Experimental Analysis of VCR Engine Operated with Prosopis Juliflora Biodiesel Blends

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Abstract- The depletion of conventional fuel reserves, ambient conditions along with increased global warming and emission standards have emerged the research interests in utilisation of advanced combustion concepts on existing engines. This also made concern over the capability of using alternate energy sources along with conventional fuels. This research work indicates the results of analysis conducted to investigate the performance, exhaust emission and combustion characteristics of a VCR diesel engine fuelled with non-edible biodiesel at a rated speed of 1500 rpm with 300 bar injection pressure at three compression ratios. The test fuel was derived from Prosopis Juliflora seed oil methyl ester blends15% (B15) and 25% (B25) by volume. Biodiesel was produced by the transesterification process using methanol along with KOH as catalyst. The combustion characteristics investigated were rise in cylinder pressures, net heat release rate, cumulative heat release rate and mass fraction of fuel burned. The lower heat release rates, increased cylinder pressures were observed for both the blends compared to diesel. Increased brake thermal efficiency observed at higher compression ratio for B25 blend. It has also been observed that the emissions were decrease in trend with increase in compression ratios.

Keywords Thermal efficiency, Compression ratio, Combustion, Heat release rate, Ignition delay.

1. Introduction

Increase in demand regarding energy utilizations lead to major dependence on fossil fuels which are in a drastic depletion state, which in turn increased concern over renewable energy sources like bio fuels. Major concern for renewable energy sources is going on for diesel engines which occupy major portion in all engineering machinery like automotive and agricultural purposes because of their comparable thermal efficiencies and power outputs. Biodiesels derived from various non-edibles and edible oils are considered as alternative sources for diesel engines and various research works are in progressive stages to adopt these biodiesels as alternate source for diesel engines [1]. The primary advantages of these biodiesels are that they are non-toxic, renewable, void of sulphur traces and biodegradable in nature compared to conventional fossil fuels. The comparisons and tests conducted by using biodiesels by different researchers [2-3] shown the suitability of biodiesels as alternative fuel sources for diesel engines. Experimental analysis was also done by changing various operating parameters like varying injection pressures, timings and compression ratios. Senthil Kumar.K and Thundil kruppa Raj [4] conducted performance test on a Diesel engine using ethanol blended biodiesel. Experimental results with increased inlet air temperatures (40^0 to 60^0 C) and advanced injection timings (12^0, 15^0, 18^0 BTDC) showed the suitability of ethanol blended biodiesel as an alternative fuel with decreased HC and CO emissions. Sathish Kumar.R et al [5] used Manilkara Zapota methyl ester blend on Variable Compression Ratio (VCR) engine and obtained the results. The results revealed its suitability as an alternative substitute to replace pure diesel. The results showed that the use of these blends resulted in increased brake thermal efficiencies and decreased brake specific fuel consumptions. Performance, combustion and emission characteristics were studied [6] by using bael oil-diesel-diethyl ether blends on VCR engine. The results reveal that there was an increase in brake thermal efficiency by 3.5% and decreased nitrogen oxides by 4.7% by injecting fuel at 23^0 BTDC. Water emulsified hybrid Pongamia biodiesel was used by Varatharaju Perumal and Ilangkumaran.M [7] on single cylinder naturally aspirated diesel engine. The results show 9% increase in BSFC, 5% decrease in BTE along with decreased emissions like smoke, CO and NOx. Senthur Prabhu.S et al [8] conducted experimental investigations using blends of pre heated palm oil (20%), diesel and n-butanol with BHT (2000ppm) on DI diesel engine. The results obtained were increased BSFC and BTE with decreased CO, smoke and EGT with 1.9% increased NOx emission. The experimental work carried by Prakash.T et al [9] using blends of bio-ethanol (30%), diesel (30%) and castor oil (40%) show that the BTE was comparable to diesel with increased smoke emission. Experimental analysis done by Houssem El Haj Youssef et al [10] using the blend of diesel and waste cooking oil (20%) with 2^0 advanced injection timing. The results had shown increased performance with a reduction in smoke opacity. The blends of diesel (70%), gasoline (15%) and n-butanol (15%) showed increased maximum pressure rise rates with decreased ignition delay and combustion duration. The BSFC and CO emissions increased when compared to diesel, whereas there was reduction in NOx levels [11]. Oxygen enriched fuels like biodiesels promote increased combustion phenomenon [12, 13] which in turn results in less pollutants, when used in conventional and HCCI engines. Tsutsumi.Y et al [14] used ignition improvers like Di Ethyl Ether (DEE), Di Methyl Ether (DME) to biodiesel fuel blends at lower percentages lead to further reduction of emissions by allowing the fuel blends to evaporate immediately and also to maintain the blend cetane value to the required extent. The Pongamia oil blends with addition of DEE lead to reduction of NOx and smoke emissions [15]. Experimental analysis on diesel engine operated with methyl ester of Sesame oil shown higher BTE with lower emissions of HC, CO and NOx [16].

