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COMBUSTION AND EMISSION ANALYSIS OF A FOUR STROKE IC ENGINE USING (Bio diesel produced from Kranja oil by transesterification process)

ABSTRACT: The performance, emission and combustion characteristics of a single cylinder four stroke variable compression ratio engine when fuelled with blends Karanja methyl ester 20%, 40%, 60% with diesel (on a volume basis) and Kranja oil 10%,20% with diesel(on a volume basis)are investigated and compared with diesel.The suitability of Karanja oil methyl ester as a bio fuel has been established in this study. Bio diesel produced from Kranja oil by transesterification process has been used in this study. Experiment has been conducted at a fixed engine speed of 1500 rpm, 0% load to 100% load and at a compression ratios of 16:1, 17:1 and 18:1. The impact of compression ratio on fuel consumption, combustion pressures and exhaust gas emissions has been investigated and presented. Optimum compression ratio which gives best performance has been identified. The results indicate longer ignition delay, maximum rate of pressure rise, lower heat release rate and higher mass fraction burnt at higher compression ratio for Karanja oil methyl ester when compared to that of diesel and Karanja oil. The brake thermal efficiency for Karanja oil methyl ester blends, Karanja oil and diesel has been calculated and the blend B40 is found to give maximum thermal efficiency. The blends when used as fuel results in reduction of carbon monoxide, hydrocarbon and increase in nitrogen oxides emissions.

NOMENCLATURE

SYMBOLS	DESCRIPTION
BP	Brake Power
BSFC	Brake Specific fuel consumed
BTE	Brake thermal efficiency
B20	20% bio diesel + 80% diesel
B40	40% bio diesel + 60% diesel
B0	60% bio diesel + 40% diesel
CA	Crank angle
CI	Compression Ignition
CO	Carbon monoxide
CO_2	Carbon dioxide
CR	Compression Ratio
HC	Hydro carbon
KOME	Karanja Oil Methyl Ester
K10	10% Karanja oil+90% diesel

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K20	20%Karanja oil+80% diesel
ME	Mechanical efficiency
NHRR	Net heat release rate
NO_x	Nitrogen oxides
Pmax	Maximum pressure
Text	Exhaust temperature
Tmax	Maximum temperature
VCR	Variable Compression Ratio

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Chapter 1

1. INTRODUCTION

1.1. GENERAL BACKGROUND

The diesel engine is a major tool in the day-to-day life of modern society. It powers much of our land and sea transport, provides electrical power, used for farming, construction and industrial activities. The fossil fuel scarcity and pollutant emissions from diesel engines have become two important problems of world today. One method to overcome the crisis is to find suitable substitute for the petroleum based fuels. Bio fuels have been gaining popularity recently as an alternative fuel for diesel engines. Bio fuels are derived from biomass, and are renewable either through agricultural processes or biological waste generation. Bio fuels can be used in any diesel engine, usually without any modifications. It boasts a reduction in toxic emissions (except NOx emissions) compared to diesel fuel. In India million tonnes non edible seeds are going in waste. Utilizing these seeds as an alternative fuel source will reduce the consumption of conventional fuels. The overall objective is to prevent waste, increase the value recovery of resource as bio fuel and minimize the amount of waste going for disposal.

1.2. ENERGY SCENARIO

Energy is one of the major sources for the economic development of any country. India being a developing country requires much higher level of energy to sustain its rate of progress. According to the International Energy Agency (IEA), hydrocarbons account for the majority of India's energy use. Together, coal and oil represent about two-thirds of total energy use. Natural gas now accounts for a seven percent share, which is expected to grow with the discovery of new gas deposits. India had approximately 5.7 billion barrels of proven oil reserves as of January 2011, the second-largest amount in the Asia-Pacific region after China. The combination of rising oil consumption and relatively flat production has left India increasingly dependent on imports to meet its petroleum demand. To combat the present energy crisis, one of the important strategies need to be adopted is to develop and promote appropriate technology for utilizing non-traditional energy resources to satisfy energy requirements. Hence to overcome all these problems most combustion devices are modified to adapt gaseous fuels in dual fuel mode .

1.3. BIODIESEL AS DIESEL SUSTITUTE

For substituting the petroleum fuels used in internal combustion engines, fuels of bio-origin provide a feasible solution to the twin crises of 'fossil fuel depletion' and 'environmental degradation'. For diesel engines, a significant research effort has been directed towards using vegetable oils and their derivatives as fuels. In India, attempts are being made for using non-edible and under-exploited oils for production of esters. Several research institutions are actively pursuing the utilization of non-edible oils for the production of Bio-diesel, additives for lubricating oils, saturated and unsaturated alcohols and fatty acids and many other value added products. Biodiesel has received a good response worldwide as an alternative fuel to diesel. Biodiesel is a cleaner burning fuel because of its own molecular oxygen content. Again in place of diesel, biodiesel can be substituted as the pilot fuel in the dual fuel due to the diminishing reserves of petroleum fuels and rising awareness for protecting our environment. Biodiesel is produced by transesterification process which involves a chemical reaction between an alcohol and triglyceride of fatty acid in the presence of a suitable catalyst leading to the formation of fatty acid alkyl esters (biodiesel) and glycerol . Biodiesel's viscosity is much closer to that of the diesel fuel than vegetable oil. Although biodiesel has many advantages over diesel fuel, there are several problems needs to be addressed such as its lower calorific value, higher flash point, higher viscosity, poor cold flow properties, etc. This can lead to the poor atomization and mixture formation with air that result in slower combustion, lower thermal efficiency and higher emissions . To overcome such limitations of biodiesel some researchers have studied the performance and emissions of the diesel engines with increased injection pressure. Fuel injection pressure and fuel injection timing play a vital role in ignition delay and combustion characteristics of the engine, as the temperature and pressure change significantly close to TDC. The fuels properties also play a significant role to increase or decrease exhaust pollutants . Various investigations clearly reported that cetane number (CN) affects exhaust emissions. The CN also affects the combustion efficiency, and this ensures starting the engine easily. However, if the CN is excessively higher than the normal value, the ignition delay will be too short for the fuel to spread into the combustion chamber. As a result of this unexpected condition, the engine performance will be reduced and the smoke value will increase .

1.4. Advantages of use of biodiesel in VCR engine

As compared to diesel fuel biodiesel has the following advantages.

It is renewable since it is produced from vegetable oils or from animal tallow. So adequate amount of biodiesel can be provided by cultivating the seeds of different vegetable oil plant.

1. It is non-toxic. It does not contain sulphur and any aromatic compounds, but contains 10% to 11% oxygen.
2. The use of biodiesel reduces CO, HC and smoke emission from the engine. The emission reduces due to the oxygen content of biodiesel.
3. It can be used CI engine by blending with diesel or it can also be directly used in the engine without any engine modification.
4. It has a higher flash point than diesel fuel hence it is less volatile, and is safer to store and transport the fuel.
5. It has good lubricant properties with respect to petroleum diesel.

Beside all these advantages, in current scenario biodiesel produced from vegetable oil has higher cost as compared to diesel fuel, which can be minimised by mass production only.

In the present study, the effects of CR on the performance, combustion and emission of a four stroke single cylinder diesel engine using alternative fuels.

Chapter 2

2. LITERATURE REVIEW

2.1 . Introduction

Several targets like reducing the exhaust emissions, reducing engine noise, reducing fuel consumption and increasing specific outputs are the main focus for the researchers. The characteristic at ignition, the affinity of the alternative fuel to mixture formation and combustion speed of the alternative fuel influences the cycle variation. Many researchers worldwide are still concentrating on the further improvement of VCR engines to improve the performance, combustion and emission.

2.2. Previous Works with Alternative Fuels

Varuvel et al. [1] Compared the performance, combustion and emission characteristics of a single cylinder four stroke compression ignition engine fuelled with oil produced from waste fat .The brake thermal efficiency of neat bio fuel is 32.4% at 80% load which was very high compared to neat diesel (29.98%). The combustion duration and ignition delay were decreased with neat bio fuel due to high oxygen content and high cetane number of bio fuel. The main problem with the use of neat bio fuel in diesel engine was high NOx emissions at all loads. Addition of diesel with bio fuel reduced the NOx emissions significantly from 917 ppm to 889 ppm at 80% load with an optimum blend of B80D20.

Swaminath et al.[2] Studied the performance and emission of single cylinder four stroke diesel engine using biodiesel(fish oil) at a constant speed 1500 rpm. Fuel was blended with oxygenate and EGR technique was also used to improve the performance. He found there is increased in break thermal efficiency. The percentage reduction was CO-91% ,CO2-62%,NOx-92% and CxHy-90% were attained when the engine was run at maximum load using BFOwith2%additive with EGR and there was reduction in all the percentages when the engine was run in other loads also. In the case of NOx, there was an increase of this emission by about 48%in the maximum load with BFO when compared with diesel, for obvious reasons and that was also reduced because of the addition of oxygenates and EGR.

K. Muralidhran et al.[3] studied the performance, emission and combustion of VCR engine using methyl esters of waste cooking oil and its blends with standard diesel in a compression ignition engine. Experiments were carried out for diesel, B20, B40, B60, B80. Maximum brake thermal efficiency for B40 at full load was found to be 4.1% higher than diesel. Brake specific fuel consumption of the blends B20, B40 at full load was found to be lower than diesel. Exhaust temperatures of blends were lower than diesel. Combustion pressure of B40 was found to be higher than that of diesel. From the analysis of exhaust emission of the blends, it was found that the hydrocarbon emissions of various blends were higher at higher loads except B20. The NOx emission for biodiesel and its blends was higher than that of standard diesel except B40 at lower loads. The CO emission of the blend B40 was closer to that of standard diesel and it was found to be higher for light and medium load.

K.Muralidhran et al.[4] studied the performance, emission and combustion of VCR engine using methyl esters of waste cooking oil and its blends with standard diesel at varying compression ratio and engine loading at a constant speed .Brake thermal efficiency of blend B40 was higher than that of diesel at higher compression ratio. The specific fuel consumption of B40 was lower than that of all other blends and diesel. Exhaust gas temperature of blends was found to be lower than diesel. Combustion pressure of waste cooking oil was found to be higher than diesel. The hydrocarbon emission of various blends was higher at higher compression ratios. The increase in compression ratio increases the HC emission for blend B40. The emission of oxides of nitrogen (NOx) from the waste cooking oil blend B40 is higher than that of diesel. The CO emission of the blend B40 is closer to the standard diesel and it is very higher at compression ratio. The CO_2 emission is also lesser at the same conditions.

Jindal et al. [5] studied that the effects of the engine design parameters viz. compression ratio and fuel injection pressure on the performance with regard to parameters such as fuel consumption, brake thermal efficiency and emissions of CO,CO_2,HC(hydrocarbon), NOx and Smoke opacity with Jatropha methyl ester as fuel. It is found that the combined increase of compression ratio and injection pressure increases the brake thermal efficiency and reduces brake specific fuel consumption while having lower emissions.

Saravanan et al. [6] studied that the combustion characteristics of a stationary diesel engine fuelled with a blend of crude rice bran oil methyl ester and diesel. It was observed that the delay period and the maximum rate of pressure rise for crude rice bran oil methyl ester blend were lower than those of diesel. The occurrence of maximum heat release rate advanced for crude rice bran oil methyl ester blend with lesser magnitude when compared to diesel. This investigation ensures that the suitability of crude rice bran oil methyl ester blend as fuel for CI engines with higher fuel cost.

Prem Anand et al. [4] calculated the combustion performance and exhaust emission Characteristics of turpentine oil fuel blended with conventional diesel fuel in a diesel engine.

Yousef Heik et al. [7] carried out an experimental study to use raw Algae oil and its methyl esters in a Ricardo E6 variable compression ratio engine. Effects of engine speed, engine load output, injection timing of the algae bio fuel and engine compression ratio on the engine output torque, combustion noise(maximum pressure rise rate), maximum pressure and maximum heat release rate have been studied. However, its use reduced the engine output torque slightly and increased the combustion noise. The engine output can be increased and the combustion noise can

be reduced by controlling the engine design parameters e.g. injection timing and compression ratio.

Kalam et al. [8] evaluated the emission and performance characteristics of a multi-cylinder diesel engine operating on waste cooking oil such as 5% palm oil with 95% ordinary diesel fuel and 5% coconut oil with 95% ordinary diesel fuel. B0 was used for comparison purposes. The results shows that there are reductions in brake power of 1.2% and 0.7% and reduction of exhaust emissions such as unburned hydrocarbon (HC), smoke, carbon mono-oxide(CO), and nitrogen oxides (NOx) is offered by the blended fuels.

Mani et al. [9] studied the effect of using waste plastic oil and diesel blends in compression ignition engine. It is observed that the engine could operate with 100% waste plastic oil and can be used as fuel in diesel engines. Oxides of nitrogen (NOx) was higher by about 25% and carbon monoxide (CO) increased by 5% for waste plastic oil operation compared to diesel fuel operation. Hydrocarbon was higher by about 15%. Smoke increased by 40% at full load with waste plastic oil compared to diesel. Engine fuelled with waste plastic oil exhibits higher thermal efficiency up to 80% of the full load and the exhaust gas temperature was higher at all loads compared to diesel fuel operation.

Gumus and Kasifoglu [10] studied the performance and emissions of a compression ignition diesel engine without any modification, using neat apricot seed kernel oil methyl ester and its blends with diesel fuel and found that lower concentration of apricot seed kernel oil methyl ester in blends give a better improvement in the engine performance and exhaust emissions.

Ismet Celikten et al. [11] compared the performance and emissions of diesel fuel from rapeseed and soybean oil methyl esters injected at different pressures (250,300 and 350 bar). It has been found that the torque and power of diesel fuel engine reduced with increasing injection pressure. Smoke level (%) and CO emission also reduced while NOx emission increased as the injection pressure is increased.

Raheman and Ghadge [12] studied the performance of Ricardo E6 engine using bio diesel obtained from mahua oil (B100) and its blend with high speed diesel at varying compression ratio, injection timing and engine loading. The brake specific fuel consumption and exhaust gas temperature increased, whereas brake thermal efficiency decreased with increase in the proportion of bio diesel in the blends for all compression ratios (18:1–20:1) and injection timings (35–45_ before TDC). The authors concluded that, bio diesel could be safely blended with HSD up to 20% at any of the compression ratio and injection timing tested for getting fairly accurate performance as that of diesel.

2.3. Conclusion of Present Review

Petroleum resources are finite and, therefore, search for their alternative non-petroleum fuels for internal combustion engines is continuing all over the world. Moreover gases emitted by petroleum fuel driven vehicles have an adverse effect on the environment and human health. There is universal acceptance of the need to reduce such emissions. The researchers have performed their experiment with different alternative fuels like CNG, LPG, bio fuels, producer gas, biogas, syngas, and acetylene etc. It is observed that biofuels are more acceptable to the diesel engines because of their better combustion and performance characteristics. At the same time while choosing the blends of biodiesel the fuel blends are being used in the engine designed exclusively for diesel fuel operation, and thus are restricted upto 40% mixing only [24].

2.4. Scope of Present Investigation

It seems that with diesel or blends of biodiesel will be a promising technique for controlling performance, combustion and emissions from the diesel engines. It is found that the lower concentrations of biodiesel blends improve the thermal efficiency. Reduction in emission and brake-specific fuel consumption (BSFC) is also observed while using B-40. From various experiments conducted, it has been found that all the emission parameters like HC, CO and soot emission decreases by the use of bio-diesel but NO_x emission increases due to the high oxygen content and proper combustibility of bio-diesel. As the engines are designed to operate with diesel, so for use of higher blends of biodiesel modifications are needed in the fuel injection system. After critical review of the available literatures on the use of biodiesel as alternative fuel it was decided to carry out detailed experiment with the different blends KOME. Varying the CR ranging from 16:1 to 18:1.with varying loads to optimize the CR to a value which would give the best performance of engine and reduces the emissions which are harmful to the environment.

2.5. Objectives

- Fuel property testing of a locally available non edible vegetable oil(karanja oil) at different temperatures to know the diesel engine compatibility.

- Effect of CR on performance of a single cylinder 4 stroke compression ignition engine using blends of Karanja oil methyl ester and neat Karanja oil with diesel.

- Effect of CR on combustion of a single cylinder 4 stroke compression ignition engine using blends of Karanja oil methyl ester and neat Karanja oil with diesel.

Effect of CR on emission of a single cylinder 4 stroke compression ignition engine using blends of Karanja oil methyl ester and neat Karanja oil with diesel.

3.EXPERIMENT

3.1Introduction

This chapter discusses the principles and methodologies that have been used during the course of several experimental investigations in the variable compression ratio engine engine test rig. A single cylinder four stroke 3.5kW, 1500 rpm, water cooled diesel engine was selected for the study. The instruments fitted to the test bed were properly calibrated to minimize the possible errors during experimentation. The results of this study can give ample guidelines to utilize bio fuel and neat Karanja in variable compression ratio engine with blends of biodiesel in variable compression ratio diesel engines.

3.2 Measurement of specific gravity.

The specific gravity of all fuel blends (neat Karanja oil, blended neat oil with diesel, 100% biodiesel and blended biodiesel with diesel) are measured as per standard ASTM D4052 at varying temperature using different hydrometers.

Fig 1 Oil bath and hydrometer

Table -1 Hydrometer specification

Range	To suit surface tension	Catlog no.	Type	Interval
0.750-0.800	Low	1010	M-50SP	0.001gm/ml
0.800-0.850	Low	1011	L-50SP	0.0005gm/ml
0.850-0.900	Low	1012	M-50SP	0.001gm/ml
0.900-0.950	Low	1013	M-50SP	0.001gm/ml
0.950-1.0	Low	1014	M-50SP	0.001gm/ml

3.3 Measurement of Viscosity:-

When a fluid is subjected to external forces, it resists flow due to internal friction. Viscosity is a measure of internal friction. The viscosity of the fuel affects atomization and fuel delivery rates. It is an important property because if it is too low and too high then atomization, mixing of air and fuel in combustion chamber gets affected. Viscosity studies are conducted for different fuel blends (neat Karanja oil, blended neat oil with diesel, 100% biodiesel and blended biodiesel with diesel). Kinematic viscosity of liquid fuel samples are measured using the viscometer at different temperatures and blendings as per specification given in ASTM D445, using Cannon-Frensky viscometer tubes in viscometer oil bath.

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Fig 2 oil bath with viscometer

3.4 Acidic value calculation:-

Step-1

10 ml of ethanol was taken in a beaker .10 drops of Phenopthalin indicator was added to it. Then it was titrated with N/100 KOH. Amount of base required for the neutralization of the sample was measured. Three times the same procedure was done and three observations were taken.

Step-2

Again 10 ml of ethanol was taken in a beaker.0.5 gm of oil was added to it .It was stirred for 15 minutes. Then it was titrated with N/100 KOH.10 drops of Phenopthalin indicator was added to it .The sample was titrated with N/100 KOH. Amount of base required for neutralization was measured. Same procedure was done for three times three observations were taken.

Table 2 Observation table for titration step-1

Observation	IBR	FBR	Difference	Mean(V_1)
1	0	2.1	2.1	
2	2.1	4.2	2.1	2.06
3	4.2	6.2	2.0	

Table 3 Observation table for titration step-2

Observation	IBR	FBR	Difference	Mean(V_2)
1	6.2	20.3	14.1	
2	20.3	36.2	15.9	15.63
3	29.0	45.9	16.9	

Acid value= $(56 \text{ x } (V_2\text{-}V_1) \text{ X } 2)/100000=15.18$

At the time of esterfication same procedure was carried out taking samples from the reactor .When the Acid value was found to below 5 .It was then transferred to the separating funnel .

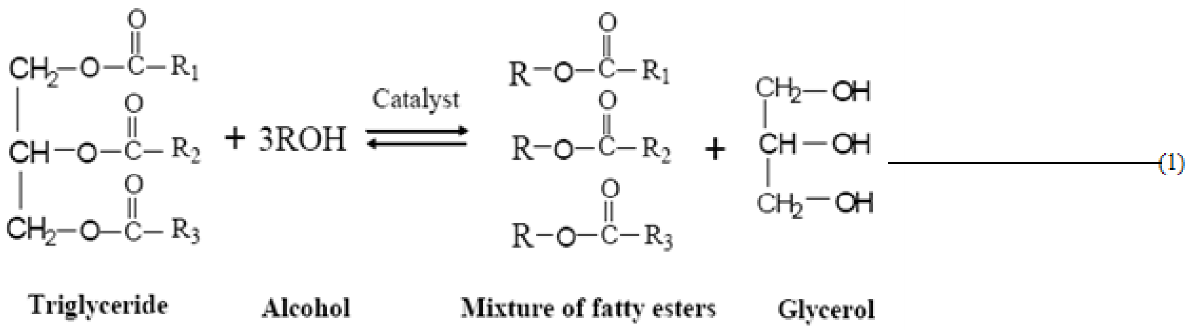
3.5 Biodiesel preparation from Karanja vegetable oil

There are two basic routes to biodiesel production from oils and fats:

1. Conversion of the oil to its triglyceride.
2. Base catalyzed transesterification of the triglyceride to its biodiesel.

To prepare Karanja biodiesel from the oil the acid value of the oil is measured using titration technique. The acid value of the oil is found to be 6.3 mg KOH/g. The acid value of the oil is decreased by esterification method. In esterification method acid catalyzed reaction of the oil with alcohol is done. Samples are taken from the products of reaction in every half an hour and acid value measured. The reaction is stopped when acid value of the oil comes below 5 mgKOH/g. The products of reaction are then allowed to be settled in a separating funnel.

After 24 hours of settling triglyceride comes to the bottom of the funnel and unreacted alcohol comes to the top. The triglyceride of Karanja oil is then collected from the bottom of the separating funnel. The triglyceride of the oil is transesterified to produce biodiesel or Karanja oil methyl ester on the same reactor. Then reaction is carried out at 60^0C with continuous stirring. The transesterification is represented as below in equation (1).



(1)

The products of the reaction is kept in the separating funnel and allowed to settle down. Glycerol is settled at the bottom of the separating funnel and collected. The top part of the funnel contains Karanja oil methyl ester or Karanja biodiesel. The dissolved water particle is removed by heating the collected Karanja biodiesel. Biodiesel are blended with diesel fuel at 10%, 20%, 30%-------100% on a volume basis.

3.5.1 Mixing of alcohol and catalyst

The catalyst is typically sodium hydroxide (caustic soda) or potassium hydroxide(potash). It is dissolved in the alcohol using a standard agitator or mixer. The alcohol/catalyst mix is then charged into a closed reaction vessel and the oil or fat is added. The system from here on is totally closed to the atmosphere to prevent the loss of alcohol. The reaction mixture is kept just above the boiling point of the alcohol (around 60 °C) to speed up the reaction and the reaction takes place. Recommended reaction time varies from 1 to 8 hours, and some systems recommend the reaction take place at room temperature. Excess alcohol is normally used to ensure total conversion of the fat or oil to its esters. Care must be taken to monitor

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the amount of water and free fatty acids in the incoming oil or fat. If the free fatty acid level or water level is too high it may cause problems with soap formation and the separation of the glycerin by-product downstream.

Fig.3. Biodiesel reactor

3.5.2 Separation

Once the reaction is complete, two major products are obtained: glycerin and biodiesel. Each has a substantial amount of the excess alcohol that was used in the reaction. The reacted mixture is sometimes neutralized at this step if needed. The glycerin is much more dense than biodiesel and the two can be gravity separated with glycerin simply drawn off the bottom of the settling vessel. In some cases, a centrifuge is used to separate the two materials faster.

Fig.4.Transesterification

3.5.3. Alcohol Removal

Once the glycerin and biodiesel phases have been separated, the excess alcohol in each phase is removed with a flash evaporation process or by distillation. In others systems, the alcohol is removed and the mixture neutralized before the glycerin and esters have been separated. In either case, the alcohol is recovered using distillation equipment and is re-used. Care must be taken to ensure no water accumulates in the recovered alcohol stream.

3.5.4. Glycerine Neutralization

The glycerin by-product contains unused catalyst and soaps that are neutralized with an acid and sent to storage as crude glycerin. In some cases the salt formed during this phase is recovered for use as fertilizer. In most cases the salt is left in the glycerin. Water and alcohol are removed to produce 80-88% pure glycerin that is ready to be sold as crude glycerin. In more sophisticated operations, the glycerin is distilled to 99% or higher purity and sold into the cosmetic and pharmaceutical markets.

