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Performance and Emission Analysis of a Variable Compression Multi Fuel Engine Filled With Karanja Bio Diesel-Diesel Blend

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

KEYWORDS:*Blended, Compression ratio, Surface methodology, Thermal efficiency.*

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I. Introduction:

The larger level of increase in the number of automobiles and fast depletion of world petroleum reserves has to be results in a great demand for petroleum products [1]. The alternative energy sources are searching towards theencourage of worlds energy demand in the last two decades. The developing country like India is desirable to produce non-edible oils produced by the bio-diesel which can be grown extensively in the waste land of the country. The use of bio-diesel have to be reduced at the tail pipe emission of Hydrocarbons (HC), carbon monoxide (CO) andparticulate matter (PM). Bio diesel acts as a promising Diesel oil from alternative fuel. Vegetable oil is a very promising alternative to diesel oil and also they are renewable and have similar properties. Many researchers has to be study at the use of vegetable oils in diesel engines. Vegetable oils offer almost the same power output with a slightly low level thermal efficiency when it is used in diesel engines. The Reduction of engine emissions is a major research aspect in engine development with the stringent exhaust gas recirculation and increasing on environmental protection. Biodiesel such as Karanja, sunflower, jatropha, rapessed are some of the popular biodiesel which are currently considered as a substitutes for Diesel. These are renewable, non-toxic, clean burning, biodegradable and environmentally friendly transportation of fuels that can be used in the neat form or blended with Petroleum derived in this diesel engines. Vegetable oil esters particularly karanja appear to be the best in alternative fuel to diesel. Diesel engines have negative effect on environment so they include high amounts of sulphur and aromatics. CO, SOX, NO X and smoke they are produced by fossil fueled diesel engine exhaust emissions [5]. It has been seen in the engine parameters such as injection timing, compression ratio that have considerable effects on the performance and emissions of the diesel engines running on biodiesel blends. There are many innovative technologies that are developed to tackle these problems. Modification is required in the current engine designs [6, 7]. Jindal metal. [8] Studied the effects of the engine design parameters such as the compression ratio, fuel injection pressure and performance parameters such as the fuel consumption, brake thermal efficiency, emissions of CO, HC, NOx, CO 2, and smoke opacity with jatropha methyl ester as fuel. The peak performance is achieved by the engine in point of 250bar injection pressure and compression ratio at 18 at which BSFC improves by 10% and BTE improves by 8.9%. With regarding emission aspects increase in the compression ratio that leads to an increase in emission of Hydro carbon and exhaust temperature whereas smoke and CO emission reduces. Muralidharan et al. [9] investigated the BTE and they found out that blend B40 with waste cooking oil is slightly more than that of standard diesel at greater compression ratios. Brake thermal efficiency of this blends increases by increase in applied load.

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 having a compression ratio at 21 and 4.1% higher than that of diesel engine. [11] Compared the combustion characteristics of single-cylinder four strokes DI variable compression ratio ,when using biodiesel-ethanol blend as fuel it takes the compression ratio of 15:1,17:1,19:1.The performance and emissions have been reduced on a variable compression ratio and the exhaust gas temperature was found to be les . 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, HRRmaxand NO emission with Jatropha biodiesel operation. In 2012 L. Labecki, et al. [13] studied the combustion and emission characteristics of rape- seed plant oil (RSO) and its blends with diesel fuel in a multi- cylinderdirectinjectiondieselengineand foundoutthe NOx emissions for 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 maximum rate of pressure rise was lower than that of diesel. [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 983ppm, 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 to be 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 the 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. 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 nonlinear 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 ThevariousblendsofKaranjaandstandarddieselfuelarepreparedandthefol- lowing 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.

II. Materials And Methods

In the study of variable compression ratio of the engine is run with the karanja biodiesel at different compression ratios are to be evaluate the performance with emissions. The different combinations of compression ratio and loads are compared by the results against the diesel fuel.

