty Analysis

Uncertainty analysis involves systematic procedures for calculating error estimates for experimental data. When estimating errors in heat engine experiments, it is usually assumed that data is gathered under fixed (known) conditions and detailed knowledge of all system components is available. Measurement errors arise from various sources, but they can be broadly classified as bias errors and precision (or random) errors. Bias errors remain constant during a set of measurements. They are often estimated from calibration procedures or past experience. Alternatively, different methods of estimating the same variable can be used, so that comparisons between those results would indicate the bias error. Elemental bias errors arise from calibration procedures or curve-fitting of calibrated data.

To quantify errors in experimental work, some calculations and estimation have to be applied on sensors, devices and machines that have been used to measure the experimental parameters. As the experiments performed as it need to express measurement uncertainty as

$$x' = x \pm u_x \qquad (P\%) \tag{VII.1}$$

where, x', x, u^x and P% are true value, tested value, uncertainty of the measurement and confidence respectively. To do this with assurance, total uncertainty of each component or portion of the experiment or procedure is determined. Total uncertainty is determined by finding error due to equipment (bias) and due to environment (precision). Wherever possible, uncertainty of each component or portion of experiment is found to determine where uncertainty may be minimized. Uncertainties are propagated in post-processing phase, to quantities that are non-linear functions of a measurement or functions of multiple measurements with uncertainties based upon the functional relationship.

Both bias and precision errors are present in an experiment. The precision is measured whereas the bias error is usually determined from equipment vendor specs. The total error is the vector sum of these errors and it is to be noted that errors in estimating each error affect the value of the total error.

$$u_X = (B_x^2 + P_x^2)^{1/2} \tag{VII.2}$$

where, B_x and P_x are bias and precision error respectively.

In case of several measurements of the same quantity like engine load, the uncertainty is estimated using statistical measures of spread. Several measurements of the same quantity are: X_1, X_2, X_3, X_4, X_5,---,X_n. Average load of the dynamometer is calculated as

$$\text{average} = (X1, X2, X3, X4, X5, ---, Xn)/n \qquad \text{(VII.3)}$$

Now, there are two ways to describe the scatter in these measurements. The mean deviation from the mean is the sum of the absolute values of the differences between each measurement and the average, divided by the number of measurements:

$$\text{mean deviation from mean} = \frac{\sum_{i=1}^{n}|X_i - \text{average}|}{n} \qquad \text{(VII.4)}$$

The standard deviation from the mean is the square root of the sum of the squares of the differences between each measurement and the average, divided by one less than the number of measurements:

$$\text{standard deviation from mean} = \frac{\sqrt{\sum_{i=1}^{n}(X_i - \text{average})^2}}{1-n} \qquad \text{(VII.5)}$$

Either the mean deviation from the mean, or the standard deviation from the mean, gives a reasonable description of the scatter of data around its mean value.

Tested load - mean deviation < true load < tested load + mean deviation (VII.6)

Tested load - mean deviation < true load < tested load + standard deviation (VII.7)

For parameter that have been evaluated depending on two or more independent parameters, propagation of uncertainty is carried out using

$$\frac{U_y}{y} = \sqrt{\left(\frac{u_{x1}}{x1}\right)^2 + \left(\frac{u_{x2}}{x2}\right)^2 + \ldots + \left(\frac{u_{xn}}{xn}\right)^2} \qquad \text{(VII.8)}$$

Where, U_y and y are uncertainty and the testing value of the evaluated parameter x1, x2,…., xn respectively.

The uncertainty analysis carried out in this Appendix is based on the lines suggested by Kline and McClintock [95]. It should be noted that the uncertainty analysis presented here considers only the errors that relate to the measurements made during testing. Δ is used here to symbolize the error in the quantity.

VII.1 Uncertainity Calculations

VII.1.1 Uncertainty in Thermal Performance Parameters

The uncertainty calculation shown here are carried out correspond to CR 18, IP 200bar and fuel as Diesel oil shown in Table III.1in Appendix III. The full load is taken as 12kg and low load is taken as 3kg.

