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Full Length Article

Investigation on performance and emission characteristics of a variable compression multi fuel engine fuelled with Karanja biodiesel–diesel blend

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ABSTRACT

The performance and emission of a single cylinder four stroke variable compression multi fuel engines when fueled with 20%, 25% and 30% of Karanja blended with diesel are investigated and compared with standard diesel. Experiment has been conducted at compression ratios of 15:1, 16:1, 17:1, and 18:1. The impact of compression ratio on fuel consumption, brake thermal efficiency and exhaust gas emissions has been investigated and presented. Experimental analysis on the performance of biodiesel over diesel was evaluated by response surface methodology to find out the optimized working condition. The overall optimum is found to be 25% biodiesel–diesel blended with a compression ratio of 18.

1. Introduction

The large increase in the number of automobiles and fast depletion of world petroleum reserves have resulted in a great demand for petroleum products [1]. The worlds energy demand in the last two decades has encouraged the world towards searching for the alternative energy sources [2]. The developing country like India is desirable to produce bio-diesel from non-edible oils which can be extensively grown in the waste land of the country [3]. The usage of bio-diesel has reduced the tail pipe emission of carbon monoxide (CO), Hydrocarbons (HC) and particulate matter (PM) [4]. Bio diesel acts as a promising alternative fuel to diesel oil. Vegetable oils are a very promising alternative to diesel oil since they are renewable and have similar properties. Many researchers have studied the use of vegetable oils in diesel engines. Vegetable oils offer almost the same power output with a slightly lower thermal efficiency when used in diesel engines. Reduction of engine emissions is a major research aspect in engine development with the increasing concern on environmental protection and the stringent exhaust gas recirculation. Biodiesel such as Jatropha, Karanja, sunflower, rapeseed are some of the popular biodiesel that are currently considered as substitutes for diesel. These are clean burning, renewable, non-toxic, biodegradable and environmentally friendly transportation fuels that can be used in neat form or blended with petroleum derived in diesel engines. Vegetable oil esters particularly karanja appear to be the best alternative fuel to diesel.

Diesel engines have a negative effect on environment since they include high amounts of sulphur and aromatics. CO, SO_X, NO_X and smoke are produced from fossil fueled diesel engine exhaust emissions [5]. It has been observed that engine parameters such as injection timing, compression ratio have considerable effects on the performance and emissions of diesel engines running on biodiesel blends. Many innovative technologies are developed to tackle these problems. Modification is required in the existing engine designs [6,7].

Jindal et al. [8] studied the effects of the engine design parameters such as compression ratio, fuel injection pressure and the performance parameters such as fuel consumption, brake thermal efficiency, emissions of CO, HC, NO_x, CO_2, and smoke opacity with jatropha methyl ester as fuel. The highest performance is achieved by the engine at 250 bar injection pressure and compression ratio of 18 at which BSFC improves by 10% and BTE improves by 8.9%. With regard to emission aspects increase in compression ratio leads to an increase in emission of HC and exhaust temperature whereas smoke and CO emission reduces.

Muralidharan et al. [9] investigated the BTE and found out that the blend B40 with waste cooking oil is slightly higher than that of standard diesel at higher compression ratios. Waste cooking oil blends give higher combustion pressure at high compression ratio due to longer ignition delay, maximum rate of pressure rise and lower heat release rate when compared with diesel. Brake thermal efficiency of the blends increases with increase in applied load.

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Nomenclature

B20	20% biodiesel + 80% diesel
B25	25% biodiesel + 75% diesel
B30	30% biodiesel + 70% diesel
BP	brake power
BTE	brake thermal efficiency
BSFC	brake specific fuel consumption
CO	carbon monoxide
CO_2	carbon dioxide
UHC	unburnt hydrocarbon
CR	compression ratio
IMEP	indicated mean effective pressure
NO_x	nitrogen oxides
VCR	variable compression ratio

Vasudevan et al. [10] Conducted experiments on variable compression ratio engine and found the maximum brake thermal efficiency at full load is 38.46% for B40 of waste cooking oil which is 4.1% higher than that of diesel using variable compression ratio engine at compression ratio 21. In 2009 Arul Mozhi Selvan et al. [11] compared the combustion characteristics of single-cylinder four strokes DI variable compression ratio engine under compression ratio of 15:1, 17:1 and 19:1 when using diesel and Biodiesel-ethanol blends as fuel. It has been observed that the cylinder gas pressure, maximum rate of pressure rise and heat release rate increase with higher ethanol concentration due to longer ignition delay. The exhaust gas temperature was found to be less and it has been observed that performance and emissions have been reduced on a variable compression ratio engine. Ganapathy et al. [12] studied the effect of injection timing along with engine operating parameters in Jatropha biodiesel engine is important as they significantly affect its performance and emissions. Advancing the injection timing (5 crank angle degree from factory settings (345 CAD) causes reduction in BSFC, CO, HC and smoke and increase in BTE, peak cylinder pressure, HRRmax and NO emission with Jatropha biodiesel operation. In 2012 L. Labecki, et al. [13] studied the combustion and emission characteristics of rapeseed plant oil (RSO) and its blends with diesel fuel in a multi-cylinder direct injection diesel engine and found out the NOx emissions for RSO and its blends are lower when compared to diesel fuel but their soot emissions are much higher than diesel. Saravanan et al. [14] analyzed the combustion characteristics of crude rice bran oil methyl ester blend in a direct injection compression ignition engine and found that the cylinder pressure was comparable whereas the delay period and the maximum rate of pressure rise was lower than that of diesel. Kasiraman et al. [15] studied the performance and combustion analysis of a neat cashew nut shell oil (CSNO) as a fuel in direct injection diesel engine The cashew nut shell oil 70% and camphor oil 30% blend (CMPRO30) Performs closer to diesel with respect to performance, emission and combustion characteristics. The brake thermal efficiency of CMPRO 30 blend is 29.1% at peak load compared to diesel brake thermal efficiency of 30.14% whereas it is 23.1% for neat CSNO. At peak load the NO emissions of CMPRO 30 blend, diesel fuel and neat CSNO are 1040 ppm, 1068 ppm and 983 ppm, respectively. The smoke emissions are higher for neat CSNO with a value of 4.22 BSU. For CMPRO 30 blend it is 3.91 BSU whereas it is 3.64 BSU for diesels. The peak pressure, maximum rate of pressure rise, ignition delay, combustion duration and heat release rates of CMPRO 30 blend and diesel fuel are comparable. Yang et al. [16] investigated the performance, combustion and emission characteristics of diesel engine fueled by biodiesel at partial load conditions Due to the lower calorific value of biodiesel, the BSFC increases with the increasing biodiesel blend ratio at all engine loads. With regard to the impact of partial loads, it is found that the percentage Increase in BSFC of B100 as compared to diesel increases with the decreasing engine load. The largest increase in BSFC is found tobe at 10% load where a 28.1% increase in BSFC is observed. As for BTE, the experimental results show that the use of biodiesel results in a reduced thermal efficiency at lower engine loads and an improved thermal efficiency at higher engine loads. Raheman and Ghadge [17] studied the performance of RicardoE6 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 biodiesel 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.