Zhang et al [17] used butanol with diesel and biodiesel blends, and observed reduced PM emissions at higher engine loads. Similar trend has been observed by Xiaoye Han et al [18] using n-butanol blends with diesel. The performance parameters obtained were comparable with diesel and shown reduction in NOx and smoke emissions. Sharifah Najiha Badar et al [19] concluded that Algae-derived bio-energy is one of the best alternative energy sources to replace fossil-based fuels. K. Vijayaraj and Sathiyagnanam.A.P [20] used blends of methyl ester of Cotton seed oil and studied combustion characteristics of a DI diesel engine. The results reveal that 25% of methyl ester of Cotton seed oil showed optimum combustion characteristics at all load conditions.

From the above literature survey it has been concluded that several edible and non-edible oils could be used as biodiesels in conventional diesel engines with little or no engine modifications. This is because of their higher cetane and oxygen percentages which can cause optimum combustion and heat release rates [21] by influencing the combustion efficiency.The Prosopis Juliflora plant is widely spread all over the world and it consists of 44 species in that 40 known species are native to America. This plant is plentifully available in arid regions of India, and is also remaining in several countries [22]. Its seed oil is non-edible in nature. Earlier reports have also suggested suitability of some non-edible oils in a diesel engine by blending with diesel. The novelty of this study is such that no analyzations are available on the use of PJME –diesel blends in a VCR CRDI diesel engine at various compression ratios. The present study aims at evaluating the performance, emission and combustion characteristics of a VCR CRDI diesel engine fuelled with PJME–diesel blends.

2. Materials and Methods

2.1. Preparation of Fatty Acid Methyl Esters

To prepare methyl esters by transesterification requires raw Juliflora seed oil, 15% of methanol & 5% of sodium hydroxide on a mass basis. The Juliflora seed oil was chemically reacted with an alcohol in the presence of a catalyst to produce methyl esters. The mixture was stirred continuously and then allowed to settle under gravity in a separating funnel. Two distinct layers form after gravity settling for 24 hours. The upper layer was of ester and the lower layer was of glycerol. The lower layer was separated out. The methyl ester was then blended with diesel in various concentrations for preparing biodiesel blends to be used in the test engine. The properties of PJME and its blends compared to diesel are shown in Table 1.

Table 1. Properties of test fuel compared to diesel

Properties	Diesel	PJME	B15	B25
Density(kg/m^3)	840	970	860	873
Calorific Value(MJ/kg)	42.80	40	42.38	42
Viscosity at 40^{0}C(mm^2/sec)	2.85	4.90	4.59	3.36
Cetane Number	46	49	46.45	46.75
Oxygen % by weight	0	12	1.8	3

2.2. Experimental Setup

The photographic view of experimental setup is shown in Fig. 1 and its technical specifications are shown in Table 2. Experimental investigations were carried out by using CRDI vertical single cylinder water cooled computerized direct injection VCR diesel engine with eddy current dynamometer at a rated rpm of 1500 with an injection pressure of 300 bar and with an injection timing of 280 BTDC. The VCR engine was provided with AVL DI GAS 444 N five gas analyzer and AVL 437C Smoke meter. This CRDI VCR engine works with programmable Open ECU for Diesel injection, fuel injector, common rail with rail pressure sensor and pressure regulating valve, crank position sensor, fuel pump and wiring harness. The setup enables study of CRDI VCR engine performance with programmable ECU at different compression ratios. Various necessary instruments were used along with experimental setup to measure crank angle, combustion rate, in cylinder pressure and temperatures and fuel flow rates. The signals from all the measuring instruments were connected to a computer through a data acquisition system.