2.1. Fuel preparation

The vegetable oils used without further purification were obtained from commercial sources. The samples were converted to methyl esters by alkali catalytic and non-catalytic super critical methanol transerterification methods. An alcohol to form esters and glycerol with the reaction of a fat or oil by the transertification process. Untreated oil is mixed with a mixture a catalyst (Methoxide) and anhydrous methanol in proper proportion. The continuously stirred for around three hours for the mixture is maintained at a temperature little below 64LC (being the B.P. of methanol) .The mixture is allowed to settle down for the time of 25 hour. 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, after completion of stirring. The experiments to ensure the mixture of homogeneity was prepared before the commencing by the fuel. The properties of the fuel blend and diesel have been determine as per the ASTM Standards in an analytical lab. The fuels properties were tested using standards measuring components shown in Table 1and results are shown in Table 2.

2.2 Experimental set-up

In this study of commercial diesel fuel used in India has to be taken for the baseline. The test engine used to the variable compression ratio of the multi fuel engine coupled with eddy current dynamometer. We analyzing the engine performance by engine performance analyze software package ‘‘Engine soft 8.0” for online performance analysis. The Germany made MRU delta 1600L type of exhaust gas analyzer are used to measure the various constituent of exhaust gases such as CO,CO2,NOx and HC. The exhaust gases like HC, CO, and CO2 are measured by the infraredmeasurement and the Electro chemical sensor used in the constituent NOx measurement. The presents of study was carried out to investigate the performance and emission of karanja blend in by volume basis in variable compression ratio and compared with diesel engine. Table 3shows the specification of the experimental setup of the engine. The engine is used to an electric dynamometer to apply the load. To tests were apply various loads, starting from no load to full load condition. The various constituents and fuel flow rate at each load for different bland in exhaust gases such as car-bon monoxide, hydrocarbon, nitrogen oxides and carbon dioxide were analyzed. The system is used to analyze, store and collect 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 by using manual crank start. When applied load at the engine reaches the operating condition. The variable speed is conducted for testing. Experiment was carried out on a tested 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 consumptionwith respect to compression ratio of 15:1, 16:1, 17:1 and

18:1 for different blend are recorded and calculated. In each the combustion parameter, the experiment operating parameter and exhaust emission are stored and analyzed in personal computer and the results are analyzed. The different blend of karanja diesel blend is same procedure is repeated. Table 4 showsthe accuracy of the measurement and 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. The final result is calculated from the primary measurements. The error in the final result is equal to the maxi-mum 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 uncertain-ties 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.

Table 2Properties of Biodiesel-Blends-karanja

Fuel blend	Kinematics Viscosity (cSt) M	Heating Value (kJ/kg) HV	Flash Point (LC) FP	Density (g/cm^3) q	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 3Test 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 Lc	±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	±0.5L CA	±0.2
Load	±1 N	±0.2

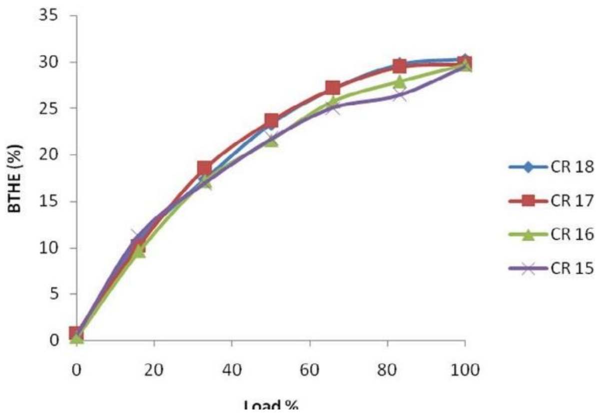


Fig. 1. Variation of BTE with load for different Compression ratio for B25.