Some optimization approach has to be followed so that the efficiency of the engine is not compressed. As far as the internal combustion engines are concerned the thermal efficiency and emission are the important parameters for which the other design and operating parameters have to be optimized. The most common optimization techniques used for engine analysis are response surface method, grey relational analysis [18–20]; artificial neural network has been employed to predict output parameters of the engine [21]. Taguchi technique has been popular for parameter optimization in design of experiments. Multi objective optimization of parameters using non linear regression has found optimum value to be 13% biodiesel–diesel blend with an injection timing of 24°Btdc [21]. Karnwal et al. [22] used the Taguchi method for analyzing the role of operating and injection system parameters on low noise, performances and emissions. Ganapathy et al. [23] reported the performance optimization of jatropha biodiesel engine model using Taguchi approach. Many researches about optimization and modification on engine low temperature performances of engine; new instrumentation and methodology for measurements should be performed when petroleum diesel is substituted completely by biodiesel [24]. Most of the research studies concluded that in the existing design of engine and parameters at which engines are operating a 20% blend of bio-diesel with diesel works well [25].

From the review of literature, it can be seen that while lot of work has been carried out to improve the performance of biodiesel fueled compression ignition engine. However, it has to be noted that the study on variable compression ratio engine using bio diesel is limited. The effect of compression ratio on engine parameters, emission and combustion characteristics have not been studied extensively. Hence this study has been devoted to find suitable compression ratio which gives optimum performance. In this research, Karanja oil and its blends with diesel is chosen as fuel for variable compression ratio multi fuel engine. The various blends of Karanja and standard diesel fuel are prepared and the following investigations are carried out. The performance, emission and combustion characteristics of variable compression ratio engine using various blends at compression ratios of 15:1, 16:1, 17:1, and 18:1, for all loads are studied and it is compared with the results of standard diesel fuel.

2. Materials and methods

In this study the variable compression ratio engine was run with karanja biodiesel at different compression ratios to evaluate the performance with emissions. The results were compared against the diesel fuel as well as for different combinations of compression ratio and loads.

2.1. Fuel preparation

The vegetable oils were obtained from commercial sources and used without further purification. The samples were converted to methyl esters by alkali catalytic and non catalytic super critical methanol transesterification methods. Transesterification (also called alcoholysis) is the reaction of a fat or oil with an alcohol to form esters and glycerol.

Untreated oil is mixed with a mixture of anhydrous methanol and a catalyst (Methoxide) in proper proportion. The mixture is maintained at a temperature little below 65 °C (being the B.P. of methanol) and continuously stirred for around three hours. After completion of stirring, the mixture is allowed to settle down for 24 h. The layer of glycerol settled at the bottom is carefully taken out and the upper layer is the ester of karanja oil which is tapped separately. The fuel blend was prepared just before commencing the experiments to ensure the mixture homogeneity. The properties of the fuel blend and diesel have been determined as per the ASTM Standards in an analytical lab. The fuels properties were tested using standard measuring devices shown in Table 1 and results are shown in Table 2.

2.2. Experimental set-up

Commercial diesel fuel used in India has been taken for base line reading for this study. The test engine used is variable compression ratio multi fuel engine coupled with eddy current dynamometer. The performance of the engine was analyzed by using engine performance analyze software package "Engine soft 8.0" for online performance analysis. The Germany made MRU delta 1600L type exhaust gas analyzer are used to measure the various constituent of exhaust gases such as HC, CO, CO_2, NO_X. The exhaust gases like HC, CO, CO_2 are measured by the infrared measurement and other constituent NO_X are measured by Electro chemical sensor. The present study was carried out to investigate the performance and emission of karanja blend in diesel by volume basis in variable compression ratio engine and compared with diesel. Table 3 shows the specification of the experimental engine setup. An electric dynamometer was used to apply load on the engine. Tests were carried out for various loads, starting from no load to full load condition. At each load for different blend the fuel flow rate and the various constituents of exhaust gases such as carbon monoxide, hydrocarbon, nitrogen oxides and carbon dioxide were analyzed. Computerized data acquisition system is used to collect, store and analyze the data during the experiment by using various sensors.

Table 1
Measuring devices and test methods for measuring fuel properties.