VII.1.1.1 Uncertainty in Brake Power

BP=2πNT/60*1000, T = W*R

At Full Loading Condition (Load is 12kg)

$$\Delta BP/BP = \sqrt{[(\Delta N/N)^2 + (\Delta W/W)^2]}$$

$$= \sqrt{[(30/1530)^2 + (0.1/12)^2]}$$

$$= \sqrt{[0.0004 + 0.0000694]}$$

$$\mathbf{= 0.021 = 2.1\%}$$

At Low Loading Condition (Load is 3kg)

$$\Delta BP/BP = \sqrt{[(\Delta N/N)^2 + (\Delta W/W)^2]}$$

$$= \sqrt{[(30/1470)^2 + (0.1/3)^2]}$$

$$= \sqrt{[0.0004 + 0.0011]}$$

$$\mathbf{= 0.038 = 3.8\ \%}$$

VII.1.1.2 Uncertainty in BMEP

The BMEP is calculated by using the formula

$$BMEP = \frac{BP\ (KW) \times 60}{L \times A \times \left(\frac{N}{n}\right) \times No\ of\ cylinders \times 100}$$

At Full Loading Condition

$$\Delta BMEP/BMEP = \sqrt{[(\Delta BP/BP)^2 + (\Delta L/L)^2 + (\Delta D/D)^2]}$$

$= \sqrt{[(0.021)^2 + (1/110)^2 + (1/87.5)^2]}$

$= 0.04413 = 4.413\%$

At Low Loading Condition

$\Delta BMEP/BMEP = \sqrt{[(\Delta BP/BP)^2 + (\Delta L/L)^2 + (\Delta D/D)^2]}$

$= \sqrt{[(0.038)^2 + (1/110)^2 + (1/87.5)^2]}$

$= 0.0407 = 4.07\%$

VII.1.1.3 Uncertainty in BTHE

The BTHE is calculated by using the formula

$$BTHE = \frac{BP \times 3600 \times 100}{Fuel\ flow\ in \frac{kg}{hr} \times Calorific\ Value}$$

At Full Loading Condition

$\Delta BTHE/BTHE = \sqrt{[(\Delta BP/BP)^2 + (\Delta m_f/m_f)^2]}$

$= \sqrt{[(0.021)^2 + (0.01/1.04)^2]}$

$= 0.023 = 2.3\%$

At Low Loading Condition

$\Delta BTHE/BTHE = \sqrt{[(\Delta BP/BP)^2 + (\Delta m_f/m_f)^2]}$

$= \sqrt{[(0.038)^2 + (0.01/0.57)^2]}$

$= 0.041 = 4.1\%$

VII.1.1.4 Uncertainty in BSFC

The BSFC is calculated by using the formula

$$BSFC = \frac{Fuel\ flow\ in \frac{kg}{hr}}{BP}$$

At Full Loading Condition

$$\Delta BSFC/BSFC = \sqrt{[(\Delta m_f / m_f)^2 + (\Delta BP/BP)^2]}$$

$$= \sqrt{[(0.01/ 1.04)^2 + (0.02)^2]}$$

$$\mathbf{= 0.022 = 2.2\%}$$

At Low Loading Condition

$$\Delta BSFC/BSFC = \sqrt{[(\Delta m_f / m_f)^2 + (\Delta BP/BP)^2]}$$

$$= \sqrt{[(0.01/ 0.57)^2 + (0.038)^2]}$$

$$\mathbf{= 0.0418 = 4.18\ \%}$$

VII.1.1.5 Uncertainty in Volumetric Efficiency (η_{vol})

The η_{vol} is calculated by using the formula

$$\eta_{vol} = \frac{Airflow\ in\ kg/hr \times 100}{\left(\frac{\pi}{4}\right) \times D^2 \times L \times \left(\frac{N}{n}\right) \times 60 \times No\ of\ cylinders \times \rho_a}$$

At Full Loading Condition

$$(\Delta\eta_{vol}/\eta_{vol}) = \sqrt{[(\Delta m_a/m_a)^2 + (\Delta D/D)^2 + (\Delta L/L)^2 + (\Delta\rho_a/\rho_a)^2]}$$

$$= \sqrt{[(0.01/26.92)^2 + (1/87.5) + (1/110)^2]}$$

$$= 0.0146 = 1.46\%$$

At Low Loading Condition

$$(\Delta\eta_{vol}/\eta_{vol}) = \sqrt{[(\Delta m_a/m_a)^2 + (\Delta D/D)^2 + (\Delta L/L)^2 + (\Delta\rho_a/\rho_a)^2]}$$

$= \sqrt{[(0.01/28.14)^2 + (1/87.5) + (1/110)^2]}$

$= 0.0146 = 1.46\%$

VII.1.1.6 Uncertainty in HGas

HGas= $(m_a + m_f) \times C_{pgas} \times (T_{exhaust} - T_{air})$

At Full Loading Condition

$\Delta HGas/HGas = \sqrt{[(\Delta m_a/m_a)^2 + (\Delta m_f/m_f)^2 + (\Delta EGT/EGT)^2]}$