Fig.5. Water washing

3.5.5. Methyl Ester Wash

Once separated from the glycerin, the biodiesel is sometimes purified by washing gently with warm water to remove residual catalyst or soaps, dried, and sent to storage. In some processes this step is unnecessary. This is normally the end of the production process resulting in a clear amber-yellow liquid with a viscosity similar to mineral diesel. In some systems the biodiesel is distilled in an additional step to remove small amounts of color bodies to produce a colorless biodiesel. After water washing the biodiesel is heated at about 80^0C to remove unwanted moisture.

3.6 Experimental set up

Fig 6 Actual Engine setup

The principles and methodologies that have been used during the course of several experimental investigations in the VCR diesel engine test rig consisting of a 1 cylinder, 4 stroke, power 3.5 kW at 1500 rpm Diesel engine connected to eddy current dynamometer in computerized mode to study the performance, emission and combustion of engine by varying its CR at different load condition from 0kg to 12kg using various blends of KOME and diesel. The detailed specification of the engine is shown in Table 1. Engine performance analysis software package “EnginesoftLV” have been employed for online performance analysis. Piezo sensor and crank angle sensor which measures the combustion pressure and the corresponding crank angle respectively are mounted into the engine head. The output shaft of the eddy current dynamometer is fixed to a strain gauge type load cell for measuring applied load to the engine.

Type K-Chromel (Nickel-Chromium Alloy)/ Alumel (Nickel- Aluminium Alloy) thermocouples is used to measure gas temperature at the engine exhaust, calorimeter exhaust, water inlet of calorimeter and water outlet of calorimeter, engine cooling water outlet and ambient temperature. The fuel flow is measured by the use of 50ml burette and stopwatch with level sensors.

3.6.1 Engine specification and its attachements.

Table :-4 Engine specification

Make	Kirloskar
General details	VCR Engine test setup 1 cylinder, 4 stroke, Water cooled, compression ignition
Rated power	3.5Kw at 1500 rpm
Speed	1500 rpm(constant)
Number of cylinder	Single cylinder
Compression ratio	16:1 to 18:1(variable)
Bore	87.5 mm
Stroke	110 mm
Ignition	Compression ignition
Loading	Eddy current dynamometer
Load sensor	Load cell, type strain gauge 0-50 Kg
Temperature sensor	Type RTD, PT100 and Thermocouple, Type K
Cooling	Water
Air flow transmitter	Pressure transmitter, Range (-) 250 mm WC
Rotameter	Engine cooling 40-400 LPH; Calorimeter 25-250 LPH
Software	“EnginesoftLV” Engine performance analysis software
Propeller shaft	With universal joints
Air box	M S fabricated with orifice meter and manometer
Fuel tank	Capacity 15 lit with glass fuel metering column
Calorimeter	Type Pipe in pipe
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.
Digital milivoltmeter	Range 0-200mV, panel mounted
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
Pump	Type Monoblock
Overall dimensions	W 2000 x D 2500 x H 1500 mm

3.6.2. Compression ratio adjustment

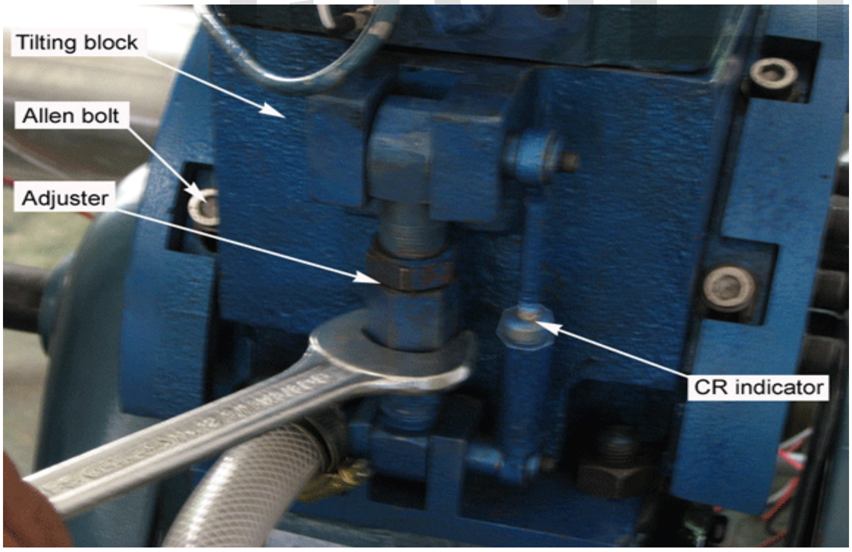

Fig 7

Slightly loosen 6 Allen bolts provided for clamping the tilting block. Loosen the lock nut on the adjuster and rotate the adjuster so that the compression ratio is set to “maximum”. Refer the marking on the CR indicator.Lock the adjuster by the lock nut.Tighten all the 6 Allen bolts gently.You may measure and note the centre distance between two pivot pins of the CR indicator. After changing the compression ratio thedifference (Δ) can be used to know new CR.

3.6.3 Dynamometer

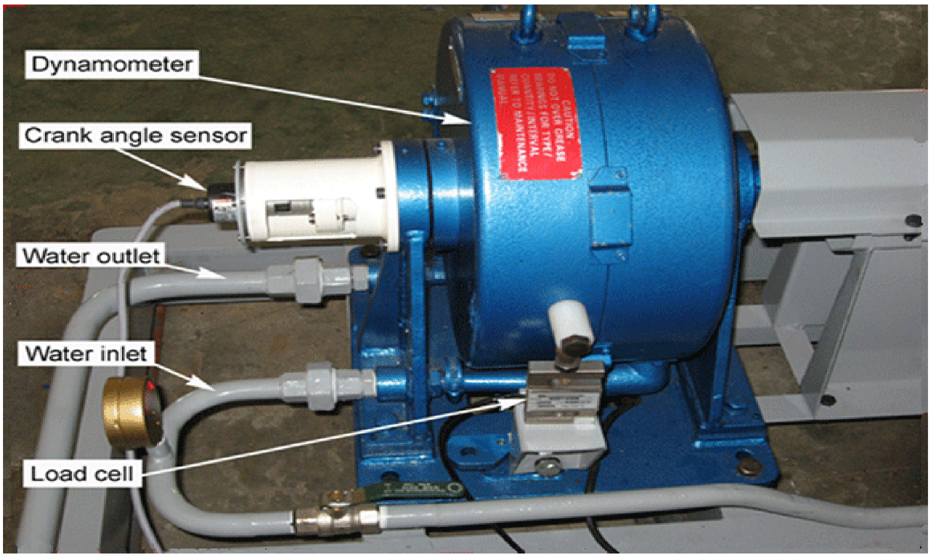

Fig 8

It is an absorption type of eddy current water cooled dynamometer. It is a device used for loading unit. Here load is measured by a strain gauge type load cell.

3.6.4 Multi gas analyser:-

Multi - Gas Analyzer capable of measuring CO, HC, CO_2, O_2 & NOx (optional) contents in the exhaust. The AVL-444 analyzer provides optimized analysis methods for different applications. It is specifically designed for the automobile industry.

Designed using latest technology, these analyzers can easily check the pollution level of various automobiles, elegant and smart in appearance. The analyzers are easy to install and known for their efficient functioning. Further, the range is tested on various parameters in order to meet the set industrial standards. The specifications of accuracy for measurement of various parameters are given in Table-1.

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Table-5: Measurement range and accuracy of AVL 444 gas analyzer

Measured Quality	Measuring Range	Resolution	Accuracy
CO	0…10% vol.	0.01% vol.	<0.6% vol: ±0.03% vol. >=0.6% vol: ±5% of ind value
CO_2	0…20% vol.	0.1% vol.	<10% vol: ±0.5% vol. >=10% vol: ±5% vol.
HC	0…20000 ppm vol	<=2000:1 ppm vol. >2000:10 ppm vol.	<200 ppm vol: ±10 ppm vol. >=200 ppm vol: ±5% of ind. val.
O_2	0…22% vol.	0.01% vol.	<2% vol: ±0.1% vol. >=2% vol: ±5% vol.
NO	0…5000 ppm vol.	1 ppm vol.	<500 ppm vol: ±50 ppm vol. >=500 ppm vol: ±10% of ind. val.
Engine Speed	400…6000 min^{-1}	1 min^{-1}	±1% of ind. val.
Oil Temperature	-30…125^{0}C	1^{0}C	±4^{0}C
Lambda	0…9.999	0.001	Calculation of CO, CO_2, HC, O_2

Fig. 9 Multi gas analyser

3.6.5 Smoke meter:- device use to measure smoke opacity.

Fig. 9 Smoke meter

Table-6: Measurement range and accuracy of Smoke meter

Measured Quality	Measuring Range	Resolution	Accuracy
Opacity	0…100%	0.1%	±%of full scale
Absorption	0…99.99m-1	0.01 m-1	Better than± 0.1 m-1
RPM	400-6000 1/ min	11 1/ min	±10
OilTemperature	00…150^{o}C	1^{o}C	±3^{o}C

3.7 Experimental lay out:-

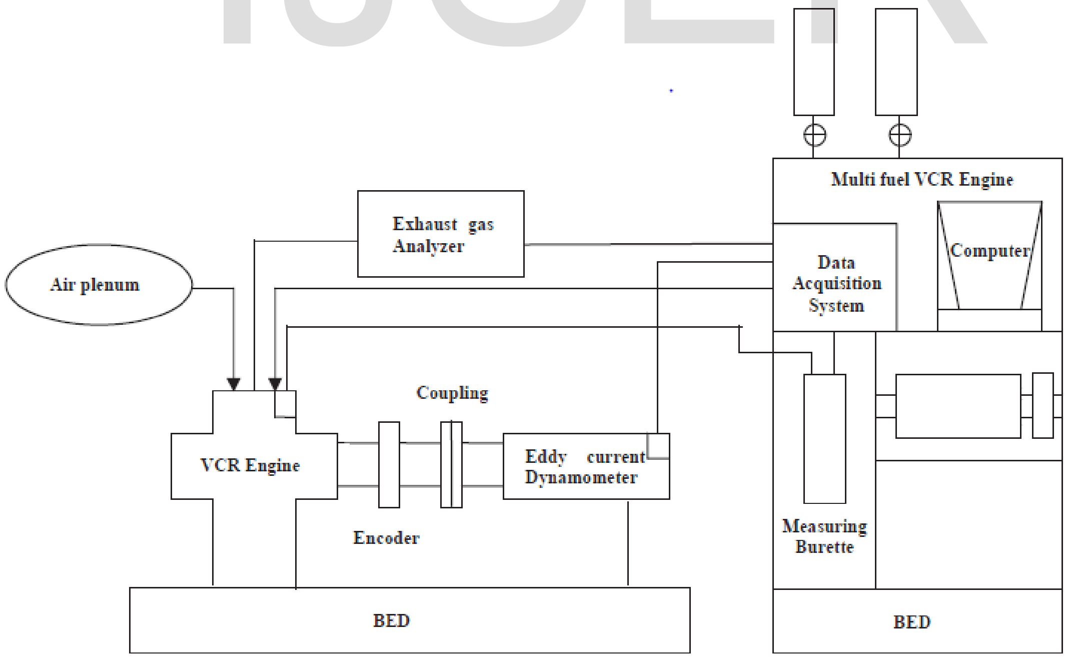

Fig.11 Lay out of VCR engine

3.8 Experimental procedure:-

The variable compression ratio engine is started by using standard diesel. It is run for 30 minutes. When the engine is warmed up then readings are taken. The tests are conducted at the rated speed of 1500 rpm. Fuel consumption is measured by the help of the measuring burette attached to the data acquisition system. In every test, brake thermal efficiency, brake specific fuel consumption, exhaust gas temperature, mechanical efficiency, torque and combustion parameters like combustion pressure, combustion temperature, ignition delay, net heat release rate, combustion duration and exhaust gas emissions such as carbon monoxide (CO), carbon dioxide (CO_2), hydrocarbon (HC), nitrogen oxides (NO_x), and smoke opacity are measured. From the initial measurement, performance, combustion and emission parameters with respect to compression ratio 16:1, 17:1 and 18:1 at 100% load for different blends are calculated and recorded. Also the engine operating parameters such as performance, combustion and emission with respect to different loads for different blends at compression ratio 18 are measured and recorded. At each operating conditions, the performance and combustion characteristics are also processed and stored in personal computer (PC) for further processing of results. The same procedure is repeated for different blends of Karanja oil methyl esters.

3.9 Conclusion

The specific gravity of biodiesel fuels is lower than that of straight vegetable oil. Therefore, the specific gravity of the blend increases with the increase of biodiesel concentration. Also, the specific gravity shows a inverse relationship with temperature. As the temperature is increased, specific gravity decreases. The viscosity of biodiesel is found lower than that of straight vegetable oil. The viscosity of the blend increases with the increasing biodiesel fraction for all. Similar to the effect of temperature on specific gravity, viscosity also shows linearly inverse trend i.e increasing temperature reduces the viscosity. This property helps in better atomisation and hence fuel burning in application of biodiesels. It has been noticed that the specific gravity and the viscosity of the biodiesel blends increase with increase of the biodiesel fraction. It is also seen that the specific gravity and viscosity of each blend decreases with increase in the temperature.

4. Result and Discussion

4.1 Fuel property testing at different blend and temperature.

4.1.1 Specific gravity blend of Karanja at different temperature:- Reffering to table number{6} it can be seen that specific gravity go on decreasing for all the blend with increase in temperature. It is found that the specific gravity of neat karanja oil (K100) varies from 0.925 to 0.878 at a temperature range of 30-100°C.

Table-7 variation of specific gravity with temperature and blend (neat karanja)

Temp / Blend	30 ^{0}C	40 ^{0}C	50 ^{0}C	60 ^{0}C	70 ^{0}C	80 ^{0}C	90 ^{0}C	100 ^{0}C
K-10	0.827	0.8215	0.814	0.807	0.800	0.793	0.785	0.780
k-20	0.8375	0.832	0.825	0.820	0.812	0.8055	0.797	0.791
k-30	0.850	0.843	0.836	0.830	0.823	0.816	0.810	0.803
k-40	0.860	0.854	0.847	0.840	0.834	0.827	0.821	0.816
k-50	0.871	0.866	0.860	0.853	0.845	0.838	0.833	0.8265
k-60	0.883	0.877	0.870	0.866	0.859	0.852	0.846	0.840
k-70	0.892	0.886	0.880	0.873	0.866	0.860	0.853	0.846
k-80	0.905	0.898	0.891	0.884	0.877	0.871	0.866	0.858
k-90	0.916	0.913	0.904	0.895	0.887	0.881	0.874	0.870
k-100	0.925	0.919	0.914	0.907	0.901	0.891	0.885	0.878

4.1.2 Viscosity of blend of Karanja at different temperature:- Reffering to the table {7} it can be observed that viscosity for all blend go on decreasing with increase in temperature. It is found that the viscosity of neat Karanja oil blends (K10) varies from 4.116 cSt to 2.2912 cSt at a temperature range of 30 to 100°C.

Table- 8 variation of viscosity with temperature and blend (neat karanja)

Temp / BLEND	30^0C	40^0C	50^0C	60^0C	70^0C	80^0C	90^0C	100^0C
K-10	4.116	3.238	2.351	2.043	1.763	1.2572	2.0572	2.2912
K-20	4.976	4.354	3.421	2.724	2.332	2.052	1.842	1.7
K-30	8.061	6.563	4.5	3.695	2.852	2.612	2.316	2.005
K-40	10.344	8.382	6.420	5.25	4.189	3.586	2.789	2.556
K-50	14.821	11.948	8.917	6.920	4.683	3.933	3.494	3.147
K-60	18.678	14.789	10.898	9.024	7.384	5.305	4.372	4.189
K-70	27.167	20.686	16.562	11.185	9.667	7.633	6.563	4.226
K-80	34.513	24.602	17.645	12.475	9.956	8.989	8.097	7.491
K-90	36.478	30.665	23.436	16.753	12.785	10.636	9.631	8.561
K-100	58.1324	42.785	32.173	22.228	13.256	11.649	7.589	9.346

4.1.3 Specific gravity of blend of bio diesel at different temperature:- The specific gravity of karanja biodiesel (B100) varies between 0.891 to 0.841 at same temperature range . Many studies show that biodiesel's specific gravity is not changed a lot, because the densities of methanol and oil is close to the specific gravity of the produced biodiesel. The specific gravity of biodiesel is found higher than that those of diesel fuel. As the specific gravity of biodiesel is high compared to diesel fuel so the energy content of it is lower . Specific gravity of biodiesel will vary with the fatty acid composition and their purity.

Table -9 variation of specific gravity with temperature and blend (karanja oil methyl ester)

	30 ^{0}C	**40 ^{0}C**	**50 ^{0}C**	**60 ^{0}C**	**70 ^{0}C**	**80 ^{0}C**	**90 ^{0}C**	**100 ^{0}C**
B-10	.826	.8205	.813	.8055	.798	.792	.785	.779
B-20	.8375	.831	.823	.817	.8105	.8035	.796	.790
B-30	.8425	.836	.830	.823	.8155	.808	.8025	.796
B-40	.849	.8435	.8355	.8285	.822	.8155	.809	.803
B-50	.854	.847	.840	.8325	.8265	.820	.8125	.806
B-60	.860	.854	.846	.840	.833	.826	.820	.814
B-70	.867	.861	.854	.846	.837	.832	.826	.820
B-80	.871	.865	.859	.852	.845	.837	.831	.824
B-90	.884	.878	.872	.865	.858	.851	.843	.837
B-100	.891	.885	.877	.869	.862	.855	.848	.841

4.1.4 Viscosity of blend of bio diesel at different temperature:- viscosity of biodiesel blends (B10-B100) at 40°C varies from 2.18 cSt to 4.5249 cSt.

Table- 10 variation of viscosity with temperature and blend (karanja oil methyl ester)

	30 ^{0}C	**40 ^{0}C**	**50 ^{0}C**	**60 ^{0}C**	**70 ^{0}C**	**80 ^{0}C**	**90 ^{0}C**	**100 ^{0}C**
B-10	2.6963	2.1831						
B-20		2.4164	1.9779					
B-30		2.7149	2.1738	1.8753				
B-40	3.452	2.8456	2.3044	1.9592				
B-50		3.1254	2.603	2.1738	1.8939			
B-60		3.3494	2.7896	2.3697	2.0245			
B-70		3.7225	3.0788	2.631	2.2951	1.9592		
B-80		3.8252	3.1628	2.7149	2.3324	2.0059		
B-90		4.2637	3.4706	2.8829	2.4537	2.1085		
B-100		4.5249	3.7225	3.0135	2.5190	2.1458		

4.2 PERFORMANCE ANALYSIS OF KARANJA OIL METHYL ESTER

4.2.1 Brake specific fuel consumption

From Fig 4.2.1.1 it can be observed that the BSFC of the engine decreased with increase in load. But the BSFC for the KOME blends is found to be higher compared to that of diesel at all the loads. The BSFC varies with diesel, B20, B40, and B60 at full load is found to be 0.34 kg/kWh, 0.34kg/kWh, 0.32kg/kWh and 0.39kg/kWh respectively. At higher percentage of the blends, the BSFC increases.

The brake specific fuel consumption of the blend B40 at the compression ratio of 18 is 0.32kg/kWh whereas for diesel it is 0.34kg/kWh. At higher percentage of blends, the specific fuel consumption increases. This is due to the decrease in calorific value for the higher blends. The specific energy consumption decreases with the increase in compression ratio.

The BSFC of the engine with the Karanja oil methyl ester blends as fuels compared with its diesel fuel operation at various loads in Fig 4.2.1.2

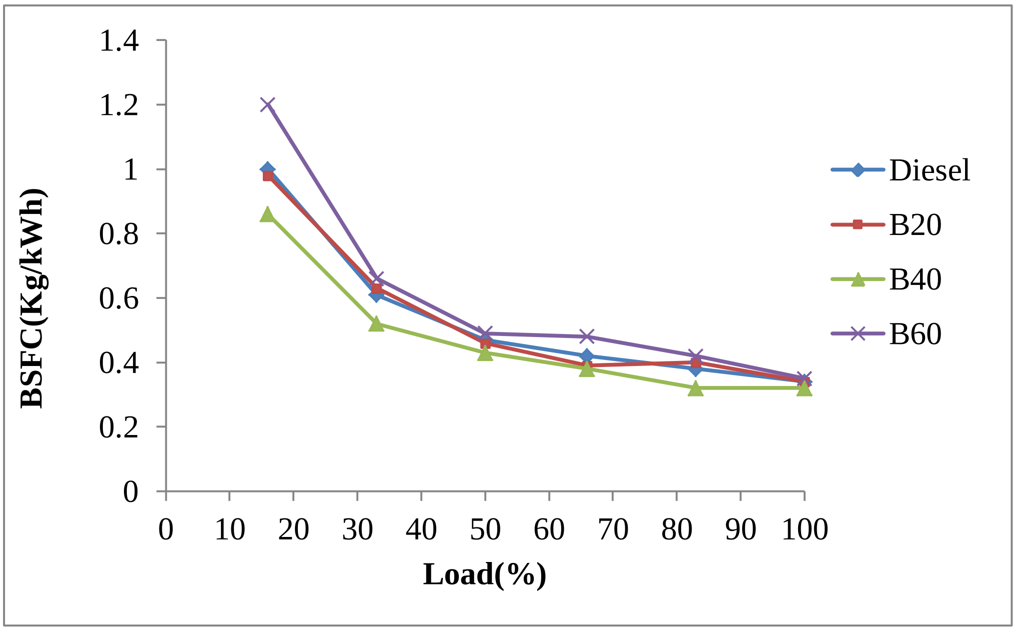

Fig 4.2.1.1

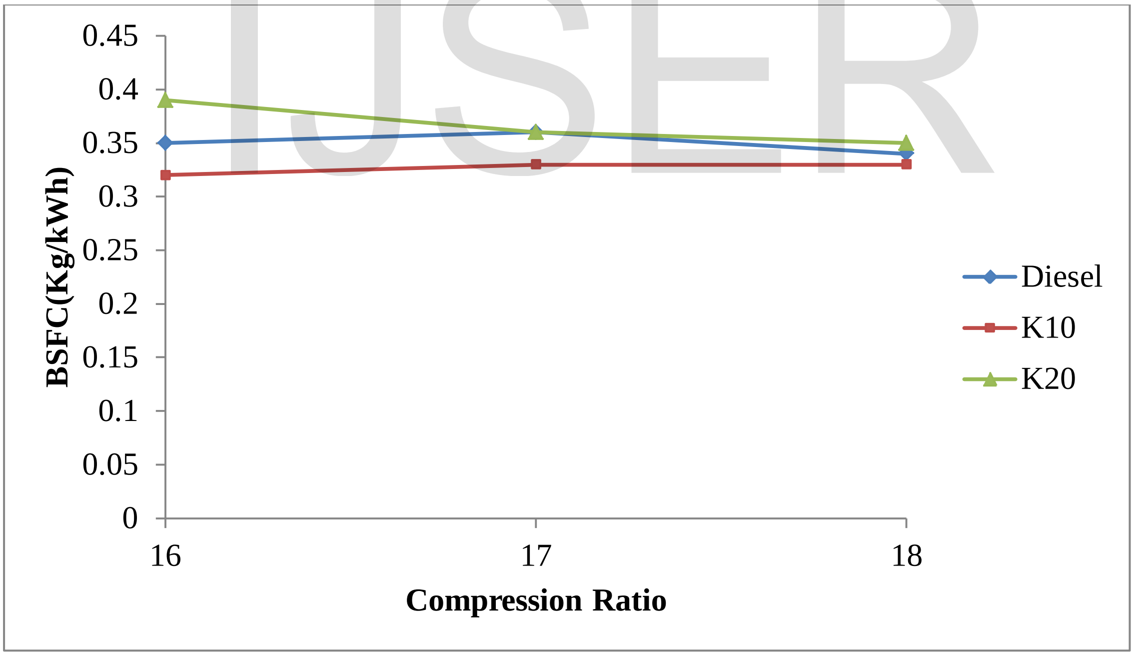

Fig 4.2.1.2

4.2.2 Brake thermal efficiency

The variation of brake thermal efficiency (BTE) for different loads and for different fuels is given in Fig 4.2.2.1 It is seen that there is a steady increase in efficiency with increases in load in all the fuel operations. It is happened due to reduction in heat loss and increase in power developed with increase in load. The engine BTE at full load for diesel, B20, B40, and B60 fuels is 24.9%, 26.02%, 28.44% and 24.1% respectively. It is also observed that the BTE of the blend B40 is slightly higher than that of the diesel at compression ratio 18.The decrease in brake thermal efficiency for higher blends may be due to the combined effect of its lower heating value and increase in fuel consumption.

