



THE PERFORMANCE AND EMISSIONS OF A VARIABLE COMPRESSION RATIO DIESEL ENGINE FUELLED WITH BIO-DIESEL FROM COTTON SEED OIL

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ABSTRACT

A methyl ester of cottonseed oil was prepared and blended with diesel in four different compositions varying from 5% to 20% in steps of 5%. Tests were conducted in a single cylinder variable compression ratio diesel engine at a constant speed of 1500 rpm. Highest brake thermal efficiency and lowest specific fuel consumption were observed for 5% biodiesel blend for compression ratio of 15 and 17 and 20% biodiesel blend for compression ratio of 19. The 20% biodiesel blend at a compression ratio of 17 had maximum nitric oxide emission as 205 ppm, while it was 155 ppm for diesel. Substantial reduction in Carbon monoxide emissions and smoke in the full range of compression ratio and loads was observed. Improved heat release characteristics were observed for the prepared biodiesels. The results reveal that the biodiesels can be used safely without any modification to the engine.

Keywords: cotton seed oil, biodiesel, emission, combustion, compression ratio.

INTRODUCTION

India is the second largest producer of cotton seed in the world next to china with the potential of 4.6 million tonnes of oil seeds per annum. With the rapid development of rural agricultural production and rapid growth of local industry in India, the discrepancy between demand and supply of energy has become an increasingly acute problem. Due to seasonality of farm work, a temporary shortage of fuel will bring about unexpected and irreparable loss to peasants. The limited (and fast diminishing) resources of fossil fuels, increasing prices of crude oil, and environmental concerns have been the diverse reasons for exploring the use of vegetable oils as alternative to diesel oil [1-4]. Vegetable oils offer almost the same output with slightly lower thermal efficiency when used in diesel engines [5-7]. Reduction of engine emissions is a major research aspect in engine development with the increasing concern over environmental protection and the stringent exhaust gas regulation [8-13]. The use of neat vegetable oils poses some problems when subjected to prolonged usage in CI engines. These problems are attributed to high viscosity, low volatility and polyunsaturated character of vegetable oils [14-17]. Some of the common problems posed by using vegetable oils in diesel engines are coking and trumpet formation on the injectors, carbon deposits, oil ring sticking and thickening and gelling of lubricating oil as a result of contamination by the vegetable oils [18-20]. Different methods such as preheating, blending and transesterification [21-25] are being used to reduce the viscosity and to produce bio-diesel, suitable for engine applications. In the present investigation, bio-diesel is prepared from cotton seed oil. The fuel properties of the

synthesized bio-diesel were determined and their performance, emission and combustion characteristics were studied on a four-stroke, single cylinder, variable compression ratio direct-injection diesel engine to ensure their suitability as CI engine fuel.

EXPERIMENTAL SYSTEM DEVELOPMENT FOR BIO-DIESEL FORMULATION

METHODOLOGY

The raw cotton seed oil was extracted by mechanical expeller in which small traces of organic matter, water and other impurities were present. Transesterification is a most common and well established chemical reaction in which alcohol reacts with triglycerides of fatty acids (vegetable oil) in presence of catalyst to form glycerol and esters [1-3, 10, 17, 22-23]. The reaction is shown in Figure-1. Experiments were conducted in a laboratory setup consisting of heating mantle, reaction flask (made of glass), separating funnel and mechanical stirrer. A round bottom flask of 2 litre was used as laboratory scale reactor for the present analysis. It consisted of three necks. One for stirrer, the others for condenser and inlet of reactants, as well as for placing the thermocouple to observe the reaction temperature. The flask has a stopcock at the bottom for collection of the final product. The progress of the reaction was observed by measuring the acid value. In the course of the test, it was observed that the appropriate quality of bio-diesel could be produced from cotton seed oil in both acid catalyst esterification and alkaline catalyst esterification.

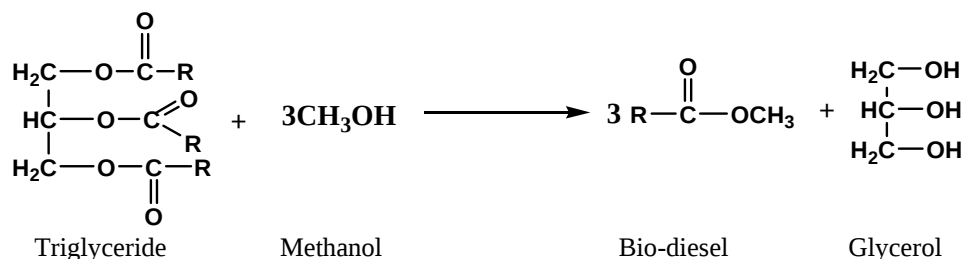


Figure-1. Transesterification reaction.

Esterification procedure

The cotton seed oil in the flask was heated on a heating mantle with a mechanical stirrer arrangement. The mixture (methanol and cotton seed oil) was continuously stirred in the air closed reaction flask for 2 h at 65°C with a stirring speed of 450 rpm. The temperature in the apparatus should be maintained just above the boiling point of alcohol i.e., 65°C to accomplish the reaction. Alcohol in vegetable oils affects the conversion efficiency of the process. For the stoichiometric transesterification, 3 mols of alcohol are required for each mole of the oil. However, in practice, the molar ratio should be higher than this theoretical ratio in order to derive the reaction towards early completion [25-27].

Sulphuric acid is used as catalyst in the acid-catalyst pretreatment. Experimentally it is optimized that 1% by volume of the sulphuric acid and a molar ratio of

6:1 gives the maximum conversion efficiency. The products of the first stage are used as input for alkaline process. A molar ratio of 9:1 and 1.5% by weight of NaOH is found to give the maximum ester yield. With the completion of the reaction, the products were allowed to separate in to two layers. The lower layers contained impurities and glycerol. The top ester layer is separated and purified by using distilled water (10% by volume). Hot distilled water is sprayed over the ester, stirred gently and allowed to settle in the separating funnel. After washing, the final product was heated up to 70°C for 15 min under vacuum condition resulting in a clear amber-light yellow liquid with a viscosity comparable to diesel and then stored for further use. The summary of esterification process, settling and washing is illustrated in Figure-2.



Figure-2. Summary of process flow diagrams for esterification process and settling after washing.

Characterization of cotton seed oil

The source of cottonseed oil is cottonseed, which is a crop by- product. It is basically a triglyceride ester with a number of branched chains of 8-18 carbon atoms. It contains 85.3% fatty acids. Fatty acid composition of the oil is essential to determine the quantity of reactants and the catalyst. FFA (Free Fatty Acids) can be determined from the acid value. It has been stated that the acid value

of the vegetable oil should be less than one for a base catalyzed transesterification process [15, 22] from earlier studies.

The significant properties of cotton seed oil are found during the present investigation. The comparisons of properties of cotton seed oil with that of other oils are given in Table-1.

**Table-1.** Fatty acid composition of cotton seed oil in comparison with other oils.

Property	Test method (ASTM)	Cotton seed oil (<i>gossypium herbaceum</i>)	Rubber seed oil ^a (<i>Ficus elastica</i>)	Polanga ^a (<i>calo phyllum ino phyllum</i>)	Karanja ^a (<i>pongamia pinnata</i>)	Jatropha ^a (<i>jatropha curcas</i>)
Fatty acid composition (%)						
(i) Palmitic acid C16:0		28.7	10.2	12.01	11.65	16
(ii) Stearic acid C18:0		0.89	8.7	12.95	7.50	6.5
(iii) Oleic acid C18:1		13.27	24.6	34.09	51.59	43.5
(iv) Linoleic acid 18:2		57.51	39.6	38.26	16.46	34.4
(v) Linoleic acid 18:3		0	16.3	0.3	2.65	0.80
Specific gravity	D4052	0.912	0.91	0.896	0.913	0.920
Viscosity (mm ² /s) at 40°C	D445	50	66.2	71.98	27.84	18.2
Flash point(°C)	D93	210	198	221	205	174
Calorific value(MJ/Kg)	D240	39.6	37.5	39.25	34	38.5
Acid value (mg KOH/g of oil)		0.11	34	44	5.06	3.8

^a Source: Ref [26].