$= \sqrt{[(0.01/26.92)^2 + (0.01/ 1.04)^2 + (0.01/423.71)^2]}$

$= 0.009622 = 0.96\%$

At Low Loading Condition

$\Delta HGas/HGas = \sqrt{[(\Delta m_a/m_a)^2 + (\Delta m_f/m_f)^2 + (\Delta EGT/EGT)^2]}$

$= \sqrt{[(0.01/28.14)^2 + (0.01/ 0.57)^2 + (0.01/196.27)^2]}$

$= 0.0175 = 1.75\%$

VII.1.2 Uncertainty in Emission Constituents (Resolution/Range)

$\Delta CO/CO$ (%) = 0.001/15 = 0.00006 = **0.006%**

$\Delta CO/CO$ (ppm) = 1/4000 = 0.00025= **0.025%**

$\Delta CO_2/CO_2$ = 0.01/20 = 0.0005 = **0.05%**

$\Delta HC/HC$ = 1/30000 = 0.00003 = **0.003%**

$\Delta O_2/O_2$ = 0.01/25 = 0.0004 = **0.04%**

$\Delta NO_x / NO_x$ = 1 / 5000 = 0.0002 = **0.02%**

$\Delta SO_x / SO_x$ = 1/2000 = 0.0002 = **0.02%**

Table VII.1 Probable errors in the estimation of thermal performance of diesel engine running on diesel biodiesel blends

Quantity	**Value**	**Probable error**
Speed (N)	1500 rpm	30 rpm
Load (W)	12 Kg	0.1 kg
Fuel flow rate (m_f)	1.04 Kg/hr	0.01 kg/hr
Bore (D)	87.5mm	1mm
Stroke Length (L)	110mm	1mm
Air flow rate (m_a)	26.92 Kg/hr	0.01 Kg/hr

Table VII.2 Resolution and Range of Gas analyzer for the emission constituents

Quantity	**Resolution**	**Range**
Carbon Monoxide (CO)	0.001 %,	0%-15%, 0ppm-4000ppm
Carbon Dioxide (CO_2)	0.01%	0%-20%
Hydrocarbon (HC)	1 ppm	0ppm-30000ppm
Oxygen (O_2)	0.01%	0%-25%
Oxides of Nitrogen (NO_x)	1ppm	0ppm-5000ppm
Oxides of Sulphur (SO_x)	1ppm	0ppm-5000ppm

Appendix VIII

Genetic Algorithm

The optimization problems reveal the fact that the formulation of engineering design problems could differ from problem to problem. Certain problems involve linear terms for constraints and objective function but certain other problems involve nonlinear terms for one or both of them. In some problems, the terms are not explicit functions of the design variables. Unfortunately, a single optimization algorithm does not exist which will work in all optimization problems equally efficiently. Some algorithms perform better on one problem, but may perform poorly on other problems. That is why, the optimization literature contains a large number of algorithms, each suitable to solve a particular type of problem. The choice of a suitable algorithm for an optimization problem is, to a large extent, dependent on the user's experience in solving similar problems. Since the optimization algorithms involve repetitive application of algorithmic steps, they need to be used with the help of a computer. The optimization algorithms are classified into a number of groups, which are now briefly discussed.

- ***Single-variable optimization algorithms***

These algorithms provide a good understanding of the properties of the minimum and maximum points in a function and how optimization algorithms work iteratively to find the optimum point in a problem. The algorithms are classified into two categories—direct methods and gradient-based methods. Direct methods do not use any derivative information of the objective function; only objective function values are used to guide the search process. However, gradient-based methods use derivative information (first and/or second-order) to guide the search process. Although engineering optimization problems usually contain more than one design variable, single-variable optimization algorithms are mainly used as unidirectional search methods in multivariable optimization algorithms.

- ***Multi-variable optimization algorithms***

These algorithms demonstrate how the search for the optimum point progresses in multiple dimensions. Depending on whether the gradient information is used or not used, these algorithms are also classified into direct and gradient-based techniques.

- ***Constrained optimization algorithms***

These algorithms use the single-variable and multivariable optimization algorithms repeatedly and simultaneously maintain the search effort inside the feasible search region. These algorithms are mostly used in engineering optimization problems.

- ***Specialized optimization algorithms***

There exist a number of structured algorithms, which are ideal for only a certain class of optimization problems. Two of these algorithms integer programming and geometric programming are often used in engineering design problem. Integer programming methods can solve optimization problems with integer design variables. Geometric programming methods solve optimization problems with objective functions and constraints written in a special form.

There exist quite a few variations of each of the above algorithms. These algorithms are being used in engineering design problems since sixties. Because of their existence and use for quite some years, these algorithms are termed as traditional optimization algorithms.