Properties	Measurement and Apparatus	Standard Test Method
Density	Hydrometer	ASTM D941
Flash & Fire Point	Penksy martins apparatus	ASTM D93
Calorific Value	Bomb Calorimeter	ASTM D240
Viscosity	Glass capillary viscometer	ASTM D445
Cetane number	Ignition Quality Tester	ASTM D613

2.3. Test procedure

The variable compression ratio engine available in the laboratory is started by using manual crank start. When the engine reaches the operating condition load is applied. The test is conducted at variable speed. Experiment was carried out on a test engine running on diesel, B20, B25 and B30in order to analyze the performance and emission. All the experiments were carried out at constant injection pressure of 200 bar by varying the load from 0 to 12 kg. After completion of each experiment the engine was run on diesel in order to flush out fuel in fuel line. Hydrocarbons(HC),Carbon Monoxide(CO),carbon dioxide(CO_2) and nitrogen oxides(NO_x), were measured with a 5 gas MRU delta exhaust gas analyzer. From the initial measurement brake thermal efficiency (BTE) and specific fuel consumption with respect to compression ratio of 15:1, 16:1, 17:1 and 18:1 for different blend are calculated and recorded. In each experiment operating parameter, the combustion parameter and exhaust emission are analyzed and stored in personal computer and the results are analyzed. The same procedure is repeated for the different blend of karanja diesel blend. Table 4 shows the accuracy of the measurement and variation in the calculated result of various parameters.

2.4. Error analysis

Errors and uncertainties in the experiments can arise from instrument selection, condition, calibration, environment, observation, reading and test planning. Errors will creep into all experiments regardless of the care which is exerted. Uncertainty analysis is needed to prove the accuracy of the experiments. In any experiment, the final result is calculated from the primary measurements. The error in the final result is equal to the maximum error in any parameter used to calculate the result(Holman) Percentage uncertainties of various parameters like total fuel consumption, brake power, brake specific fuel consumption and brake thermal efficiency was calculated using the percentage uncertainties of various instruments used in the experiment. For the typical values of errors of various parameters given in Table 4, using the principle of propagation of errors, the total percentage uncertainty of an experimental trial can be computed

$$
\begin{aligned}
&= \text{Square root of}[(\text{uncertainty of brake power})^2 \\
&\quad + (\text{uncertainty of SFC})^2 + (\text{uncertainty of TFC})^2 \\
&\quad + (\text{uncertainty of BTE})^2 + (\text{uncertainty of HC})^2 \\
&\quad + (\text{uncertainty of CO})^2 + (\text{uncertainty of NOx})^2 \\
&\quad + (\text{uncertainty of Pressure pick up})^2] \\
&\quad = 1.85\%
\end{aligned}
$$

3. Result and discussion

3.1. Brake thermal efficiency (BTE)

Brake thermal efficiency evaluates how efficient the engine transforms the chemical energy of the fuel into useful work. This parameter is determined by dividing the brake power of the engine by the amount of energy input to the system.

Fig. 1 shows the variation of brake thermal efficiency for the blend B25 at all loads. From the experiment it was observed that BTE increases with increasing in load for both diesel as well as Bio-Diesel blends. It was due to increase in power developed with

Table 2
Properties of Biodiesel-blends-Karanja.

Fuel blend	Kinematics Viscosity (cSt) ν	Heating Value (kJ/kg) HV	Flash Point (°C) FP	Density (g/cm³) ρ	Cetane Number
Diesel	2.71	44,800	55	0.836	51.00
B20	3.04	43,690	96	0.851	51.70
B40	3.51	43,150	99	0.854	52.82
B50	3.62	43,307	106	0.856	53.15
B60	3.81	42,937	123	0.859	53.86
B100	4.37	42,133	163	0.900	54.53

Table 3
Test engine and instrument details.

Specification of variable compression ratio engine	
General details	4-Stroke, water cooled, VCR
Rated power	4.5 kW at 1500 rpm
Speed	1500 rpm (variable)
Number of cylinder	Single cylinder
Compression ratio	12:1–18:1
Bore	87.5 mm
Stroke	110 mm
Ignition	Compression ignition
Loading	Eddy current dynamometer
Load sensor	Strain gauge load sensor
Temperature sensor	Type-K chromel
Starting	Manual crank start
Cooling	Water
Air flow transmitter	Pressure transmitter
Rotometer	Pressure transmitter

Table 4
The accuracies of the measurement.

Measurements	Accuracy	Percentage uncertainty
Engine speed	±2 rpm	±0.2
Temperatures	±1 °c	±0.1
Carbon monoxide	±0.02%	±0.2
Hydrocarbon	±10 ppm	±0.2
Carbon dioxide	±0.5%	±1.0
Nitrogen oxides	±15 ppm	±0.2
Burette fuel measurement	±2 CC	±1.5
Crank angle encoder	±0.5° CA	±0.2
Load	±1 N	±0.2

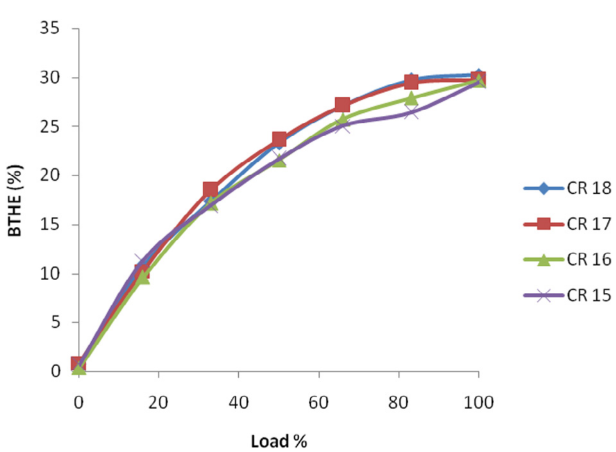

Fig. 1. Variation of BTE with load for different compression ratio for B25.

increase in load. By increasing the compression ratio of the engine, the brake thermal efficiency also gets increased for all the fuel types tested. Brake thermal efficiency is directly proportionate to the compression ratio [26]. Fig. 2 shows the variation of brake thermal efficiency for different compression ratios and for different blends at maximum load. It has been observed that the brake thermal efficiency of the blend B25 is higher for the compression ratio 18. The brake thermal efficiency of the standard diesel, B20, B25 and B30 for compression ratio 18 is 28.9%, 29.2% and 30.26% and 29.76%. Increase in thermal efficiency is due to increase in peak pressure and increase in combustion temperature. The decrease in BTE for higher blend percentage may be due to lower heating value and increase in fuel consumption [27] of pungam methyl ester blend.

