

Research Article

Combustion and Emission Characteristics of Variable Compression Ignition Engine Fueled with *Jatropha curcas* Ethyl Ester Blends at Different Compression Ratio

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Engine performance and emission characteristics of unmodified biodiesel fueled diesel engines are highly influenced by their ignition and combustion behavior. In this study, emission and combustion characteristics were studied when the engine operated using the different blends (B10, B20, B30, and B40) and normal diesel fuel (B0) as well as when varying the compression ratio from 16.5 : 1 to 17.5 : 1 to 18.5 : 1. The change of compression ratio from 16.5 : 1 to 18.5 : 1 resulted in 27.1%, 27.29%, 26.38%, 28.48%, and 34.68% increase in cylinder pressure for the blends B0, B10, B20, B30, and B40, respectively, at 75% of rated load conditions. Higher peak heat release rate increased by 23.19%, 14.03%, 26.32%, 21.87%, and 25.53% for the blends B0, B10, B20, B30, and B40, respectively, at 75% of rated load conditions, when compression ratio was increased from 16.5 : 1 to 18.5 : 1. The delay period decreased by 21.26%, CO emission reduced by 14.28%, and NO_x emission increased by 22.84% for B40 blends at 75% of rated load conditions, when compression ratio was increased from 16.5 : 1 to 18.5 : 1. It is concluded that *Jatropha* oil ester can be used as fuel in diesel engine by blending it with diesel fuel.

1. Introduction

The world is presently confronted with the twin crises of fossil fuel depletion and environmental degradation. Indiscriminate extraction and lavish consumption of fossil fuels have led to reduction in underground-based carbon resources. The search for alternative fuels, which promise a harmonious correlation with sustainable development, energy conservation, efficiency, and environmental preservation, has become very important today. Intensive research is going on throughout the globe for a suitable diesel substitute. In this race among different alternatives, vegetable oils have attained primary place as some of their physical, chemical, and combustion related properties are nearly similar to those of diesel fuel. A lot of research work has been carried out to use vegetable oil in its neat form. Since India is net importer of vegetable oils, edible oils cannot be used for substitution of diesel fuel. So, major concentration has been focused on nonedible oils as the fuel alternative to diesel fuel.

Many efforts have been made by several researchers to use nonedible oil as an alternative fuel in CI engine. Nonedible oil from the plant seeds is the most promising alternative fuel for CI engine, because it is renewable, environment friendly, nontoxic, biodegradable, also has no sulphur and aromatics, and has favorable heating value and higher cetane number. Its chemical structure contains long chain saturated and unbranched hydrocarbons that are the most favorable property for the use in conventional diesel engine [1–6].

Available literature indicates that plant oils are possible alternative fuel for diesel engine. But it was reported that CI engines that run on plant oils achieve lower peak power and torque, as well as lower engine speeds, and these fuels cause injector coking, dilution of engine oil, and carbon deposits in various parts of the engine, filter clogging, and ring sticking, when it is used directly in an engine as a diesel substitute fuel [7]. These problems adversely affect the performance of the direct injection CI engines. These are all due to large molecular mass, chemical structure of oil, higher

TABLE 1: Comparison of fuel properties of different fuels.

Fuel properties	Diesel	Jatropha oil	Ethyl Ester	Jatropha Ethyl Ester Blends				Test Method
				B10	B20	B30	B40	
Viscosity at 37°C, cS	4.38	38.33	7.33	5.16	5.66	5.83	6.00	D-445
Density at 37°C, g/cc	0.832	0.931	0.875	0.843	0.850	0.856	0.861	D-1298
Calorific value, MJ/kg	42.90	32.62	35.77	41.47	40.39	39.52	39.08	D-4868
Cloud Point, °C	0.5	8	1.7	0.7	0.8	1.3	1.5	D-2500
Pour point, °C	-7.8	4	-2.8	-7.2	-6.8	-6.8	-5.3	D-97
Flash Point, °C	58	287.7	111.7	61.7	68.7	76.3	83.7	D-93

viscosity, low volatility, and polyunsaturated character of oil [8–10]. High viscosity of plant oils is the major constraint which adversely affects the engine performance. The high viscosity of plant oil (more than 10 times that diesel fuel) leads to poor fuel atomization and inefficient mixing with air, which contribute to incomplete combustion. Different methods have been suggested by researchers to modify the plant oils [11–15]. These include

- (i) cracking the plant oils,
- (ii) dilution of plant oils with diesel fuel,
- (iii) microemulsification,
- (iv) heating the plant oils before injecting into the combustion chamber,
- (v) chemically transforming the plant oils to biodiesel by alcoholysis (transesterification).

Among these, chemically transforming the plant oils to biodiesel by alcoholysis (transesterification) was considered as the most suitable modification because technical properties of esters are nearly similar to diesel [16]. Through transesterification, plant oils are converted to the alkyl esters of the fatty acids present in the oil [17]. Furthermore, the methyl or ethyl esters of fatty acids can be burned directly in unmodified diesel engines with very low deposit formation.

Biodiesel has a higher cetane number than petroleum diesel fuel, no aromatics, and no sulfur and contains 10% to 11% oxygen by weight [18]. Some of the researchers suggested that this higher oxygen content of oil leads to complete combustion [19–22]. These characteristics of biodiesel reduce the emissions of carbon monoxide, hydrocarbon, and particulate matter in the exhaust gas compared with diesel fuel [23]. But the differences in physical properties between diesel and biodiesel fuels affect the combustion and heat release characteristics. Because the combustion and HRR characteristics of biodiesel must be known in order to achieve the reduction of brake specific fuel consumption (BSFC) and emission while keeping other engine performance parameters at an acceptable level. However, most of the researchers so far have correlated the performance and emission characteristics of biodiesel against test parameters such as biodiesel fraction blend, engine-speed, engine load, injection timing, injection pressure, and engine compression ratio. But there are very few works that have been reported on the engine combustion characteristics and heat release phenomena corresponding

to different biodiesels and its blends. Moreover, much more research has been done using methyl ester than ethyl ester. Therefore, the objective of this study is to investigate the combustion characteristics and heat release rate phenomena of a compression ignition engine running with Jatropha ethyl ester blends at different compression ratios.

2. Materials and Methods

2.1. Preparation of Ester and Blends. In this study, a two-step “acid-base” process, that is, acid-pretreatment followed by main base-trans-esterification reaction using ethanol as reagent and H₂SO₄ as catalysts for acid and KOH for base reaction was followed to produce biodiesel from *Jatropha curcas* oil. The description of the blends is as follows: B0: pure diesel, B10: 10% Jatropha ethyl ester + 90% diesel, B20: 20% jatropha ethyl ester + 80% diesel, B30: 30% Jatropha ethyl ester + 70% diesel, and B40: 40% Jatropha ethyl ester + 60% diesel. The physicochemical properties of the diesel, Jatropha oil, Jatropha ethyl ester, and their blends with diesel were evaluated as per the ASTM standards. All the measurements were repeated three times, and the arithmetic mean of these three readings was employed for analysis. The fuel properties data of all fuel are summarized in a tabular form as shown in Table 1.