The variation of brake thermal efficiency (BTE) for different compression ratio and for different blends is given in Fig 4.2.2.2. It is observed that the BTE of the blend B40 is slightly higher than that of the diesel at higher compression ratio. It appears that the BTE of the blend B40 is higher for the compression ratio 18 and full load. The BTE of the diesel and blend B40 for compression ratio 18 is 24.9% and 28.44% respectively. By increasing the compression ratio of the engine, the BTE also gets increased for all the fuel types tested. BTE is directly proportional to the compression ratio.

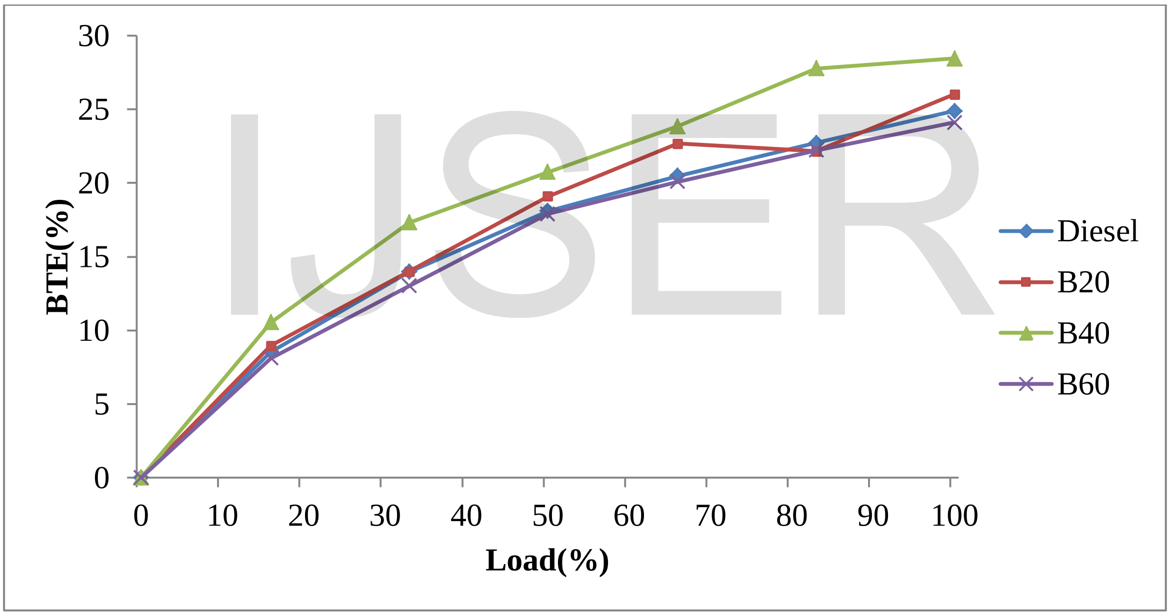

Fig 4.2.2.1

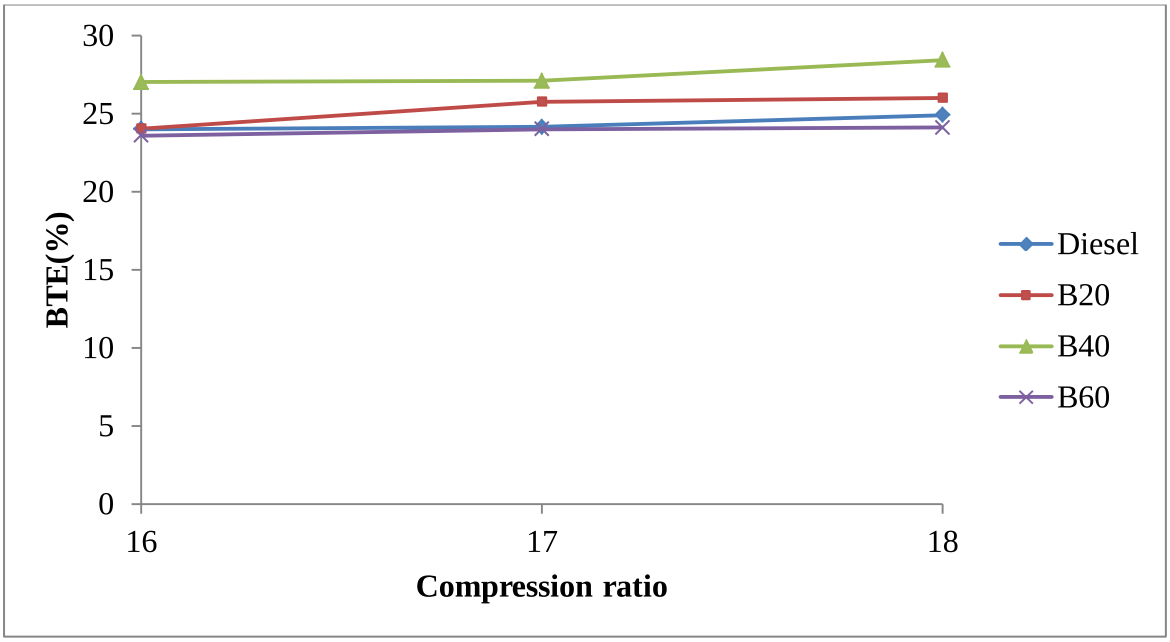

Fig 4.2.2.2

4.2.3 Mechanical efficiency

The variations of mechanical efficiency with compression ratio for various blends are shown in Fig.4.2.3.1 It has been observed that the mechanical efficiency of the blends is lesser in lower compression ratio and higher in higher compression ratio. The mechanical efficiency of the blend B20 and B40 increases with increase in compression ratio, when compared to that of diesel. The maximum mechanical efficiency obtained from blend B20 and B40 for compression ratio 18 is 54.61% and 53.56%. Mechanical efficiency increases with increasing compression ratio for all the blends.

Fig.4.2.3.2 shows that the variation of mechanical efficiency with load ratio for various blends. It has been observed that there is a steady increase in mechanical efficiency for all the blends as the load increases. Maximum mechanical efficiency has been obtained from blend B20 and B40 for full load and it is 54.61% and 53.56% respectively. Mechanical efficiency increases with increasing load for all the blends. The efficiency of the fuel blends is in general very closer to that of diesel. The increase in efficiency for all the blends may be due to improved quality of spray, high reaction activity in the fuel rich zone and decrease in heat loss due to lower flame temperature of the blends than that of diesel.

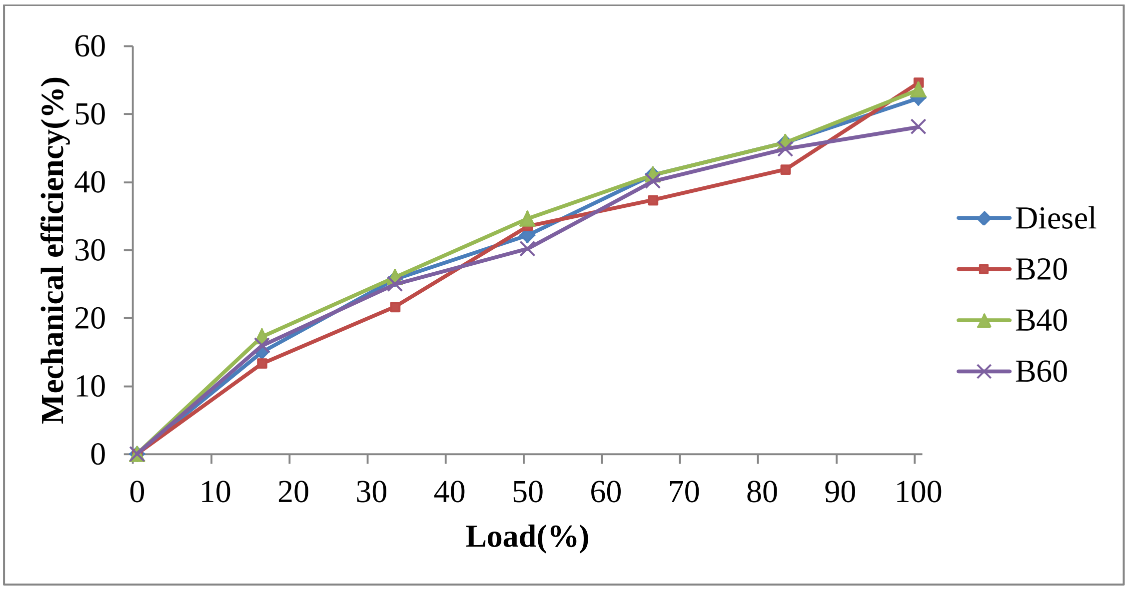

Fig 4.2.3.1

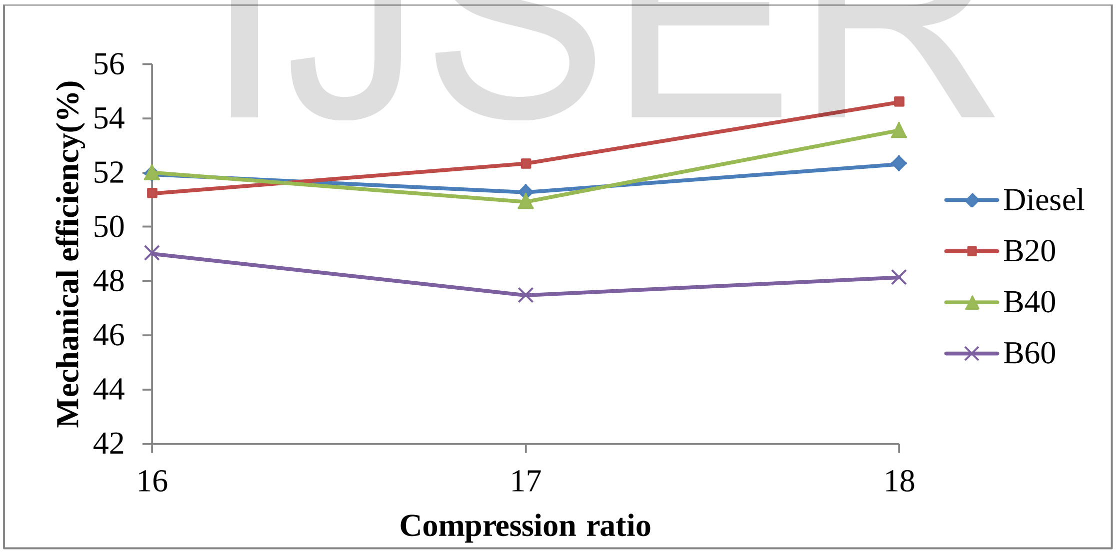

Fig 4.2.3.2

4.2.4 Exhaust gas temperature

The variation of exhaust gas temperature for different compression ratio and for different blends is shown in Fig.4.2.4.1. The result indicates that exhaust gas temperature decreases for different blends when compared to that of diesel. As the compression ratio increases, the exhaust gas temperature of various blends is lesser than that of diesel. The highest temperature obtained is 324.53ºC for diesel for a compression ratio 18, whereas the temperature is only 301.63ºC for the blend B40. The reason for the reduction in exhaust gas temperature at increased compression ratio is due to the lower calorific value of blended fuel as compared to the diesel and lower temperature, at the end of compression,

The variation of exhaust gas temperature with applied load for different bends is shown in Fig 4.2.4.2.The exhaust gas temperature decrease for different blends when compared to that of diesel. The highest temperature obtained is 324.53ºC for diesel for full load, whereas the temperature is only 324.11ºC and 310.63ºC for the blend B20 and B40.

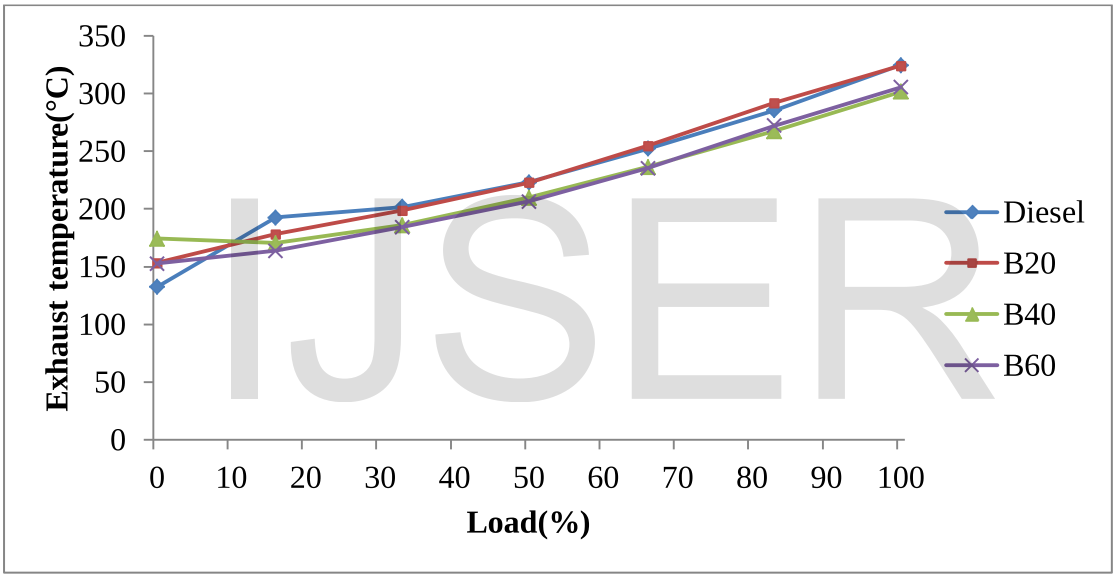

Fig 4.2.4.1

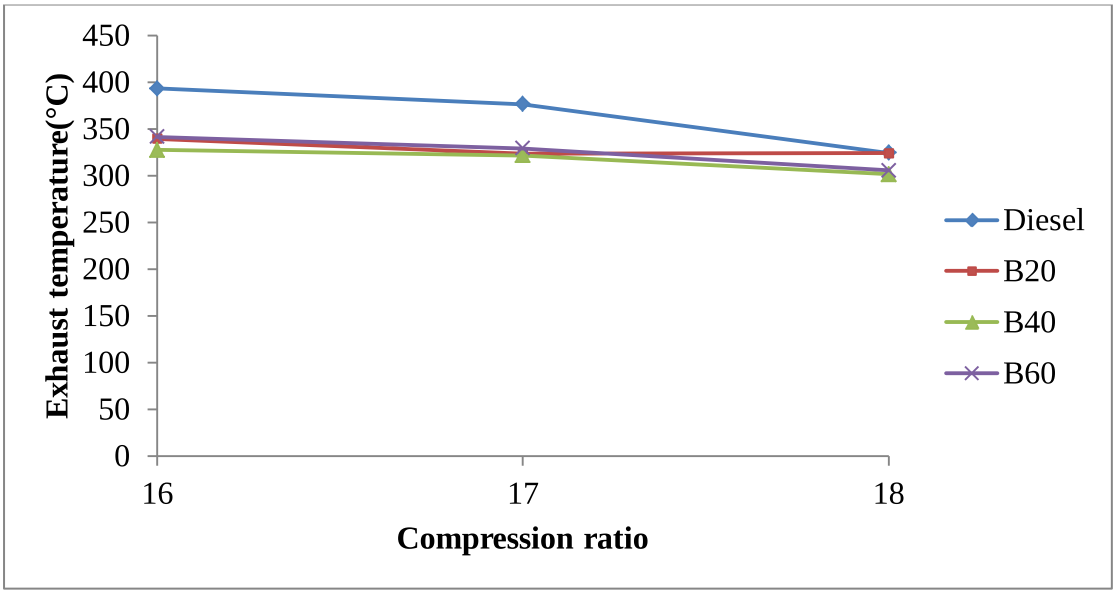

Fig 4.2.4.2

4.3 Combustion Analysis of Karanja oil methyl ester

4.3.1 Maximum combustion pressure.

The variation of combustion pressure with respect to crank angle for compression ratio 18 and for different blends is shown in Fig.4.3.1.1. It shows almost the same pressure crank angle characteristics for KOME blends and diesel.

The variation of peak pressure with respect to load for different compression ratio and for different blends is shown in Fig.4.3.1.2 The peak pressure depends on the amount of fuel taking part in the uncontrolled phase of combustion, which is governed by the delay period and spray envelop of the injected fuel. Larger the ignition delay more will be the fuel accumulation, which finally results in a higher peak pressure. It is seen that the peak pressure for diesel as well as KOME blends is almost the same at full load, the peak pressure value for diesel and blends B20, B40, and B60 being 61.2 bar, 59.6 bar, 60.15 bar and 60.55 bar for

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respectively. It is observed that the cylinder pressure increased with load for all the fuel operations. With increasing load, the amount of fuel supplied to the engine increase and due to combustion of relatively more amount of fuel, the peak pressure of the in-cylinder gas is more at higher load.

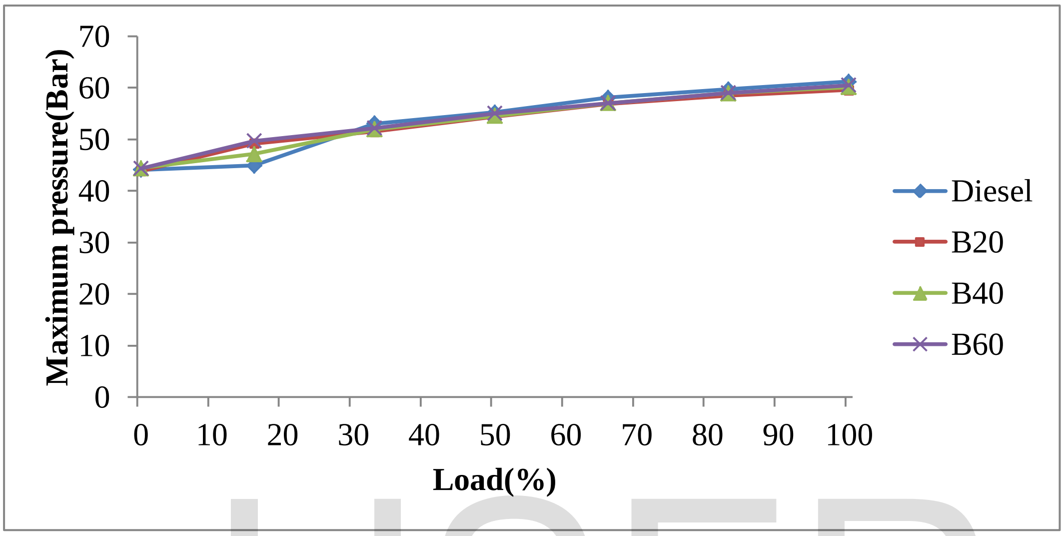

Fig 4.3.1.1

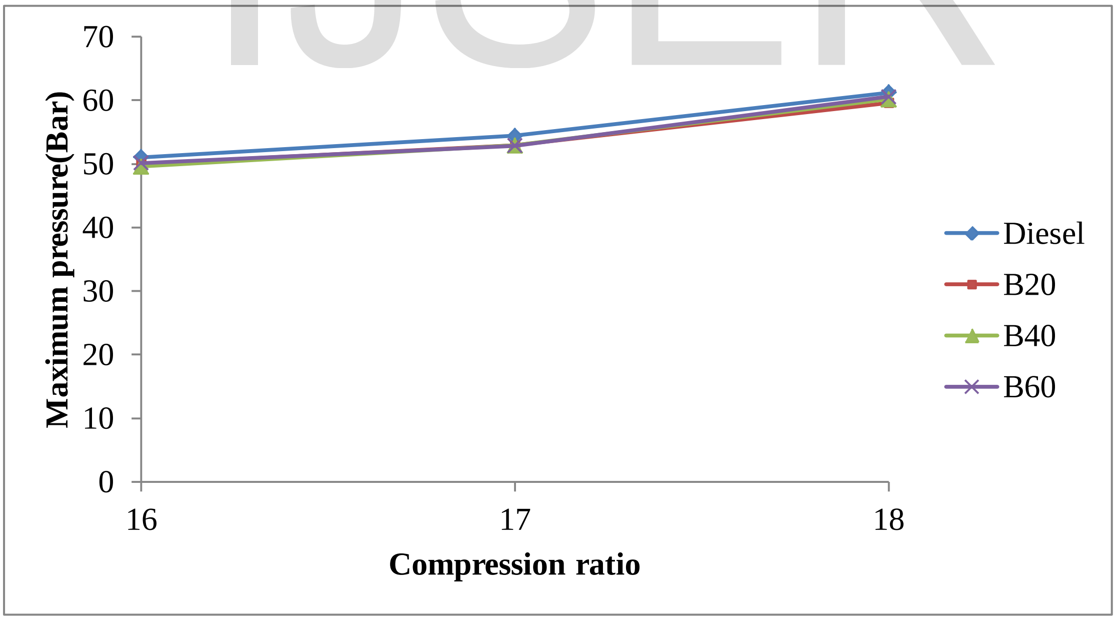

Fig 4.3.1.2

4.3.2 Combustion duration

It is difficult to define exactly the combustion duration of a diesel engine as the total combustion process consists of the rapid premixed combustion, mixing controlled combustion and the late combustion of fuel present in the fuel rich combustion products. The combustion duration in general increases with load. At compression ratio 18, B60 gives better combustion duration as compared to diesel. The variation of the total combustion duration with different loads for different blends is shown in Fig 4.3.2.1. At full load, the combustion duration for the fuel blends B20, B40,B60 and diesel is 45, 38,51 and 19 ºCA respectively. As the calorific value of the biodiesel blend is lower than diesel, a higher quantity of fuel is consumed to keep the engine speed stable at different loads. The decrease in combustion duration is due to the efficient combustion of the injected fuel.

Fig 4.3.2.2 shows the variation of combustion duration with compression ratio for different blends. With increase in compression ratio combustion duration increases. The oil blends causes longer duration for combustion at lower compression ratio and less duration for combustion at higher compression ratio.