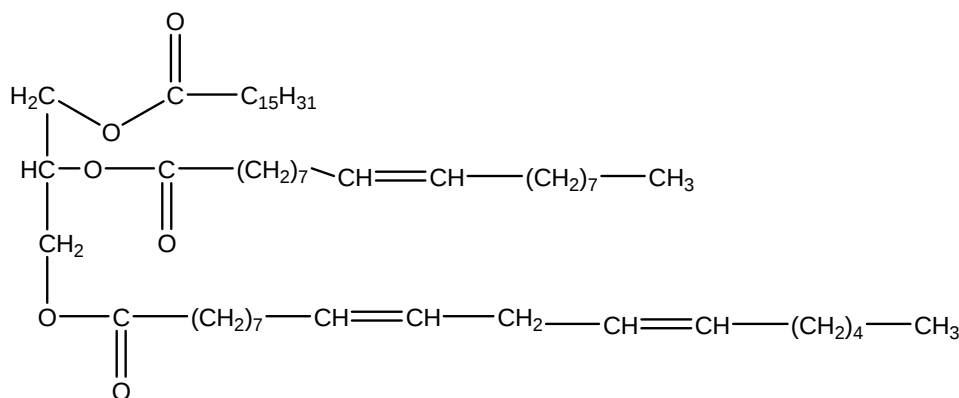
Some of these properties are taken from existing literature [26], while most of the others determined in the laboratory. The important properties of cotton seed oil (CSO), cotton seed oil methyl ester (COME) are found and compared with that of diesel (Table-2).

Table-2. Fuel specifications of diesel, cotton seed oil and cotton seed methyl ester.

Properties	Test method (ASTM)	Diesel	Cotton seed oil	Cottonseed methyl ester
Specific gravity	D4052	0.835	0.912	0.874
Kinematic viscosity (mm ² /s) at 40°C	D445	4	50	4.2
Cetane Index	D4737	47	48.1	51.2
Cloud point (°C)	D2500	4	0	4
Pour point (°C)	D189	-6	-5	0
Flash point (°C)	D93	44	220	142
Fire point (°C)		85	253	176
Calorific Value (MJ/Kg)	D240	42.5	39.6	40.6
Sulphur content (mg/Kg)	D5453	496	3	51
Copper strip corrosion (3h, 100°C)	D130	1A	1A	1A
Elemental analysis (% W/W)				
C		83	81.42	82.11
H		13	11.28	11.83
O		0.19	7.22	5.98
N		1.76	0.028	0.042
S		0.25	0.053	0.040

The result indicates that, transesterification has improved the important fuel properties like specific gravity; viscosity; flash point; and acid value. The viscosity substantially got reduced from a value of 50 to 4.2 mm²/s (approximately one 11th of initial value). The calorific value of methyl ester is lower than that of diesel because of its oxygen content. The flash point temperature of CSO and COME is higher than the pure diesel fuel. The high flash point temperature of COME is a beneficial safety feature, as the fuel can be safely stored and

transported at the room temperature. The cetane number (CN) of COME prohibits its direct use as the alternative fuel in diesel engines, but it could be used in blends with the pure diesel fuel, because most of the determined properties of the a COME-diesel blended fuels were very close to the diesel fuel. The final structure along with chemical formula, exact mass, molecular weight and elemental analysis is given by CHEMOFFICE software as shown in Figure-3.

Chemical Formula: $C_{55}H_{100}O_6$

Exact Mass: 856.75

Molecular Weight: 857.38

m/z: 856.75 (100.0%), 857.76 (60.9%), 858.76 (19.4%), 859.76 (4.3%)

Elemental Analysis: C, 77.05; H, 11.76; O, 11.20

Figure- 3. Schematic diagram showing molecular structure of raw cotton seed oil.**Experimental setup and measurements**

The specifications of the engine used are given in Table-3 and the experimental set-up is shown in Figure-4. The engine was directly coupled to an eddy current

dynamometer using flexible coupling and a stub shaft assembly; the output of the eddy current dynamometer was fixed to a strain gauge load cell for measuring load applied to the engine.

Table-3. Engine specifications.

S. No.	Parameters	Specifications
	Engine	
1	Nozzle opening pressure	200 bar
2	No. of Cylinders	Single cylinder
3	No. of Strokes	Four-stroke
4	Rated Power	3.7 kW at 1500 RPM
5	Cylinder Bore	80 mm
6	Stroke Length	110 mm
7	Compression Ratio	5:1 to 20:1
	Dynamometer	
8	Type	Eddy current
9	Maximum	5.6 kW at 1500-3000 RPM
10	Load Measurement	Strain Gauge

A KISTLER piezoelectric transducer in conjunction with a KISTLER charge amplifier and data acquisition system was used to measure cylinder pressure. Before mounting on the cylinder head, the transducer charge amplifier combination was statically calibrated using a dead weight pressure tester. An optical encoder using photo emitter and detector was used to detect TDC. A non-dispersive infrared analyser (NDIR), AVL DIGAS 444 was used for the measurement of carbon monoxide (CO), nitric oxide (NO), unburnt hydrocarbon (HC). CO

was measured as percentage volume and NO, HC was measured as n-hexane equivalent, ppm. Smoke was measured as percentage opacity using an AVL 437 opacimeter. Fuel flow rate was measured by means of mass flow meter. Solenoid valve with glass burette are provided for diesel and bio-diesel fuel measurement separately. A hot wire mass airflow sensor determines the mass of air flowing into the engine's air intake system. Temperatures were measured using K-type thermocouples.

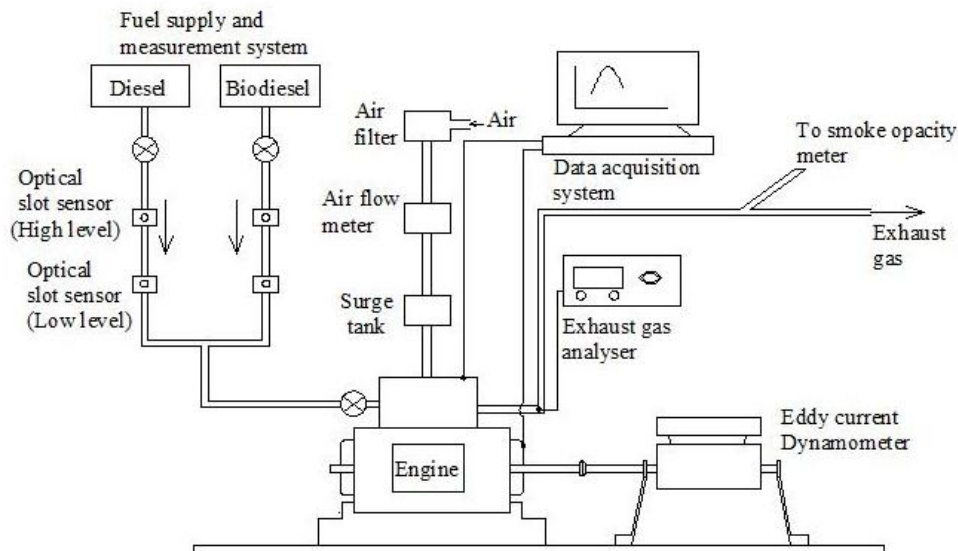


Figure-4. Schematic diagram of experimental setup.