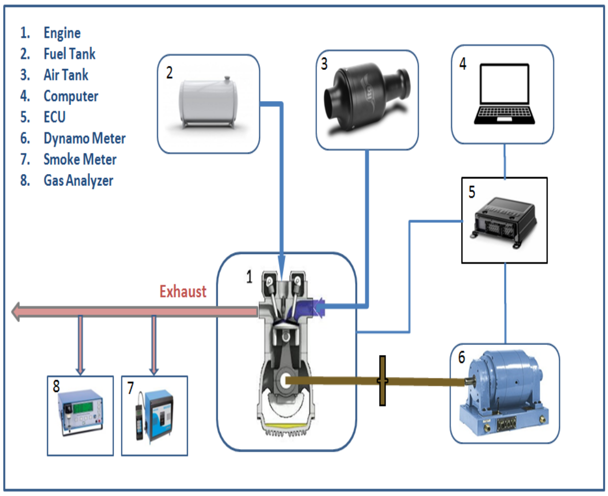

Fig. 1. Schematic view of Experimental setup

Table 2. Test engine specifications

Parameter	Specification
Type of engine	Kirloskar, CRDI, Four Stroke, Single Cylinder, Naturally aspirated diesel engine with eddy current dynamometer
Power	3.5 kW at 1500 rpm
Bore * Stroke	87.5 mm * 110mm
Compression Ratio	17.5 (VCR 12 to 22)
Injection pressure	300 bar
Injection timing	28^0 BTDC
Swept volume	661.45 (cc)

2.3. Experimental Procedure

The test fuel PJME was mixed with diesel in volume ratios of 15% and 25% and named as B15 and B25 blends. These blends were tested on VCR engine with compression ratios of 16, 18 and 20. A series of experimental trails were conducted on the test engine with test fuels and average value of all trail readings from no load to full load conditions was obtained. After allowing the engine to get stabilized with applied loads during each experimental trail, the emission, combustion and performance details were recorded by using the data acquisition system connected between experimental setup and computer.

3. Results and Discussion

3.1. Engine Performance Analysis

3.1.1. Brake thermal efficiency

This efficiency is described as the ratio of amount of heat actually converted to the brake power to the total heat supplied. There was no considerable variation in BTE at compression ratio 16 for B15 and B25 blends when compared to diesel, but it was improved in lateral stages with increase in compression ratios to 18 and 20. The BTE values for diesel, B15 and B25 at CR16 were 34.5%, 34.8% and 33.8% respectively. The variation of BTE with BP at CR16 is shown in Fig. 2, and it is clear that the BTE was more for the blends when load reaches the maximum. The increased in-cylinder pressures and temperatures at maximum loads for biodiesel blends lead to better evaporation and mixing with air, which resulted in the maximum BTE [23].

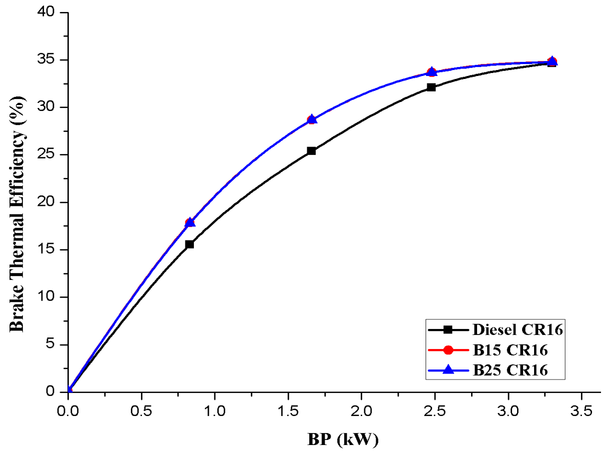

Fig. 2. Variation of BTE with BP at CR16

With increased compression ratios, the operating temperatures of the test engine were improved. The evaporation and mixing rates of blends into air were also promoted, resulting in improved BTE values. The BTE values at CR20 for diesel, B15 and B25 were 27.4%, 34.7% and 34.5% respectively at full load conditions. The BTE was improved by 20.5% compared to diesel for the blend B25 at CR20 at full load condition, because of higher compression ratio and improved oxygen percentage with this blend strength. It was also observed that BTEs were closer to each other for all blends with diesel at full load conditions because of closer heating values and cetane number. Higher cetane number and oxygen percentages in biodiesel blends resulted in better ignition quality which in turn increased the BTE. Variation of BTEs with brake powers at CR18 and CR20 is shown in Figs. 3 and 4.