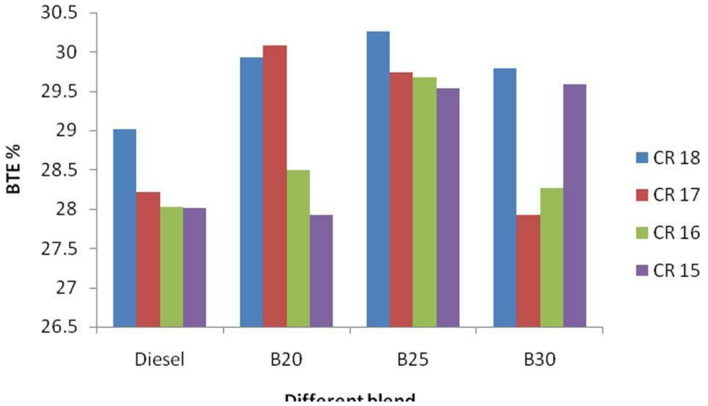


Fig.2.Variation of BTE with fuels blends for different Compression ratio at Max load

3.2. Brake specific fuel consumption

The variation in the Brake specific fuel consumptionfor B25 at all loads for different compression ratios are shown in fig.3. It is the obvious from the figure that BSFC of the engine is 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 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. 2shows the variation of brake thermal efficiency for different compression ratios and for different Kilo watt is lesser than that of lower compression ratio.

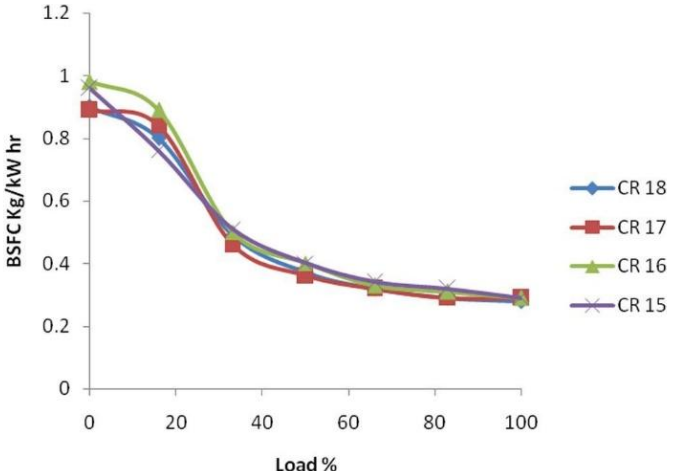


Fig. 3. Variation of BSFC with load for different compression ratio for B25

The BSFC for the blend B20, B25, B30 at higher load are 0.29, 0.28 and 0.29 kg/kWh whereas for diesel it is 0.31 kg/k Wh. Fig. 4shows 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 showsthat the indicated mean effective pressures forblend B25 is higher loads and lower loads than standard diesel. The variation of indicated mean effective pressure with different load for different blend is shown in Fig. 6the blend B25 and diesel closely follows standard at compression ratio of 15 and 16. The indicated mean effective pressure for blend diesel and B25 at full load is7.48 bar and 5.5 bar respectively.

III. Emission Characteristics

The engine operating parameters such as fuel type, atomization ratio, air-fuel equivalence ratio and combustion chamber design affect all emissions emitted by internal combustion engines. Especially, the exhaust are very important, emissions of CO and unburned HC exhaust they are 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 had evaluated in terms of CO, CO2, HC and NO_X at various CR at different loading conditions of the engine.

4.1. Nitrogen oxides

Fig. 7 showsthe variation of NO_xemission for B25 at all loadsand 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. 8shows 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

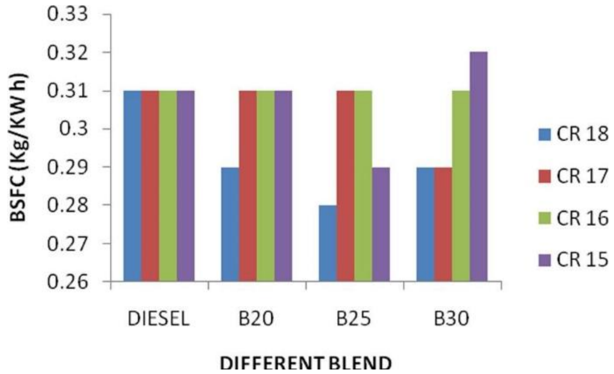


Fig. 4. Variation of BSFC with fuels blends for different compression ratio at Max load