2.2. Experimental Setup. A single cylinder, water cooled, 3.73 kW power, variable compression ratio engine was used for the test as shown in Figure 1. This test bed has a provision to change its compression ratio by rising or lowering bore head of the engine. The test bed is also equipped with all the control electrical, electronic computer, and data acquisition system. For running the engine, the compression ratio of the engine was changed to the desired ratio. Engine was started manually. Loading and unloading were done through computer. Various sensors are mounted on the engine to measure different parameters. A temperature thermocouple was installed on the surface of high pressure fuel pipe. A precision crank angle encoder was coupled with the main shaft of the engine. k-type of thermocouples is placed at different points to note the temperatures at the inlet, exhaust of the engine, engine head, cooling water inlet, cooling water outlet, and lubricating oil temperatures, and so forth. The software stores the data of pressures and volumes corresponding to a particular crank angle location for plotting the P - V

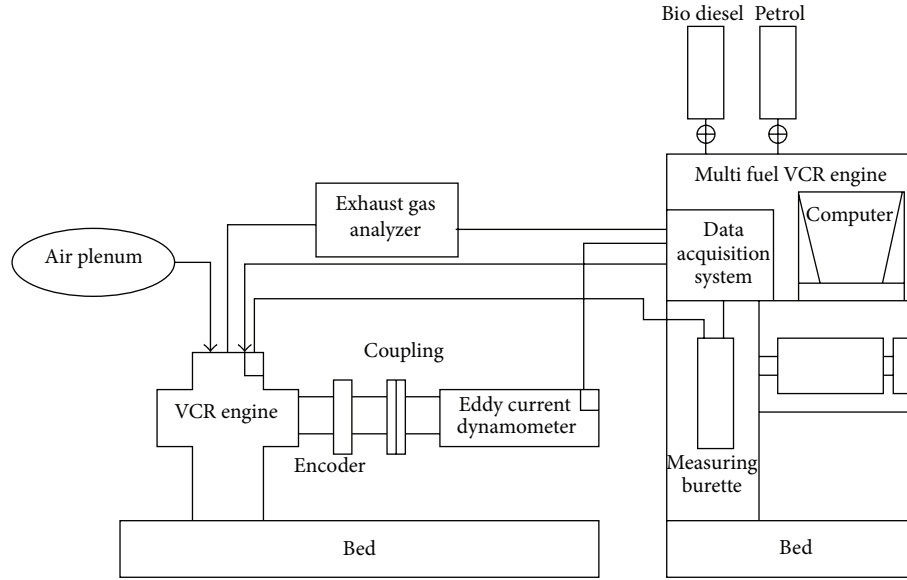


FIGURE 1: Schematic diagram of the experimental setup.

TABLE 2: Brief specification of variable compression ratio (VCR) engine.

S. No.	Parameter	Specification
1	Engine power	5 HP
2	Engine speed	1350 to 1600 rpm variable governed speed
3	Number of cylinders	One
4	Compression ratio	5 : 1 to 20 : 1
5	Bore, mm	80
6	Stroke, mm	110
7	Type of ignition	Spark ignition or compression ignition
8	Method of loading	Eddy current dynamometer
9	Method of starting	Manual crank start

and $P-\theta$ curves. The software also provides the facility of analyzing the combustion data such as the rate of heat release, ignition delay, combustion duration period in degrees, and peak pressure and stores it separately for analysis in the data acquisition system. The technical specifications of the engine are given in Table 2. A constant level of engine cooling water flow was maintained at more than 60 mL sec^{-1} . The standard fuel injection timing for the test engine was 23° BTDC . Engine test was done using software "Engine Test Express." This software is highly integrated "C" language based software.

Nucon multigas analyzer was used to measure the concentration of carbon monoxide (CO) and nitric oxide (NO_x) in the exhaust gases. A nominal flow rate of 500 to 1000 mL/min was maintained throughout the experiment as recommended by the manufacturer for an acceptable response time consistent with low consumption of sample gas. The digital meters were present on the instrument to directly display the reading. The range of carbon monoxide

meter was 0 to 2 percent (least count 0.001 percent) and for nitric oxide meter was 0 to 2000 ppm (least count 1 ppm).

2.3. Evaluation Procedure. The engine was evaluated using different fuel blends of Jatropa ethyl ester and diesel fuel at loads of 0% (no load), 25%, 50%, and 75% of rated load at compression ratio of 16.5 : 1, 17.5 : 1, and 18.5 : 1. The engine was warmed up prior to data acquisition. Initially the test engine was operated with base fuel diesel for about 10 minutes to attain the normal working temperature conditions. After that the baseline data was generated and the corresponding results were obtained. The engine was then operated with blends of Jatropa ethyl ester. During the tests with Jatropa ethyl ester blends, the engine was started with diesel until it was warmed up and then fuel was switched to various ester blends. After finishing the tests with diesel-ester blends, the engine was always switched back to diesel fuel and the engine was run until the ester blends had been purged from the fuel line, injection pump, and injector. This was done to prevent starting difficulties at the later time. Combustion and emission parameters such as peak pressure, heat release rate, ignition delay, and NO_x and CO emissions were evaluated (Table 3).

2.4. Theoretical Consideration. The heat release rate (HRR) is an important parameter to analyze the combustion phenomena in the engine cylinder. The important combustion phenomena parameters such as combustion duration and intensity can be easily estimated from the heat release rate diagram. The HRR diagram also provides key input parameters in the modeling of the NO_x emission. The heat release rate is modeled by applying the first law of thermodynamics. The simplified model is shown in (1) as

$$Q_n = \frac{\lambda}{\lambda - 1} P \frac{dV}{d\theta} + \frac{1}{\lambda - 1} V \frac{dP}{d\theta}, \quad (1)$$

TABLE 3: Test matrix for combustion and emissions study on diesel engine.

Sr. No.	Variables	Types of variables studied	Details of variables studied
1	Independent	(1) Fuels used	B0, B10, B20, B30 and B40
		(2) Load (%)	0, 25, 50 and 75
		(3) Compression ratio	16.5 : 1, 17.5 : 1, 18.5 : 1
2	Dependent	(1) Heat release rate (HRR)	At 0%, 25%, 50%, 75% of rated load
		(2) Ignition delay	At 0%, 25%, 50%, 75% of rated load
		(3) Peak pressure	At 0%, 25%, 50%, 75% of rated load
		(4) CO emissions	At 0%, 25%, 50%, 75% of rated load
		(5) NO _x emissions	At 0%, 25%, 50%, 75% of rated load