All the experiments were conducted at a rated speed of 1500 rpm. No adjustment was made at the fuel injection timing that is at 23° BTDC (static) was used for diesel and bio-diesel respectively. The cooling temperature was maintained at 70°C during all the experiments. The experiments were conducted by using diesel, B5 (5% COME+95% diesel), B10 (10% COME+90% diesel), B15 (15% COME+85% diesel), and B20 (20% COME+80% diesel), at different load conditions on the engine from 0% to 100% in appropriate steps of 25% at different compression ratios (CR) of 15:1, 17:1 and 19:1. The experimental procedures started with a preliminary investigation of the engine running on the pure diesel fuel in order to determine the engine's operating characteristics and exhaust emission levels, constituting the 'baseline' that is compared with the corresponding cases when the blended fuels were used. For every fuel change, the fuel lines were cleaned and the engine was left to operate for 30 min to stabilize at its new condition. In each experiment, engine parameters related to thermal performance of the engine such as fuel consumption and applied load are measured.

RESULTS AND DISCUSSIONS

Performance parameters

Graphs of the brake specific fuel consumption (bsfc) as a function of load obtained during engine operation on COME-diesel (bio-diesel) blends and Diesel fuel at compression ratios of 15:1, 17: and 19:1 have been superimposed as shown in Figure-5. For all fuels tested, bsfc decreased with increase in load at all compression ratios. One possible explanation for this reduction could be due to higher percentage of increase in brake power with load compared to fuel consumption. At the compression

ratio 15: 1, the bsfc of 0.305 kg/kWh was obtained for the Diesel fuel. In spite of the different calorific values, the fuel blends B5 and B20 maintain about the same bsfc as that of Diesel fuel, whereas the B10 and B15 blends shows a bsfc of increase by 9.5% and 6.2%, respectively, at equivalent loads. The lower bsfc can be related, reasonably, to the higher, 1.45% and 2.50%, amounts of oxygen present in the considered blends. Fuel based oxygen, because of its indigenous property, accelerates reactions from within the extremely fuel rich spray patterns themselves, leading to more complete combustion at low compression ratio.

At a higher compression ratio of 17:1, the bsfc of the fully loaded engine for Diesel fuel is 0.295 kg/kWh, whereas that of B15 and B20 shows a bsfc increase by 5% and 6.1%, respectively. The bsfc of blend B5, which differs itself as having the lowest, 0.925% amount of oxygen, appeared to be nearly the same as that of blend B10, both of them having higher bsfc by 3.7% relative to Diesel fuel. This result indicates a diminishing role of fuel oxygen at high compression ratio and, hence, high air-fuel mixture turbulence intensity, allowing residual air borne oxygen to be available at the final combustion stages. This point of view is further supported by data obtained for the engine runs under full load at the compression ratio of 19:1, the fuel blend B20 maintain about the same bsfc as that of Diesel fuel, 0.291kg/kWh, whereas blends B5 and B15 maintain their bsfc higher by 7.6% and blend B15, suggests the bsfc is higher by 5.1%. The reason for the above trend is more likely due to lower calorific values of bio-diesel blends compared to those of neat diesel [9]. So in order to produce equivalent power to that of diesel more bio-diesel blended fuel is required. The reason for increase in bsfc for all loads was due to calorific value of bio-diesel blends being lower than pure diesel.

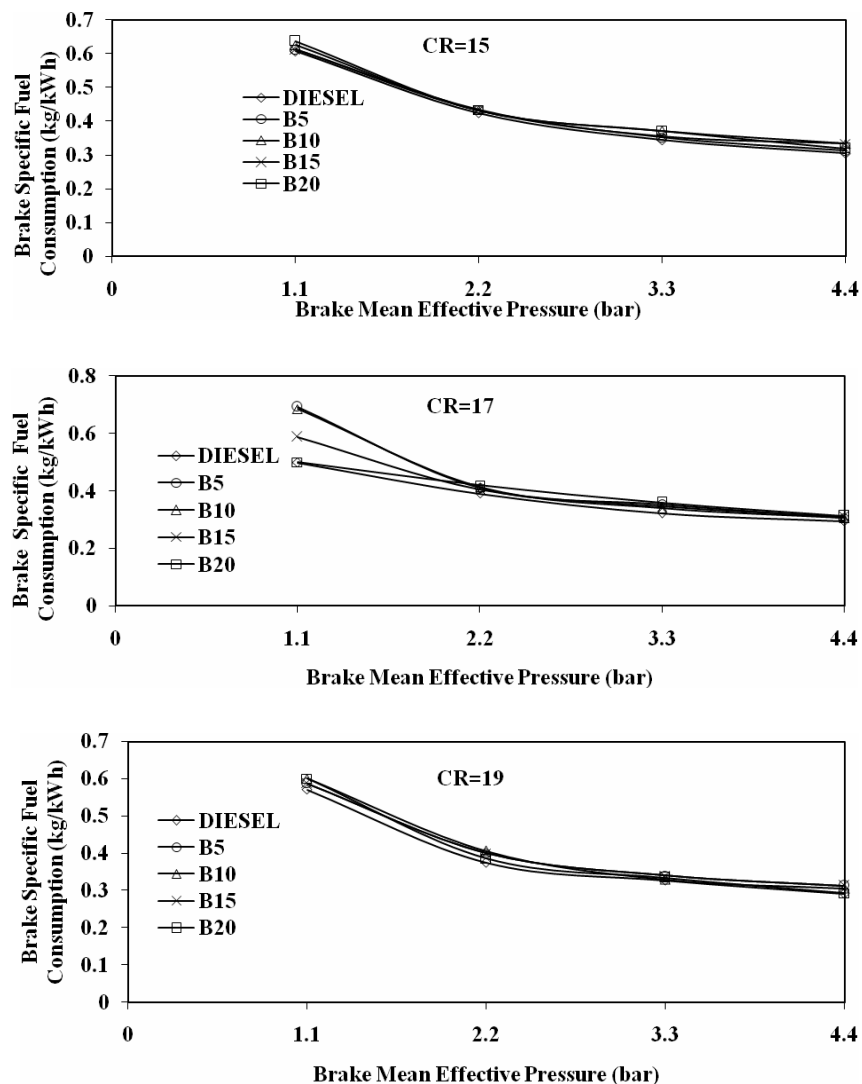


Figure-5. Brake specific fuel consumption (bsfc) as a function of engine load (bmep) at various compression ratios.

The variation of brake thermal efficiency with load for different fuels and various compression ratios are represented in Figure-6. In all cases, it increases with an increase in load. This can be attributed to reduction in heat loss and increase in power with increase in load. The maximum brake thermal efficiencies obtained is about 27.37% for B5 at 15:1 CR (26.65% diesel), 27.98% for B5 and B10 at 17:1 CR (26.65% diesel), and 29.28% for B20 at 19:1 CR (27.92% diesel), which is quite higher compared to that of diesel. The mixing of bio-diesel in

diesel oil yields good thermal efficiency curves. Initially the thermal efficiency of the engine is improved by increasing concentration of the bio-diesel in the blend. The possible reason for this is the additional lubricity provided by bio-diesel. The molecules of bio-diesel (i.e. methyl ester of the oil) contain some amount of oxygen, which takes part in the combustion process. It is noticed that after a certain limit with respect to diesel ester blend, the thermal efficiency trend is reversed and it starts decreasing as a function of the concentration of blend.