- ***Nontraditional optimization algorithms***

There exist a number of other search and optimization algorithms which are comparatively new and are becoming popular in engineering design optimization problems in the recent years. Two such algorithms are genetic algorithms and simulated annealing.

Investigators have put together about 34 different optimization algorithms. Over the years, investigators and practitioners have modified these algorithms to suit their problems and to increase the efficiency of the algorithms. However, there exist a few other optimization algorithms like stochastic programming methods and dynamic programming method which are very different than the above algorithms.

Many engineering optimization problems contain multiple optimum solutions, among which one or more may be the absolute minimum or maximum solutions. These absolute optimum solutions are known as global optimal solutions and other optimum solutions are known as local optimum solutions. Ideally, we are interested in the global optimal solutions because they correspond to the absolute optimum objective function value. Unfortunately, none of the traditional algorithms are guaranteed to find the global optimal solution, but genetic algorithms and simulated annealing algorithm are found to have a better global perspective than the traditional methods. Moreover, when an optimal design problem contains multiple global solutions, designers are not only interested in finding just one global optimum solution, but as many as possible for three main reasons. Firstly, a design suitable in one situation may not be valid in another situation. Secondly, it is also not possible to include all aspects of the design in the optimization problem formulation. Thus, there always remains some uncertainty about the obtained optimal solution. Thirdly, designers may not be interested in finding the absolute global solution, instead may be interested in a solution which corresponds to a marginally inferior objective function value but is more amenable to fabrication and operation. Thus, it is always prudent to know about other equally good solutions for later use. However, if the traditional methods are used to find multiple optimal solutions, they need to be applied a number of times, each time starting from a different initial solution and hoping to achieve a different optimal solution each time. Genetic algorithms allow an easier way to find multiple optimal solutions simultaneously in a single simulation.

Another class of optimization problems deals with simultaneous optimization of multiple objective functions. In formulating an optimal design problem, designers are often faced with a number of objective functions. Multi objective optimization problems give rise to a set of optimal solutions known as *Pareto-optimal* solutions all of which are equally important as far as all objectives are concerned. Thus, the aim in these problems is to find as many Pareto-optimal solutions as possible. Because of the complexity involved in the multi objective optimization algorithms, designers usually choose to consider only one objective and formulate other objectives as constraints. Genetic algorithms handle multiple objectives and help find multiple Pareto-optimal solutions simultaneously.

In many engineering design problems, a good solution is usually known either from the previous studies or from experience. After formulating the optimal problem and applying the optimization algorithm if a better solution is obtained, the new solution becomes the current best solution. The optimality of the obtained solution is usually confirmed by applying the optimization algorithms a number of times from different initial solutions.

Two nontraditional search and optimization methods have gained popularity in engineering optimization problems not because they are new but because they are found to be potential search and optimization algorithms for complex engineering optimization problems. *Genetic algorithms* (GAs) mimic the principles of natural genetics and natural selection to constitute search and optimization procedures, *Simulated annealing* mimics the cooling phenomenon of molten metals to constitute a search procedure. Since both these algorithms are abstractions from a natural phenomenon, they are very different search methods than other procedures.

Genetic algorithms are computerized search and optimization algorithms based on the mechanics of natural genetics and natural selection. GAs is radically different from most of the traditional optimization methods. GAs works with a string-coding of variables instead of the variables itself. The advantage of working with coding of variables is that the coding discretizes the search space, even though the function may be continuous. On the other hand, since GAs require only function values at various discrete points, a discrete or discontinuous function can be handled with no extra cost. This allows GAs to be applied to a wide variety of problems. Another advantage is that the GA operators exploit the similarities in string-structures to make an effective search.

The most striking difference between GAs and many traditional optimization methods is that GAs work with a population of points instead of a single point. Because there are more than one string being processed simultaneously, it is very likely that the expected GA solution may be a global solution. Even though some traditional algorithms are population-based, like Box's evolutionary optimization and complex search methods, those methods do not use previously obtained information efficiently. In GAs, previously found good information is emphasized using reproduction operator and propagated adaptively through crossover and mutation operators. Another advantage with a population-based search algorithm is that multiple optimal solutions can be captured in the population easily, thereby reducing the effort to use the same algorithm many times.

GAs does not require any auxiliary information except the objective function values. Although the direct search methods used in traditional optimization methods do not explicitly require the gradient information, some of those methods use search directions that are similar in concept to the gradient of the function. Moreover, some direct search methods work under the assumption that the function to be optimized is unimodal and continuous. In GAs, no such assumption is necessary.