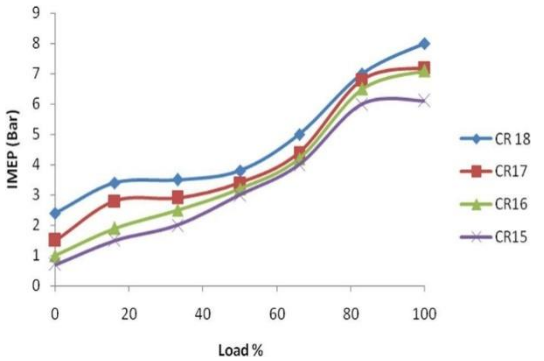
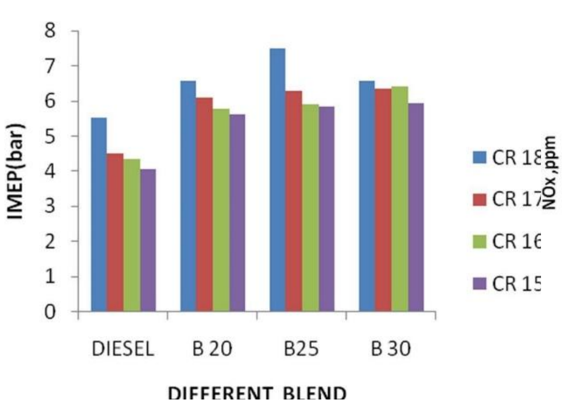


Fig. 5. Variation of IMEP with load for different Compression ratio for B25



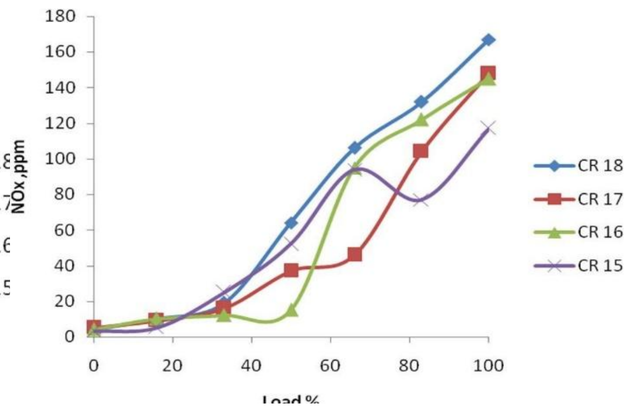


Fig. 6. Variation of IMEP with fuels blends for

Fig. 7. Variation of $NO_X$with load for different

Compression ratio for B25. Different compression ratio at Max load

Of NOx 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 con-tributes towards NO_x production [28]. NO_x emissions were also higher at part loads for biodiesel. the 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 been aggravated the situation [29]