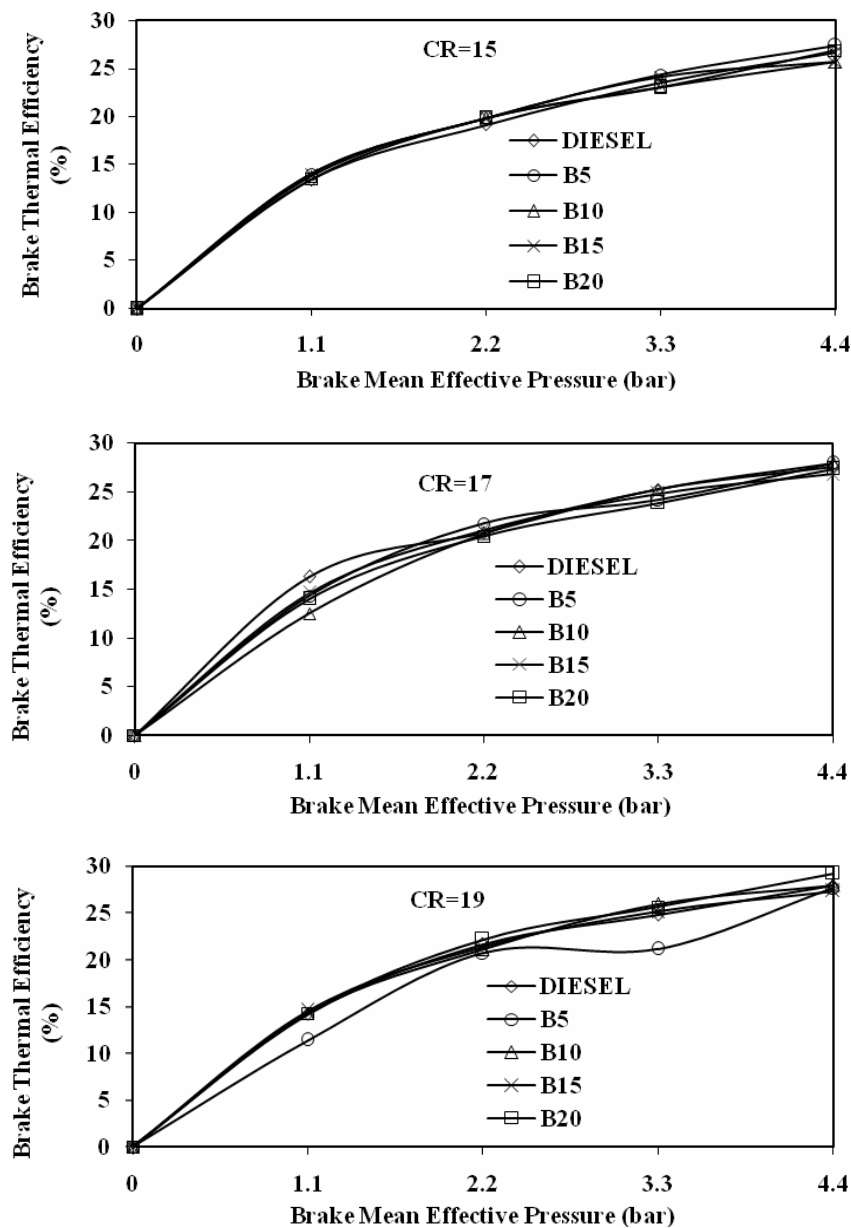


Figure-6. Variation of brake thermal efficiency as a function of engine load (bmep) at various compression ratios.

Test results indicate that when the mass percent of fuel oxygen exceeds 1.45%, the oxygen loses its positive influence on the fuel energy conversion efficiency in this particular engine. New evidence related to possible causes of such behaviour must await further investigations, but the worsening of the process can be linked to the blend's structure, its physical properties, changes in injection timing advance, fuel evaporation and the start of combustion [16] and [19]. Because of the different structures of COME, its evaporation proceeds at much high temperatures, ranging between 325 and 366 °C than those of Diesel fuel (210–360 °C). The higher flash point, poor volatility and flammability may have an influence on the auto-ignition and combustion processes of biofuels. This is especially important at high compression

ratios where the extent of evaporation and the combustion processes actually play a key role.

Emission parameters

Figure-7 Shows the plots of carbon monoxide (CO) emissions of COME blends and diesel fuel operation at different load conditions and compression ratios. The plots showed reduced CO emissions at higher loads when running on bio-diesel blends at increasing compression ratios from 15: 1 to 19:1. This is typical with all internal combustion engines since air/fuel (A/F) ratio decreases with increased load. Also the plots showed increase in CO emissions as the load was increased up to 50% when running on bio-diesel blends and then decreased at the bmep of 3.3 bar and 4.4 bar. The diesel fuel operation



produced the highest CO emissions at high load levels and when the engine load reaches a certain bmep (= 4.4 bar), the CO emissions start to increase more rapidly for both diesel and bio-diesel blends. At high compression ratio of 19:1, the highest CO emission of 0.69% by volume was measured for Diesel fuel, and the lowest of 0.13% was

obtained for blend B10. The CO emissions for blends B5, B15, and B20 were 0.27%, 0.18% and 0.37% by volume, respectively. Reduced CO emissions were maintained, probably, owing to the oxygen inherently present in biofuels.

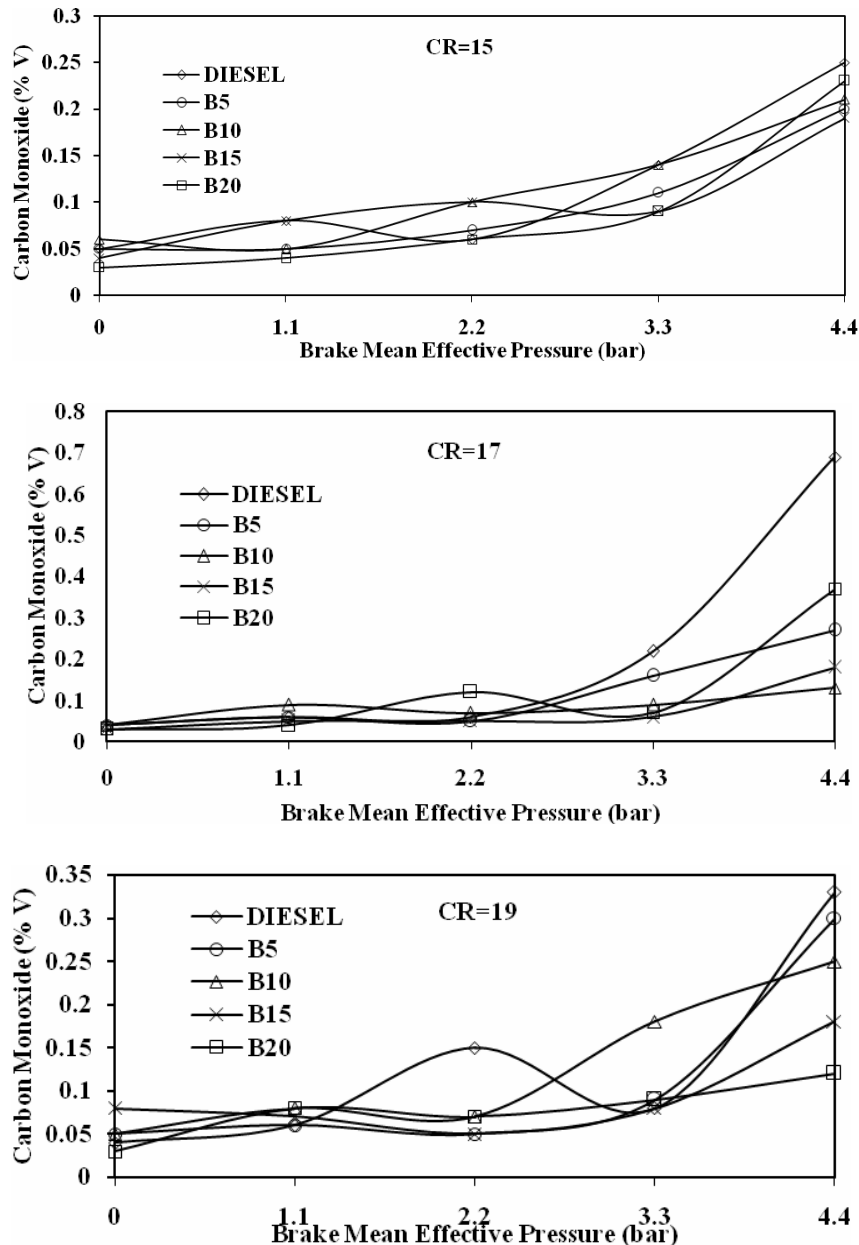


Figure-7. Carbon monoxide emissions as a function of engine load (bmep) at various compression ratios.