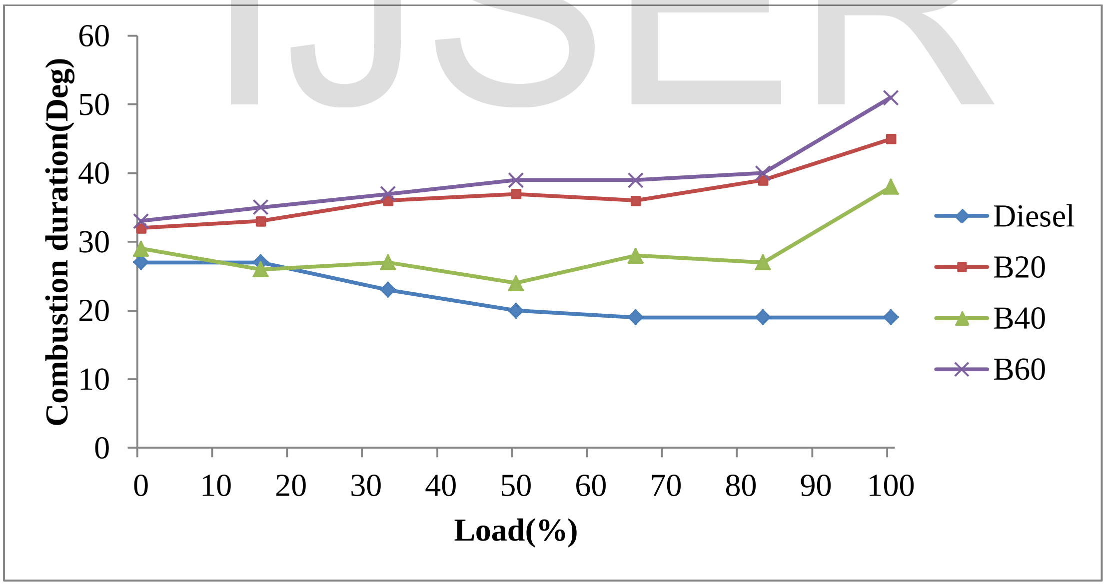

Fig 4.3.2.1

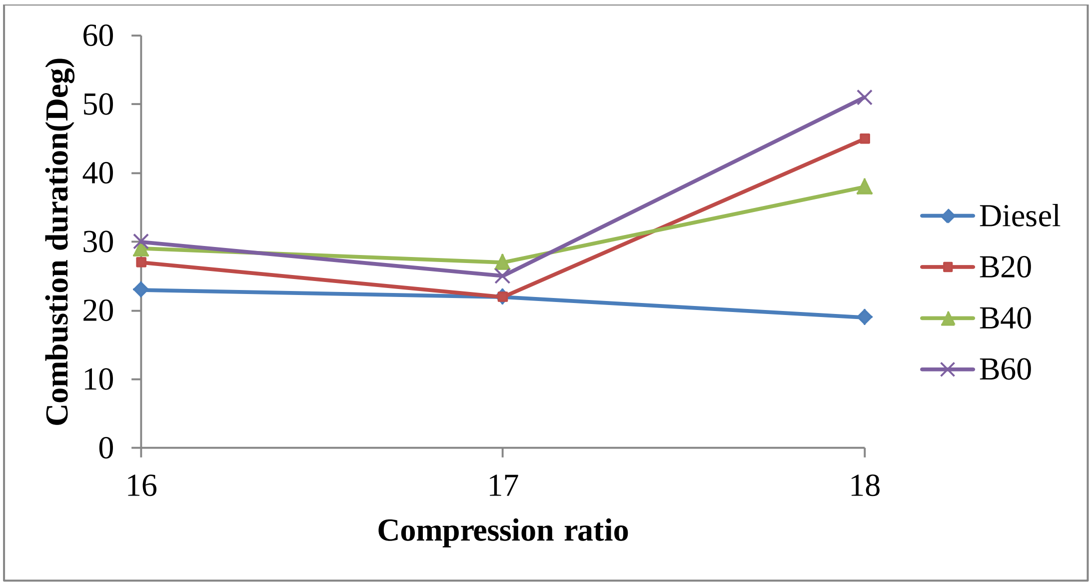

Fig 4.3.2.2

4.3.3. Net heat release rate

The net heat release rate at compression ratio 18 at 100% load with crank angle for different KOME blends is given in Fig 4.3.3.1. The maximum heat release rate of diesel, B20, B40, and B60 has been observed to be 53.2, 48.8, 47.7 and 45.07 J/ ºCA. The heat release rate is analyzed based on the changes in crank angle variation of the cylinder. The heat release rate of KOME blends decreases compared to that of diesel at full load. The heat release rate of diesel is higher than oil blend due to its reduced viscosity and better spray formation.

Fig 4.3.3.2.shows the variation of heat release rate with compression ratio for different blends. Low compression ratio Heat release rate increases with the low compression ratios and slightly decreases at higher compression ratio. This may be due to the air entrainment and lower air/fuel mixing rate and effect of viscosity blends. The heat release rate of diesel is higher than oil blend due to its reduced viscosity and better spray formation.

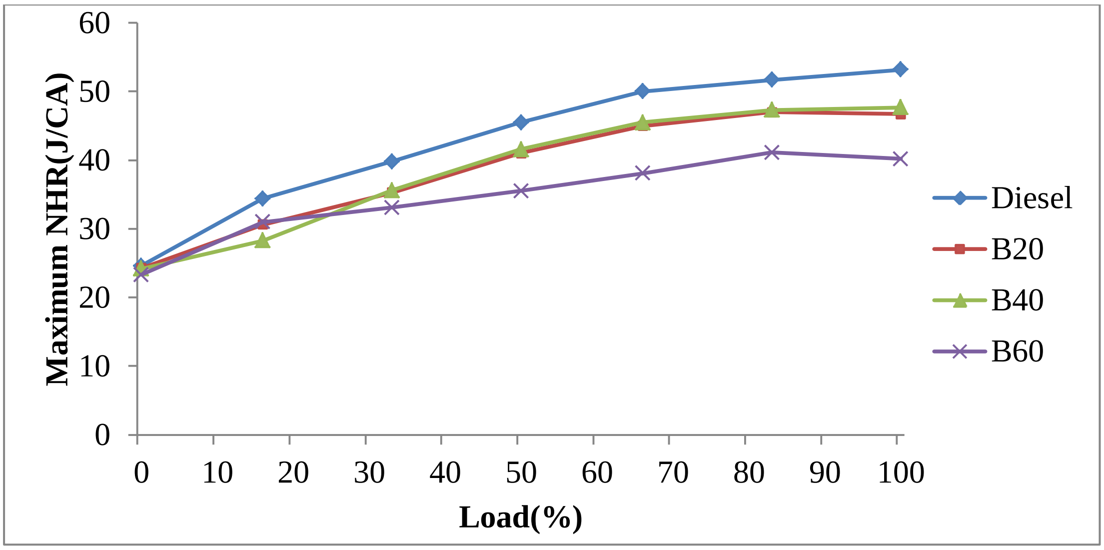

Fig 4.3.3.1

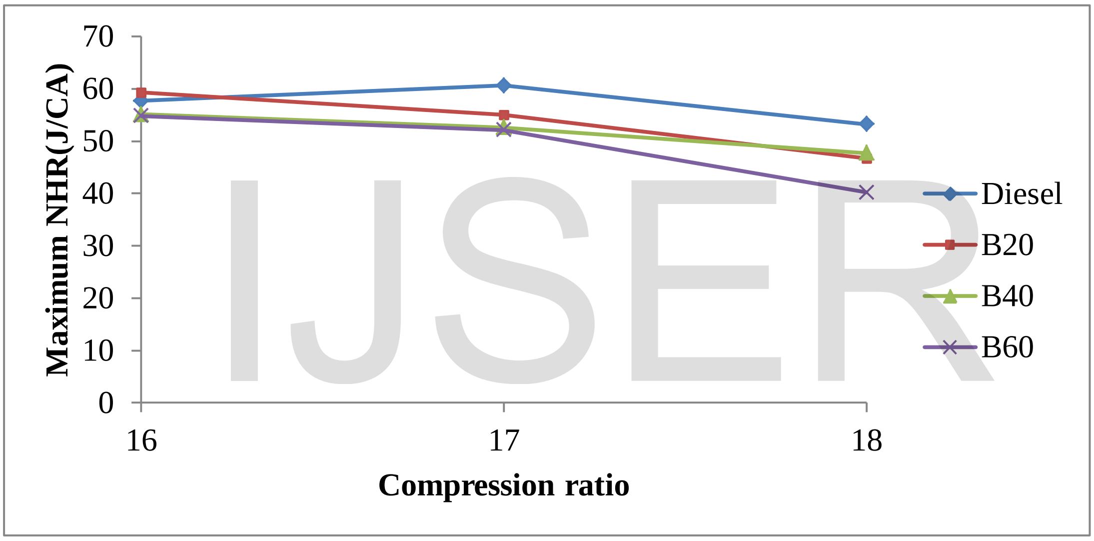

Fig 4.3.3.2

4.3.4 Mass fraction burnt

The variations of the mass fraction burnt with the crank angle for KOME blends and diesel at compression ratio 18 at full load is given in Fig 4.3.4.1. The mass fraction burnt of blends is slightly higher than that of diesel at full load. Due to the oxygen content of the blend, the combustion is sustained in the diffusive combustion phase. The mass fraction burnt for the fuel blend B40 is higher than that diesel for crank angle 350º-360º and it is closer for crank angle 365º-380º. The highest rate of burning shows that the efficient rate of combustion. The

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engine operates in rich mixture and it reaches stoichiometric region at higher compression ratio. More fuel is accumulated in the combustion phase and it causes rapid heat release.

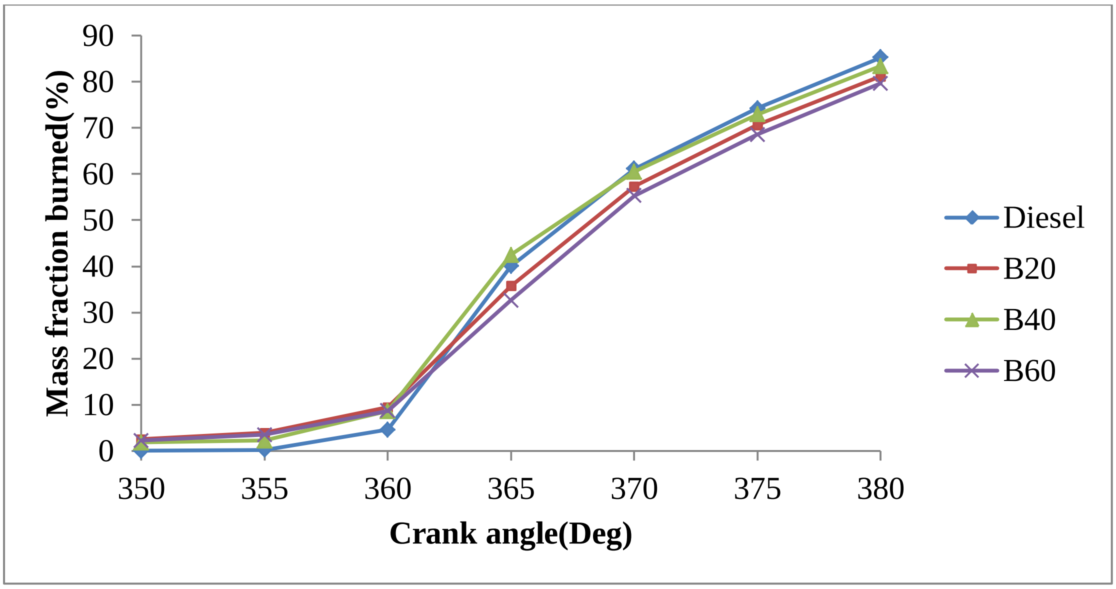

Fig 4.3.4.1

4.3.5 Ignition delay

The most vital parameter in combustion analysis is ignition delay. The variation of ignition delay with load is shown in Fig 4.3.5.1. It has observed that the ignition delay decreases with biodiesel in the diesel blend with increase in load and increases in compression ratio. At compression ratio 18 and 100% load condition, the ignition delay period of B20, B40 and B60 is 5.5, 6.2, 6.5 ºCA higher than diesel. It is due to the higher cetane number, maximum cylinder pressure and higher temperature.

Fig 4.3.5.2.shows the variation of ignition delay with compression ratio for different blends. Ignition delay decrease with increase in compression ratio.

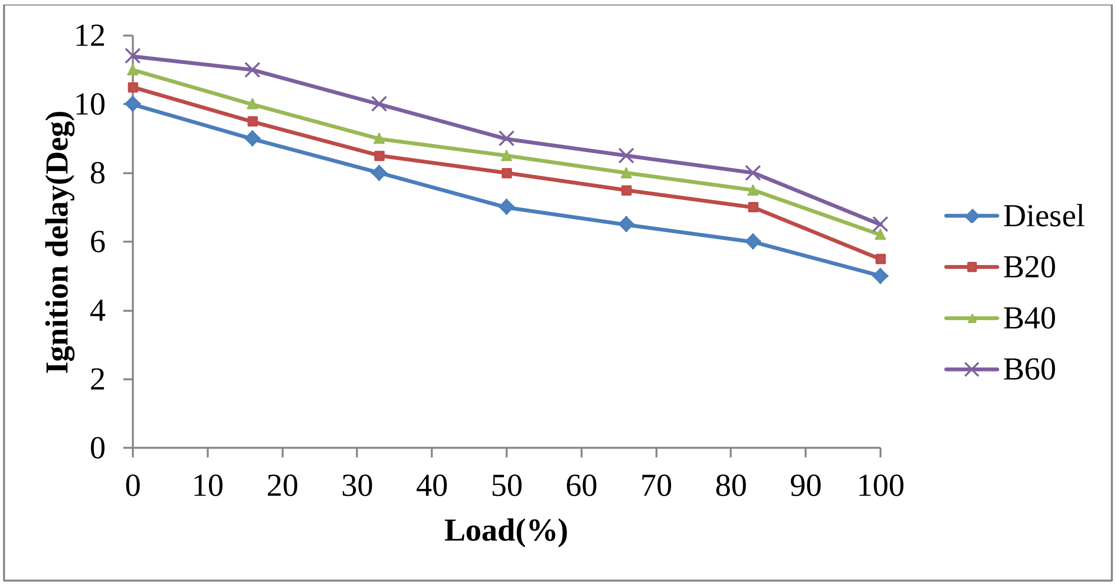

Fig 4.3.5.1

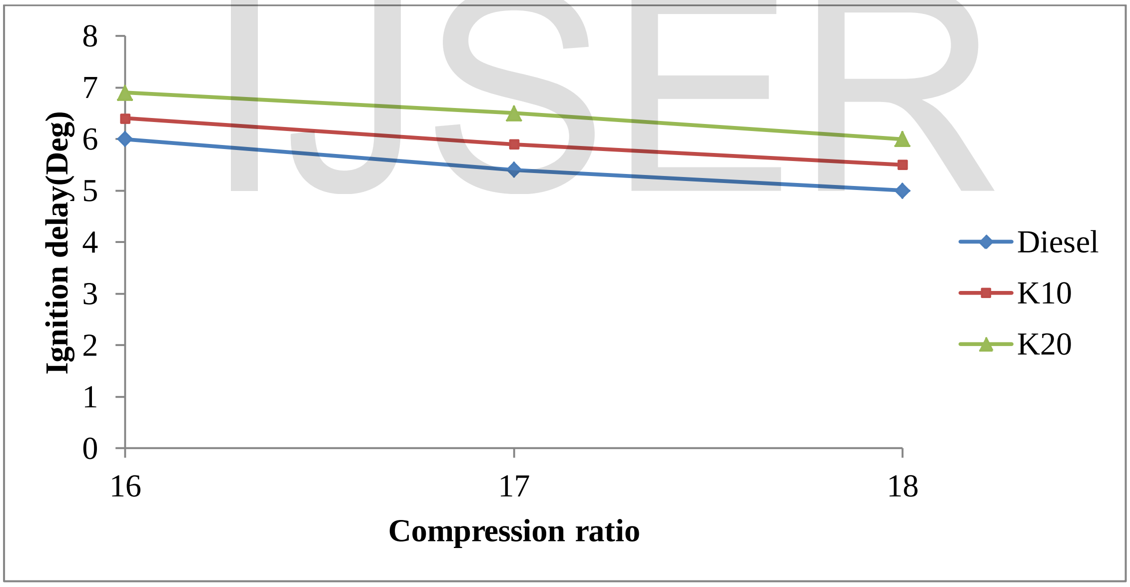

Fig 4.3.5.2

4.3.6 Maximum combustion temperature

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The variations of the maximum combustion temperature with loads for different blends and diesel at compression ratio18 are given in Fig 4.3.6.1. B20 gives better combustion temperature than diesel. Increasing load combustion temperature increases for all cases.

Fig 4.3.6.2 shows the variation of maximum combustion temperature with compression ratio for different blends. It has been observed that increasing compression ratio combustion temperature increases. B20 gives better combustion temperature than all other blends.

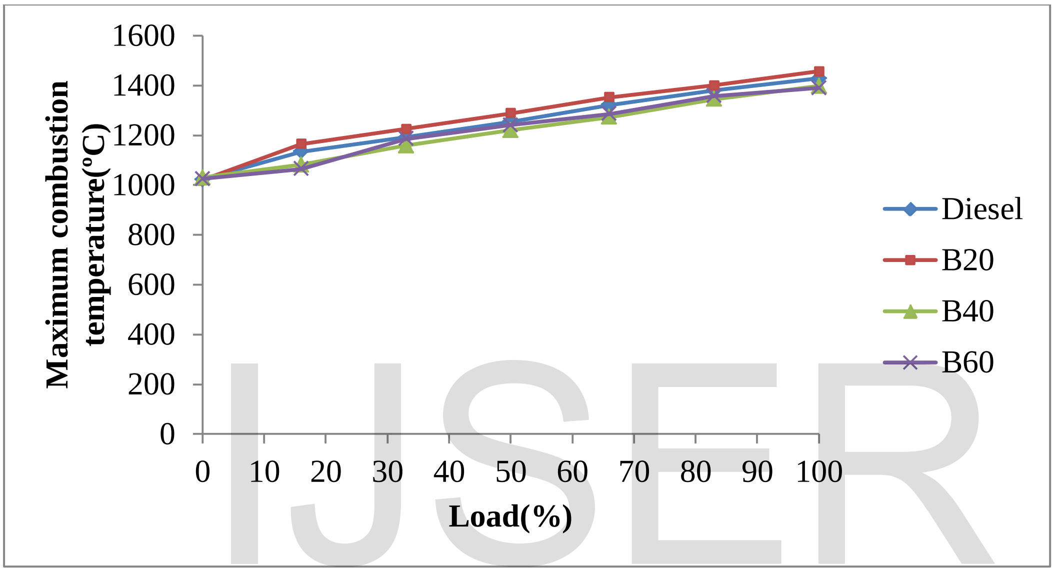

Fig 4.3.6.1

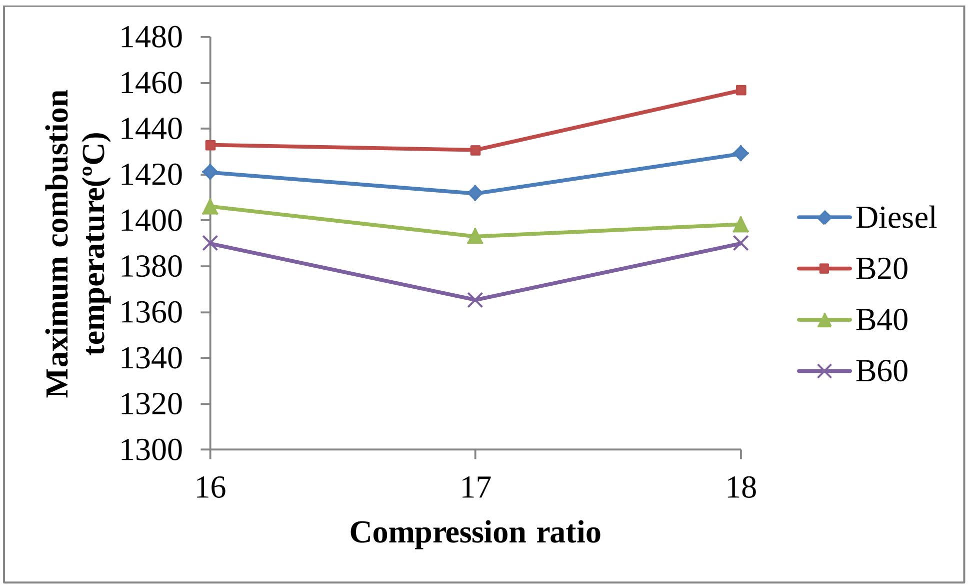

Fig 4.3.6.2

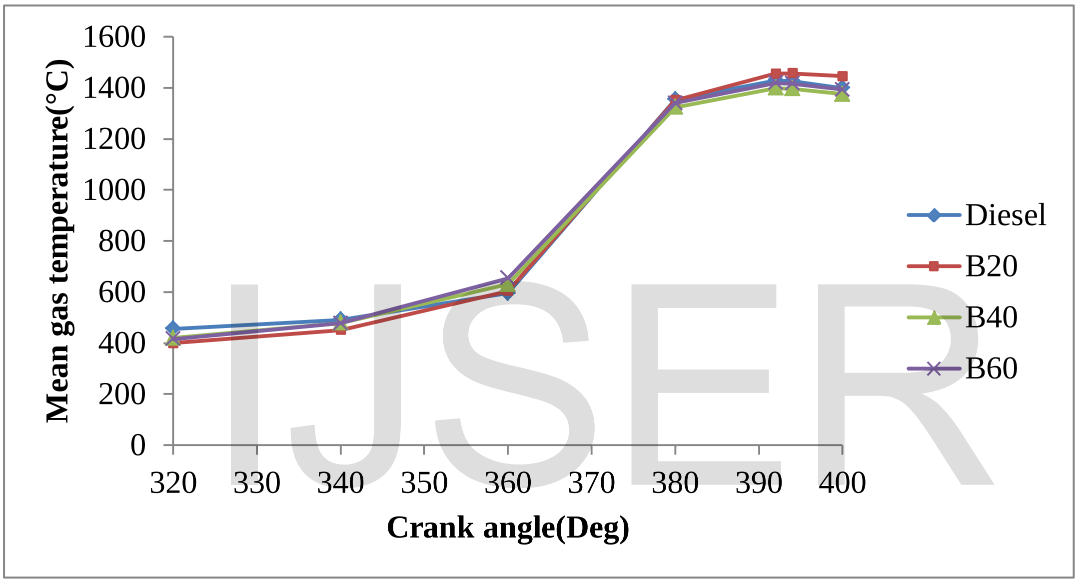

Fig 4.3.6.3

4.4 Emission analysis

4.4.1 **Carbon monoxide emission**

Fig 4.4.1.1 shows the variation of carbon monoxide emission of the blends and diesel for various loads. The CO emission of the blend B40 and B20 is slightly less than diesel and it is found to be higher for light and medium load. The CO emission of the blend B60 is more than diesel. The proportion of CO increase due to rising temperature in the combustion chamber, physical and chemical properties of the fuel, air-fuel ratio, lack of oxygen at high speed, and smaller amount of time available for complete combustion

Fig 4.4.1.2 shows the variation of carbon monoxide emission of the blends and diesel with various compression ratios. The CO emission of the blend B40 is close to diesel and it is found to be higher for compression ratio 16. The other blends B20, B60 have slightly lesser CO emission for compression ratio 16. It observed that by increasing compression ratio CO emission decrease.

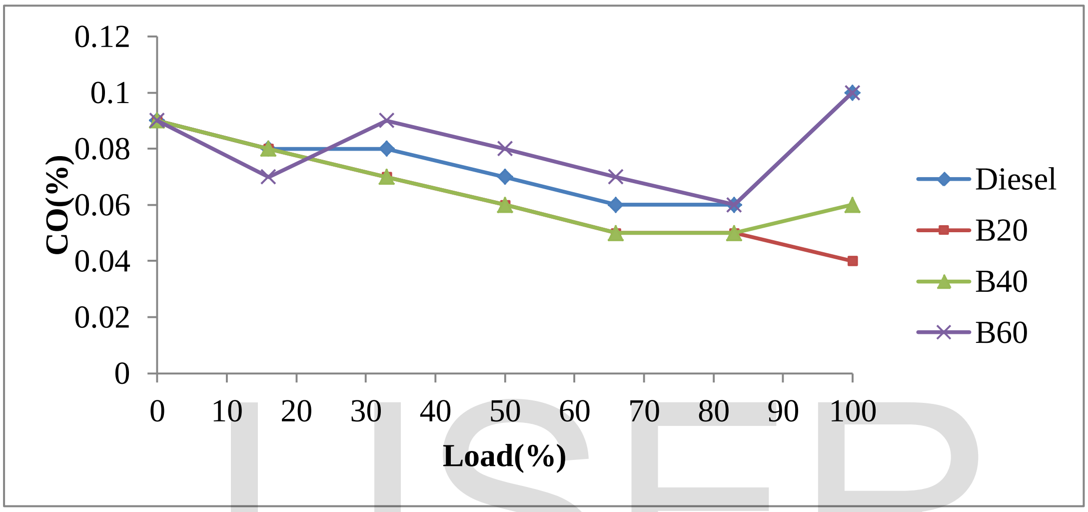

Fig 4.4.1.1

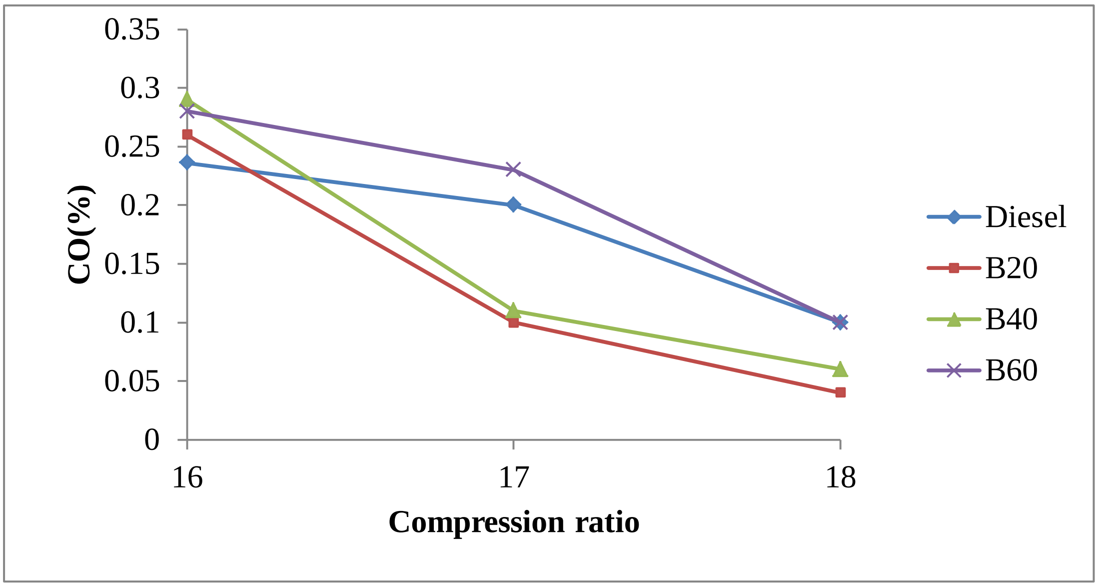

Fig 4.4.1.2

4.4.2 Carbon dioxide emission

The variation of carbon dioxide emission with different loads is shown in Fig 4.4.2.1. More amount of CO_2 is an indication of complete combustion of fuel in the combustion chamber. It also related to the exhaust gas temperature. CO_2 emission of the blend B40 is slightly higher than diesel. By increasing load CO_2 emission increases.