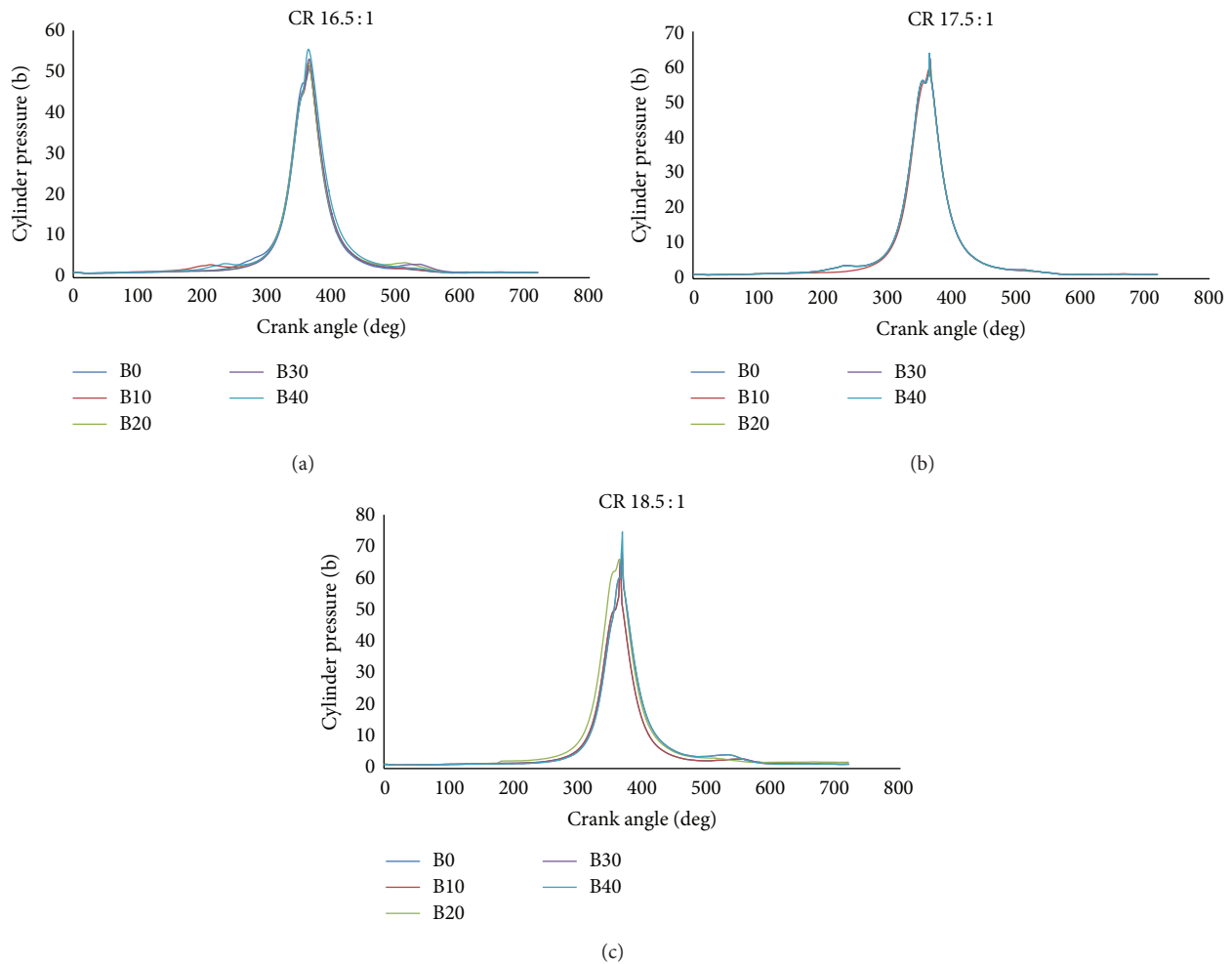


FIGURE 2: Pressure versus crank angle for all fuels at 75% of rated load for various compression ratios.

where λ is the ratio of specific heats which was taken as 1.35, θ is crank angle, P is cylinder gas pressure, and V is cylinder volume.

3. Result and Discussion

3.1. Combustion Characteristics. The combustion characteristics of the biodiesels can be compared by the means of cylinder gas pressure, heat release rate, and ignition delay.

3.1.1. Cylinder Pressure

(1) Effect of Blend. In a CI engine, cylinder pressure depends on the burnt fuel fraction during the premixed burning phase, that is, initial stage of combustion. Cylinder pressure characterizes the ability of the fuel to mix well with air and burn condition. Figure 2 shows the comparison of cylinder pressures with crank angle for the fuels tested at all compression ratios under the 75% of rated load. The results show that the peak cylinder pressure of the engine running with

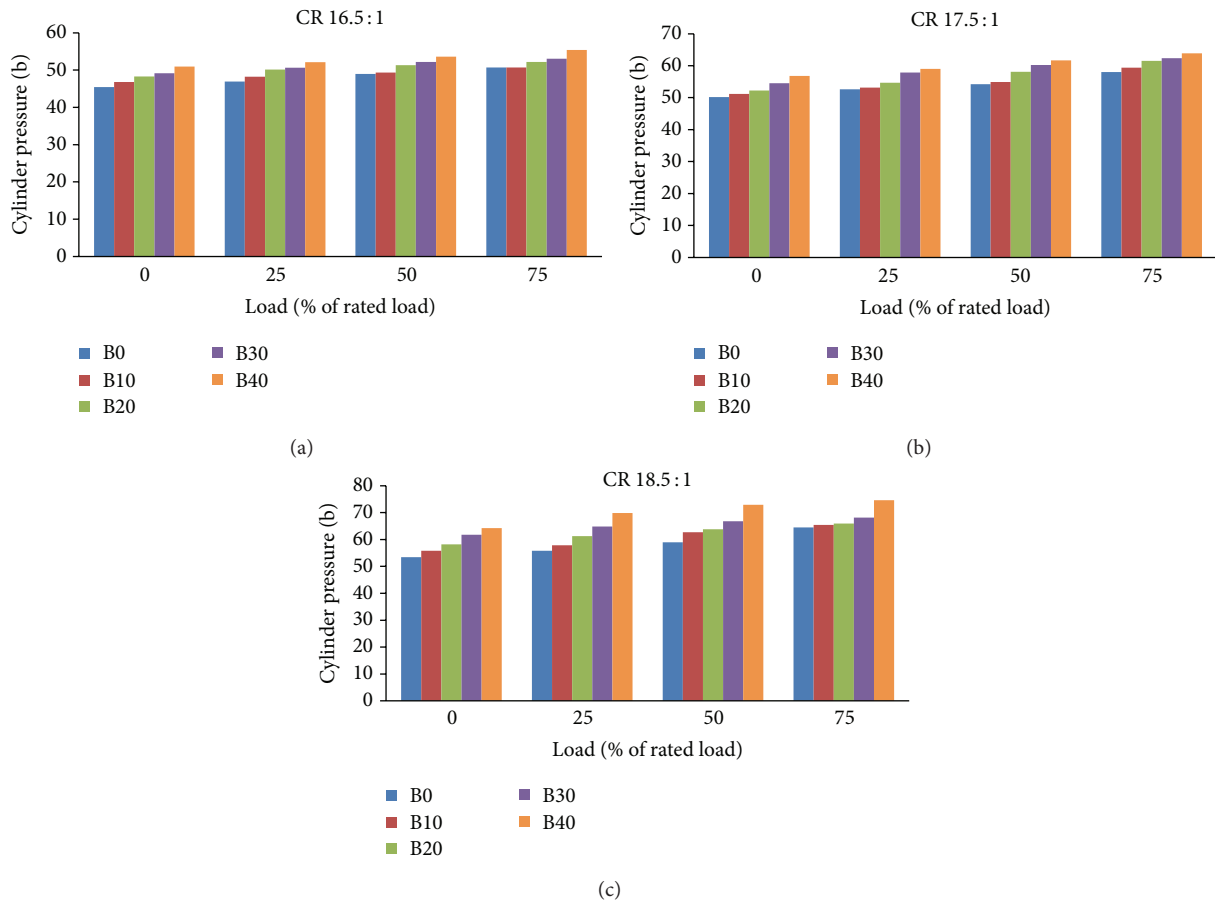


FIGURE 3: Variation of peak cylinder pressure with engine load for all fuel at compression ratios of 16.5, 17.5, and 18.5, respectively.

ester blends is slightly higher than the engine running with diesel at 75% of rated load and compression ratios. There were some reasons for this behavior: (1) due to the high viscosity, low volatility, and higher cetane number of biodiesel blends, there is occurrence of a short ignition delay and advanced injection timing for esters blend than diesel fuel. As a result, combustion starts later for diesel fuel and the peak cylinder pressure attains a lower value as it is further away from the TDC in the expansion stroke. (2) Due to the presence of oxygen molecule in biodiesel, the hydrocarbons achieve better combustion resulting in higher cylinder pressure [24].

The effect of the load on the cylinder pressure has also been investigated and the results are shown in Figure 3. It can be seen that the in-cylinder pressure increases with increasing load for both diesel and ester blends. It is observed that the peak pressure of 50.67, 51.36, 52.16, 53.04, and 55.41 bar was recorded for standard diesel, B10, B20, B30, and B40, respectively, at 75% of rated load conditions for compression ratio of 16.5:1. For compression ratio of 17.5:1, the peak pressures of 58.03, 59.42, 61.54, 62.37, and 63.89 bar were recorded for standard diesel, B10, B20, B30, and B40, respectively, at 75% of rated load conditions. For compression ratio of 18.5:1, the peak pressure of 64.45, 65.38, 65.92, 68.15, and 74.63 bar was recorded for standard diesel, B10, B20, B30, and B40, respectively, at 75% of rated load conditions. Similar conclusions were drawn by other authors in the literature [25].