As seen in Figure-8, the emission of unburnt hydrocarbons (HC) for all fuels and compression ratios is small, 15-88 ppm, increasing slightly with load and proportion of fuel injected. It is quite difficult to determine any reliable dependencies; however, the HC emissions for biofuels proceed in the graphs at a bit lower levels than that of Diesel fuel. The COME operation showed very low (about a factor of 2) HC emissions throughout the load

range. The plots indicate that HC emissions peaked at a particular A/F ratio and then dropped with further increase in load. The plots showed increase in HC emissions as the quantity of diesel fuel in the blend increases. A very significant difference was noted at low and high load conditions at the compression ratio 19:1. The increase in diesel/COME blend showed highest HC concentration in the exhaust when compared with the results of increase in



COME/diesel blend. However at light load, the COME diesel blends did not show any marked difference in HC emissions irrespective of compression ratios. For minimum load the reduction was practically unaffected by

the addition of bio-diesel in the diesel fuel at all Compression ratios. The most beneficial reduction appeared at intermediate loads.

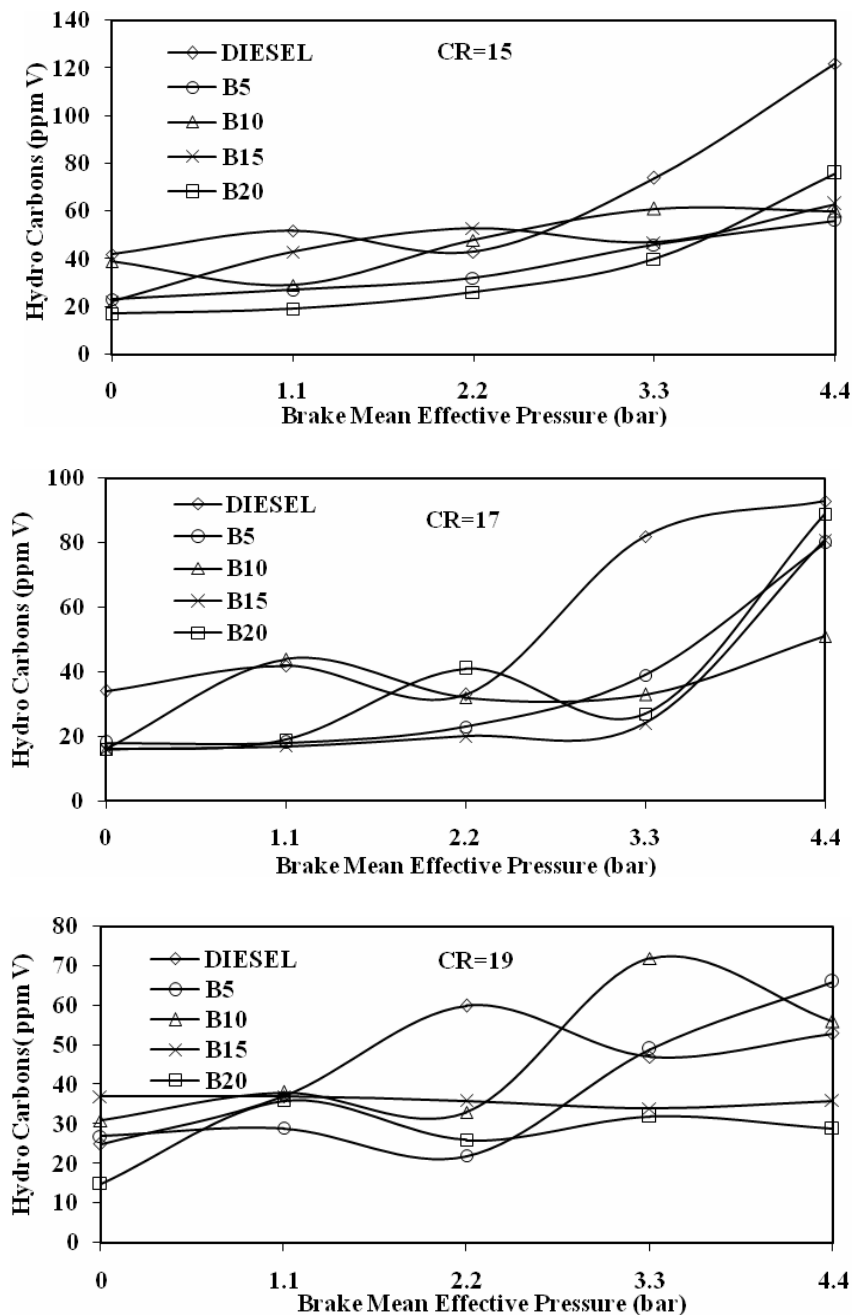


Figure-8. Hydro carbon emissions as a function of engine load (bmep) at various compression ratios.

As seen in Figure-9, the emission of nitric oxide (NO) increases gradually with load, reaching the maximum values of 10-205 ppm for COME-diesel blend and 16-153 ppm for diesel fuel at high loads. When operating at the low CR of 15:1, the minimum NO emissions at adequate loads were maintained by the B20 blend. This blend suggests NO emissions lower by 6.12%

to 37.5% relative to diesel fuel. Biofuels with higher oxygen contents, 1.45% to 4.075%, and neat bio-diesel produce higher or a bit lower NO emissions, depending on the load. The reason for the decrease in NO was that the cetane number of the bio-diesels was higher than that for diesel fuel and this is associated with lower NO emissions [8]. Increasing cetane number reduces the size of the



premixed combustion by reducing the ignition delay. This results in lower NO formation rates since the combustion pressure rises more slowly, giving more time for cooling through heat transfer and dilution and leading to localized gas temperatures [9]. By reducing aromatics the flame temperature will drop, leading to a lower NO production rate. As a result, the aromatics have high carbon-hydrogen ratios and thus fuels with lower aromatics will lead to a smaller amount of CO₂ and larger amount of H₂O being formed compared to high aromatic fuels. Since H₂O has a

lower tendency to dissociate at high temperatures (compared to CO₂), this will lead to low aromatic fuels having lower concentrations of O radical and O₂ (from radical-radical recombination) which will further reduce the kinetic production of NO. Furthermore, if the final yield of NO is determined by the high-temperature equilibrium between oxygen, nitrogen versus NO, the lower high temperature O₂ concentrations will lead to lower equilibrium concentration of NO.