One other difference in the operation of GAs is the use of probabilities in their operators. None of the genetic operators work deterministically. In the reproduction operator, even though a string is expected to have fixed copies in the mating pool, a simulation of the roulette-wheel selection scheme is used to assign the true number of copies. In the crossover operator, even though good strings (obtained from the mating pool) are crossed, strings to be crossed are created at random and cross-sites are created at random. In the mutation operator, a random bit is suddenly altered. The action of these operators may appear to be naive, but careful studies provide some interesting insights about this type of search. The basic problem with most of the traditional methods is that they use fixed transition rules to move from one point to another. For instance, in the steepest descent method, the search direction is always calculated as the negative of the gradient at any point, because in that direction the reduction in the function value is maximum. In trying to solve a multimodal problem with many local optimum points, search procedures may easily get trapped in one of the local optimum points. Consider the bimodal function shown Figure VIII.1. The objective function has one local minimum and one global minimum. If the initial point is chosen to be a point in the local basin (point $x^{(t)}$ in the figure), the steepest descent algorithm will eventually find the local optimum point. Since the transition rules are rigid, there is no escape from these local optima. The only way to solve the above problem to global optimality is to have a starting point in the global basin.

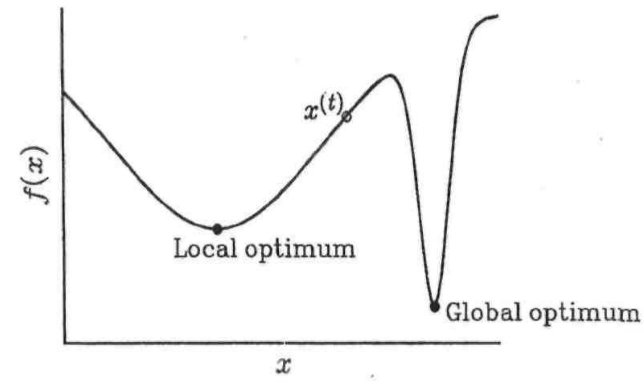


Figure VIII.1 An Objective Function with One Local Optimum and One Global Optimum

Since this information is usually not known in any problem, the steepest-descent method (and for that matter most traditional methods) fails to locate the global optimum. This phenomenon is termed as trapping in local optima. However, these traditional methods can be best applied to a special class of problems suitable for those methods.

For example, the gradient search methods will outperform almost any algorithm in solving continuous, unimodal problems, but they are not suitable for multimodal problem. Thus, in general, traditional methods are not robust. A robust algorithm can be designed in such a way that it uses the steepest descent direction most of the time, but also uses the steepest ascent direction (or any other direction) with some probability. Such a mixed strategy may require more number of function evaluations to solve continuous, unimodal problems, because of the extra computations involved in trying with non-descent directions. But this strategy may be able to solve complex, multimodal problems to global optimality. In the multimodal problem, shown in the above figure, the mixed strategy may take the point $x^{(t)}$ into the global basin (when tried with non-descent directions) and finally find the global optimum point. GAs use similar search strategies by using probability in all their operators. Since an initial random population is used, to start with, the search can proceed in any direction and no major decisions are made in the beginning. Later on, when the population begins to converge in some bit positions, the search direction narrows and a near-optimal solution is achieved. This nature of narrowing the search space as the search progresses is adaptive and is a unique characteristic of genetic algorithms.

Appendix IX

Artificial Neural Network

One efficient way of solving complex problems is following the lemma “divide and conquer”. A complex system may be decomposed into simpler elements, in order to be able to understand it. Also simple elements may be gathered to produce a complex system (Bar Yam, 1997). Networks are one approach for achieving this. There are a large number of different types of networks, but they all are characterized by the following components: a set of nodes, and connections between nodes.

The nodes can be seen as computational units. They receive inputs, and process them to obtain an output. This processing might be very simple (such as summing the inputs), or quite complex (a node might contain another network...). The connections determine the information flow between nodes. They can be unidirectional, when the information flows only in one sense, and bidirectional, when the information flows in either sense. The interactions of nodes though the connections lead to a global behaviour of the network, which cannot be observed in the elements of the network. This global behaviour is said to be emergent. This means that the abilities of the network supercede the ones of its elements, making networks a very powerful tool.