(2) *Effect of Compression Ratio.* In general, increasing the compression ratio improved cylinder pressure of the engine. On average, the cylinder pressure increased by 27.1%, 27.29%, 26.38%, 28.48%, and 34.68% for the blends B0, B10, B20, B30, and B40, respectively; when compression ratio was increased from 16.5:1 to 18.5:1, it was increased by 14.52% and 11.06%; 15.69% and 10.03%; 17.98% and 7.71%; 17.59% and 9.26%; 15.30% and 16.81% when the compression ratio was raised from 16.5:1 to 17.5:1 and further to 18.5:1, respectively, for the blends B0, B10, B20, B30, and B40 respectively, as it can be seen from Figure 4. These increased values of cylinder pressure with compression ratio were observed at 75% of rated load for all blends. This shows that increasing the compression ratio had more benefits with ester blends than with pure diesel. Due to their low volatility and higher viscosity and cetane number, biodiesel might be performing relatively better at higher compression ratios. Also the oxygen content of biodiesel may be a cause for this better performance.

3.1.2. Heat Release Rate

(1) *Effect of Blends.* The heat release rate is used to identify the start of combustion, the fraction of fuel burned in the premixed mode, and differences in combustion rates of fuels. Figure 5 shows the heat release rates for CI engine running with ester blends and diesel fuel at 75% of rated load. It can

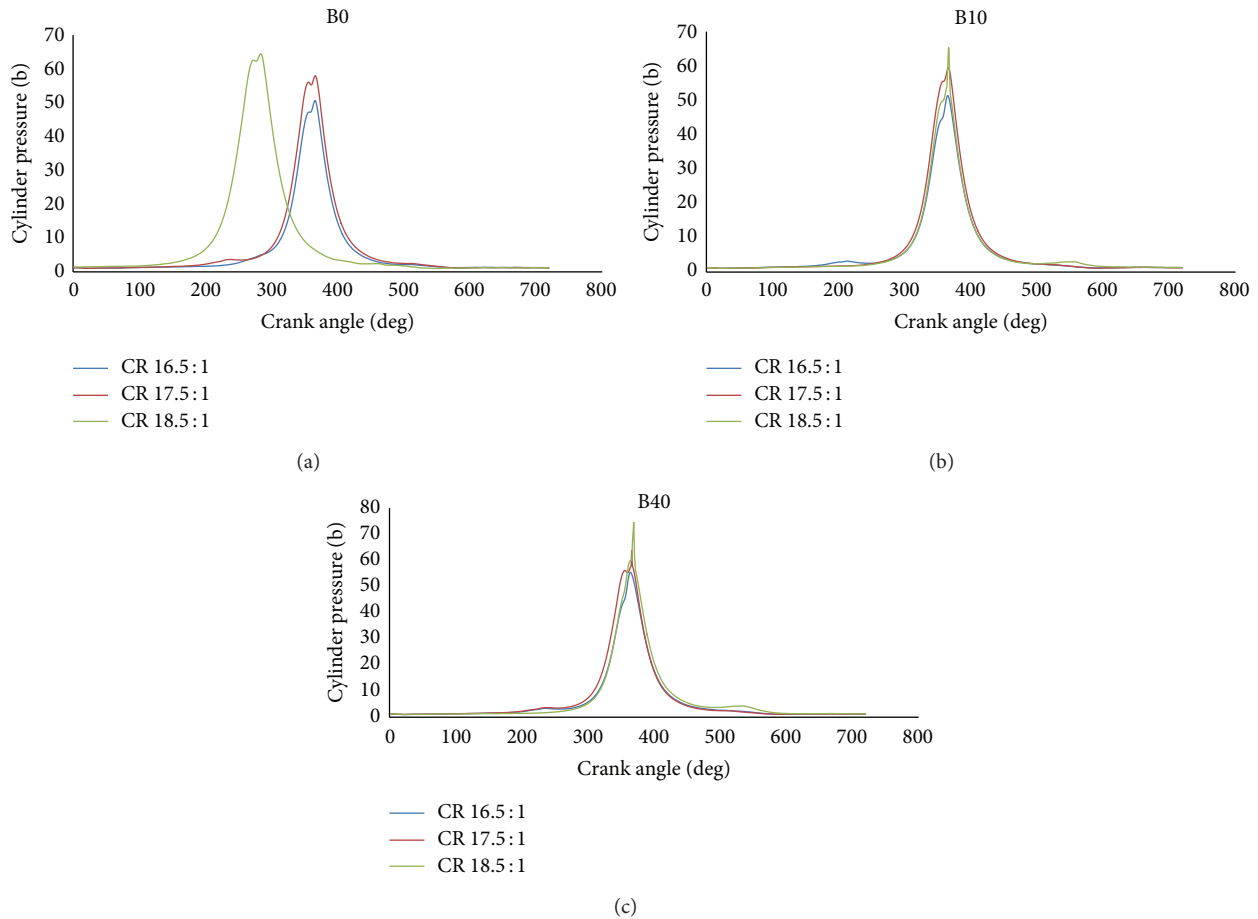


FIGURE 4: Pressure versus crank angle for B0, B10, and B40 fuel at 75% of rated load for different compression ratios.

be seen that the CI engine running with blends has a higher peak in the heat release rate diagram than the diesel. This phenomenon can be explained on the basis of the presence of the oxygen molecule in biodiesel fuel that results in the air-mixed fuel in the cylinder to burn completely and increase the heat release rate. Higher boiling point of ester blends can also result in higher heat release rate [26]. The maximum heat release rate of standard diesel, B10, B20, B30, and B40 were 13.58, 16.89, 19.56, 23.73, and 27.69 J/deg respectively, at 75% of rated load for compression ratio of 16.5 : 1. For compression ratio of 17.5 : 1, the maximum heat release rates of 15.46, 17.72, 21.57, 26.86, and 32.13 J/deg were recorded for standard diesel, B10, B20, B30, and B40, respectively, at 75% of rated load conditions. For compression ratio of 18.5 : 1, the maximum heat release rates of 16.73, 19.26, 24.71, 28.92, and 34.76 J/deg were recorded for standard diesel, B10, B20, B30, and B40, respectively, at 75% of rated load conditions.

(2) *Effect of Compression Ratio.* On average, higher peak HRR increased by 23.19%, 14.03%, 26.32%, 21.87%, and 25.53% for the blends B0, B10, B20, B30, and B40, respectively; when compression ratio was increased from 16.5 : 1 to 18.5 : 1, it was increased by 13.84% and 08.21%; 04.89% and 08.69%; 10.27% and 14.55%, 13.19% and 07.66%, 16.03%, and 08.18% when the

compression ratio was raised from 16.5 : 1 to 17.5 : 1 and further to 18.5 : 1, respectively, for the blends B0, B10, B20, B30, and B40, respectively, under 75% of rated load conditions. Higher HRR for biodiesel blends is probably due to excess oxygen present in its structure and dynamic injection advance apart from static injection advance. Increase in HRR is an indication of better premixed combustion and is probably the reason for increased NO_x emissions.

3.1.3. *Cylinder Pressure and Volume.* Figure 6 depicts that P - V diagram of compression ignition engine, which was examined at 75% of rated load conditions. The results show that the P - V diagram does not show any significant change for different fuels, namely, B0, B10, B20, B30, and B40, at different compression ratios.

3.1.4. *Ignition Delay Period.* Ignition delay of fuel is a significant parameter in determining the knocking characteristics of C.I. engines. The cetane number of a fuel, which indicates the self-igniting capability, has a direct impact on ignition delay. The higher the cetane number, the shorter the ignition delay and vice versa. The ignition delay period was determined by the Engine Test Express' software installed in computer attached to the engine.