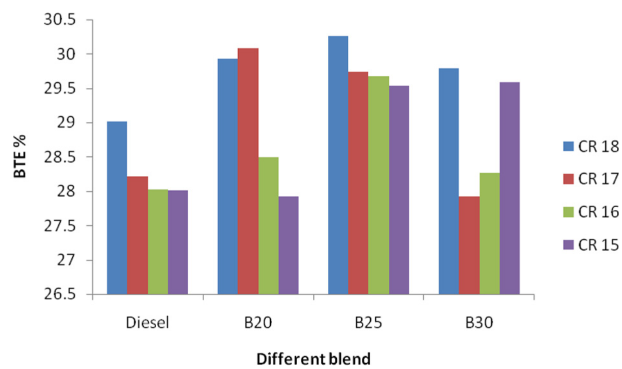

Fig. 2. Variation of BTE with fuels blends for different compression ratio at Max load.

3.2. Brake specific fuel consumption

Fig. 3 shows the variation in the Brake specific fuel consumption for B25 at all loads for different compression ratios. It is obvious from the figure that BSFC of the engine gradually decreases with the increase in load. BSFC for compression ratio 18:1 is comparatively lower than other compression ratios of 15:1, 16:1, 17:1. At higher compression ratio and higher load the energy required per

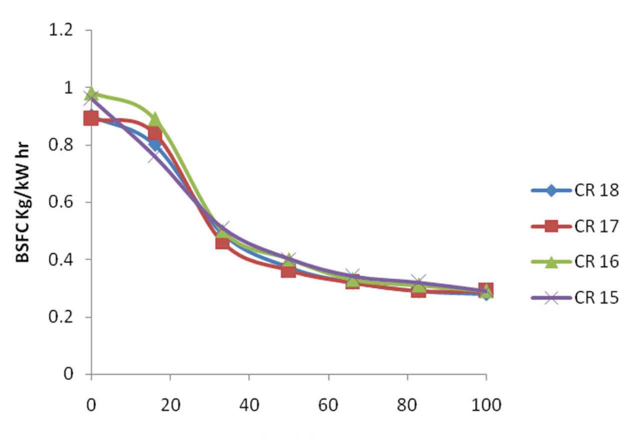

Fig. 3. Variation of BSFC with load for different compression ratio for B25.

kilo watt is lesser than that of lower compression ratio. The BSFC for the blend B20, B25, B30 at higher load are 0.29, 0.28 and 0.29 kg/k Wh whereas for diesel it is 0.31 kg/k Wh. Fig. 4 shows the variation of BSFC for diesel and bio diesel blend for different compression ratio at full load operation. For the blend of B25 it is lower than other blend. At higher percentage of blends, the BSFC increases. This may be due to fuel density, viscosity and heating value of the fuels. B30 has higher energy content than B20, B25 but lower than diesel. At higher percentage of blends, the specific fuel consumption increases. This is due to the decrease in calorific value at higher blends [9].

3.3. Indicated mean effective pressure

Fig. 5 shows that the indicated mean effective pressures for blend B25 is higher at higher loads and lower at lower loads than standard diesel. The variation of indicated mean effective pressure with load for different blend is shown in Fig. 6 the blend B25 closely follows standard diesel at compression ratio of 16 and 15. The indicated mean effective pressure for blend B25 and diesel at full load is 5.5 bar and 7.48 bar and respectively.

4. Emission characteristics

The engine operating parameters such as air-fuel equivalence ratio, fuel type, combustion chamber design and atomization ratio affect all emissions emitted by internal combustion engines. Especially, emissions of CO and unburned HC in the exhaust are very important since they represent the low chemical energy that cannot be totally used in the engine. Emissions such as CO_2, NO_x emitted by diesel engine have important effects on ozone layer and human health.

The engine emissions with Karanja biodiesel have been evaluated in terms of CO, HC, CO_2 and NO_X at various CR at different loading conditions of the engine.

4.1. Nitrogen oxides

Fig. 7 shows the variation of NO_x emission for B25 at all loads and different compression ratio. From the experiment it was observed that NO_X emission increases with load. The NO_x emission for compression ratio 18, 17, 16 and 15 are 167 ppm, 148 ppm, 145 ppm and117 ppm respectively. The minimum value of NO_x emission was found at the compression ratio of 15:1 and it increases as the compression ratio increases. Fig. 8 shows the variation of NO_x emission for diesel, B20, B25 and B30. Compared to diesel No_x emission is higher for bio diesel blend. The increase of NO_x emission for biodiesel operation may be due to the less intensity of premixed combustion compared to diesel. Also No_x emission increases with increase in blend percentage because vegetable based fuel contains small amount of nitrogen. This contributes towards NO_x production [28]. NO_x emissions were also higher at part loads for biodiesel. This is probably due to higher bulk modulus of bio-diesel resulting in a dynamic injection advance apart from static injection advance provided for optimum efficiency. Excess oxygen (10%) present in bio-diesel would have aggravated the situation [29].