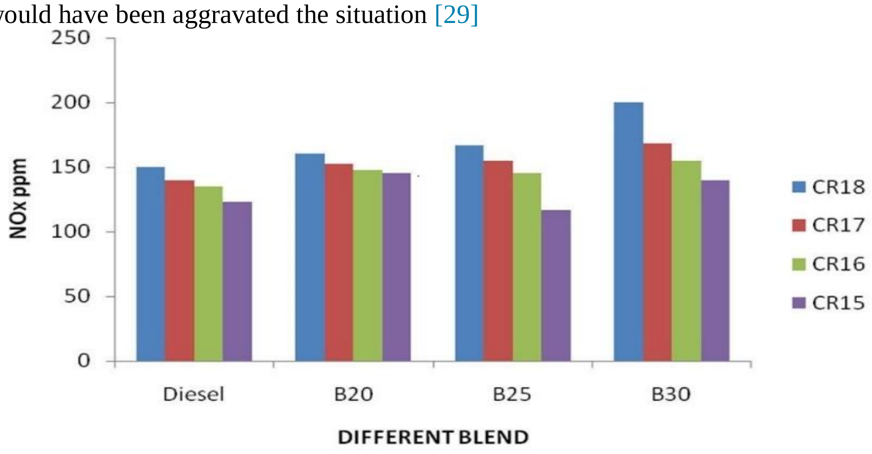


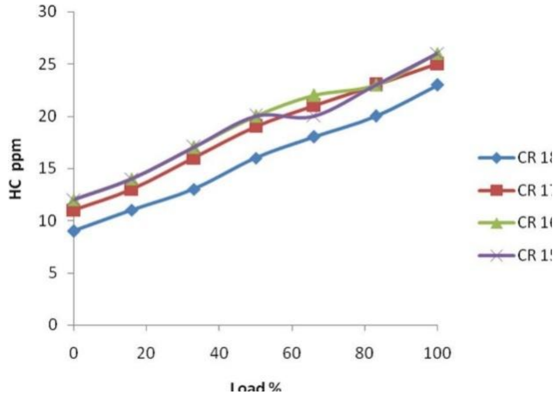
Fig. 8.Variation of $NO_X$with fuels blends for different compression ratio at Max load

4.2. Unburnt hydro carbons

Emission product that is produced by the diesel engine is UHC. It consists of fuel that is only partially (or) completely unburned. The amount of UHC depends on the engine operating fuel properties and condition. Fig. 9shows the variation of UHC for B25 of different compression ratios. At the higher compression ratio UHC were low. This may be due to increased pressure and temperature at higher compression ratio and better combustion ratio can be ensured. Fig. 10shows UBC emission for bio diesel and diesel blend at full load condition. The UHC emissions are lower than the biodiesel blend. UHC emission in exhaust had to decrease with increasing amount of biodiesel in the blend. This may be due to the inbuilt oxygen content in the molecular structure this may be responsible for complete combustion and this reducing the UHC levels. Due to the longer ignition delay, the accumulation of fuel in the combustion chamber may cause higher Hydrocarbon Emission, the accumulation of fuel. HC concentration decreases with biodiesel addition and this suggests that adding oxygenate fuels can be decrease HC from the locally over rich mixture. The oxidation of HC in the expansion and exhaust process, further more oxygen enrichment is also favorable. [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 stoichio-metric ratio CO emission are low. Fig. 11shows 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. 12shows 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].



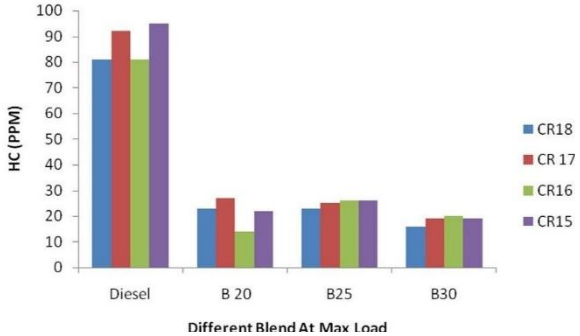
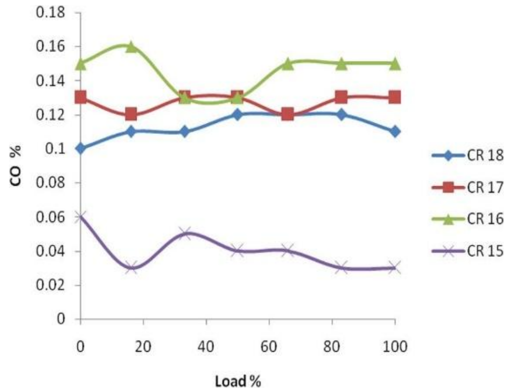


Fig. 9. Variation of HC emission with load for **Fig.** **10.** Variation of HC emission with fuels blends

Different compression ratio for B25. For different compression ratio at Max load

4.4. Carbon dioxide

The different loads of Carbon dioxide emission 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 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 time consumption high to burning of fuel leading to incomplete oxidation of CO. [34]. The CO_2 emission from the combustion of bio fuel can be reverted by the plant and the carbon dioxide level and is kept constant in the temperature.