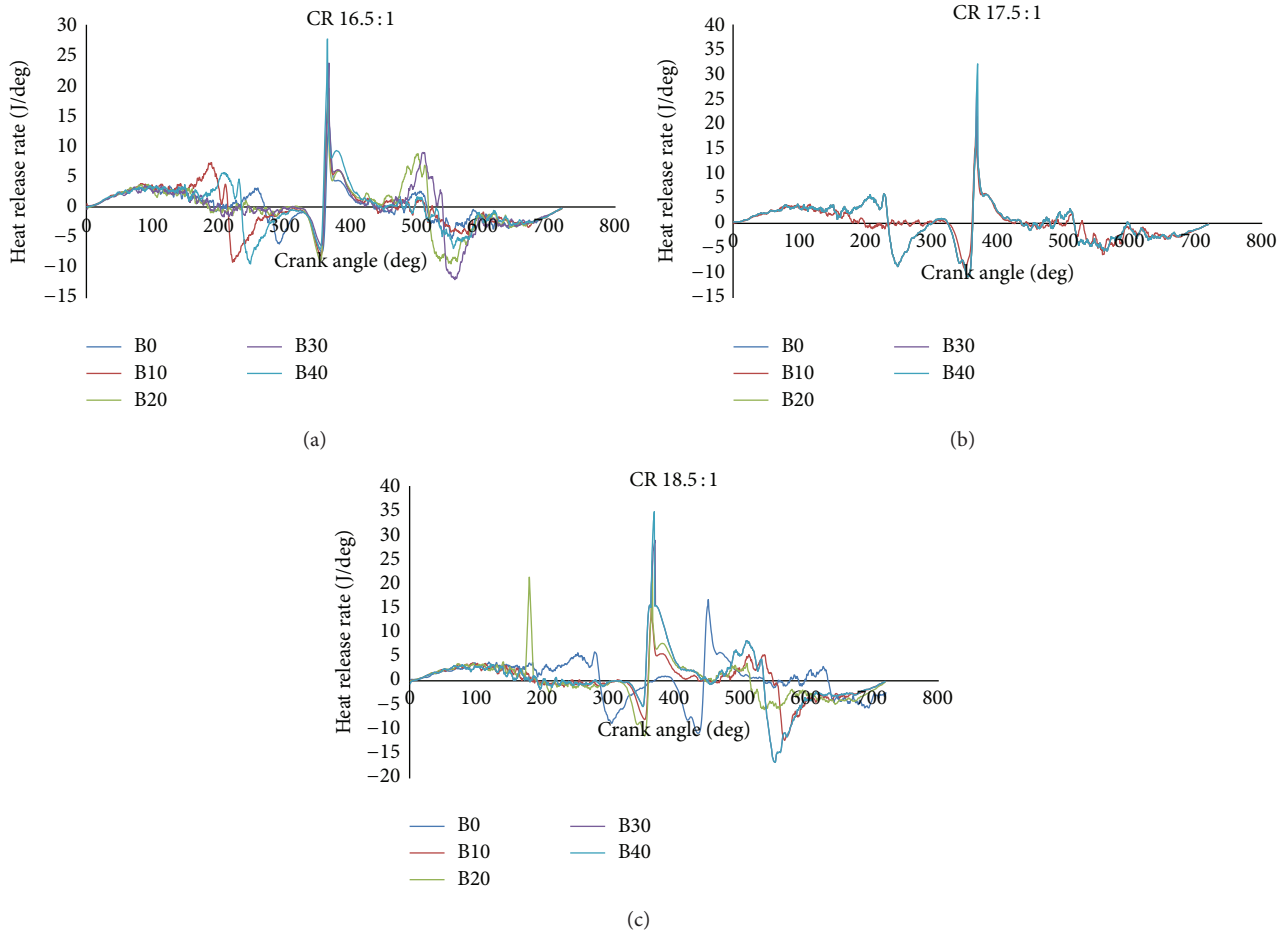


FIGURE 5: Comparison of the heat release rate for diesel and different ethyl ester blends at 75% of rated load for different compression ratios.

(1) *Effect of Blend.* Figure 7 compares the delays between diesel and ester blends at different load for the three compression ratios. As shown in the figure, as load increases as the delay period decreases for all blends for the three compression ratios. This behavior is because of the fact that as the engine speed decreases, the residual gas temperature and wall temperature decrease, which results in lower charge temperature at injection time and lengthening the ignition delay. The delays are consistently shortest for the blend B40. In spite of the slightly higher viscosity and lower volatility of biodiesel, the ignition delay seems to be lower for ester blends than for diesel. The reason may be that a complex and rapid preflame chemical reaction takes place at high temperatures. As a result of the high cylinder temperature existing during fuel injection, biodiesel may undergo thermal cracking and lighter compounds are produced, which might have ignited earlier to result in a shorter ignition delay [27]. Another reason may be due to the fact that Oleic and Linoleic fatty acid esters present in the biodiesel split into smaller compounds when it enters the combustion chamber resulting in higher spray angles and hence causes earlier ignition.

(2) *Effect of Compression Ratio.* As it is clear from Figure 7 that as the compression ratio increase, the delay period will

decrease for all blends at all loads. These results are proved clearly in Figure 8 for the blend B40 at the three compression ratios 16.5 : 1, 17.5 : 1, and 18.5 : 1, respectively. On average, the delay period decreased by 19.88%, 24.28%, 21.87%, 23.52%, and 21.26% for B0, B10, B20, B30, and B40 at 75% of rated load conditions when compression ratio was increased from 16.5 : 1 to 18.5 : 1. It was decreased by 8.77% and 12.17%, 4.34% and 20.84%, 4.31% and 18.34%, 8.19% and 16.69%, 10.67% and 11.84% for B0, B10, B20, B30, and B40 at 75% of rated load conditions when the compression ratio was raised from 14 to 16 and further to 18, respectively. The possible reason for this trend could be that the increased compression ratio actually increases the air temperature inside the cylinder helping for early combustion consequently reducing the ignition delay.

3.2. Emissions

3.2.1. NO_x Emission

(1) *Effect of Blend.* The emissions of NO_x against engine load for various blends are compared in Figure 9 at three compression ratios. As seen from the figure, all blends produced higher NO_x than pure diesel for all engine loads at all compression ratios. For all of the blends, the curves