The variation of carbon dioxide emission with different compression ratio is shown in Fig 4.4.2.2. The blend emits higher percentage of CO_2 than diesel at lower compression ratios and vice versa. CO_2 emission of the blend B40 for compression ratio 16 is lesser due to incomplete combustion and inadequate supply of oxygen. The accumulation of CO_2 in the atmosphere leads to many environmental problems like ozone depletion and global warming. The CO_2 emission from the combustion of bio fuels can be absorbed by the plants and the carbon dioxide level and is kept constant in the atmosphere.

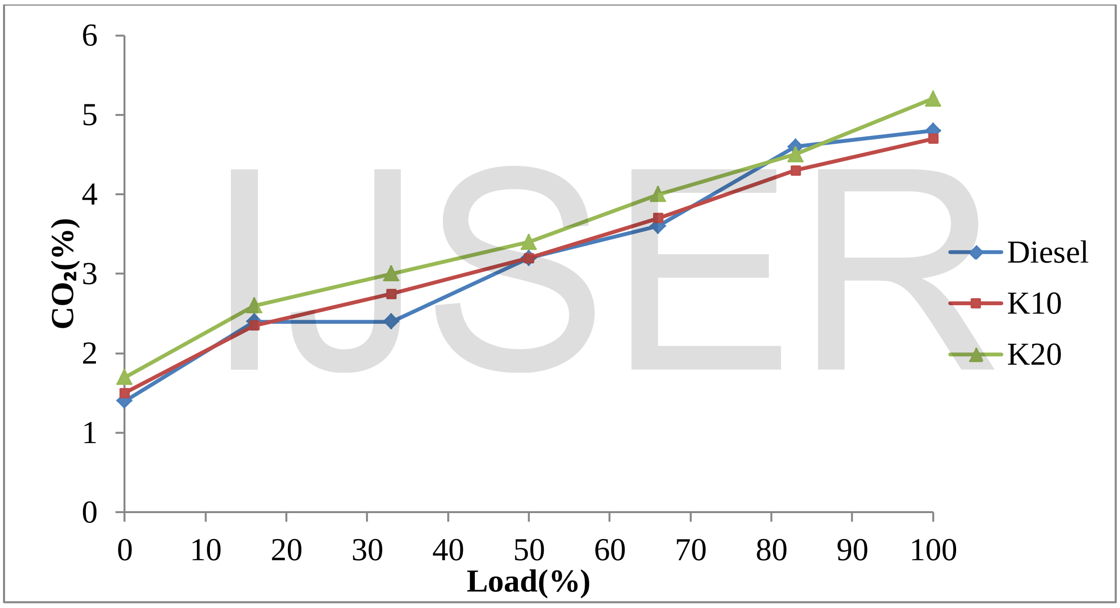

Fig 4.4.2.1

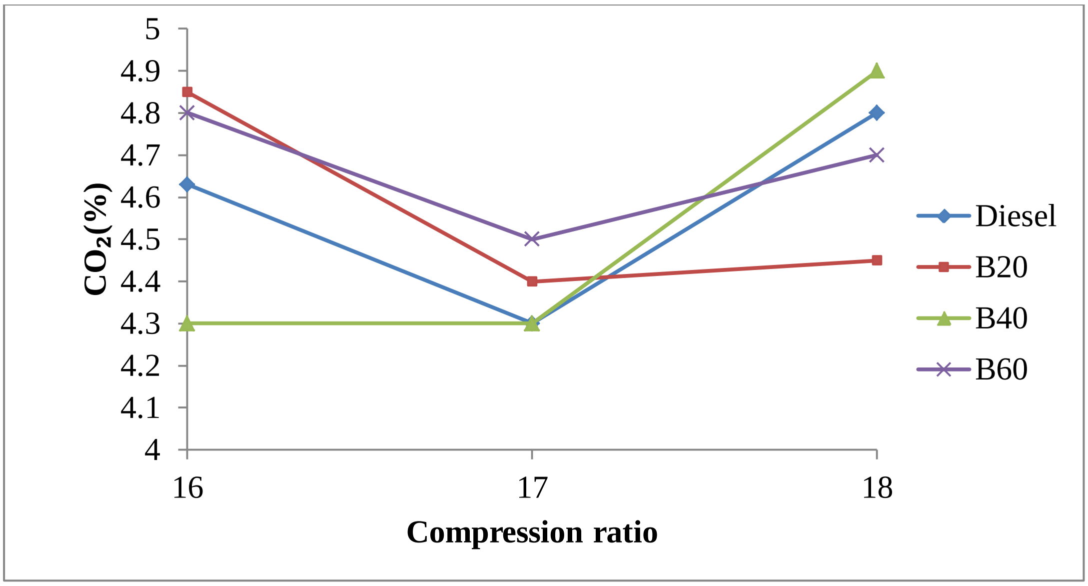

Fig 4.4.2.2

4.4.3 Hydrocarbon emission

The variation of hydrocarbon emissions with load for different blends is plotted in Fig 4.4.3.1. The hydrocarbon emissions of various blends are higher at higher loads. The effect of fuel viscosity and the fuel spray quality has been expected to produce some hydrocarbon increases with vegetable oil fuels. It shows that the increase in load increase the hydrocarbon emission for the blend B40. The other blends B20 and B60 lesser hydrocarbon emissions at 50% load than the diesel. Due to longer ignition delay, the accumulation of fuel in the combustion chamber may cause higher hydrocarbon emission.

The variation of hydrocarbon emission with different compression ratios for different blends is given in Fig 4.4.3.2. It shows that the hydrocarbon emission of various blends is higher at higher compression ratios. In this research, it shows that the increase in compression ratio increases the HC emission for blend B60. The other blends B20 and B40 produce lesser hydrocarbon emissions at higher compression ratio than diesel. Due to longer ignition delay, the accumulation of fuel in the combustion chamber may cause higher hydrocarbon emission.

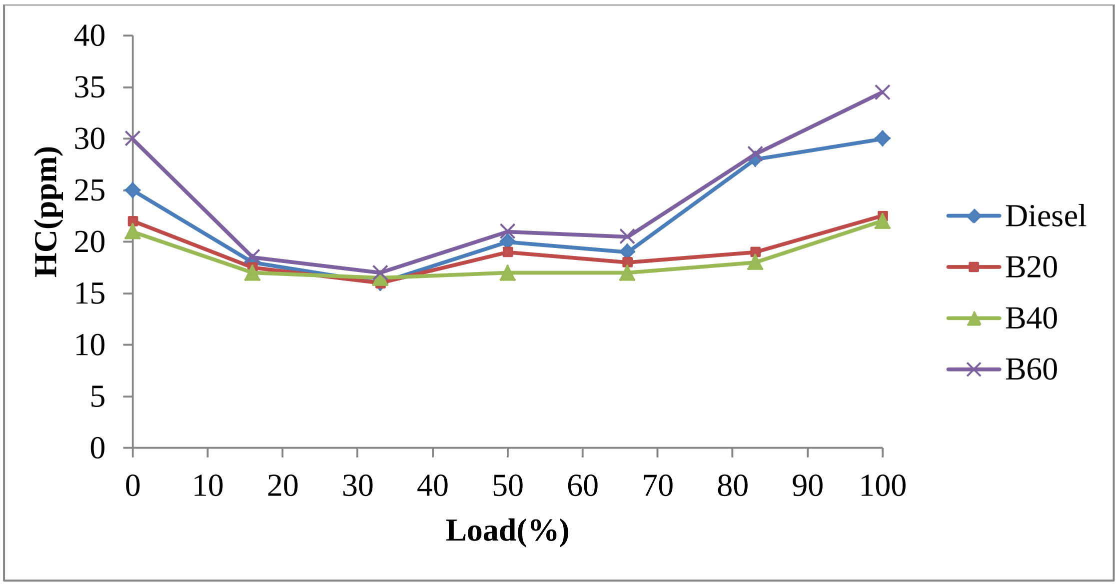

Fig 4.4.3.1

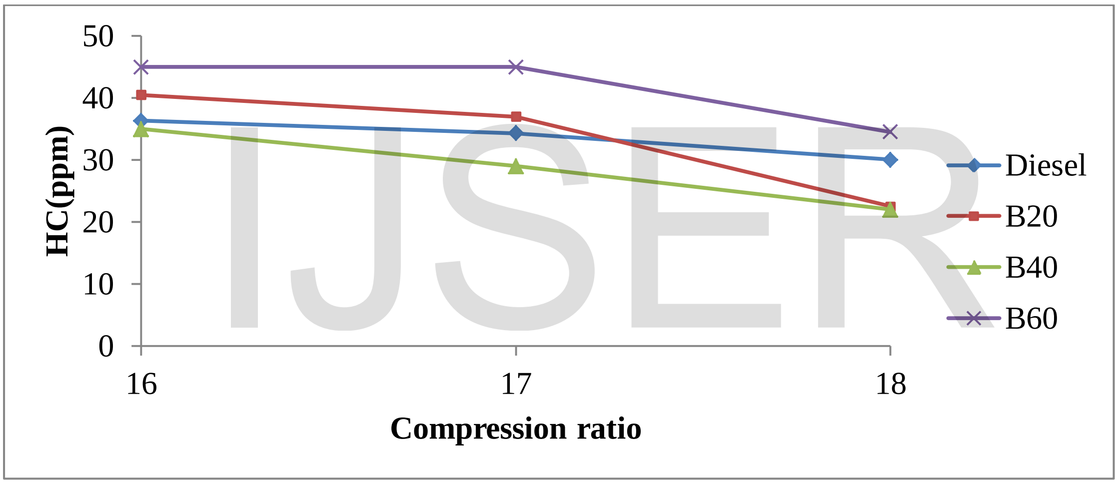

Fig 4.4.3.2

4.4.4 Nitrogen oxides emission

Fig 4.4.4.1 shows that the variations of Nitrogen oxides (NO_x) emission with load for different blends. The NO_x emission for biodiesel and its blends is higher than that of diesel except B60. For 50% load, NOx emission from blend B60 is lesser than that of diesel. But for full load the NO_x emission from blend B20 and B40 is higher than that of diesel. The other blends closely follow diesel. The reason for higher NOx emission for blends is due to the higher peak temperature. The NOx emission for diesel, B20,B40 and B60 for 50% load is 329 ppm, 384 ppm, 330 ppm and 250 ppm respectively.

The variation of nitrogen oxides (NO_x) emission with respect to different compression ratio for different blends is shown in Fig 4.4.4.2. The NO_x emission for diesel and other blends increase with increase of compression ratio. From the figure, it is obvious that for compression ratio 18, NO_x emission from the blendB20 and B40 is higher than that of diesel. The other blends closely follow diesel. The reason for higher NOx emission for blends is due to the higher peak temperature. The NO_x emission for diesel, B20, B40 and B60 for compression ratio 18 is 550ppm, 634.5ppm, 552ppm and 490ppm respectively.

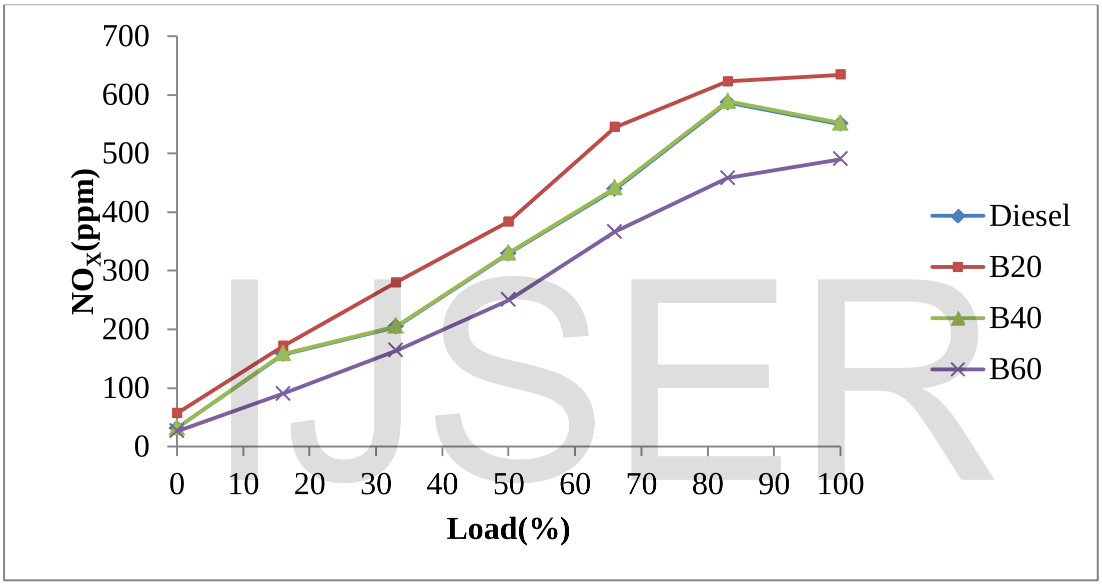

Fig 4.4.4.1

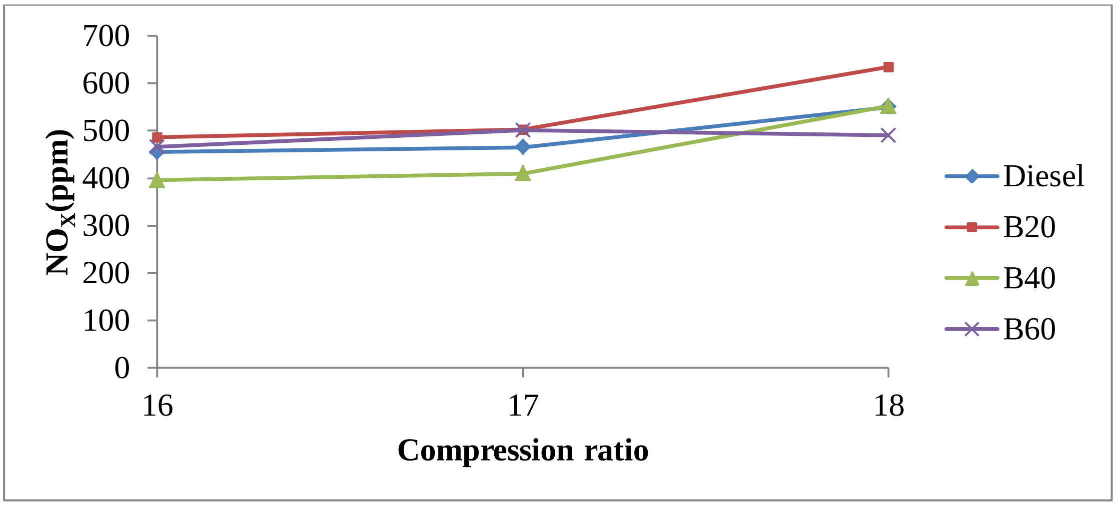

Fig 4.4.4.2

4.4.5Smoke opacity

Fig 4.4.5.1.shows that the variations of smoke opacity with load for different blends. Smoke opacity increase with increase in load. B40 and B60 give higher smoke opacity than diesel.

The variation of smoke opacity with respect to different compression ratio for different blends is shown in Fig 4.4.5.2. Smoke opacity decrease with increase in compression ratio. B40 and B60 give higher smoke opacity than that of B20 and diesel. The smoke opacity for diesel, B20, B40 and B60 for compression ratio 18 is 88.75%, 71.5%, 90% and 94% respectively.

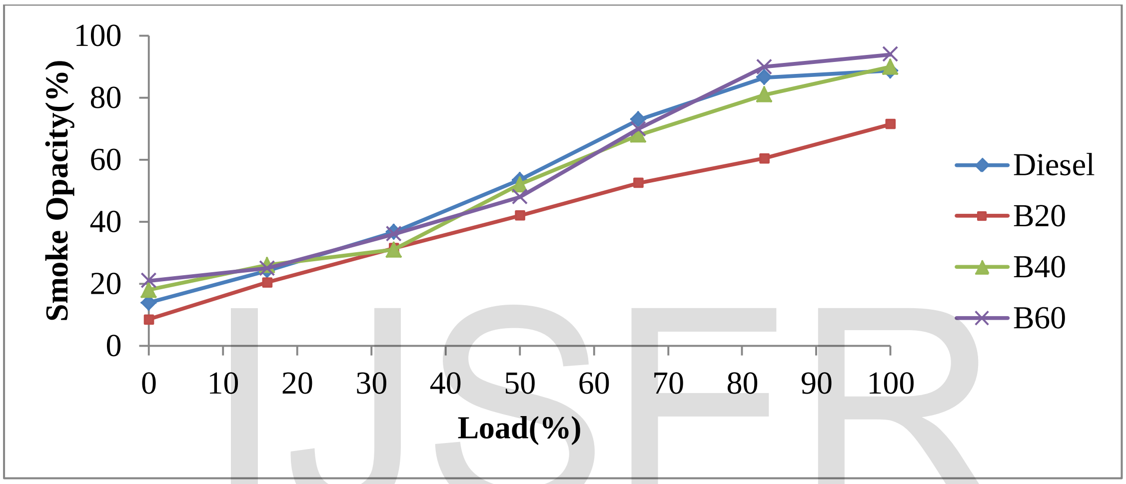

Fig 4.4.5.1

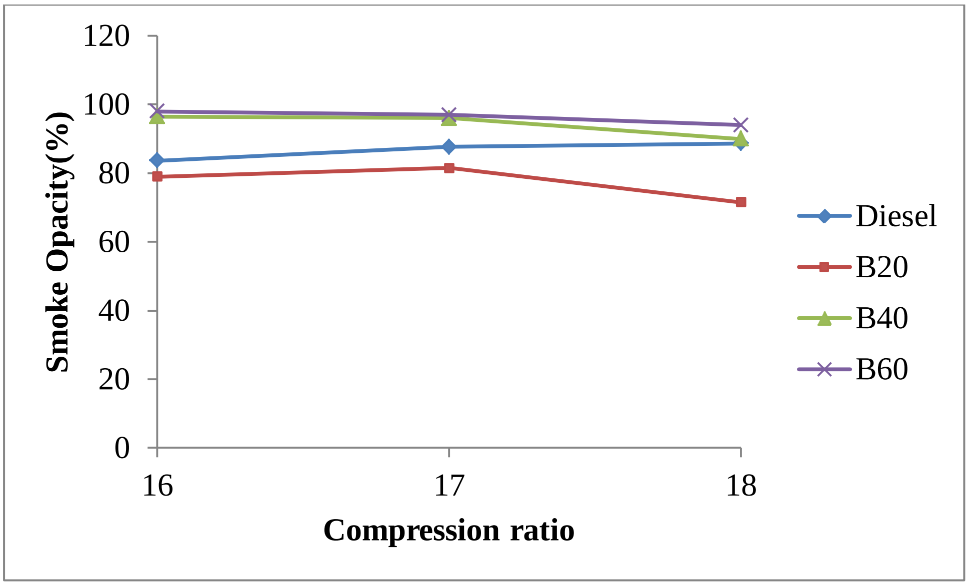

Fig 4.4.5.2

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4.5 PERFORMANCE ANALYSIS OF NEAT KARANJA

4.5.1 Brake specific fuel consumption

The brake specific fuel consumption decreases with increase in load and K10 gives less BSFC compared to K20 and diesel.. it can be seen from Fig 4.5.1.1 that with increase in blend %age of karanja oil BSFC increasing this is due to the decrease in calorific value and higher density of karanja oil for the higher blends. The BSFC varies with diesel, K10 and K20 at full load is found to be 0.34 kg/kWh, 0.33kg/kWh, 0.35kg/kWh respectively.

Form Fig 4.5.1.2 it can be observed that the brake specific energy consumption decreases with the increase in compression ratio. The BSFC for blend K20 is found to be higher compared to that of diesel. K10 shows less BSFC as compared to K10 and diesel at all compression ratio.

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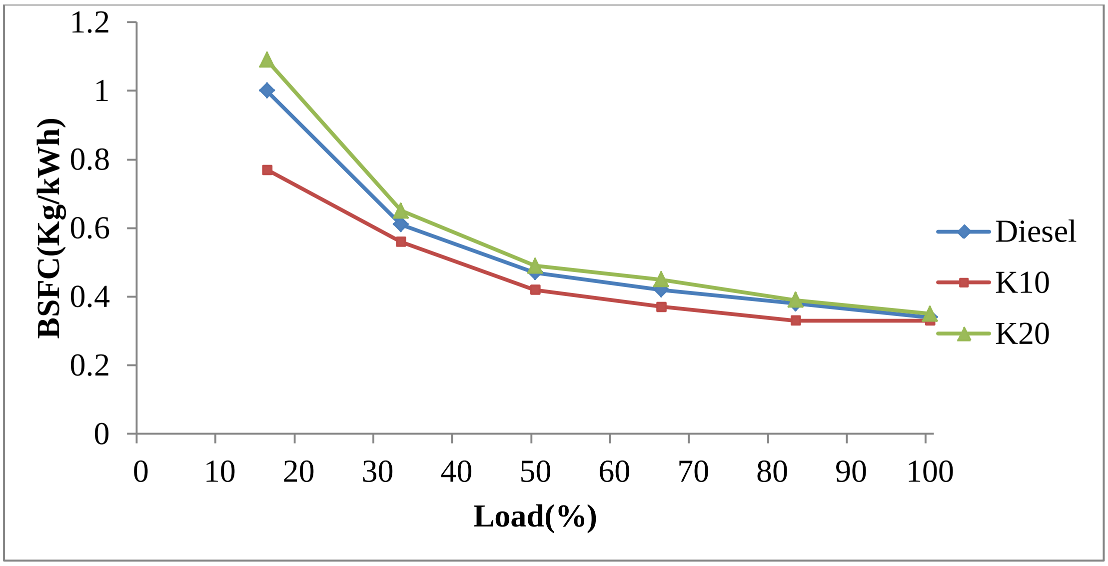

Fig 4.5.1.1

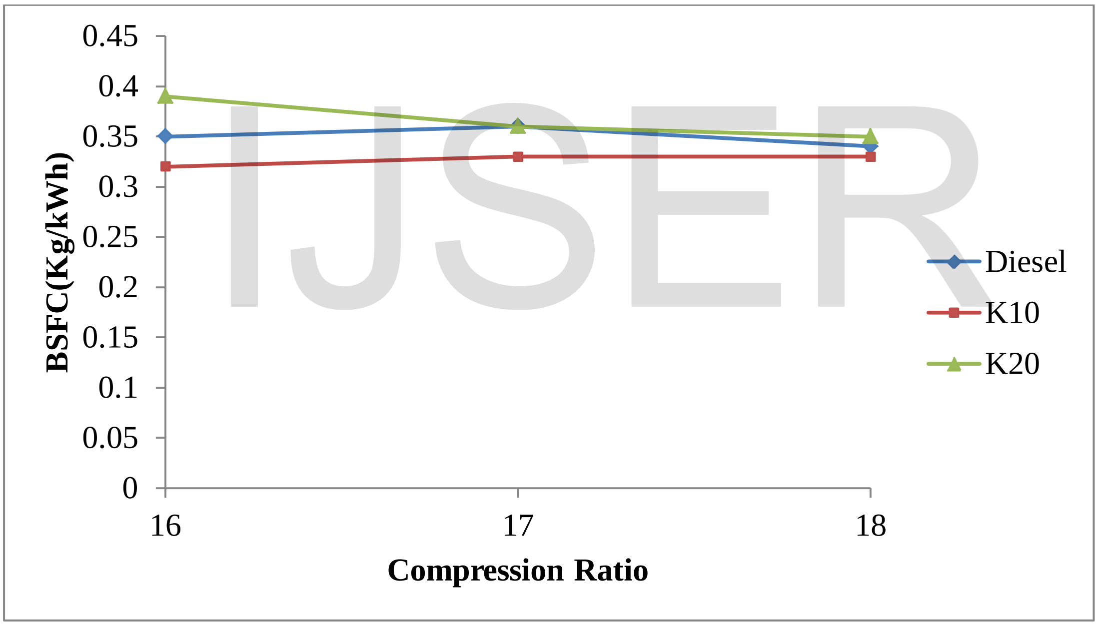

Fig 4.5.1.2

4.5.2 Brake thermal efficiency

The variation of brake thermal efficiency (BTE) for different loads and for different fuels is given in Fig 4.5.2.1. It is seen that there is a steady increase in efficiency with increases in load in all the fuel operations. It is happened due to reduction in heat loss and increase in power developed with increase in load. The engine BTE at full load for diesel, K10, and K20 fuels is 24.9%, 26.63%, and 24.1% respectively. It is also observed that the BTE of the blend K20 is slightly lower than that of the diesel and K10 is higher than diesel.