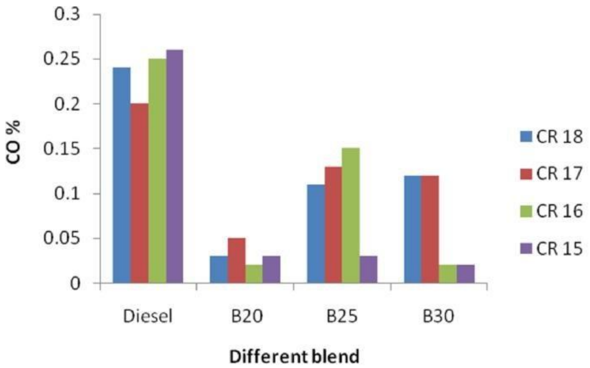
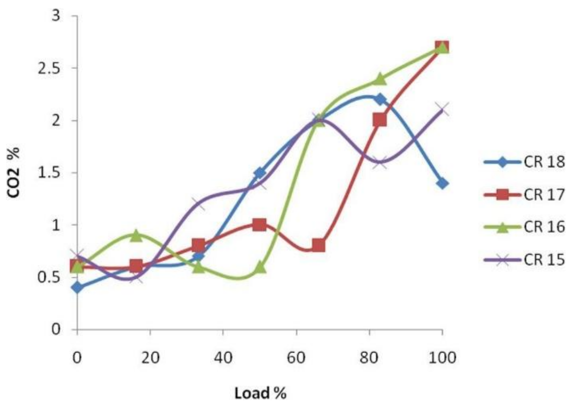


Fig. 11. Variation of CO emission with load for**Fig.** **12.**Variation of CO with fuels blends for different
Different compression ratio for B25 Compression ratio at max load



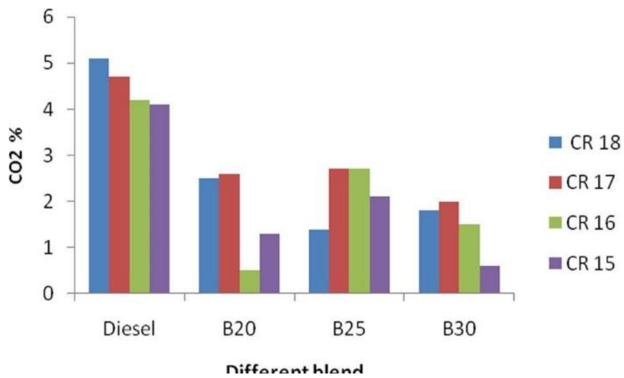
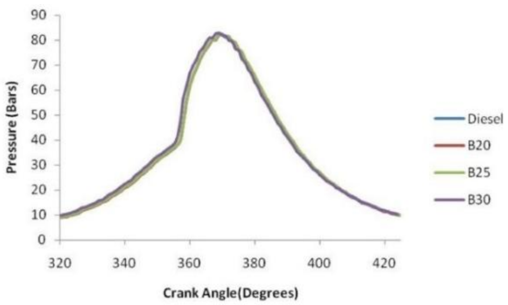


Fig. 13. Variation of $CO_2$with load for different**Fig.** **14.**Variation of $CO_2$with fuels blends for
Compression ratio for B25. Different compression Ratio at Max load

IV. Combustion Characteristics

Combustion characteristic is heavily influenced by the proper-ties 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 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].



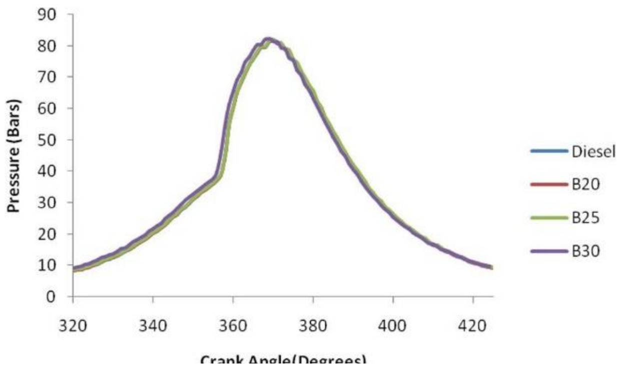
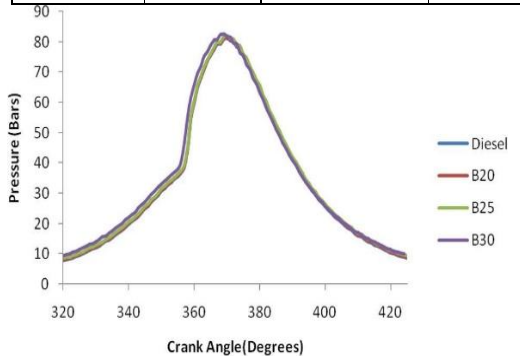


Fig. 15. Variation of combustion Pressure with crank angle **Fig.16**.Variation of combustion pressure with For compression ratio 18at Max load.Crank angle for compression ratio 17 atMax load.