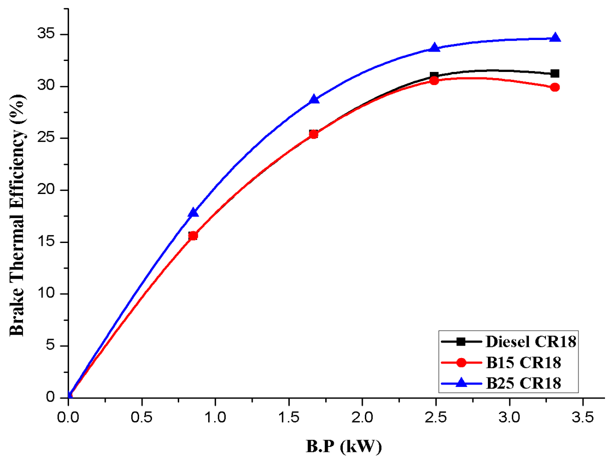

Fig. 3. Variation of BTE with BP at CR18

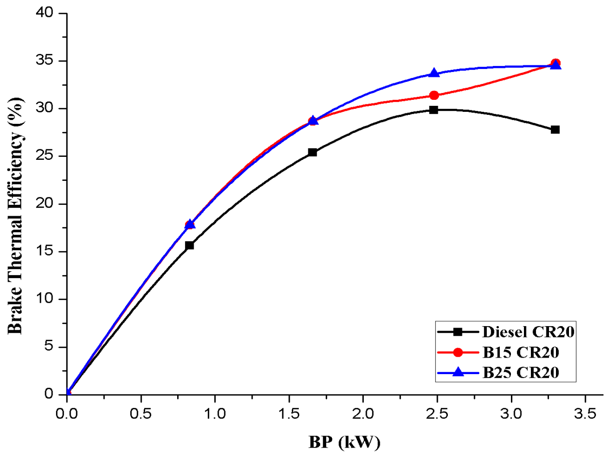

Fig. 4. Variation of BTE with BP at CR20

3.1.2. Brake specific fuel consumption

It is defined as the amount of fuel consumed per unit brake power output or it is the effectiveness to convert chemical energy of the fuel into useful work and it depends on fuel properties like viscosity, oxygen content, density, cetane number and heating value.

From the graph, it is observed that BSFC was reduced with an increase in load or at higher brake powers. The results also reveal that there was no considerable variation of BSFC for all compression ratios. The decrease in BSFC with increase in load was mainly due to variation of mixture strength from leaner to stoichiometric ratio. The lower heating value of all the blends is also considered to be another factor for increase of BSFC [24, 25]. The BSFC slightly increased for both the blends at CR16 when compared with diesel. The BSFC values were observed as 0.24, 0.28 and 0.28 kg/kW-hr for diesel, B15 and B25 respectively as shown in Fig. 5.