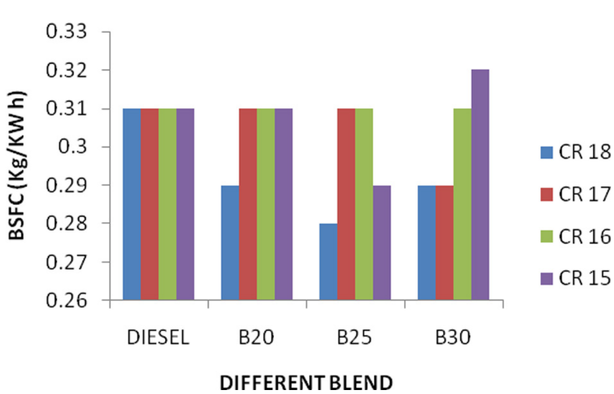

Fig. 4. Variation of BSFC with fuels blends for different compression ratio at Max load.

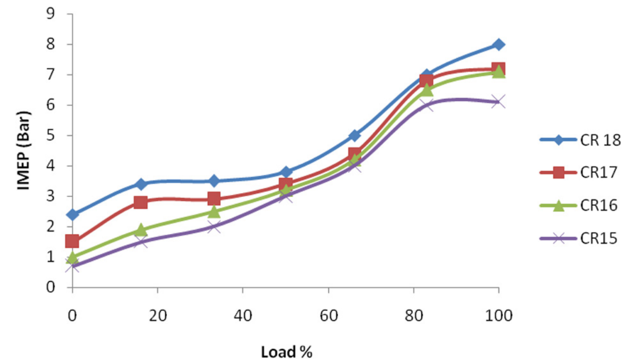

Fig. 5. Variation of IMEP with load for different compression ratio for B25.

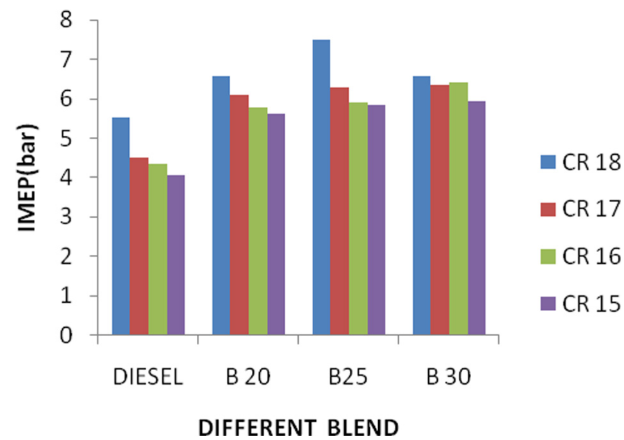

Fig. 6. Variation of IMEP with fuels blends for different compression ratio at Max load.

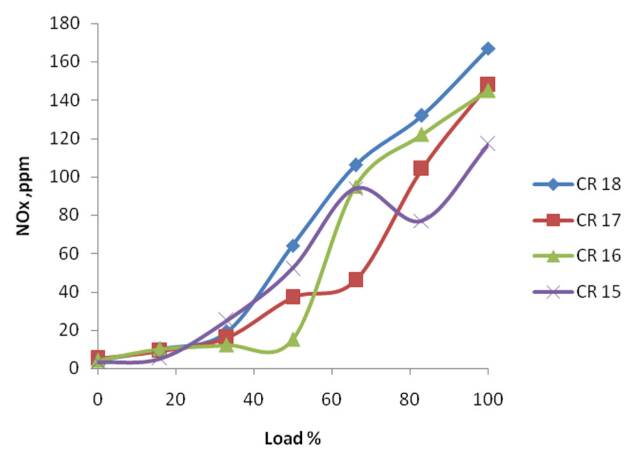

Fig. 7. Variation of NO_X with load for different compression ratio for B25.

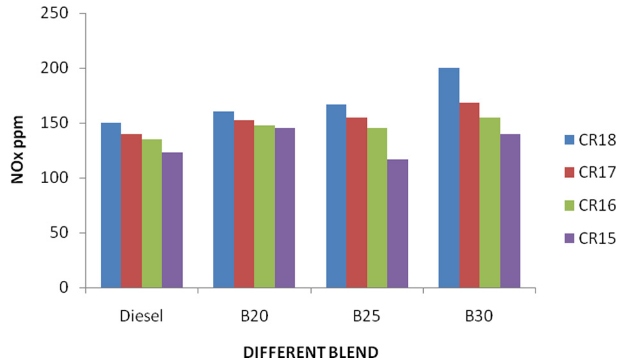

Fig. 8. Variation of NO_X with fuels blends for different compression ratio at Max load.

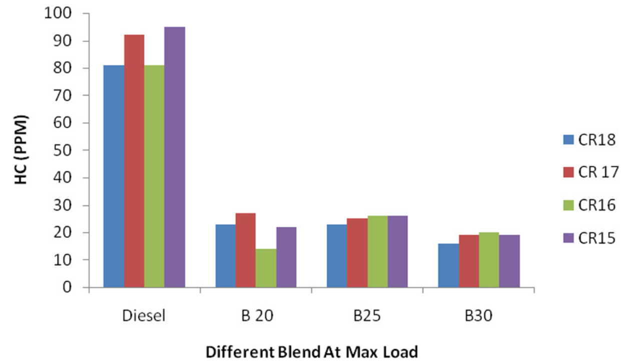

Fig. 10. Variation of HC emission with fuels blends for different compression ratio at Max load.

4.2. Unburnt hydro carbons

Another emission product that is produced by the diesel engine is UHC. It consists of fuel that is completely unburned (or) only partially burned. The amount of UHC depends on the engine operating condition and fuel properties. Fig. 9 shows variation of UHC for B25 of different compression ratios. At the higher compression ratio UHC were low. This may be due to increased temperature and pressure at higher compression ratio and better combustion can be ensured. Fig. 10 shows UBC emission for diesel and bio diesel blend at full load condition. The UHC emissions were lower for biodiesel blend. UHC emission in the exhaust had decreased with increasing amount of biodiesel in the blend. This may be due to the inbuilt oxygen content in its molecular structure this may be responsible for complete combustion and thus reducing the UHC levels. Due to the longer ignition delay, the accumulation of fuel in the combustion chamber may cause higher Hydrocarbon Emission. HC concentration decreases with biodiesel addition and this suggests that adding oxygenate fuels can decrease HC from the locally over rich mixture. Furthermore, oxygen enrichment is also favorable to the oxidation of HC in the expansion and exhaust process [30].