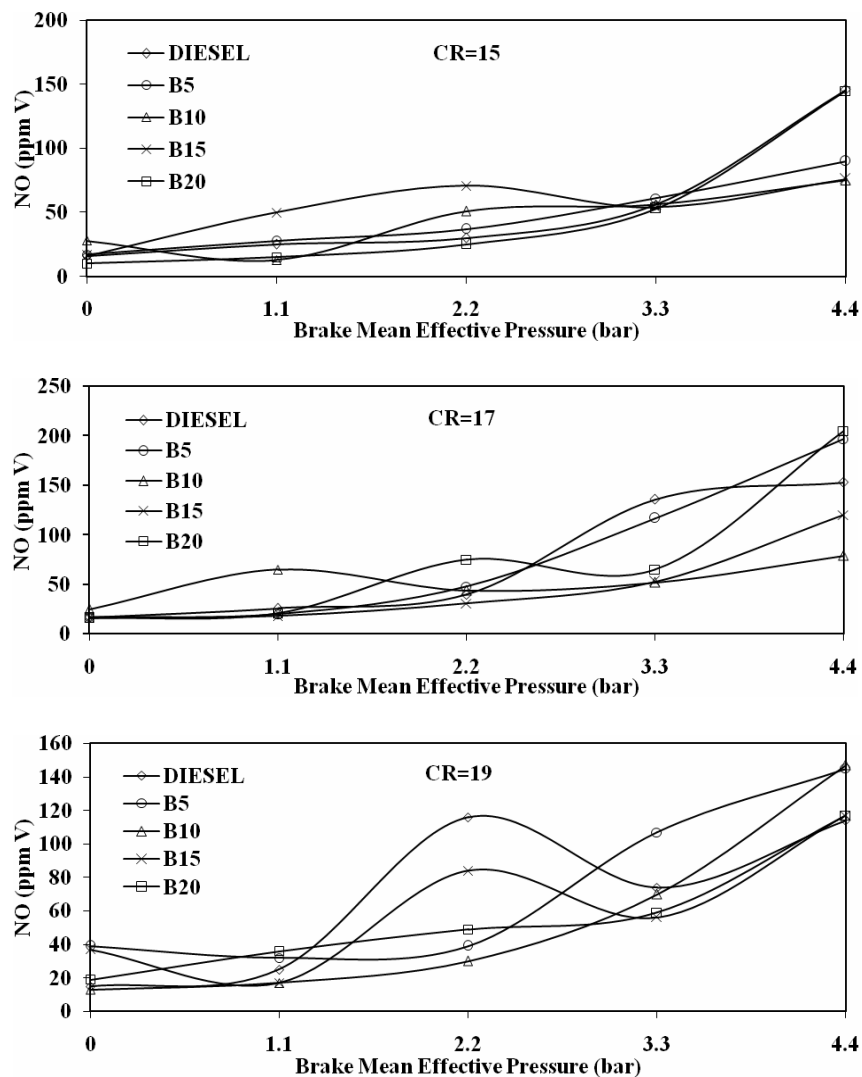


Figure-9. Nitric oxide (NO) as a function of engine load (bmep) at various compression ratios.

Figure-10 shows the smoke (soot) opacity with brake mean effective pressure of exhaust gas for neat diesel fuel and bio-diesel blends. One can observe that soot emitted by all bio-diesel blends is lower than neat diesel at low loads and lower compression ratio. This is attributed to the combustion being mixed controlled for these blends, as is also the case for neat diesel, which is however assisted by the presence of the fuel bound oxygen [9]. However, there is no definite trend observed in smoke

density with increase in blend percentage of bio-diesel. At all compression ratios, B15 was found to emit maximum smoke at full load. The minimum and maximum smoke opacities produced for B5 and B20 at 15:1 were 10.3% and 65.8% with a maximum and minimum reduction of 84% and 33% respectively, as compared to diesel. Also this trend continues for 17:1 and 19:1 CR for all the loads but the decrease in smoke opacity is comparatively higher for both compression ratios and this is due to increase in



cylinder temperature. At maximum load reduction was low, where as the most beneficial reductions appeared for the 75% load. The reason for this behaviour is the different amount of sulphur between the diesel and bio-diesel blends. Literature verifies that smoke emissions generally increase or decrease in relation to the sulphur

concentration. Sulphur in the fuel, results in sulphates that are absorbed on soot particles and increase the smoke emitted from diesel engines. In addition, the increase of oxygen content in the fuel contributes to a complete fuel oxidation even in locally rich zones, leading to a significant decrease of smoke [9-10].

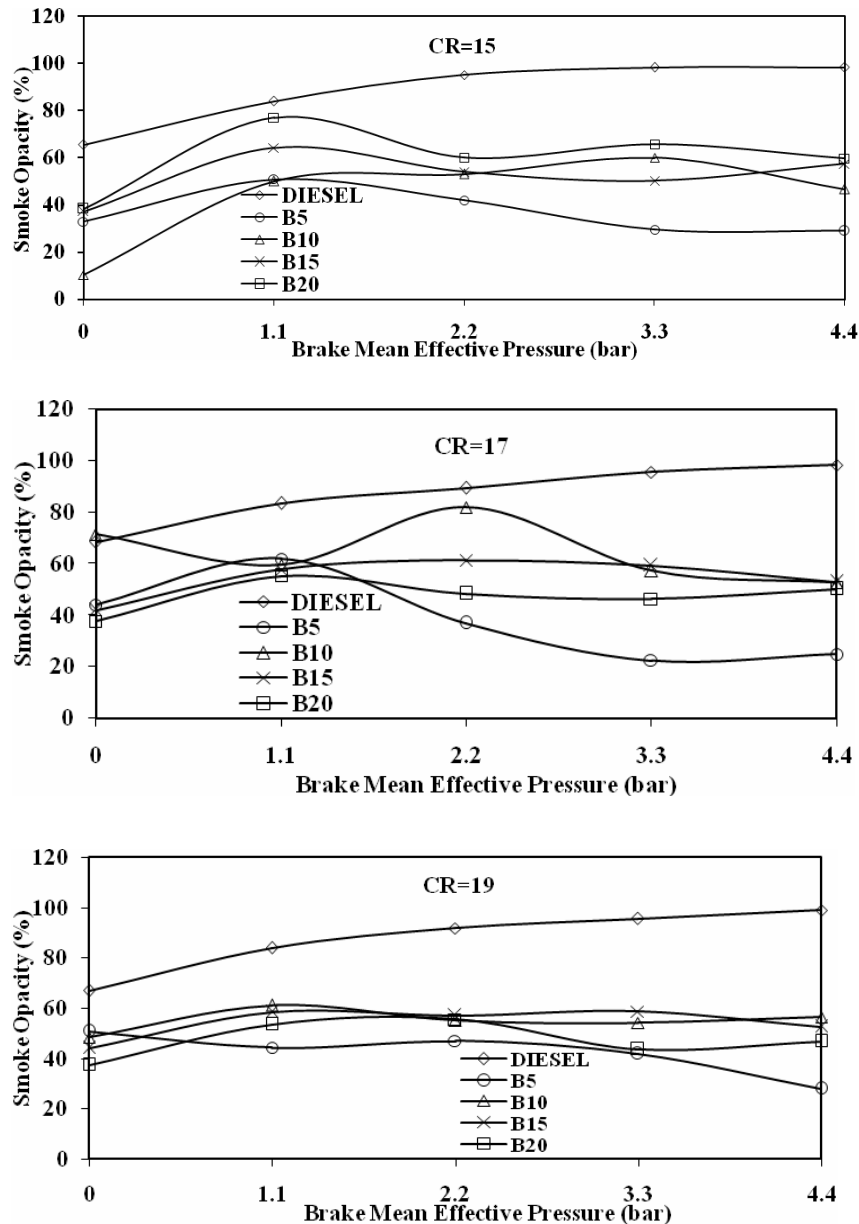


Figure-10. Smoke opacity emissions as a function of engine load (bmep) at various compression ratios.

Combustion parameters

Figure-11 shows the Pressure-crank angle diagram for all the fuel blends and neat diesel at compression ratio 15:1, 17:1 and 19:1 at full load. Overall combustion characteristics for all blends were found quite similar to diesel. In a CI engine, cylinder pressure depends on the burnt fuel fraction during the premixed burning phase, i.e., initial stage of combustion [29]. Cylinder

pressure characterizes the ability of the fuel to mix well with air and burn. From this Figure it is clear that peak pressure increases with increase in compression ratio. Since cetane number for bio-diesel is 49 which is higher than neat diesel i.e. 45 hence ignition delay is shorter for all bio-diesel blends than neat diesel and it can be observed from the figure.