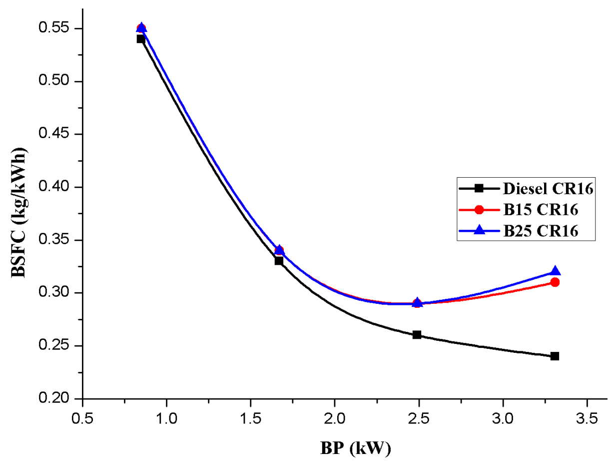

Fig. 5. Variation of BSFC with BP at CR16

It was also observed that BSFC for diesel slightly increased with increase in compression ratio, whereas no considerable variation in BSFC for B15 and B25 blends from CR16 to CR20. This opposite trend with increase in compression ratio may be due to higher operating temperatures at higher compression ratios that may lead to increased energy losses with increased frictional losses. Variation of BSFCs with BP at CR18 and CR20 is shown in Figs. 6 and 7.

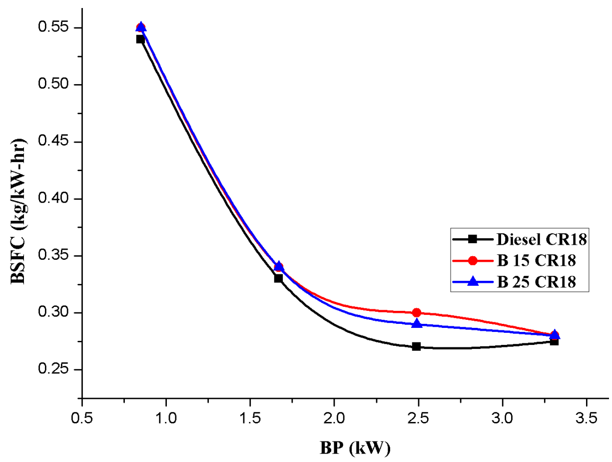

Fig. 6. Variation of BSFC with BP at CR18

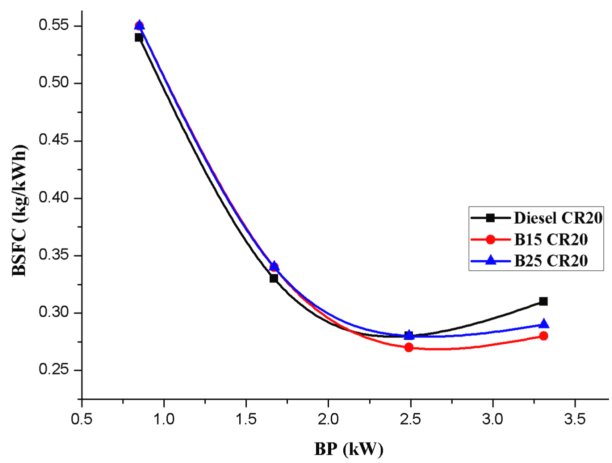

Fig. 7. Variation of BSFC with BP at CR20

The SFA content in PJME can also influence its cetane number near to that of diesel, since the cetane number depends upon the molecular structure of the fuel [26]. The SFAs like palmitic acid (C16:0) and stearic acid (C18:0) have straight chain molecular structure when compared to unsaturated fatty acids. The SFA content in PJME was 23%. In addition, the increased density of the blend can cause earlier injection of fuel allowing better mixing time and evaporation. It can influence better combustion in lateral stages of combustion with improved compression ratios.

3.2. Emission Analysis

3.2.1. Analysis of carbon monoxide

The main reason for CO emission is due to heterogeneous mixture formation, decreased reactive rates or slow flame propagation. The decreased oxygen percentage, rich mixture and incomplete combustion are also considered as collective reason for the formation of this emission. Variation of carbon monoxide emission with respect to brake power is shown in Fig. 8. Intermediate results were obtained at CR18.

The emission of CO increased with increase in blend strength and also increased with compression ratios compared to diesel at full loads. The higher unsaturated fatty acids led to poor oxidation and accumulation of more fuel droplets increased the carbon monoxide quantity. The increased blend strength also resulted in increase of fuel density and viscosity, which requires increased injection pressures. The same injection pressure and increased compression ratio resulted in decreased penetration of spray fuel droplets leading to poor combustion compared to diesel.