4.3. Carbon monoxide

Carbon monoxide emission is mainly due to the lack of oxygen, poor air entrainment, mixture preparation and incomplete combustion during the combustion process [31–32]. Carbon monoxide in the diesel emission is formed due to intermediate combustion stage. In Diesel engine which operates on the lean side of stoichiometric ratio CO emission are low. Fig. 11 shows the variation of CO emission for B25 at different blend and different compression ratio. The emission at part loads will be lower and its increase for full load which is shown in graph. Fig. 12 shows the CO emission at full load for diesel as well as bio-diesel blends at different compression ratio. For Biodiesel operation CO emission were low compared to the diesel. The emission has decreased with increase in amount of bio diesel. In the blend the additional amount of oxygen in the bio diesel accounts for better combustion inside the cylinder and hence for reduced CO emission. At lower CR, insufficient heat of compression delays ignition and so CO emissions increase. The possible reason for this trend could be that the increased CR actually increases the air temperature inside the cylinder therefore reducing the ignition lag which causes better and more complete burning of the fuel [33].

4.4. Carbon dioxide

The variation of Carbon dioxide emission with different loads is shown in Figs. 13 and 14. More amount of CO_2 is an indication of complete combustion of fuel in the combustion chamber. It also relates to the exhaust gas temperature. CO_2 emission of the blend B25 and diesel is 2.1% and 4.1% respectively for full load operation. It is observed that the amount of CO_2 produced while using karanja-diesel blends is lower than diesel at all loads. This may be due to the late burning of fuel leading to incomplete oxidation of CO. [34]. The accrual of CO_2 in the atmosphere leads to several environmental problems like ozone depletion and global warming. The CO_2 emission from the combustion of bio fuel can be reverted

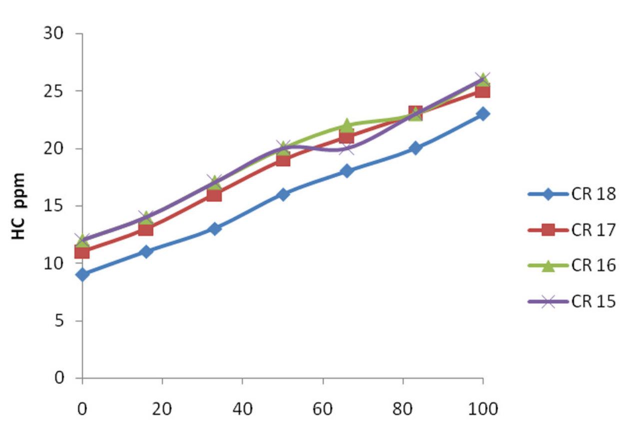

Fig. 9. Variation of HC emission with load for different compression ratio for B25.

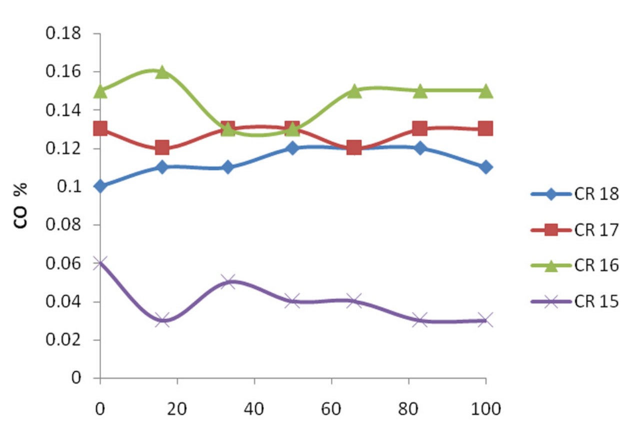

Fig. 11. Variation of CO emission with load for different compression ratio for B25.

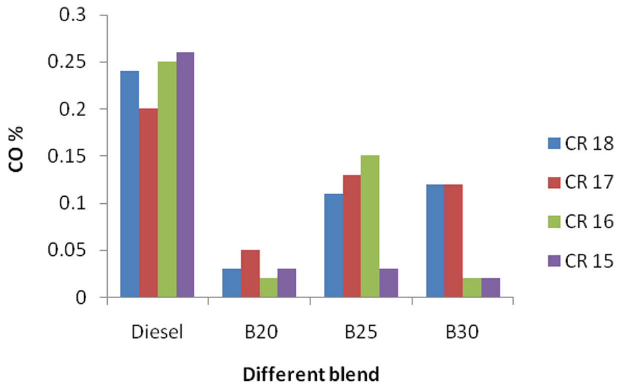

Fig. 12. Variation of CO with fuels blends for different compression ratio at Max load.

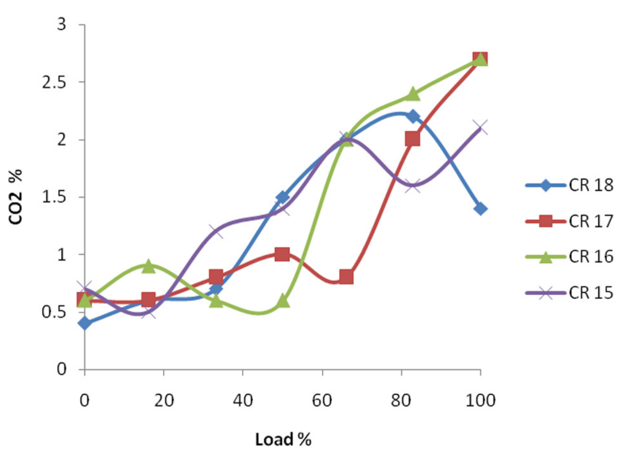

Fig. 13. Variation of CO_2 with load for different compression ratio for B25.