This may be due to higher viscosity of the blend k20 resulting in poorly formed fuel spray and air entrainment affecting the combustion in the engine and further due to lower volatility of vegetable oil.

The variation of brake thermal efficiency (BTE) for different compression ratio and for different blends is given in Fig 4.5.2.2 . It is observed that the BTE of the blend K10 is higher than that of the diesel at all compression ratio. BTE also gets increased for all the fuel types tested. BTE is directly proportional to the compression ratio.

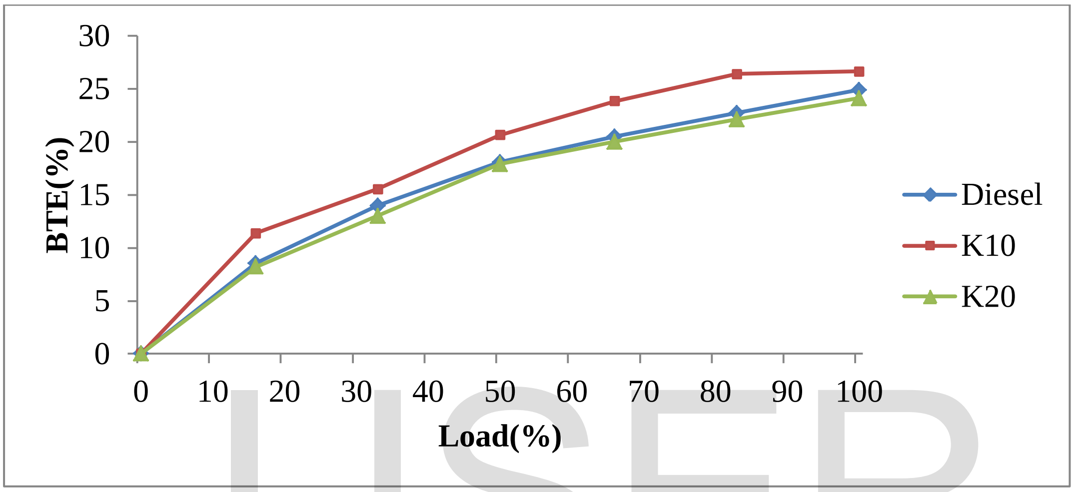

Fig 4.5.2.1

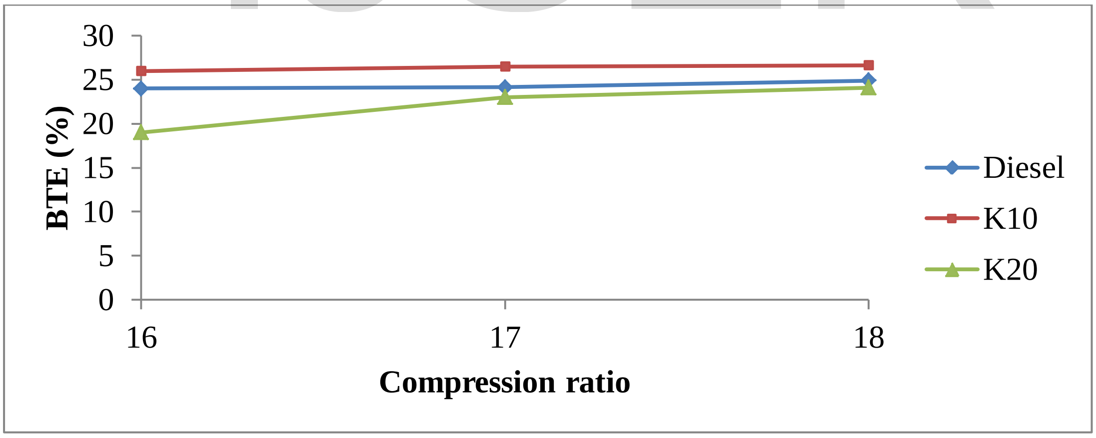

Fig 4.5.2.2

4.5.3 Mechanical efficiency

Fig 4.5.3.1 shows that the variation of mechanical efficiency with load for various blends. It has been observed that there is a steady increase in mechanical efficiency for diesel and blends as the load increases. Maximum mechanical efficiency has been obtained from blend

K10 at 50% load i.e. 35.41%. The efficiency of the fuel blends is in general very closer to that of diesel. The increase in efficiency for all the blends may be due to improved quality of spray, high reaction activity in the fuel rich zone and decrease in heat loss due to lower flame temperature of the blends than that of diesel. At full load diesel gives maximum mechanical efficiency as compared to K10 and K20. The mechanical efficiency at full load for diesel, K10, and K20 fuels are 52.32%, 35.41%,and 32.56% respectively.

The variations of mechanical efficiency with compression ratio for various blends are shown in Fig 4.5.3.2. It has been observed that the mechanical efficiency increases with compression ratio and higher in higher compression ratio. Mechanical efficiency of diesel is higher than K10 and K20 at all compression ratios. Mechanical efficiency increases with increasing compression ratio for all the blends.

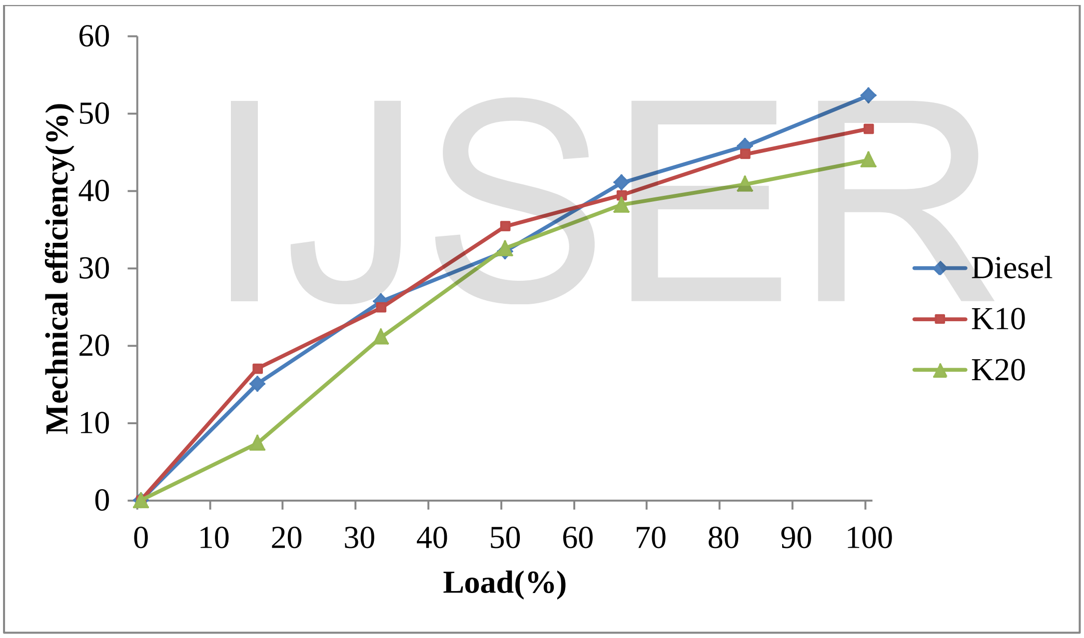

Fig 4.5.3.1

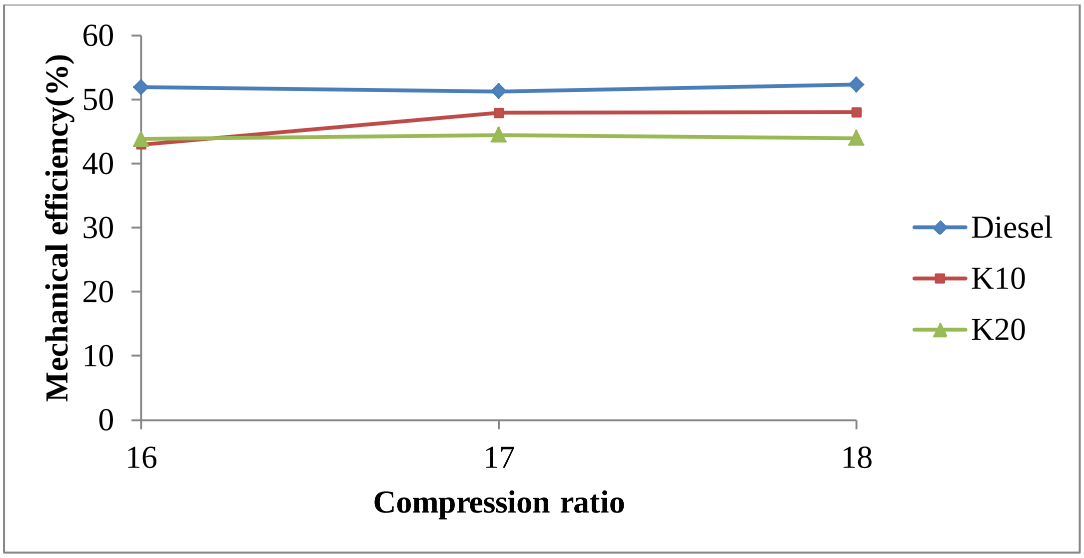

Fig 4.5.3.2

4.5.4 Exhaust gas temperature

The variation of exhaust gas temperature with applied load for different bends is shown in Fig 4.5.4.1. The exhaust gas temperature increases with increase in loadThe exhaust gas temperature decrease for different blends when compared to that of diesel. The highest temperature obtained is 324.53ºC for diesel for full load, whereas the temperature is only 317.93ºC and 310.39ºC for the blend K10 and K20.It may be due to energy content in diesel is higher as compared to K10 and K20.

The variation of exhaust gas temperature for different compression ratio and for different blends is shown in Fig 4.5.4.2.The exhaust gas temperature decreases with increase in compression ratio. The result indicates that exhaust gas temperature decreases for different blends when compared to that of diesel. As the compression ratio increases, the exhaust gas temperature of various blends is lesser than that of diesel. The reason for the reduction in exhaust gas temperature at increased compression ratio is due to lower temperature, at the end of compression,

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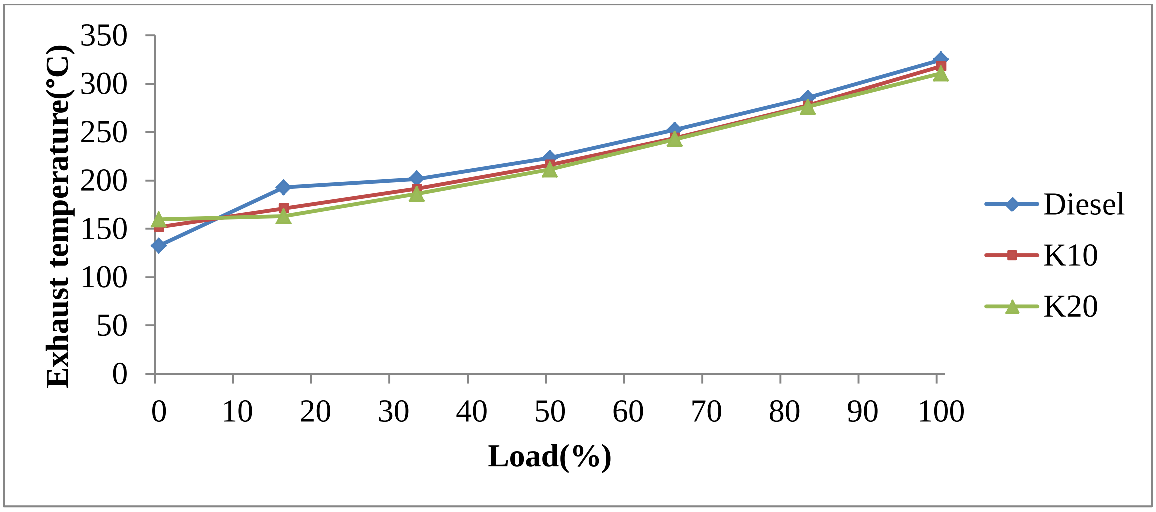

Fig 4.5.4.1

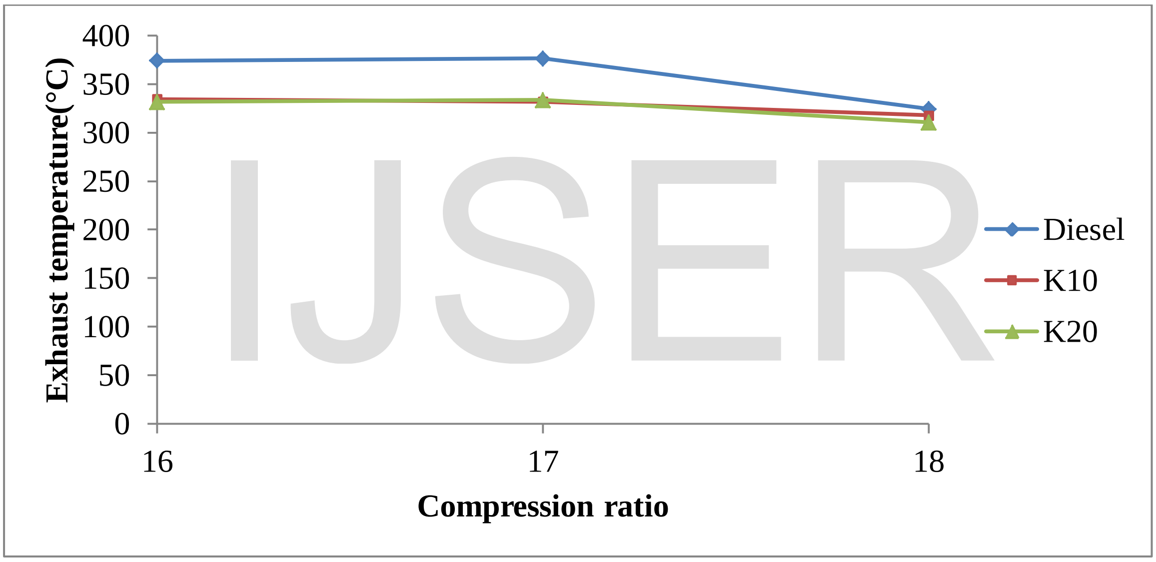

Fig 4.5.4.2

4.6 Combustion analysis of neat Karanja

4.6.1 Combustion pressure

The variation of combustion pressure with load for different blends is shown in Fig 4.6.1.1. It shows that increasing load combustion pressure increases. It shows that diesel gives maximum pressure as compared toK10 and K20. It is seen that the maximum pressure for diesel as well as karanja oil blends is almost the same at full load, the maximum pressure value for diesel and blends K10 and K20 being 61.2 bar, 58.93 bar, and 59.19 bar respectively. The peak pressure depends on the amount of fuel taking part in the uncontrolled phase of combustion, which is governed by the delay period and spray envelop of the injected fuel.

The variation of combustion pressure for different compression ratio and for different blends is shown in Fig 4.6.1.2.It shows that increasing compression ratio, combustion pressure increases. . It shows that diesel gives maximum pressure as compared toK10 and K20.

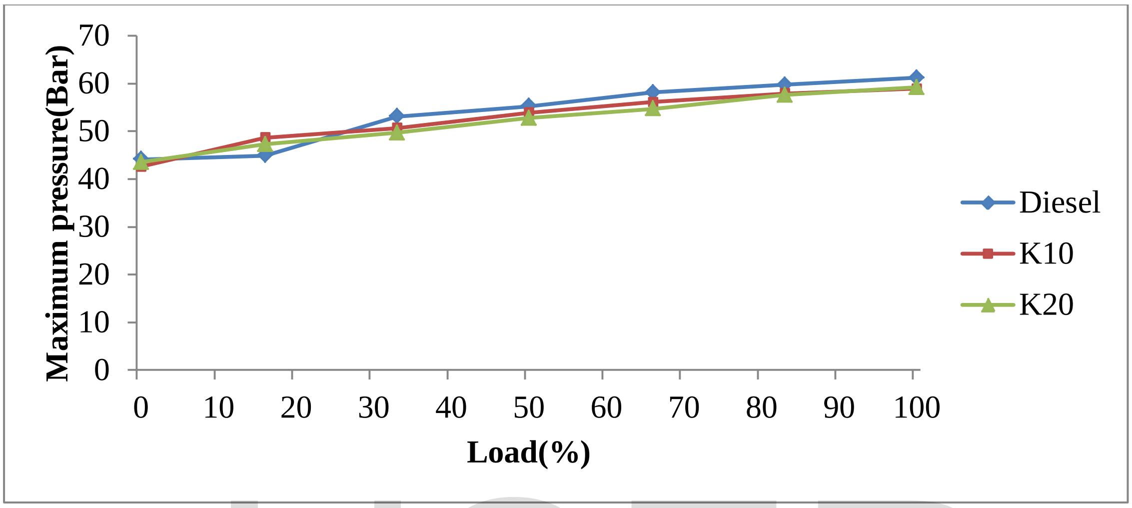

Fig 4.6.1.1

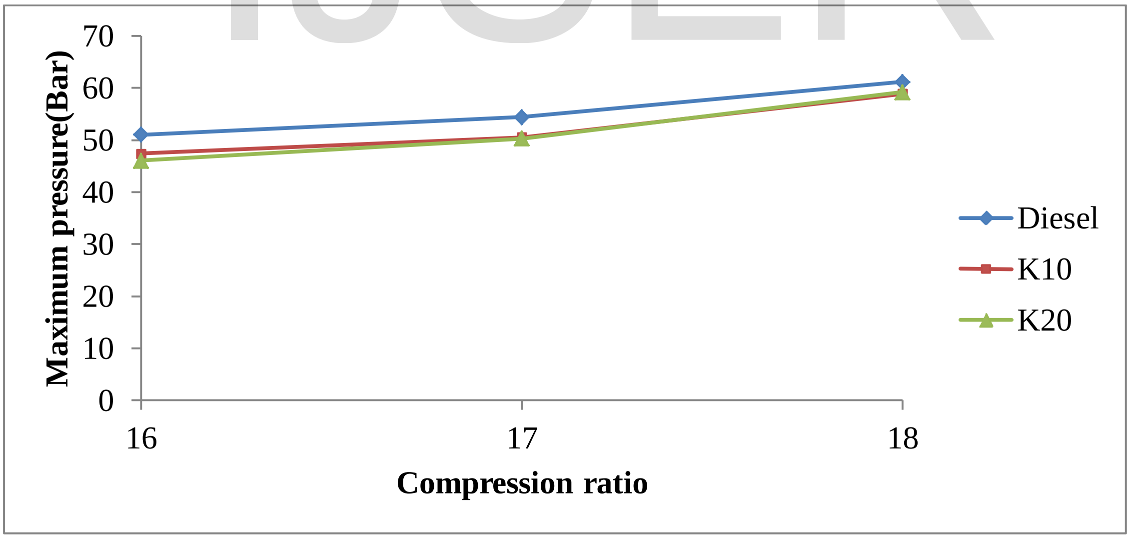

Fig 4.6.1.2

4.6.2 . Combustion duration

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It is difficult to define exactly the combustion duration of a diesel engine as the total combustion process consists of the rapid premixed combustion, mixing controlled combustion and the late combustion of fuel present in the fuel rich combustion products. The combustion duration in general increases with load. The variation of the total combustion duration with different loads for different blends is shown in Fig 4.6.2.1. At full load, the combustion duration for the fuel blends K10, K20 and diesel are 47, 77 and 19 °CA respectively. As the calorific value of the karanja oil blend is lower than diesel, a higher quantity of fuel is consumed to keep the engine speed stable at different loads. The decrease in combustion duration is due to the efficient combustion of the injected fuel. K20 gives higher combustion duration than other.

Fig 4.6.2.2 shows the variation of combustion duration with compression ratio for different blends. Increase in compression ratio combustion duration increases. The oil blends causes longer duration for combustion at lower compression ratio and less duration for combustion at higher compression ratio. K20 gives higher combustion duration than other.

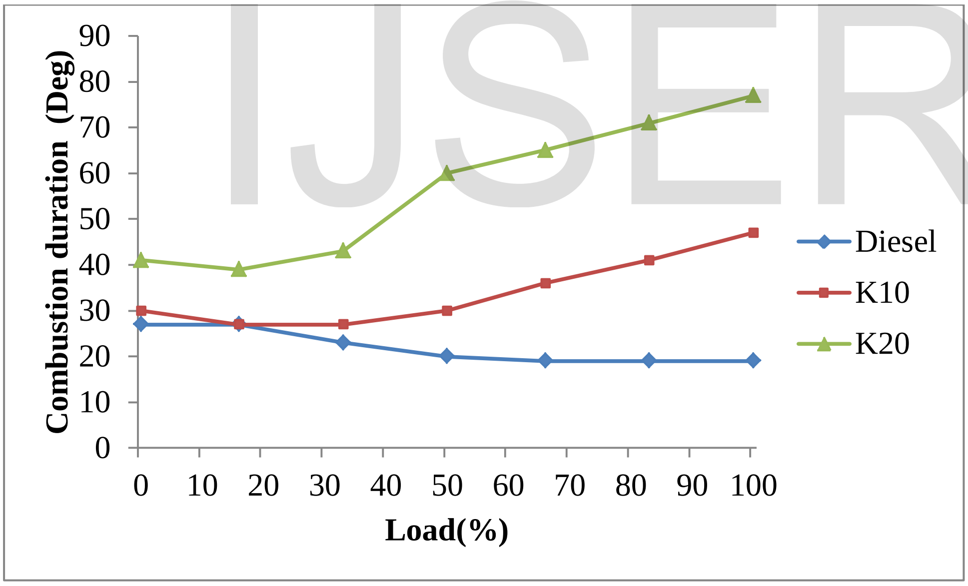

Fig 4.6.2.1

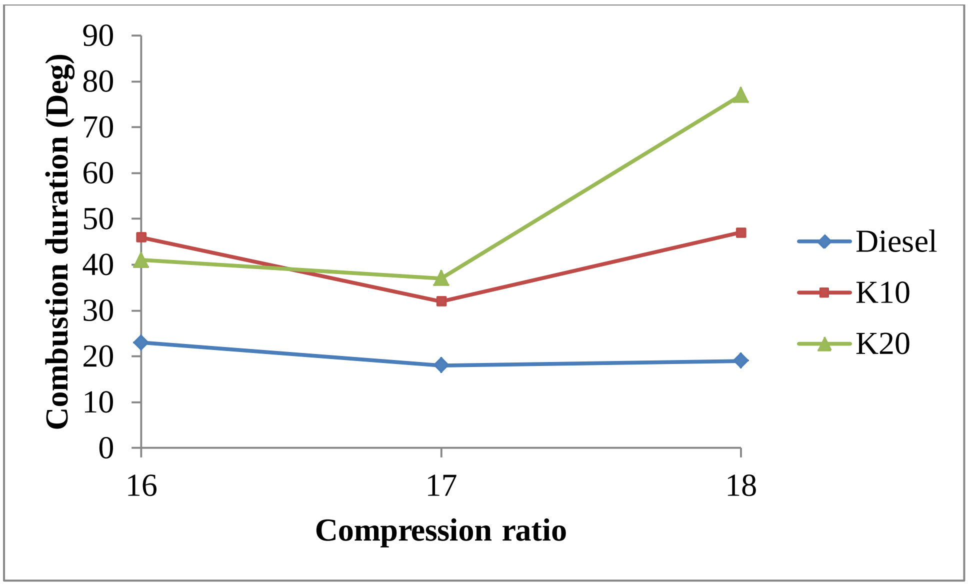

Fig 4.6.2.2

4.6.3.Net Heat release rate

The variation of the total combustion duration with different loads for different blends is shown in Fig 4.6.3.1. Increasing load heat release rate increases. The maximum heat release rate of diesel, K10, and K20 at full load has been observed to be 53.2, 47.2 and 41.5J/ ºCA. The heat release rate is analyzed based on the changes in crank angle variation of the cylinder. The heat release rate of karanja oil blends decreases compared to that of diesel at all load. The heat release rate of diesel is higher than oil blend due to its reduced viscosity and reduction of air entrainment and fuel-air mixing rates.