One type of network sees the nodes as ‘artificial neurons’. These are called artificial neural networks (ANNs). An artificial neuron is a computational model inspired

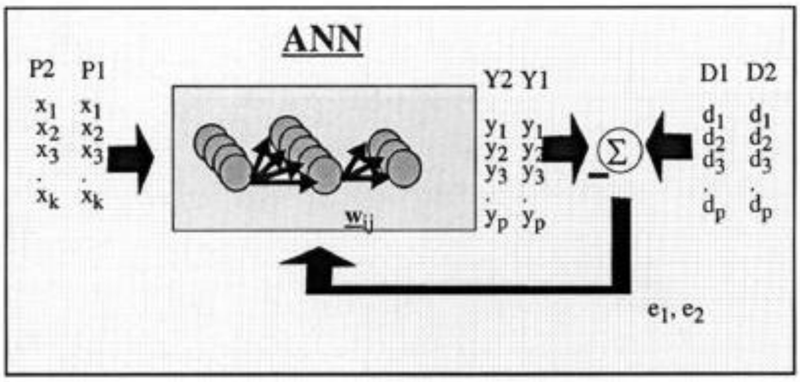


Figure IX.1 The Style of Neural Computation

in the natural neurons. Natural neuronsreceive signals through synapses located on the dendrites or membrane of the neuron. When the signals received are strong enough (surpass a certain threshold), the neuron is activated and emits a signal though the axon. This signal might be sent to another synapse, and might activate other neurons.

An ANN is developed with a systematic step-by-step procedure which optimizes a criterion commonly known as the learning rule. The input/output training data is fundamental for these networks as it conveys the information which is necessary to discover the optimal operating point. In addition, its non linear nature makes neural network processing elements a very flexible system.

Basically, an ANN is a system that receives an input, process the data, and provides an output (Refer Figure IX.1). Once an input is presented to the neural network, and a corresponding desired or target response is set at the output, an error is composed from the difference of the desired response and the real system output.

In neural network design, the engineer chooses the network topology, the trigger function or performance function, learning rule and the criteria for stopping the training phase. So, it is difficult to determine the size and parameters of the network as there is no rule or formula to do the same. The best that can be done for success with the network is to play with it. The problem with this method is when the system does not work properly it is hard to refine the solution. Despite this issue, neural networks based solution is very efficient in terms of development, time and resources.

There are many ways to implement ANNs. It is difficult to find optimal network architecture, considering the uniqueness of each system or problem. There are numerous neural network simulation softwares which allow fast development of neural networks. These softwares provide menus and graphics to define the network in terms of layers and cells in each layers, the propagation rule, activation rule, output function and learning algorithm. They allow feeding of input/output matched pairs termed as patterns for learning and validation.

The permissible error for the validation set can also be specified. The weights and bias are updated and the network is tuned. The learning terminates either on the basis of number of cycles permitted for learning and validation or on achievement of error values less than the target values specified.

Such simulation softwares can be either executable or open source softwares. The executables like Easy NN, Neural Works etc. come with binary code and provide predefined functionality. On the other hand the open source softwares, in addition to providing some predefined functionality, come with the source code and permit the modification and extension of the software definition. SANN, Genesis etc. are examples of such neural network simulation softwares with open source.

The Biological model

ANNs born after McCulloc and Pitts introduced a set of simplified neurons in 1943. These neurons were represented as models of biological networks into conceptual components for circuits that could perform computational tasks. The basic model of the artificial neuron is founded upon the functionality of the biological neuron. By definition, “Neurons are basic signaling units of the nervous system of a living being in which each neuron is a discrete cell whose several processes are from its cell body”

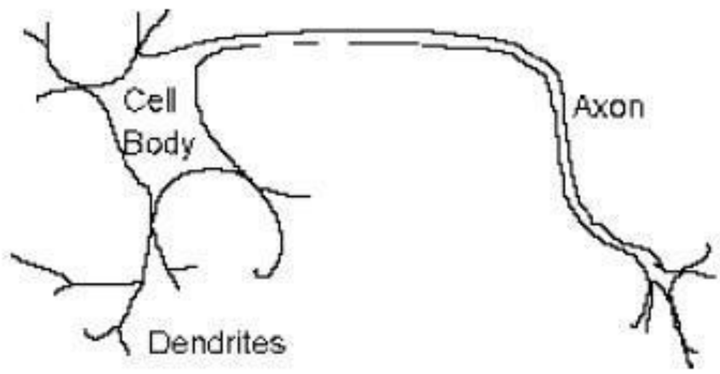


Figure IX.2 Natural Neuron

The biological neuron (Figure IX.2) has four main regions to its structure. The cell body, or soma, has two offshoots from it. The dendrites and the axon end in pre-synaptic terminals. The cell body is the heart of the cell. It contains the nucleolus and maintains protein synthesis. A neuron has many dendrites, which look like a tree structure, receives signals from other neurons.