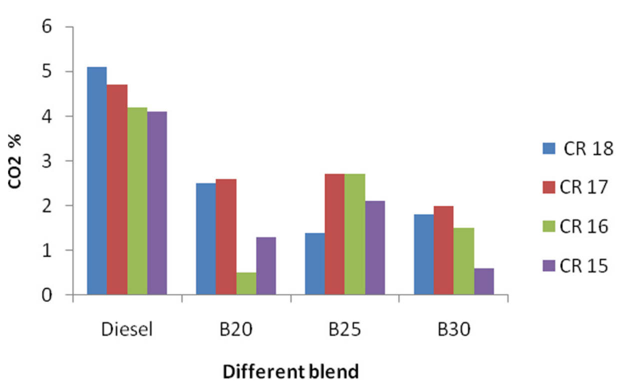

Fig. 14. Variation of CO_2 with fuels blends for different compression ratio at Max load.

by the plant and the carbon dioxide level and is kept constant in the temperature.

5. Combustion characteristics

Combustion characteristic is heavily influenced by the properties of the fuel such as cetane number, heat of combustion, oxygen content and bulk modulus. A marked difference in the combustion characteristic is expected due to the distinct variations in fuel properties between biodiesel and fossil diesel. With higher cetane number, biodiesel is expected to combust earlier from the shorter

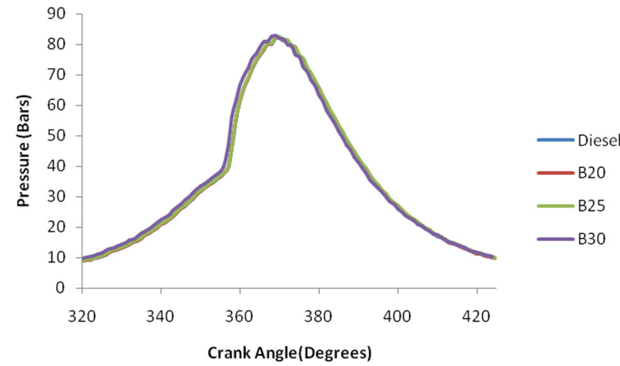

Fig. 15. Variation of combustion Pressure with crank angle for compression ratio 18 at Max load.

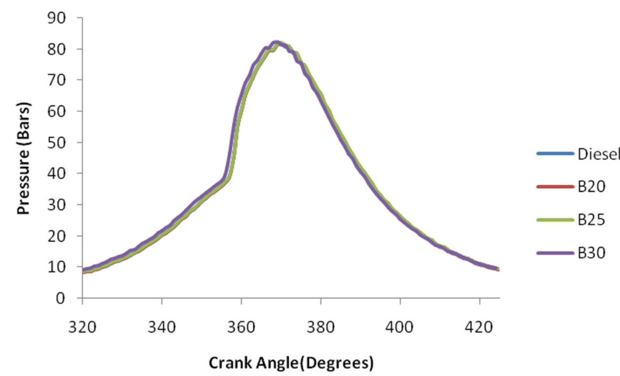

Fig. 16. Variation of combustion Pressure with crank angle for compression ratio 17 at Max load.

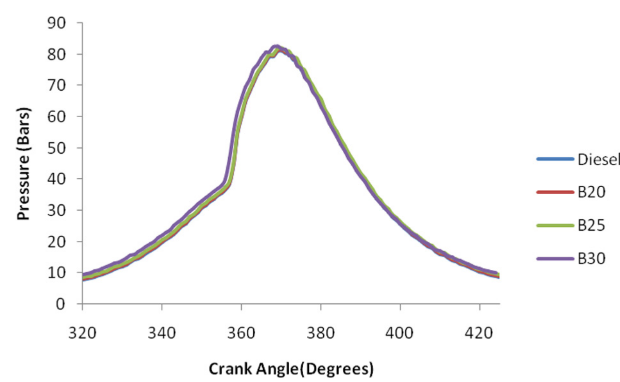

Fig. 17. Variation of combustion Pressure with crank angle for compression ratio 16 at Max load.

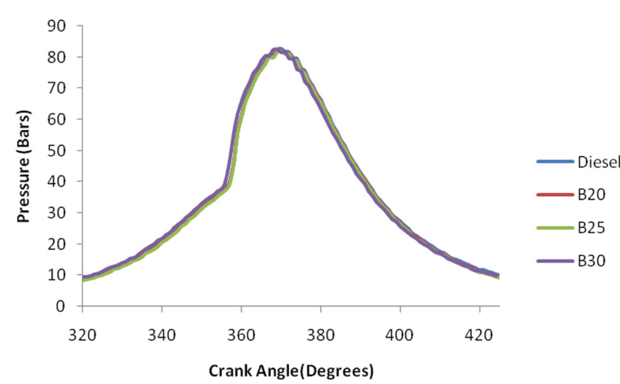

Fig. 18. Variation of combustion Pressure with crank angle for compression ratio 15 at Max load.

Table 5
Experimental results for maximum load.

Fuel	Load kg	CR	BTHE %	SFC kg/k Whr	HC ppm	CO %	CO_2 %	NO_X ppm
Diesel	12	18	29.01	0.29	92	0.24	5.1	151
Diesel	12	17	28.22	0.31	95	0.20	4.7	145
Diesel	12	16	28.03	0.30	92	0.25	4.2	132
Diesel	12	15	28.01	0.30	95	0.26	4.1	115
B20	12	18	29.93	0.29	23	0.03	2.5	160
B20	12	17	30.08	0.28	27	0.05	2.6	152
B20	12	16	28.49	0.30	14	0.02	0.5	148
B20	12	15	27.92	0.31	22	0.03	1.3	145
B25	12	18	30.26	0.28	23	0.11	1.4	167
B25	12	17	29.74	0.29	25	0.13	2.7	155
B25	12	16	29.67	0.29	26	0.15	2.7	145
B25	12	15	29.53	0.29	26	0.03	2.1	117
B30	12	18	29.79	0.29	16	0.12	1.8	200
B30	12	17	27.92	0.31	19	0.12	2	168
B30	12	16	28.27	0.30	20	0.02	1.5	155
B30	12	15	29.59	0.29	19	0.02	0.6	140

Table 6
Optimization criteria and desirability response.