Fig 4.6.3.2 shows the variation of heat release rate with compression ratio for different blends. Low compression ratio Heat release rate increases with the low compression ratios and slightly decreases at higher compression ratio. This may be due to the air entrainment and lower air/fuel mixing rate and effect of viscosity blends. The heat release rate of diesel is higher than oil blend due to its reduced viscosity and better spray formation.

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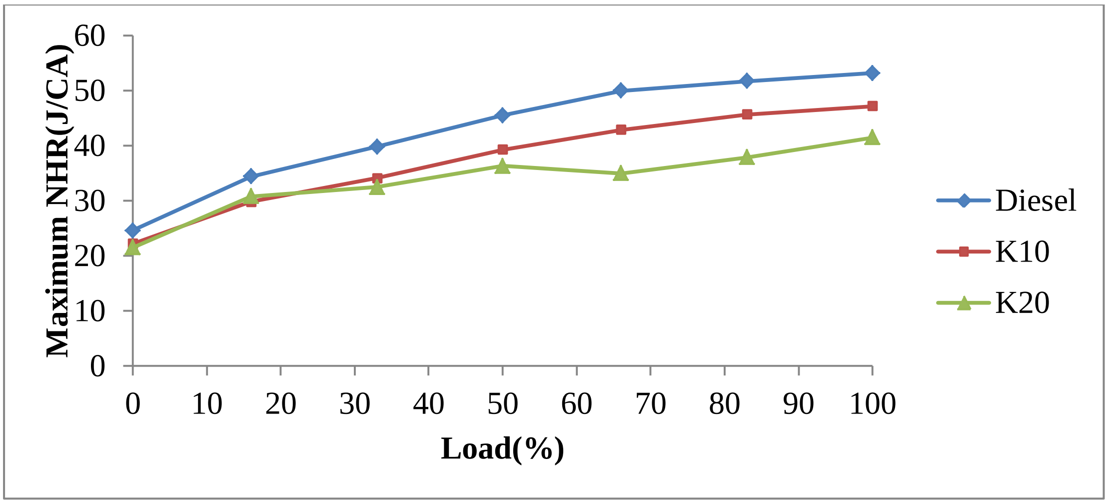

Fig 4.6.3.1

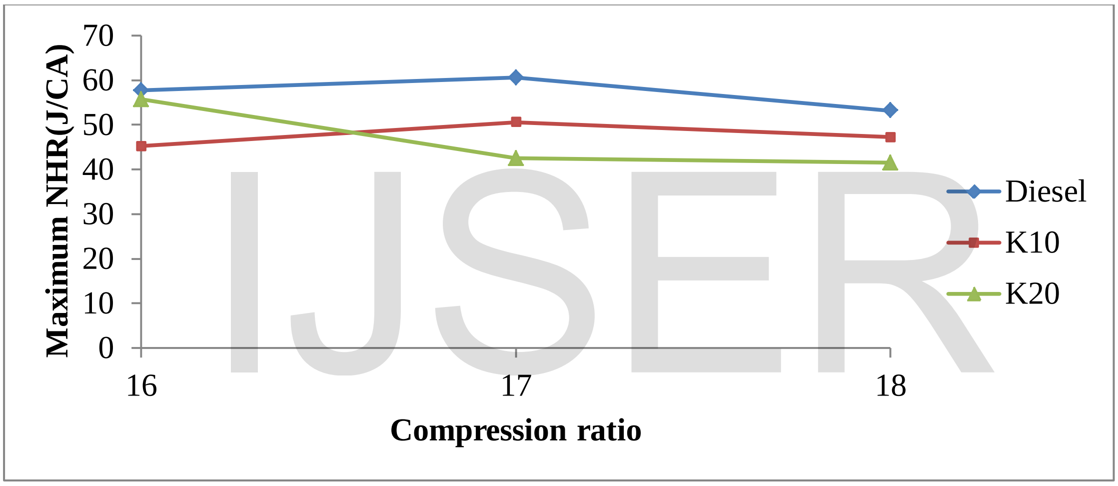

Fig 4.6.3.2

4.6.4 Mass fraction burnt

The variations of the mass fraction burnt with the crank angle for karanja oil blends and diesel at compression ratio 18 at full load is given in Fig 4.6.4.1, due to the oxygen content of the blend, the combustion is sustained in the diffusive combustion phase. Diesel gives higher mass fraction burnt than other blends. The highest rate of burning shows that the efficient rate of combustion. The engine operates in rich mixture and it reaches stoichiometric region at higher compression ratio. More fuel is accumulated in the combustion phase and it causes rapid heat release.

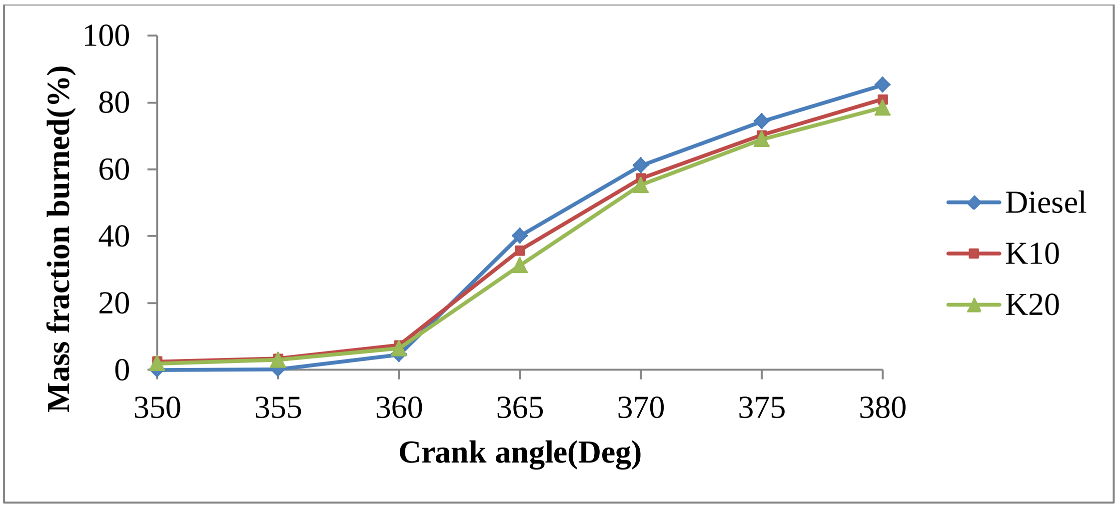

Fig 4.6.4.1

4.6.5 Ignition delay

The most vital parameter in combustion analysis is ignition delay. The variation of the ignition delay with different loads for different blends is shown in Fig 4.6.5.1. It has observed that the ignition delay decreases with karanja oil in the diesel blend with increase in load and increases in compression ratio. K20 give higher ignition delay than diesel. It is because more fuel required due to lower calorific value.

Fig 4.6.5.2 shows the variation of ignition delay with compression ratio for different blends. Ignitiondelay decrease with increase in compression ratio.

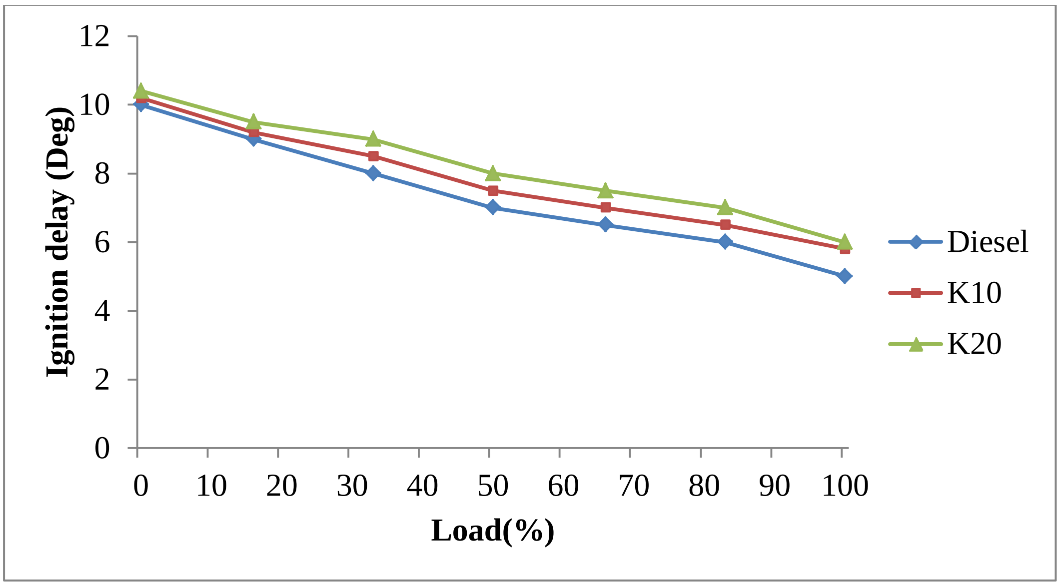

Fig 4.6.5.1

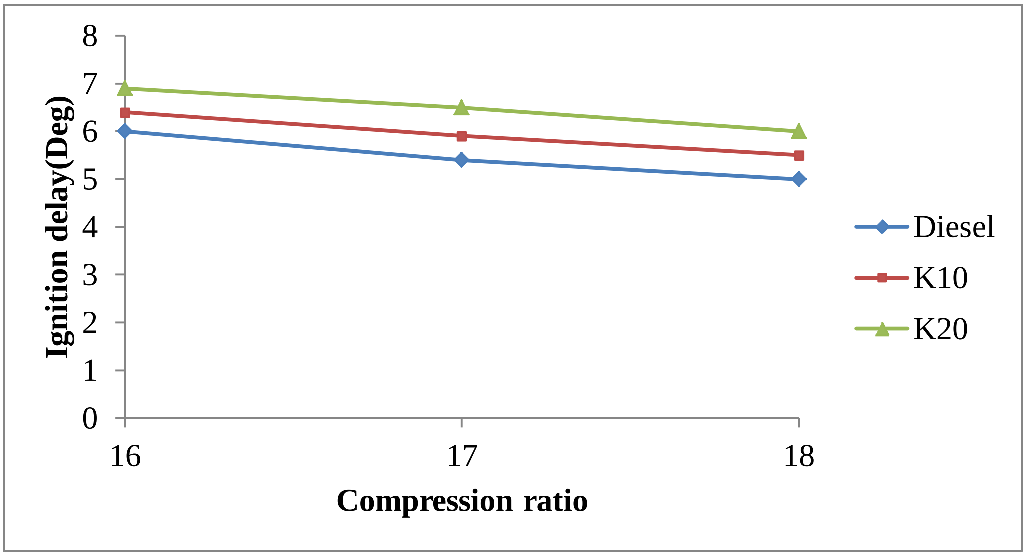

Fig 4.6.5.2

4.6.6 Maximum combustion temperature

The variations of the maximum combustion temperature with loads for different blends are given in Fig 4.6.6.1. Increasing load combustion temperature increases for all cases. Diesel gives better combustion temperature than other blends. The maximum combustion temperature of diesel, K10, and K20 at full load has been observed to be 1429.04, 1425.74 and 1400.01°C respectively.

Fig 4.6.6.2 shows the variation of maximum combustion temperature with compression ratio for different blends. It has been observed that increasing compression ratio combustion temperature increases. Diesel gives better combustion temperature than all other blends. Due to more fuel accumulated in the combustion chamber.

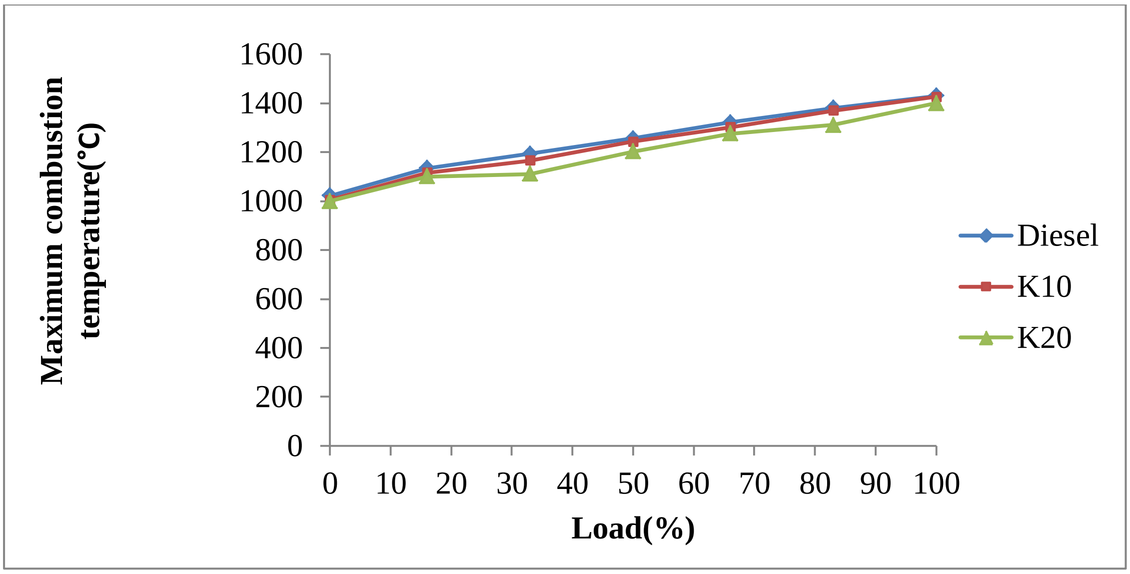

Fig 4.6.6.1

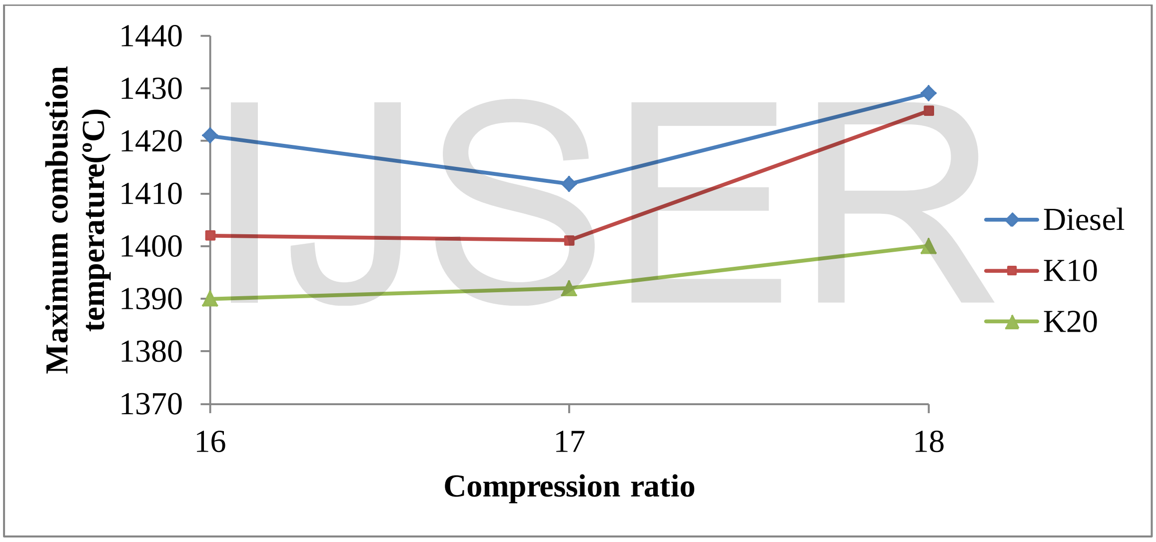

Fig 4.6.6.2

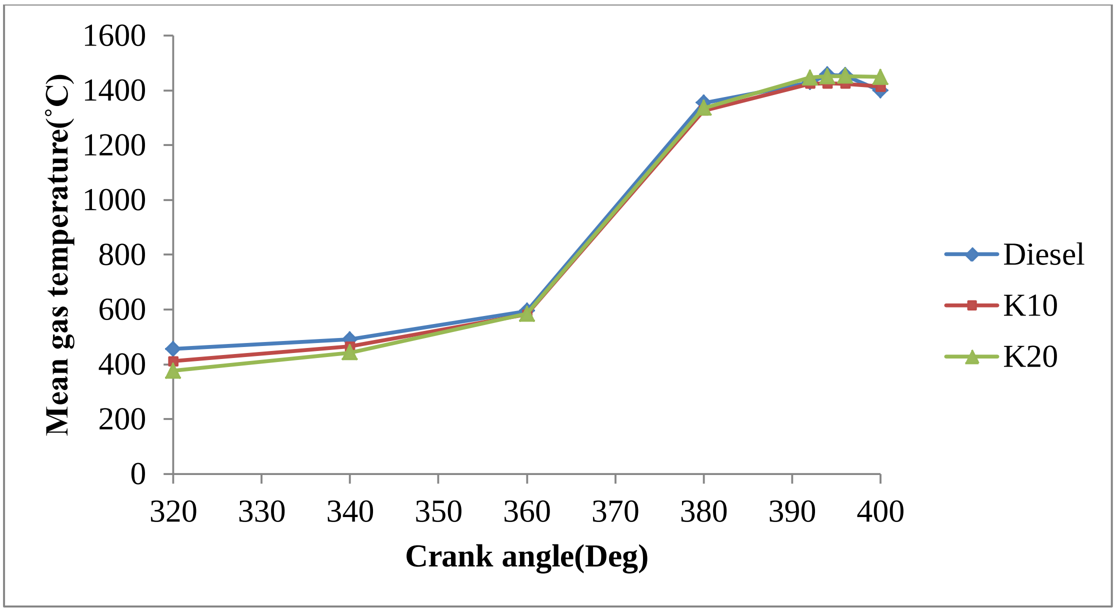

Fig 4.6.6.3

4.7 **Emission analysis of Karanja oil**

4.7.1 Carbon monoxide emission

Fig 4.7.1.1 shows the variation of carbon monoxide emission of the blends and diesel for various loads. CO emission is higher at lower load then decreases with increase in load and at higher load it again increases. The CO emission of the blend K10 and K20 is more at lower load compared to diesel, it may be due to higher viscosity and improper spray pattern resulting in incomplete combustion. At full load diesel gives highest CO emission.

Fig 4.7.1.2 shows the variation of carbon monoxide emission of the blends and diesel with various compression ratios. . CO emission decreases with increase in compression ratio. The CO emission of diesel is minimum compared to K10 and K20. This may be due to, at higher compression ratio air fuel mixing is better.

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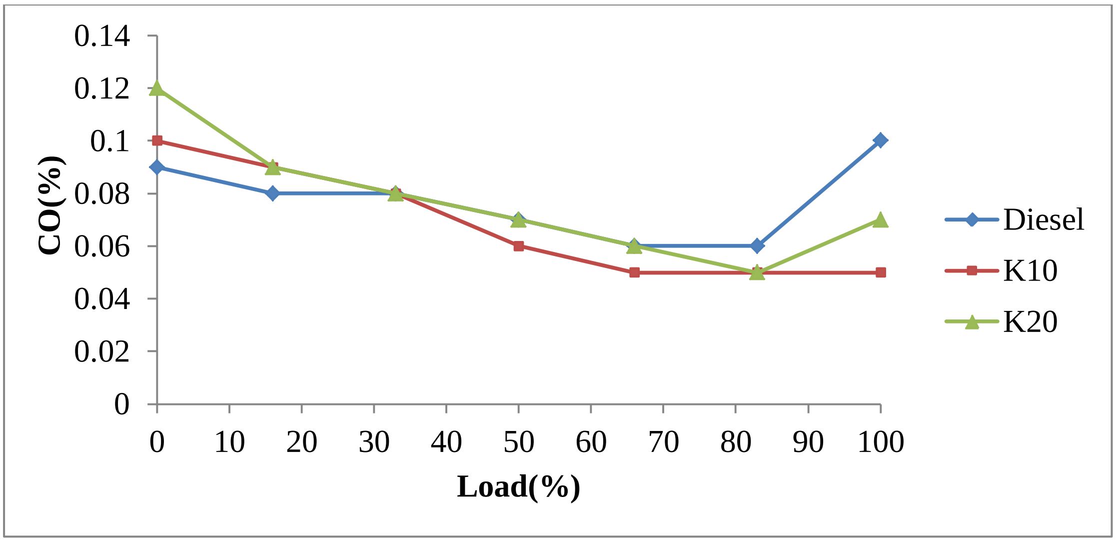

Fig 4.7.1.1

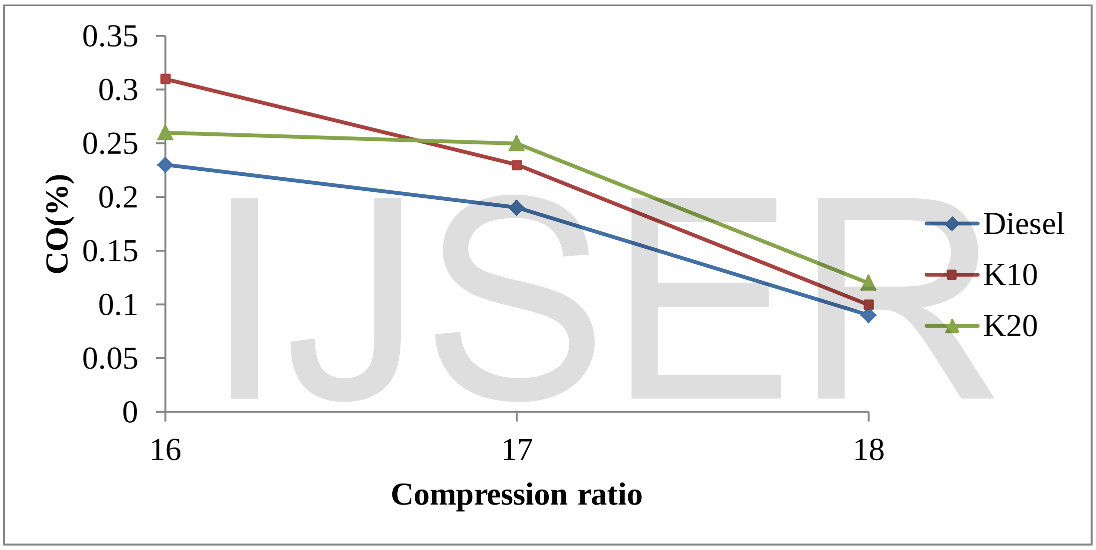

Fig 4.7.1.2

4.7.2 Carbon dioxide emission

The variation of carbon dioxide emission with different loads is shown in Fig 4.7.2.1. CO_2 emission increases with increase in load. In the range of whole engine load CO_2 emission of diesel fuel is lower than other fuel .This is because vegetable oil contains oxygen element,carbon content is relatively lower in the same volume of fuel consumed at the same engine load. More amount of CO_2 is an indication of complete combustion of fuel in the combustion chamber. CO_2 emission of the blend K20 is slightly higher than diesel at all loads. It is probably due to higher oxygen availability.

The variation of carbon dioxide emission with different compression ratio is shown in Fig 4.7.2.2. The blend emits higher percentage of CO_2 than diesel at lower compression ratios and vice versa. The CO_2 emission from the combustion of bio fuels can be absorbed by the plants and the carbon dioxide level and is kept constant in the atmosphere.