A single neuron usually has one axon, which expands off from a part of the cell body. This is called the axon hillock. The axon’s main purpose is to conduct electrical signals generated at the axon hillock down its length. These signals are called action potentials.

The other end of the axon may split into several branches, which end in a pre-synaptic terminal. The electrical signals (action potential) that the neurons use to convey the information of the brain are all identical. The brain can determine which type of information is being received based on the path of the signal.

The brain analyzes all patterns of signals sent, and from that information it interprets the type of information received. The myelin is a fatty issue that insulates the axon. The non-insulated parts of the axon area are called Nodes of Ranvier. At these nodes, the signal traveling down the axon is regenerated. This ensures that the signal travel down the axon to be fast and constant.

The synapse is the area of contact between two neurons. They do not physically touch because they are separated by a cleft. The electric signals are sent through chemical interaction. The neuron sending the signal is called pre-synaptic cell and the neuron receiving the electrical signal is called postsynaptic cell.

The electrical signals are generated by the membrane potential which is based on differences in concentration of sodium and potassium ions and outside the cell membrane.

Biological neurons can be classified by their function or by the quantity of processes they carry out. When they are classified by processes, they fall into three categories: Unipolar neurons, bipolar neurons and multipolar neurons.

Unipolar neurons have a single process. Their dendrites and axon are located on the same stem. These neurons are found in invertebrates.

Bipolar neurons have two processes. Their dendrites and axon have two separated processes too.

Multipolar neurons: These are commonly found in mammals. Some examples of these neurons are spinal motor neurons, pyramidal cells and purkinje cells.

When biological neurons are classified by function they fall into three categories. The first group is sensory neurons. These neurons provide all information for perception and motor coordination. The second group provides information to muscles, and glands. There are called motor neurons. The last group, the interneuronal, contains all other neurons and has two subclasses. One group called relay or protection interneurons. They are usually found in the brain and connect different parts of it. The other group called local interneurons are only used in local circuits

The Mathematical Model

When modeling an artificial functional model from the biological neuron, three basic components must be taken into consideration. First, the synapses of the biological neuron are modeled as weights. The synapse of the biological neuron is the one which interconnects the neural network and gives the strength of the connection. For an artificial neuron, the weight is a number, and represents the synapse. A negative weight reflects an inhibitory connection, while positive values designate excitatory connections. The following components of the model represent the actual activity of the neuron cell. All inputs are summed altogether and modified by the weights. This activity is referred as a linear combination. Finally, an activation function controls the amplitude of the output. For example, an acceptable range of output is usually between 0 and 1, or it could be -1 and 1. Mathematically, this process is described in Figure IX.3

From this model the interval activity of the neuron can be shown to be:

$$v_k = \sum_{j=1}^{p} w_{kj} x_j$$

The output of the neuron, y_k, would therefore be the outcome of some activation function on the value of v_k.

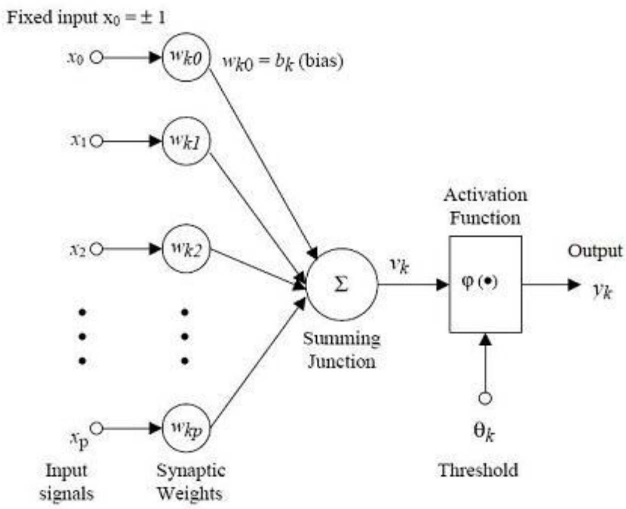


Figure IX.3 Mathematical Model of ANN

Neural Network Topologies

The pattern of connections between the units and the propagation of data can be categorized into two categories.

- **Feed-forward neural networks** where the data ow from input to output units is strictly feed-forward. The data processing can extend over multiple (layers of) units, but no feedback connections are present, that is, connections extending from outputs of units to inputs of units in the same layer or previous layers.
- **Recurrent neural networks** that do contain feedback connections. Contrary to feed-forward networks, the dynamical properties of the network are important. In some cases, the activation values of the units undergo a relaxation process such that the neural network will evolve to a stable state in which these activations do not change anymore. In other applications, the change of the activation values of the output neurons is significant, such that the dynamical behaviour constitutes the output of the neural network.