Source	Lower limits	Upper Limits	Weight		Importance	Goal	Desirability
			Upper	Lower			
Compression Ratio	15	18	1	1	5	In range	1
Fuel fraction	0	30	1	1	5	In range	1
BTE	27.92	30.26	1	1	5	Maximize	0.9963
BSFC	0.28	0.31	1	1	5	Minimize	0.9574
CO	0.02	0.26	1	1	5	Minimize	0.994
HC	14	95	1	1	5	Minimize	0.979
NO_x	117	200	1	1	5	Minimize	0.9648
CO_2	0.5	5.1	1	1	5	Minimize	0.9467
Combined							0.978

Table 7
Comparison of actual and predicted values.

S. No.	Value	Compression ratio	Fuel fraction	BTHE %	BSFC kg/kW h	CO %	CO_2	HC ppm	NO_x ppm
1	Actual	18	25	30.26	0.280	0.11	1.4	23	167
2	Predicted	18	25	29.7	0.289	0.10	1.4	22.85	170
3	Error %			1.86	3.2	9	0	0.65	5

ignition delay. This was proven in a comparative study on combustion characteristic of rice bran oil-derived biodiesel blend (B20) against fossil diesel in a naturally aspirated diesel engine [35].

5.1. Impact on cylinder pressure

The variation of combustion pressure with respect to crank angle for different compression ratios and for different blends is shown in Figs. 15–18. It has been observed from the variation of cylinder pressure for various compression ratios 15:1, 16:1, 17:1 and 18:1 that the Karanja oil blends give high combustion pressure compared to that of standard diesel due to longer ignition delay and may be due to the lower cetane number of the blend. The fuel absorbs more amount of heat from the cylinder immediately after injection and resulting in longer ignition delay [36]. It is observed that 78.72 bar, 80.4 bar, 82.31 bar and 84.8 bar for standard diesel and Karanja blends B20, B25, B30. It can been seen From the Fig that the combustion pressure for diesel is higher for lower compression ratios and the combustion Pressure for blends is higher for higher compression ratios. At a compression ratio 18:1, maximum pressure rise of the blend B30 is very different from B20. This is due to the faster and complete combustion of fuel inside the combustion chamber. At lower compression ratios, the maximum combustion pressure for diesel is higher than that of diesel–bio diesel blends.

The maximum rate of increase in pressure is increasing with increase in compression ratio for different blends. Therefore the peak pressure rise for Karanja biodiesel (B30) is 6.02 bar higher than diesel at compression ratio 19:1.

6. Optimization

6.1. Response surface methodology

Response surface methodology was employed in the present study for modeling and analysis of response parameters in order to obtain the characteristics of the engine. The design and analysis of experiment involved the following steps:

The first step was the selection of the parameters that influence the performance and emission characteristics. In this study, the compression ratio, fuel blends and power were considered as the input parameters.

The compression ratio (denoted by ‘Cr’) was varied at four levels from 15 to 18. The fuel blends (denoted by ‘B’) too was varied from 20% to 30%.

The advantage of using Design of Experiments is to evaluate the performance of the engine over the entire range of variation of compression ratio and other parameters with minimum number of experiments. The design matrix was selected based on the 3 level factor design of response surface methodology generated from the software ''Design Expert'' version 8.0.7.1 of stat ease, US, which contained 16 experimental runs as shown in Table 5

As per the run order, the experiments were conducted on the engine and the responses were fed on the responses column.

Finally, the optimal values of the compression ratio and fuel blends were obtained by using the desirability approach of the response surface methodology which is shown in Table 6.

7. Validation of optimized results

In order to validate the optimized result, the experiments were performed thrice at the optimum compression system parameters. For the actual responses, the average of three measured results was calculated. Table 7, summarizes the average of experimental values, predicted values and the percentage of error. The validation results indicated that the model developed were quite accurate as the percentage of error in prediction were in a good agreement.

From this study it is observed that at compression ratio 18 and Blend (B 25) there is an increase in Brake thermal efficiency, decrease in Brake specific fuel consumption and decrease in CO, HC and CO_2 Emissions.

8. Conclusion

The performance and emission characteristics of multifuel variable compression ratio engine fueled with karanja oil biodiesel and diesel blends have been investigated and compared with that of standard diesel

1. Brake thermal efficiency of the blends increases as compression ratio increases. The maximum brake thermal efficiency is 30.46% for B25 at full load at compression ratio 18, which is 5% higher than that of diesel
2. The Hydrocarbon Emissions of various blends have been reduced compared to diesel. The minimum emission is 16 ppm at B20 whereas it is 96 ppm for diesel at CR 16.
3. The CO emission of the blend B25 is lesser than that of the standard diesel.
4. The specific fuel consumption decreased with increase in compression ratio. B25 at CR 18 gave brake specific fuel consumption of 0.28 kg/k Wh as against 0.29 kg/k Wh of diesel
5. The Karanja oil blends gives higher combustion pressure at high compression ratio due to longer ignition delay and lower heat release rate when compared to diesel.

From the above observations, it has been found that the performance of the B25 blend is superior when compared with the conventional diesel at compression ratio 18. The research also proves that Karanja oil methyl ester can be used instead of diesel fuel in a diesel engine at compression ratio 18 with an injection pressure of 200 Bar.

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