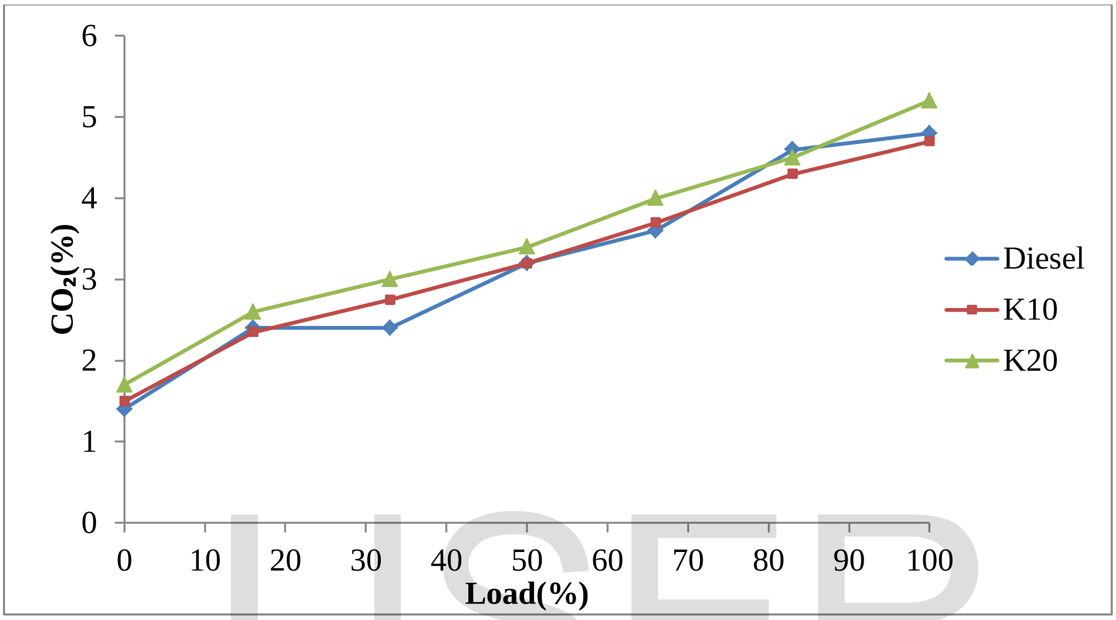

Fig 4.7.2.1

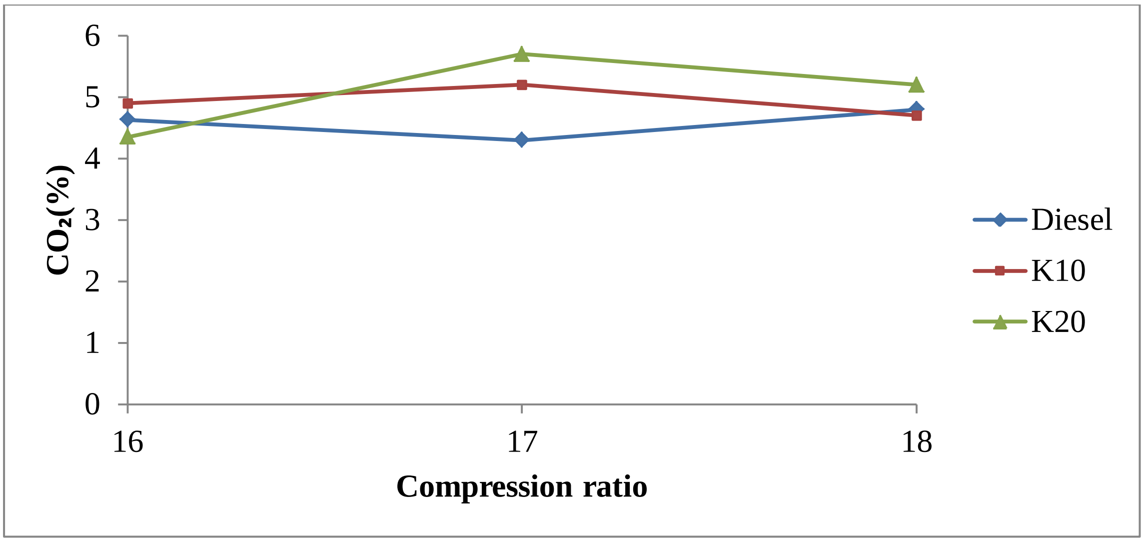

Fig 4.7.2.2

4.7.3 Hydrocarbon emission

The variation of hydrocarbon emissions with load for different blends is plotted in Fig 4.7.3.1. Increased HC emissions clearly show that the combustion in the engine is not proper. It is very clear that increasing the blend percentage of karanja oil increase the HC emissions. All blends have shown higher HC emissions at 50% load. This may be due to poor atomization of the blended fuel because of higher viscosity. Physical properties of fuels such as density and viscosity influence the HC emissions. The blend K10 has higher HC emissions at full load.

The variation of hydrocarbon emission with different compression ratios for different blends is given in Fig 4.7.3.2. It shows that the hydrocarbon emissions of various blends are lower at higher compression ratios. Blend K20 gives higher HC emission at lower compression ratio but at higher compression ratio K10 gives higher.

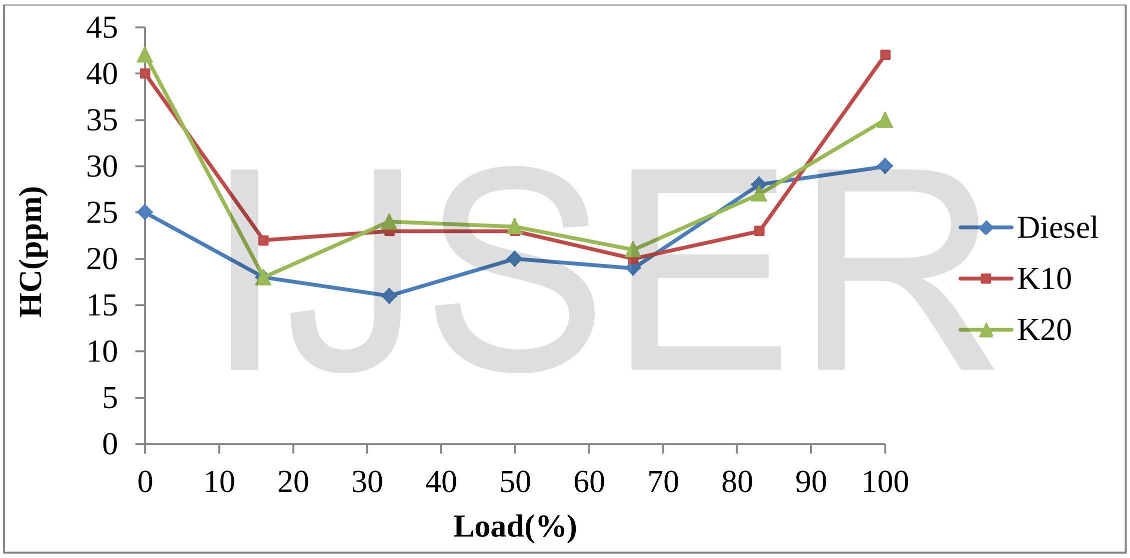

Fig 4.7.3.1

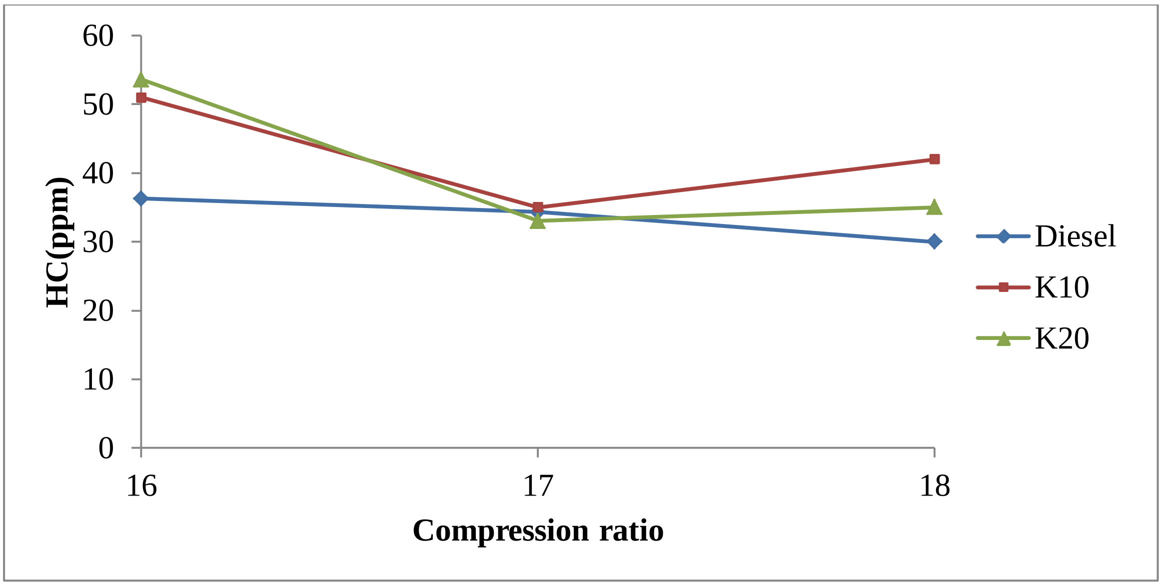

Fig 4.7.3.2

4.7.4 Nitrogen oxides emission

Fig 4.7.4.1 shows that the variations of Nitrogen oxides (NO_x) emission with load for different blends. NO_x emission increases with increase in load. This is properly due to higher combustion temperature in the engine cylinder with increasing load. It is also observed that with increasing the percentage of Karanjo oil blend there is a trend of decreasing NO_x emission. NO_x emission for diesel, K20, and K20 are 550ppm, 493ppm, and 524ppm at full load.The limitation is higher viscosity of these higher karanja oil blends.

The variation of nitrogen oxides (NO_x) emission with respect to different compression ratio for different blends is shown in Fig 4.7.4.2. The NO_x emission for diesel and other blends increase with increase of compression ratio. Diesel gives higher NOx emission than other blends. The other blends closely follow diesel.

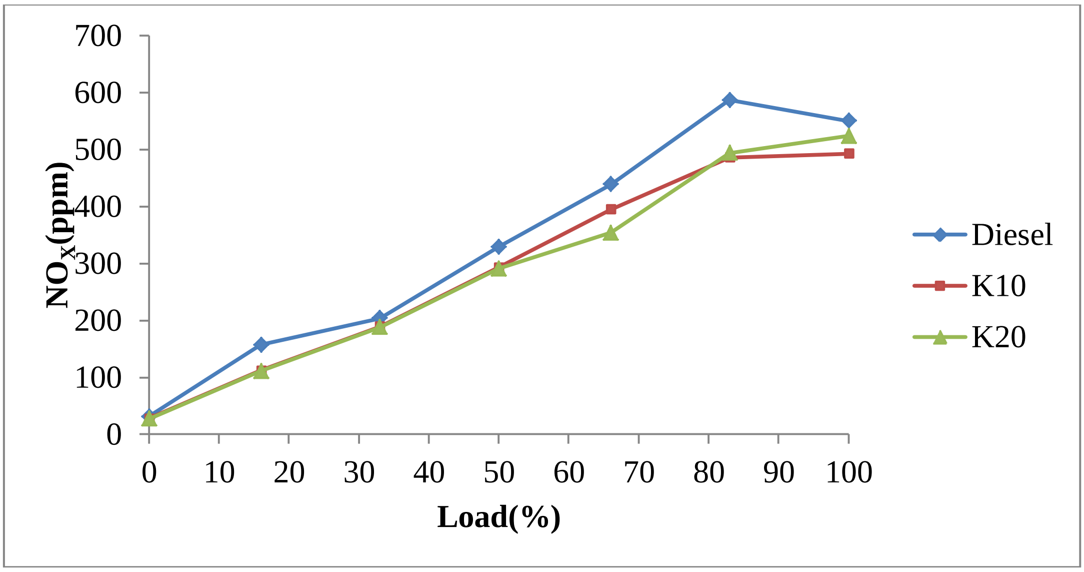

Fig 4.7.4.1

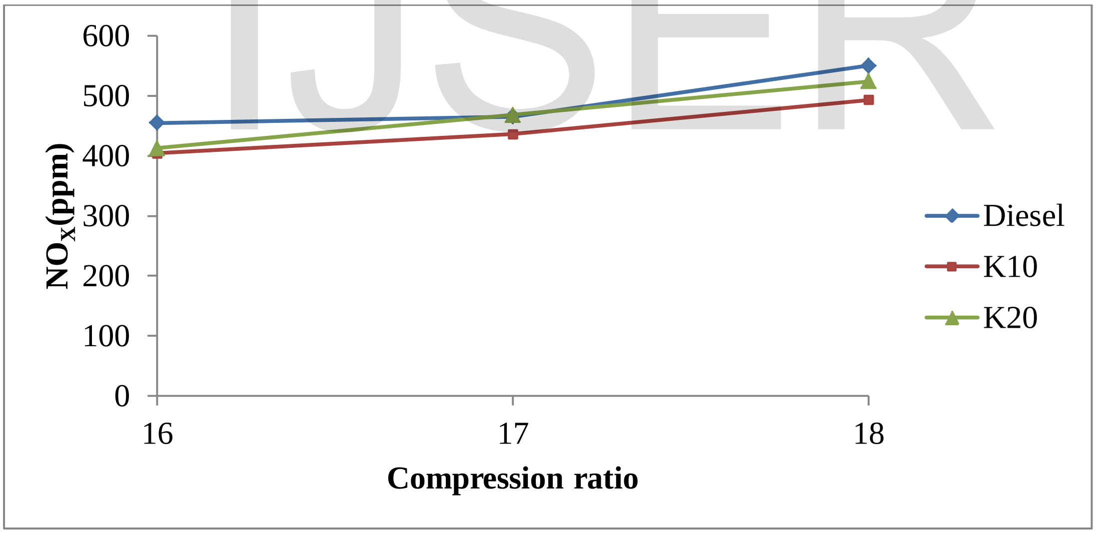

Fig 4.7.4.2

4.7.5 Smoke opacity

Fig 4.7.5.1 shows that the variations of smoke opacity with load for different blends. Smoke opacity increase with increase in load. K10 and K20 give higher smoke opacity than diesel. It is observed that K10 and K20 have smoke opacity less than diesel at nearly 70% load. Hence it can be conclude that K20 would be better blend from other. The smoke opacity for diesel, K10, and K20 at full load is 88.7%, 96%, and 97.8% respectively.

The variation of smoke opacity with respect to different compression ratio for different blends is shown in Fig 4.7.5.2. Smoke opacity increase with increase in compression ratio. K20 give higher smoke opacity than that of K10 and diesel.

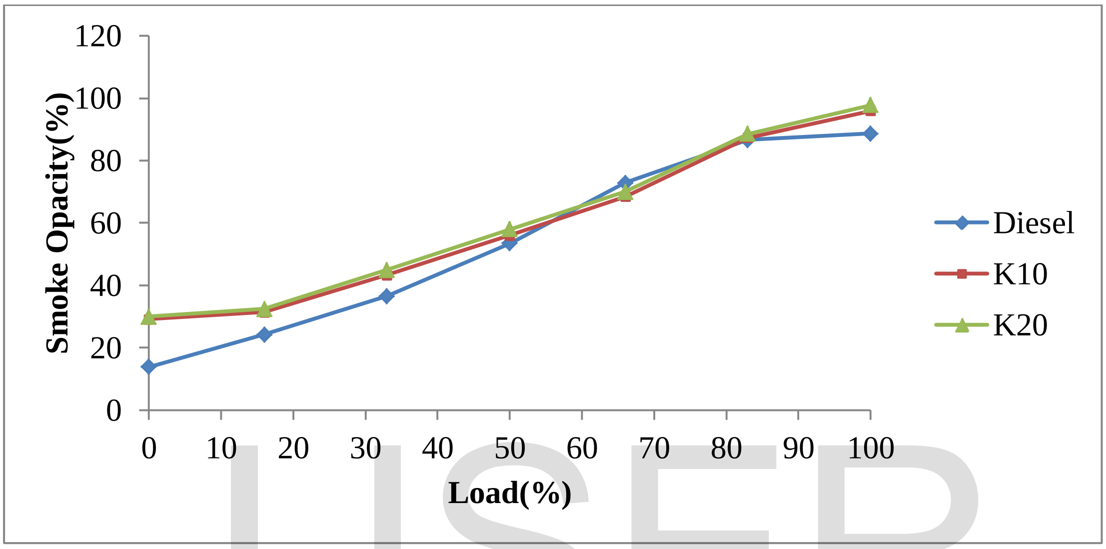

Fig 4.7.5.1

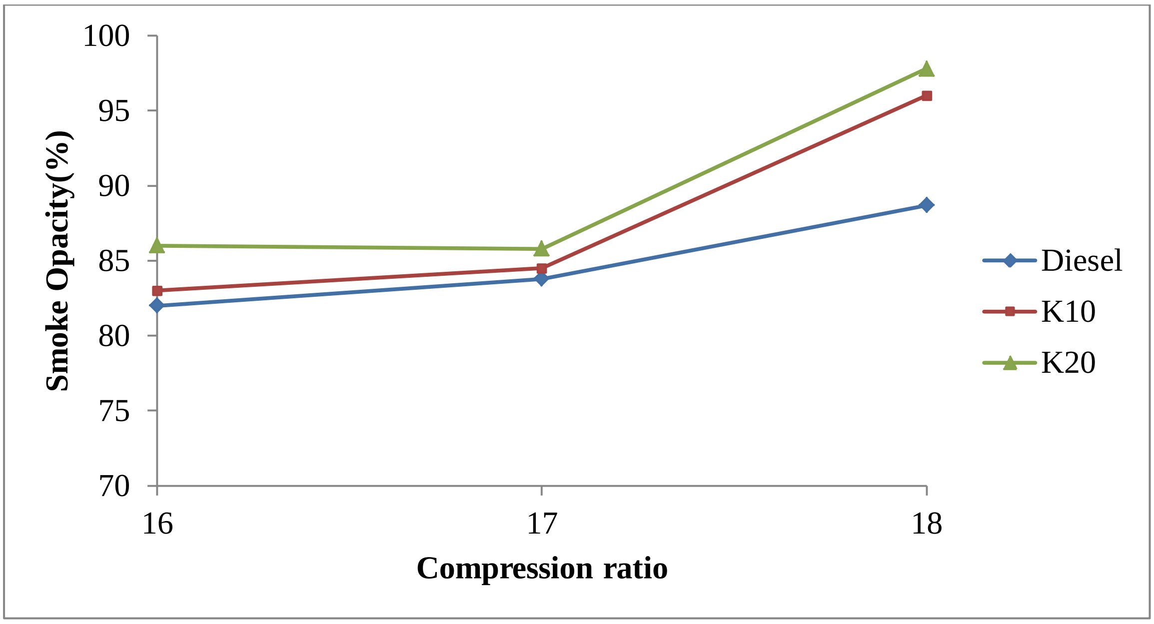

Fig 4.7.5.2

5. CONCLUSION:

The performance, emission and combustion characteristics of a dual fuel variable compression ratio engine with Karanja oil bio-diesel, Karanja oil and diesel blends have been investigated and compared with that of diesel. The experimental results confirm that the BTE, SFC, exhaust gas temperature, mechanical efficiency and torque of variable compression ratio engine, is a function of bio diesel blend, load and compression ratio. For the similar operating conditions, engine performance reduced with increase in bio-diesel percentage in the blend. However by increasing the compression ratio the engine performance varied and it becomes comparable with that of diesel. The following conclusions is drawn from this investigation:

- The brake thermal efficiency of the blend B40 is slightly higher than that of diesel at compression ratio 18. The specific fuel consumption of the blend B40 is lower than that of all other blends and diesel. The engine BTE at full load for diesel, B20, B40,and B60 fuels is 24.9%, 26.02%, 28.44% and 24.1% respectively. The brake specific fuel consumption of the blend B40 at the compression ratio of 18 is 0.32kg/kWh whereas for diesel it is 0.34kg/kWh.
- There is a steady increase in mechanical efficiency for all the blends as the load increases. Maximum mechanical efficiency has been obtained from blend B20 and B40 for full load and it is 54.61% and 53.56% respectively. The exhaust gas temperature decreases for different blends when compared to that of diesel. As the compression ratio increases, the exhaust gas temperature of various blends is lesser than that of diesel.

- Increasing compression ratio torque increases. B20 gives better torque than diesel. The highest torque obtained is 22.22Nm for B20 for a compression ratio 18, whereas the torque is only 21.93Nm for diesel.
- Combustion analysis reveals almost the same pressure crank angle characteristics for KOME blends and diesel. The peak pressure has been observed to be 61.2 bar, 59.6 bar, 60.15 bar and 60.55 bar for diesel and blends B20, B40, and B60 respectively at full load.
- The combustion duration in general increases with load. At compression ratio 18, B60 gives better combustion duration as compared to diesel. With increase in compression ratio combustion duration increases.
- The maximum heat release rate of diesel, B20, B40,and B60 has been observed to be 53.2, 48.8, 47.7 and 45.07 J/ ºCA. The heat release rate of KOME blends decreases compared to that of diesel at full load.
- The mass fraction burnt of blends is slightly higher than that of diesel at full load. The mass fraction burnt for the fuel blend B40 is higher than that diesel for crank angle 350º-360º and it is closer for crank angle 365º-380º.
- It has observed that the ignition delay decreases with biodiesel in the diesel blend with increase in load and increases in compression ratio. At compression ratio 18 and 100% load condition, the ignition delay period of B20, B40 and B60 is 5.5, 6.2, 6.5 ºCA higher than diesel.
- B20 gives better combustion temperature than diesel. Increasing load combustion temperature increases for all cases. It has been observed that increasing compression ratio combustion temperature increases. B20 gives better combustion temperature than all other blends.
- The CO emission of the blend B40 and B20 is slightly less than diesel and it is found to be higher for light and medium load. The CO emission of the blend B60 is more than diesel. The CO emission of the blend B40 is close to diesel and it is found to be higher for compression ratio 16. The other blends B20, B60 have slightly lesser CO emission for compression ratio 16. It observed that by increasing compression ratio CO emission decreases.
- CO_2 emission of the blend B40 is slightly higher than diesel. By increasing load CO_2 emission increases. The blend emits higher percentage of CO_2 than diesel at lower

compression ratios and vice versa. CO_2 emission of the blend B40 for compression ratio 16.

- The hydrocarbon emissions of various blends is higher at higher loads. It shoes that the increase in load increase the hydrocarbon emission for the blend B40. It shoes that the hydrocarbon emission of various blends is lower at higher compression ratios. The blends B20 and B40 produce lesser hydrocarbon emissions at higher compression ratio than diesel except B60.
- The NO_x emission for biodiesel and its blends is higher than that of diesel except B60. . The NO_x emission for diesel and other blends increase with increase of compression ratio.
- Smoke opacity increase with increase in load. B40 and B60 gives higher smoke opacity than diesel. Smoke opacity decrease with increase in compression ratio. B40 and B60 gives higher smoke opacity than that of B20 and diesel.
-

- The performance, emission and combustion characteristics of a dual fuel variable compression ratio engine with Karanja oil and diesel blends have been investigated and compared with that of diesel.
- The experimental results confirm that the BTE, SFC, exhaust gas temperature, mechanical efficiency and torque of variable compression ratio engine, is a function of bio diesel blend, load and compression ratio. For the similar operating conditions, engine performance reduced with increase in bio-diesel percentage in the blend. However by increasing the compression ratio the engine performance varied and it becomes comparable with that of diesel.
- The following conclusions is drawn from this investigation was found out. K10 gives less BSFC compared to K20 and diesel. . The engine BTE at full load for diesel, K10, and K20 fuels is 24.9%, 26.63%,and 24.1% respectively. It is also observed that the BTE of the blend K20 is slightly lower than that of the diesel and K10 is higher than diesel.
- The highest temperature obtained is 324.53ºC for diesel for full load, whereas the temperature is only 317.93ºC and 310.39ºC for the blend K10 and K20.It may be due to energy content in diesel is higher as compared to K10 and K20. CO, HC

emission of K10 and K20 is lower than diesel and NOx emission was higher than diesel.

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