Training of ANNs

A neural network has to be configured such that the application of a set of inputs produces (either 'direct' or via a relaxation process) the desired set of outputs. Various methods to set the strengths of the connections exist. One way is to set the weights explicitly, using a priori knowledge. Another way is to 'train' the neural network by feeding it teaching patterns and letting it change its weights according to some learning rule. (Refer Figure IX.4)

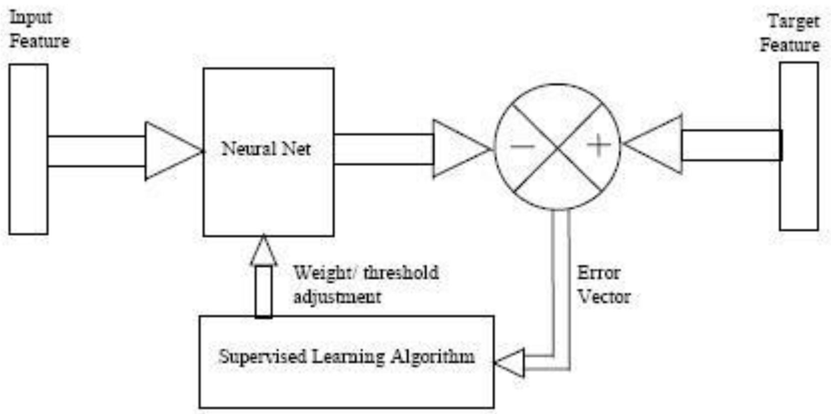


Figure IX.4 Training of ANN

We can categorise the learning situations in two distinct sorts. These are:

- **Supervised learning or Associative learning** in which the network is trained by providing it with input and matching output patterns. These input-output pairs can be provided by an external teacher, or by the system which contains the neural network (self-supervised).

- **Unsupervised learning or Self-organisation** in which an (output) unit is trained to respond to clusters of pattern within the input. In this paradigm the system is supposed to discover statistically salient features of the input population. Unlike the supervised learning paradigm, there is no a priori set of categories into which the patterns are to be classified; rather the system must develop its own representation of the input stimuli.

- **Reinforcement Learning** This type of learning may be considered as an intermediate form of the above two types of learning. Here the learning machine does some action on the environment and gets a feedback response from the environment. The learning system grades its action good (rewarding) or bad (punishable) based on the environmental response and accordingly adjusts its parameters. Generally, parameter adjustment is continued until an equilibrium state occurs, following which there will be no more changes in its parameters. The self organizing neural learning may be categorized under this type of learning.

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PUBLICATIONS FROM THE THESIS (RELATED TO THE PRESENT STUDY)

International Journals

1. Amarnath H. K., P. Prabhakaran, "A Study on The Thermal Performance and Emissions of a Variable Compression Ratio Diesel Engine Fuelled with Karanja Biodiesel and The Optimization of Parameters Based on Experimental Data", *International Journal of Green Energy*, 9: 841–863, 2012,Copyright © Taylor & Francis Group, LLC,
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2. Amarnath H. K., P. Prabhakaran, Sachin Bhat, Ruturraj Paatil, "Comparative Analysis of Thermal Performance and Emission Characteristics of Methyl Esters of Karanja and Jatropha Oils Based Variable Compression Ratio Diesel Engine", International Journal of Green Energy, Taylor & Francis Publication., Accepted for publication, Awaiting Editorial office processing.
3. Amarnath H. K., P. Prabhakaran, "Combustion Analysis of a Single Cylinder Direct Injection Variable Compression Ratio Diesel Engine Fuelled with Karanja Biodiesel", FUEL, Elsevier publication, Under Review.
4. Amarnath H. K., P. Prabhakaran, "An Experimental Investigation on The Performance and Emissions of Biodiesel on a Variable Compression Ratio Diesel Engine Aided By Optimization and Modeling", FUEL, Elsevier publication, Under Review.
5. H.K.Amarnath., P. Prabhakaran., Sachin Bhat, Ruturraj Paatil, "A Comparitive Experimental Study Between The Biodiesels of Karanja, Jatropha And Palm Oils Based on Their Performance and Emissions In a Four Stroke Diesel Engine" Vol. 7, No. 4, April 2012, ISSN 1819-6608, Page 407-413.
6. H.K.Amarnath., P. Prabhakaran,"Comparative Analysis of Karanja and Palm Biodiesels Based on The Thermal Performance and Emissions of a Variable Compression Ratio Diesel Engine" FUEL, Elsevier publication, Submitted to Journal.

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\- **H. K. Amarnath**