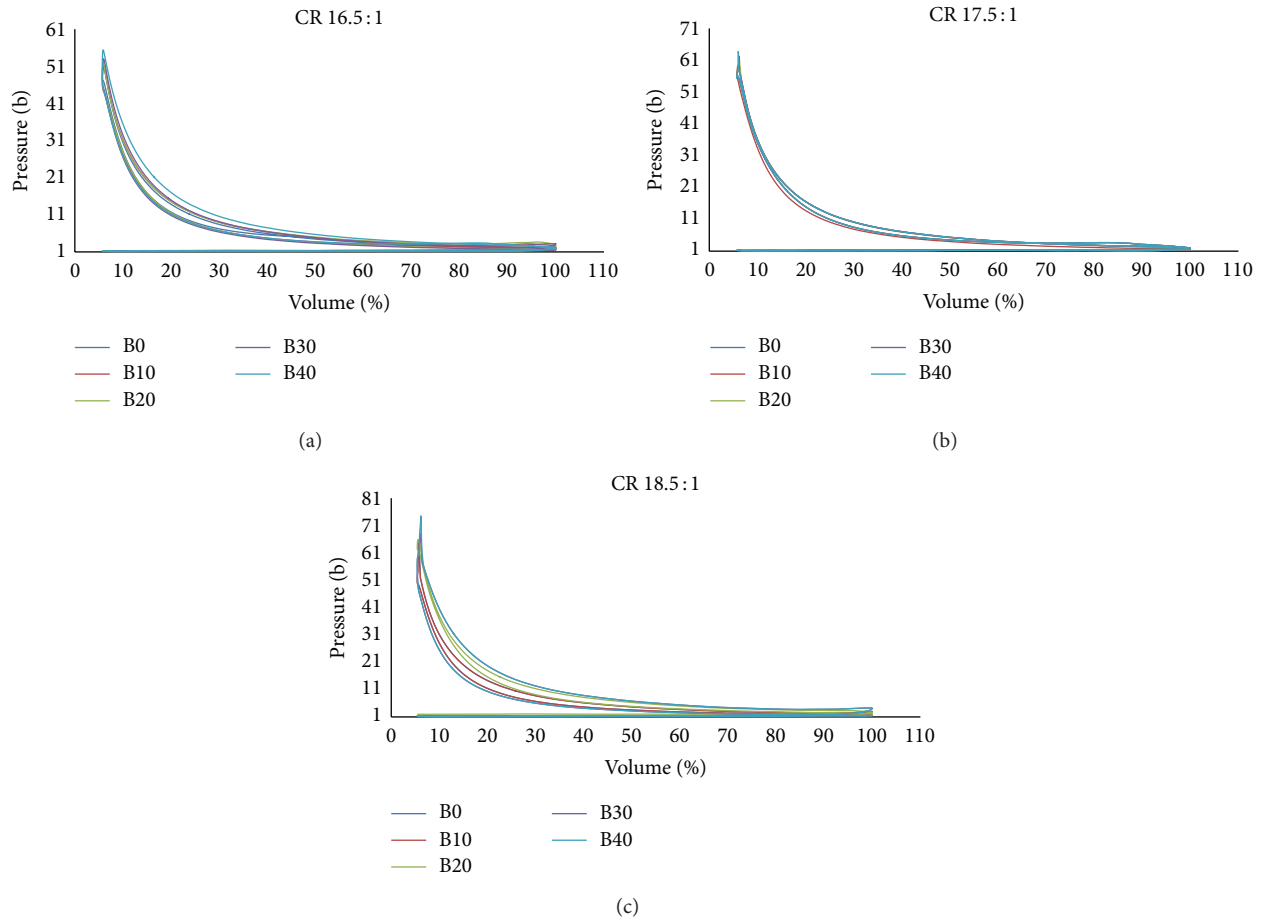


FIGURE 6: P - V diagram for diesel and ethyl ester blends at different compression ratios.

for each blend remain over the curve of pure diesel because NO_x emission in exhaust gases was very much dependent on combustion chamber temperature. The combustion chamber temperature depends on load. Increase in load results in more fuel supply into the combustion chamber, thus producing high flame temperature. At high temperature, the reaction $\text{N}_2 + \text{O}_2 = 2\text{NO}$ takes place. There were some other reasons for this behavior. (1) The NO_x emission increases for blends may be associated with the oxygen content of the ester, since the oxygen present in the fuel may provide additional oxygen for NO_x formation. Peterson et al. 1992 proposed a theory for the slight NO_x increase of biodiesel. They considered that biodiesel typically contains more double bonded molecules than petroleum derived diesel. These double bonded molecules have a slightly higher adiabatic flame temperature, which leads to the increase in NO_x production for biodiesel. (2) Another factor causing the increase in NO could be the possibility of higher combustion temperatures arising from improved combustion because larger part of the combustion is completed before TDC for ester blends compared to diesel due to their lower ignition delay. So it is highly possible that higher peak cycle temperatures are reached for ester blends compared to diesel. (3) The emission of NO_x from diesel engines depends also on iodine number

of fuel. The emission of NO_x increases with increase in iodine number. Hence, the emission of NO_x was found higher in blends compared to diesel.

(2) *Effect of Compression Ratio.* On an average, the NO_x emission increased by 1.45%, 14.38%, 38.79%, 29.02%, and 22.84% for B0, B10, B20, B30, and B40 at 75% of rated load conditions, when compression ratio was increased from 16.5:1 to 18.5:1. It was increased by 0% and 1.45%, 13.38% and 0.8%, 14.00% and 21.74%, 8.61% and 18.78%, and 9.09% and 12.60% when the compression ratio was raised from 16.5:1 to 17.5:1 and further to 18.5:1, respectively, for the blends B0, B10, B20, B30, and B40 at 75% of rated load conditions (Figure 10). This increased amount of NO_x emission with compression ratio was observed at all engine loads for all blends. Hence the most significant factor that causes NO_x formation is high combustion temperatures and the combustion temperature increases as the compression ratio increases, so as the compression ratio increases, the amount of NO_x will increase. Another reason for increased emission of NO_x with increase in compression ratio is that with lower compression ratio, the premixed burning is high due to longer delay resulting in lesser production of NO_x in engines. With increase in compression ratio, ignition delay reduces and

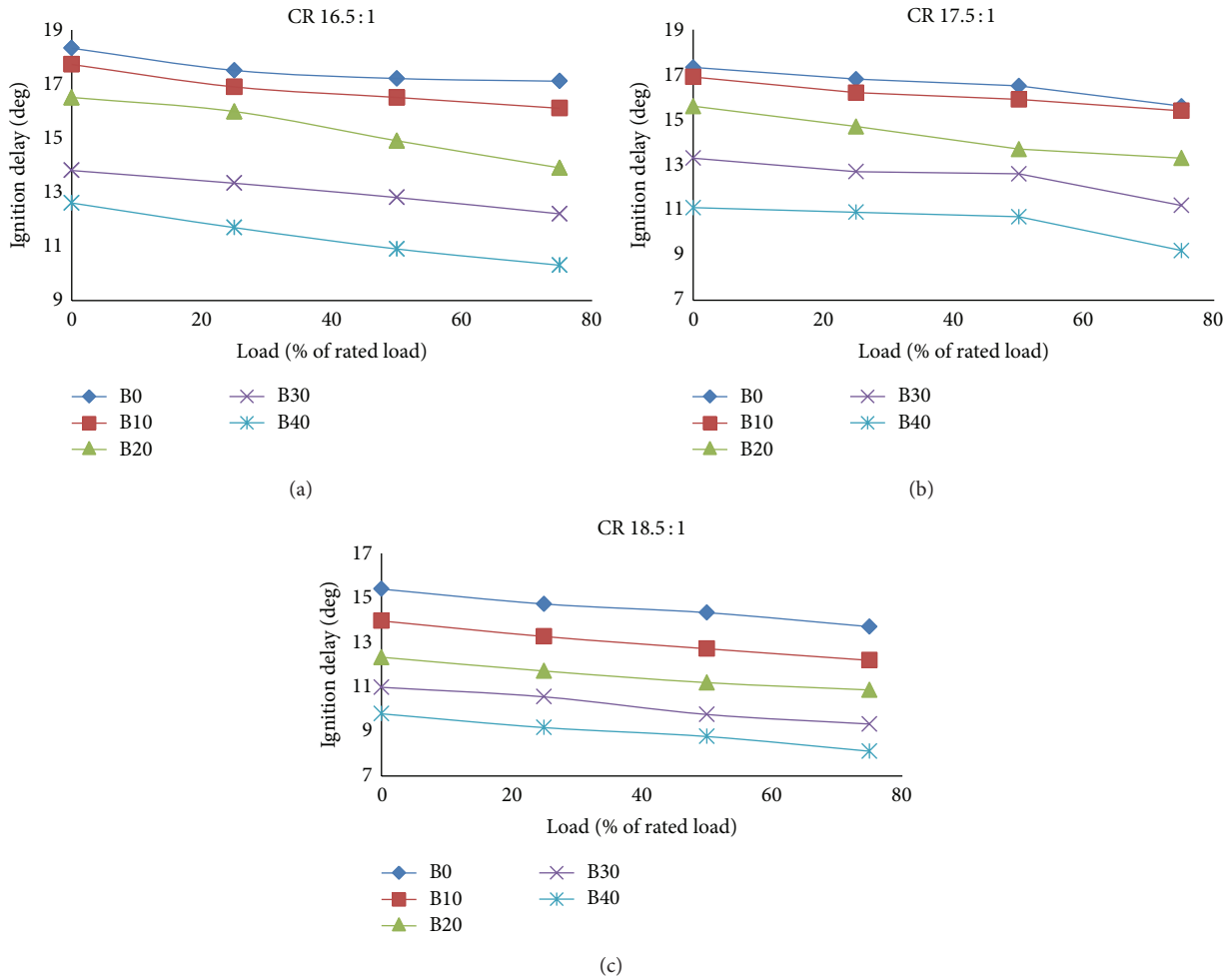


FIGURE 7: Effect of engine load on the ignition delay period for diesel and ethyl ester blends at different compression ratios.