Fuel	Load kg	CR	BTHE%	SFC kg/kW/hr	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



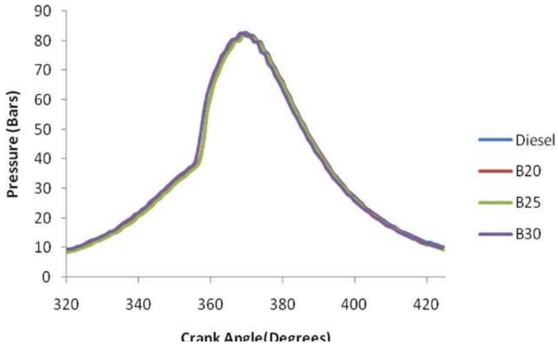


Fig. 17.Variation of combustion Pressure with crank Angle for compression ratio 16at Max load.

Fig.18. Variation of combustion pressure with crank Angle for compression ratio 15 at Max load.

TABLE 5 Experimental Results for Max Load

Table6 optimization criteria desirability

Source	**Lower limits**	**Upper Limits**	**Weight**		**Importance**	**Goal**	**Desirability**
			Upper	**Lower**			
Compression Ratio	15	15	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	13	0.25	1	1	5	Minimize	0.978

Table 7Comparison of actual and predicted values.

S. No	**Value**	**Compression ratio**	**Fuel fraction**	**BTHE%**	**BSFC kg/kWh**	**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 %**	18	25	1.86	3.2	9	0	0.65	5

5.1. Impact on cylinder pressure

The variation of combustion pressure with respect to crank angle and for different blends and for different compression ratios is shown in Figs. 15–18. It have been observed from the variation of compression ratio and for various cylinder pressure is 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 time 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 time 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 and 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, maxi-mum 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 compression ratio with increase in pressure for different blends. Therefore the peak pressure rise for Karanja biodiesel (B30) is 6.02 bar higher than diesel at compression ratio 19:1.

V. Optimization

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

1. The first step is the selection of the performance and emission characteristics the parameters that influence. In this study, the fuel blends and power were considered as the input parameters, the compression ratio.
2. The fuel blends (denoted by 'B') too was varied from 20% to 30%. The compression ratio (denoted by 'Cr') was varied at four levels from 15 to 18.
3. 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, 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. which contained 16 experimental runs as shown in Table 5
4. As per the run order, the experiments has conducted on the engine and the responses were fed on the responses column.
5. 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.

VI. Validation Of Optimized Results

The experiments were performed thrice at the optimum compression system parameters In order to validate the optimized result. The average of three measured results was calculated, for the actual responses. Table 7, summarizes the average of predicted values, the experimental 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, CO_2 and HC Emissions.

VII. Conclusion

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

1. Compression ratio increases as brake thermal efficiency of the blends increases. When full load at compression ratio 18, which is 5% higher than that of diesel the maximum brake thermal efficiency is 30.46% for B25.
2. The Hydrocarbon Emissions of various blends has been reduced compared to diesel. It is ppm for diesel at CR 16,the minimum emission is 16 ppm at B20
3. The CO emission of the blend B25 is less than of the standard diesel.
4. B25 at CR 18 gave brake specific fuel consumption of 0.28 kg/kWh as against 0.29 kg/kWh of diesel. The specific increase in compression ratio with fuel consumption decreased.
5. The Karanja oil blends gives high compression ratio at higher combustion pressure due to longer ignition delay and lower heat release rate when compared to diesel.
6. From the above observations, it was 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.
7.

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