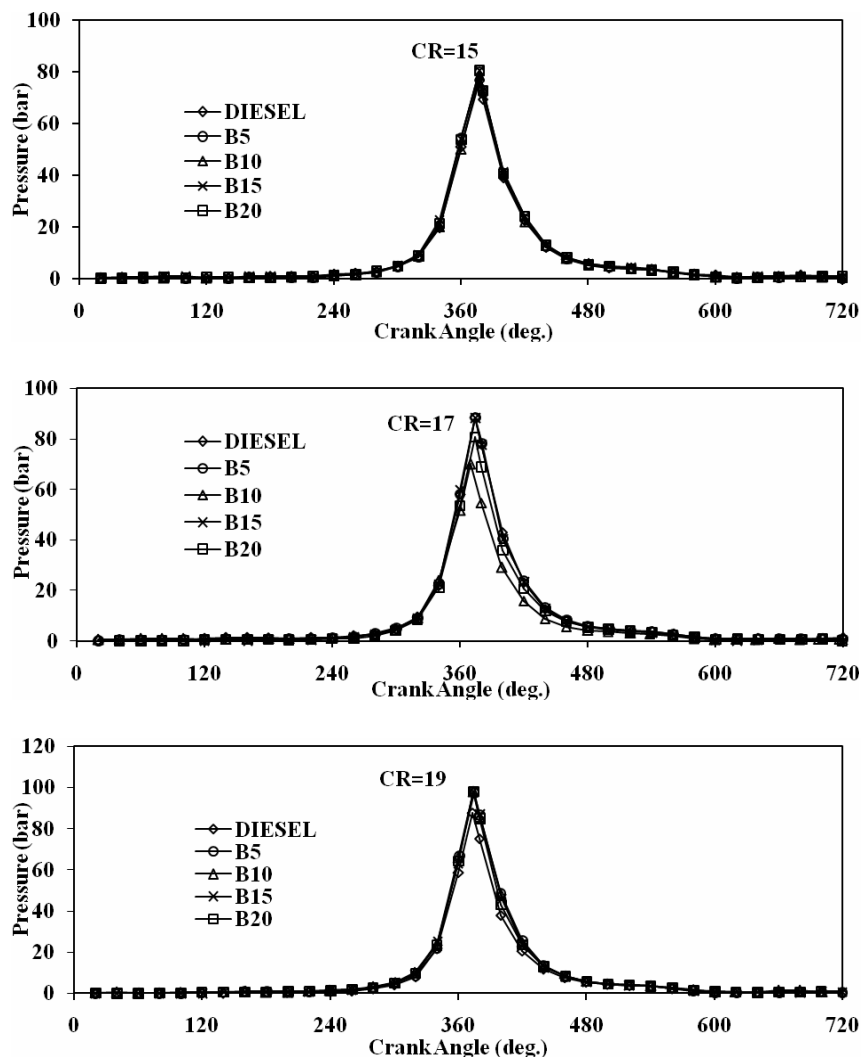


Figure-11. Variation of pressure with crank angle for different blends at various compression ratios.

Effect of engine load (bmep) on engine peak pressure for diesel and bio-diesel blends is shown in Figure-12. Peak pressure increases with increase in loads and bio-diesel blends at all compression ratios. Blends have more peak pressure than neat diesel at higher compression ratio (17:1 and 19:1). The reason for this is the presence of oxygen in the bio-diesel results complete combustion of fuel leading to increase in peak temperature and peak pressure. Also it can be observed that blend B20 has maximum peak pressure and it decreases with decrease in bio-diesel percentages. While at compression ratio of 15:1, blend B15 and B20 have highest peak pressure at full load. At 19:1 CR, the maximum peak pressure for B20 is about 90.66 bar, B15 is about 90.61 bar, B10 is 90.14 bar, B5 is 89.65 bar and for diesel is 81.23 bar. In a compression ignition engine, peak pressure depends on the combustion rate in initial stages, which in turn is influenced by the amount of fuel taking part in the uncontrolled combustion phase. The premixed or uncontrolled combustion phase is generated by the ignition

delay period and by the mixture preparation during the delay period. Thus higher viscosity and lower volatility of the COME which lead to poor atomization and mixture preparation with air and fast burning nature of COME blends during the ignition delay period are the reasons for this trend of peak pressure. Peak pressure is highest for neat diesel and all the fuel blends have nearly same peak pressures and ignition delay at compression ratio of 15:1. But in after burning, period rate of decrease in pressure is highest for diesel and least for B20. It can also be observed from Figure-12 that peak pressure is highest for B20 at 75% load and rate of pressure rise is highest for B5. This might be due to presence of oxygen in the bio-diesel that makes complete combustion of fuel possible, thereby producing more CO_2 and hence releasing more heat from the gases. Thus, both the peak temperature and pressure of bio-diesel-fueled engine is higher than that of diesel fueled engine.

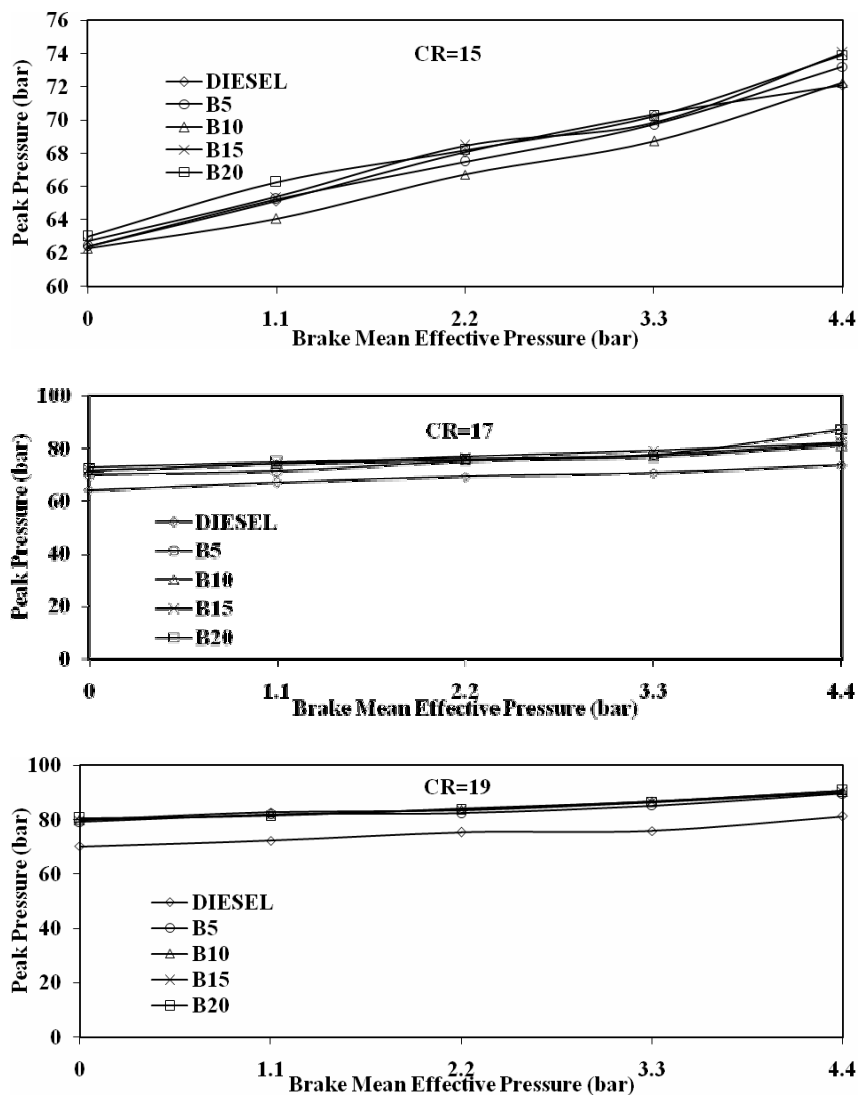


Figure-12. Variation of peak combustion pressure with crank angle for different blends at various compression ratios.