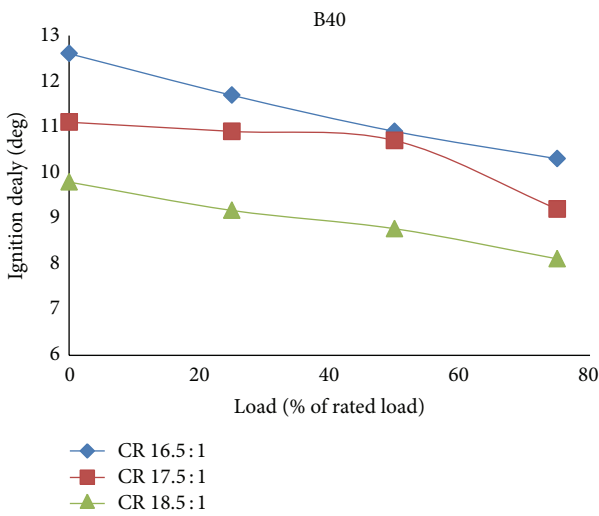


FIGURE 8: Effect of different compression ratios for B40 fuel at different load.

peak pressure increases resulting in high temperature which causes larger amounts of NO_x formation.

3.2.2. CO Emission

(1) *Effect of Blend.* As shown in Figure 11, CO concentration in exhaust gases increased with increase in load. This is due to the fact that as the load increased to the maximum value, the fuel consumption with higher oxygen content also proportionately increased which lead to the better combustion of fuel and produce higher CO emission. As it is clear from the figure that the curves of CO emissions for all biodiesel blends remain under the curve of pure diesel and decrease as the biodiesel percent increases at all compression ratios. This reduced emission of carbon monoxide may have resulted due to increased combustion efficiency reflected in terms of higher brake thermal efficiency because of presence of the oxygen molecules in the blended fuels. Several other reasons have been reported to explain the decrease of CO when substituting conventional diesel for biodiesel: (1) the increased biodiesel cetane number. The higher the cetane number, the lower the probability of fuel-rich zones formation, usually related to CO emissions; (2) the advanced injection and combustion when using biodiesel may also justify the CO reduction with this fuel.

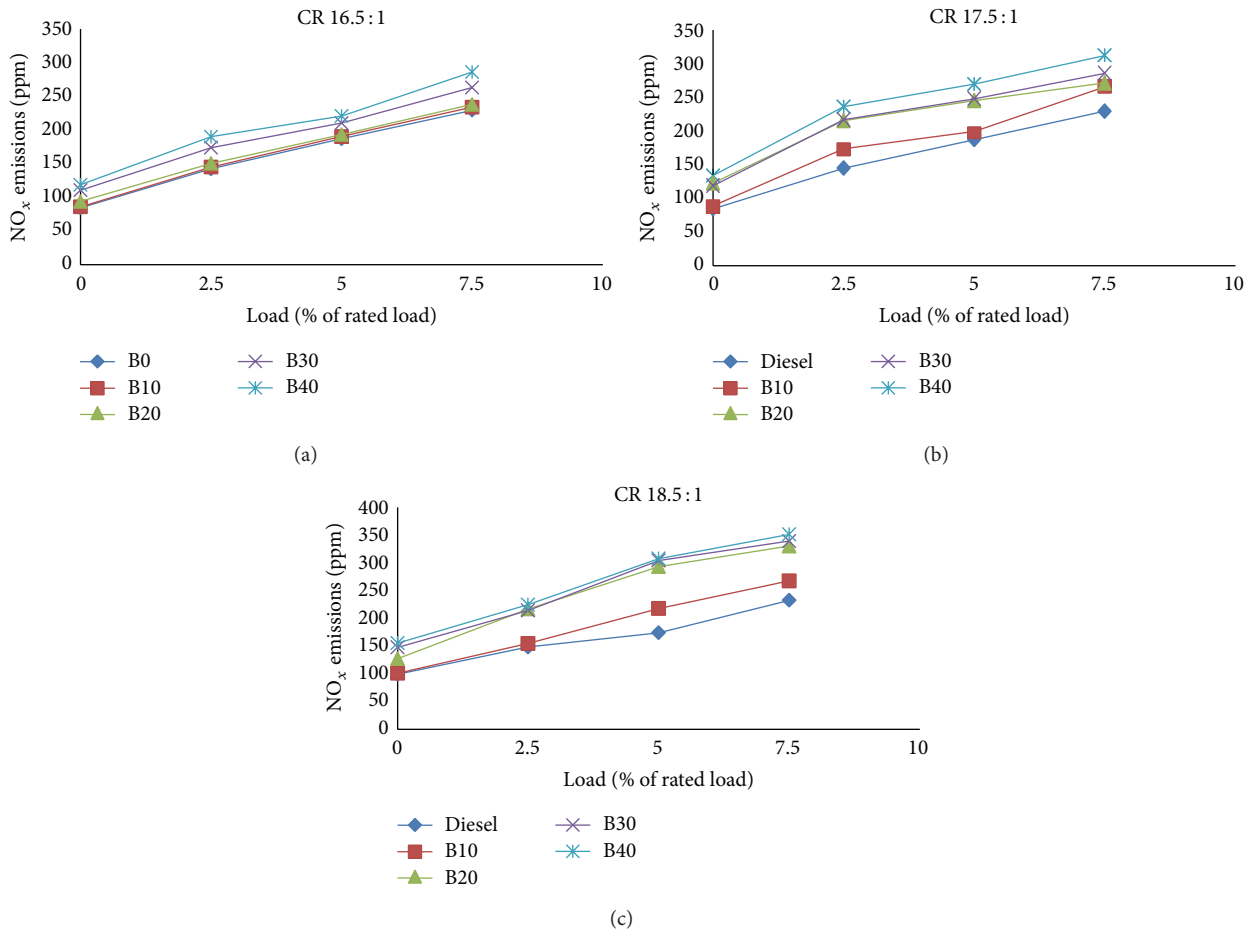


FIGURE 9: Variation of NO_x emission with engine load for diesel and all blends of Jatropha ethyl ester at different compression ratios.

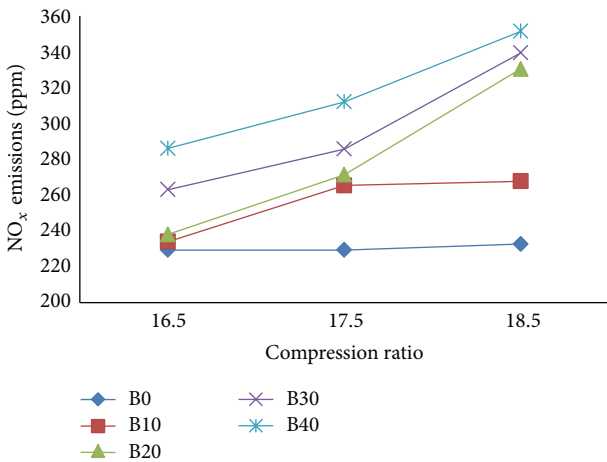


FIGURE 10: Variation of NO_x emission with compression ratio for diesel and all ester blends at 75% of rated load conditions.