Figure-13 presents comparative data pertains to heat release rate for COME-diesel blends and diesel at various compression ratio. The premixed burning phase

associated with a high heat release rate is significant with bio-diesel blends operation.

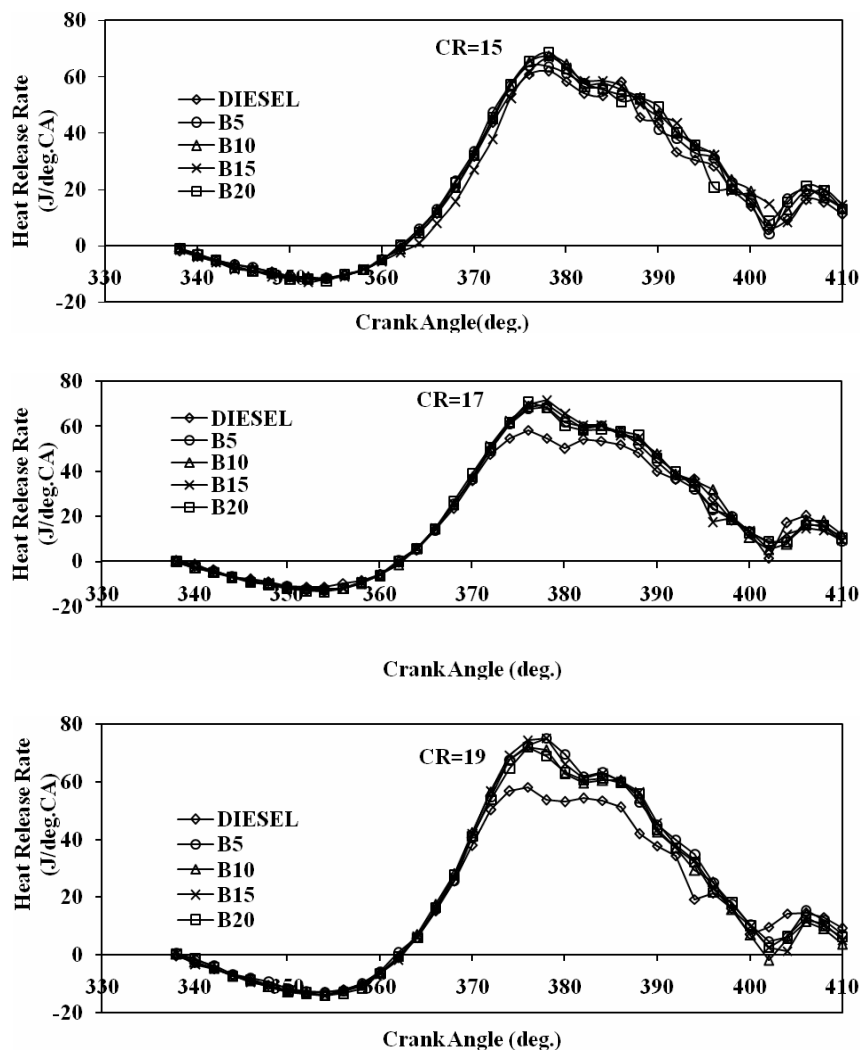


Figure-13. Variation of heat release rate with crank angle for different blends at various compression ratios.

Slightly higher peak HRR of 60 J/deg. (17:1 CR) and 65 J/deg. (19:1 CR) were obtained for bio-diesel under full load. It was found to be 50 J/deg. (17:1 CR) and 48 J/deg. (19:1 CR) for diesel under similar conditions. Increase in HRR is an indication of better premixed combustion and is probably the reason for increased NO emissions. Higher HRR for bio-diesel blends is probably due to excess oxygen present in its structure and dynamic injection advance apart from static injection advance. Higher boiling point of bio-diesel can also result in higher HRR [31]. However for 15:1 CR there is no much variation in peak HRR in compared with diesel at full load. Increase in diffusion combustion without increase in premixed combustion is indicative of poor combustion and is probably the reason for decreased NO emissions at full load.

CONCLUSIONS

The experimental conclusions of this investigation can be summarized as follows:

- Brake specific fuel consumption was found to have minimum for neat diesel as compared to biodiesel blends at all loads. However at lower compression ratios (15:1, 17:1) biodiesel blend B5 has minimum specific fuel consumption and at compression ratio of 19:1, B20 has least specific fuel consumption compare to diesel. At 15: 1 compression ratio, the bsfc is increased by 2.5% and 4.92% for the B5 and B20 blends. At 17:1 compression ratio, it is increased by 3.5% for both the B5 and B10 blends, whereas at 19:1 compression ratio, the B20 blend suggests slightly better fuel economy;
- The brake thermal efficiency was found to increase with increase in compression ratio and there is no large difference in the brake thermal efficiency of bio diesel blends and neat diesel. However at high loads B5 was found to have maximum thermal efficiency at compression ratio 15:1 and B20 at compression ratio 19:1. At compression ratio 17:1, B10 and B5 have almost equal efficiencies at full load. The maximum brake thermal efficiency values vary between 27.37-



29.28% for COME-diesel blends and 26.65-27.92% for Diesel fuel;

- The maximum NO emissions increase proportionally with the mass percent of oxygen in the biofuel and compression ratio, at 17: 1 CR reaching the highest, 205 ppm, value for the B20 blend and 197 ppm for B5. At the 15:1 CR, the NO emissions for all fuels are slightly lower, ranging from 75 ppm (B10) to 146 ppm (diesel fuel);
 - Smoke opacity emitted by biodiesel blends is lower than neat diesel at low load and lower compression ratios. The visible smoke and carbon monoxide emissions emerging from the biodiesel over all loads and compression ratios are lowered by up to 71.7% and 24% to 63.6%, respectively. The carbon dioxide CO₂ emissions, along with the fuel consumption are slightly higher for the B5 and B10 blends. The emission of unburned hydrocarbons HC for all fuels is low, 15–80 ppm, showing slightly milder values for COME-diesel blends compared with diesel fuel;
 - Overall combustion characteristics for all blends were found quite similar to diesel at all compression ratios. Peak pressure increases with increase in compression ratio for all biodiesel blends and neat diesel. At compression ratio of 17:1, ignition delay is shorter for all biodiesel blends than neat diesel due to higher cetane number and peak pressure is highest for B20;
 - Biodiesel blends have more heat release rate than mineral diesel at compression ratios of 17:1 and 19:1. And diesel fuel shows lowest heat release rate at initial stage and longer combustion duration at 75% load and at compression ratio of 15:1. However, maximum heat release is same for neat diesel and B10 at that compression ratio. Also biodiesel blends have large negative heat release rate due to cooling effect of the liquid fuel injected into the cylinder at all compression ratios;
 - Peak pressure increases with increase in loads for all fuels at all compression ratios. Blends have more peak pressure than neat diesel at higher compression ratio(17:1 and 19:1) and blend B20 has maximum peak pressure and it decreases with decrease in biodiesel percentages at all compression ratios; and
 - A general practical conclusion is that, all tested biodiesel blends can be used safely without any modification in engine. So, blends of methyl ester of cotton seed could be successfully used.
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