(2) *Effect of Compression Ratio.* On an average, the CO emission reduced by 14.28% when compression ratio was increased from 16.5:1 to 18.5:1 for the blend B40 at 75% of rated load conditions and as it can be seen from Figure 12,

similar values were obtained for the other blends. The possible reason for this trend could be that the increased compression ratio actually increases the air temperature inside the cylinder consequently reducing the delay period causing better and more complete burning of the fuel and so lower CO emission.

4. Conclusions

The combustion and emission characteristics of ethyl ester derived from *Jatropha curcas* oils have been experimentally investigated using a variable compression ignition engine. The effects of ester blend ratio, engine load, and compression ratio on the engine combustion and emissions parameters were investigated. The key findings of this experiment are as follows.

- (i) The engine running with ester blends has produced higher peak heat release rate than the engine running with normal diesel at 75% of rated load conditions.
- (ii) In general, increasing the compression ratio improved the performance and cylinder pressure of the engine and had more benefits with ester blends than with diesel fuel.

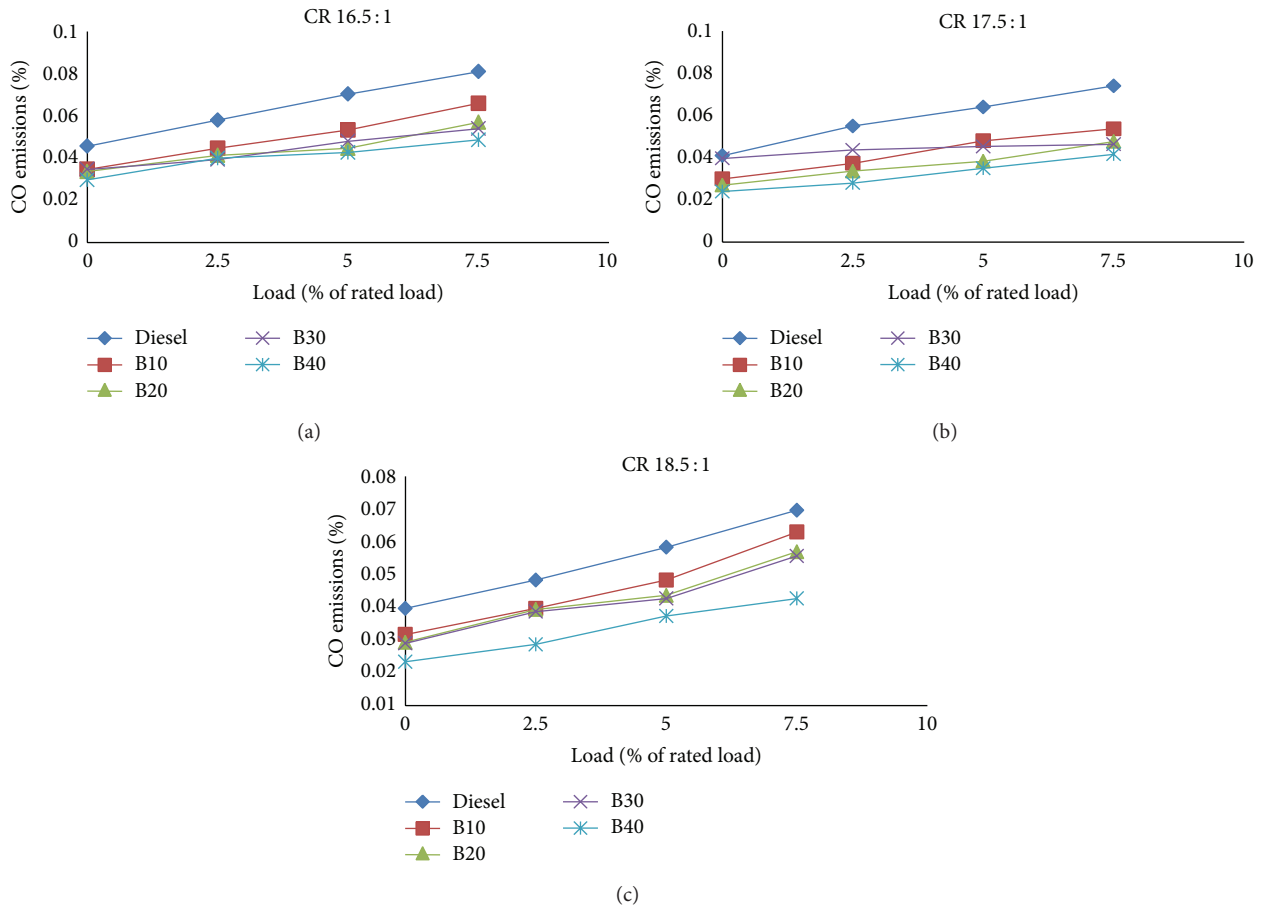


FIGURE 11: Variation of CO emission with engine load for diesel and all blends of Jatropha ethyl ester at different compression ratios.

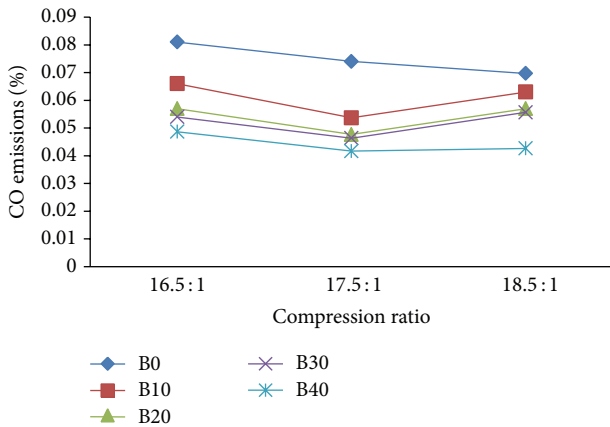


FIGURE 12: Variation of CO emission with compression ratio for all blends at 75% of rated load conditions.

- (iii) In spite of the slightly higher viscosity and lower volatility of ester blends, the ignition delay seems to be lower for ester blends than for diesel. On average, the delay period decreased by 21.26% for B40 blends at 75% of rated load conditions, when compression ratio was increased from 16.5 : 1 to 18.5 : 1.
- (iv) CO emission reduced by 14.28% and NO_x emission increased by 22.84% for B40 blend at 75% of

rated load conditions, when compression ratio was increased from 16.5 : 1 to 18.5 : 1.

- (v) A practical conclusion can be drawn that all tested fuel blends can be used safely without any modification in engine. So blends of ethyl esters of Jatropha oil could be used successfully.
- (vi) On the whole it is concluded that Jatropha oil ester can be used as fuel in diesel engine by blending it with diesel fuel. Use of Jatropha oil can give better performance and reduced CO emission.

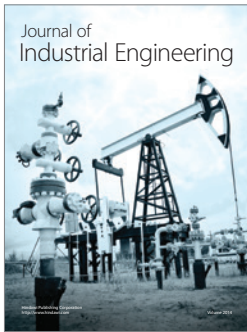
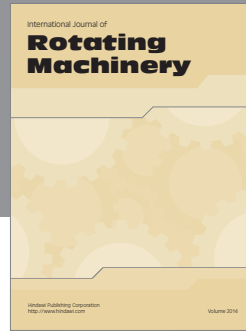
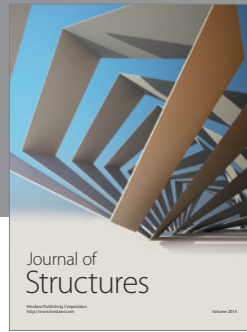
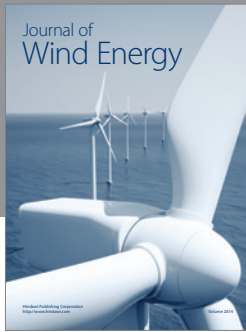
Conflict of Interests

The authors declare that there is no conflict of interests regarding the publication of this paper.

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