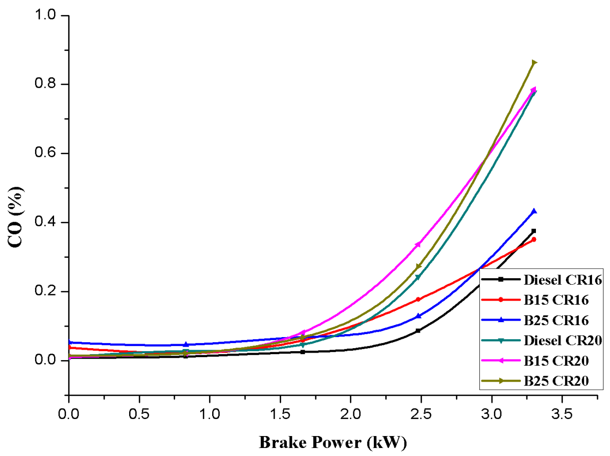

Fig 8. Variation of CO with BP at CR16 and CR20

The emission of CO at peak load conditions with CR16 for diesel, B15 and B25 was observed as 0.375%, 0.351% and 0.433%, and with CR20 it was observed as 0.777%, 0.786% and 0.864% respectively.

3.2.2. Analysis of hydrocarbons

The influencing parameters for this emission are fuel atomization, quality of air fuel mixture, combustion quality and operating parameters. At lower compression ratios the unburnt hydrocarbons (UHC) emissions were more for diesel and test blends, and at higher compression ratios, there was no considerable variation in UHC emissions. The UHC emissions for diesel, B15 and B25 blends at CR16 were 72ppm, 77ppm and 81ppm respectively. For increased compression ratios of 18 and 20, slight decrease in UHC emissions was observed. At CR20 the UHC emissions for diesel, B15 and B25 were observed as 69ppm, 68ppm and 80ppm respectively at full load conditions and intermediate results were obtained at CR18. Variation of hydrocarbon emission with respect to brake power is shown in Fig. 9 for CR16 and 20. Lower compression ratios led to lower operating temperatures which in turn reduces evaporating capacity of fuels causing increased UHC emissions particularly at higher blend strengths, and also the blend strength reduces the calorific value of the fuel and cetane number.

Increase in blend strength also increases the kinematic viscosity which in turn requires higher injection pressures. It was also observed that increase in compression ratios increased the operating temperatures leading to better evaporation and combustion rates, which further resulted in reduction of UHC emissions.

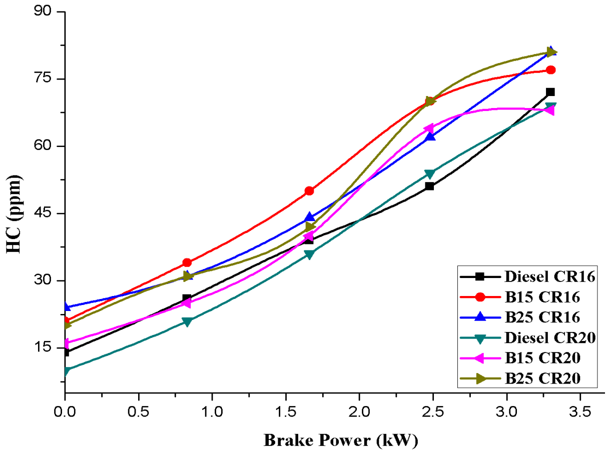

Fig. 9. Variation of HC with BP at CR16 and 20

3.2.3. Analysis of carbon dioxide

Higher carbon dioxide emission is possible either because of higher Oxygen quantity available in biodiesels [27] or because of carbon monoxide conversion to carbon dioxide after complete combustion of fuel at higher temperatures. The emission of carbon dioxide was slightly more for diesel compared to other blends at compression ratio 16, and it was almost same at compression ratio 20 meaning increase of compression ratio led to increased operating temperatures, resulting in increased combustion rate for both the blends. At lower compression ratios the evaporation rate for diesel was more compared to other blends. The increased viscosity and density of other blends led to incomplete evaporation rates causing heterogeneous combustion.

The CO_2 emission for diesel, B15 and B25 fuels at CR16 was observed as 7.84%, 7.6% and 7.69% respectively. At CR20, the CO_2 emission for diesel was decreased to 7.63% and for B25 it was decreased to 7.64%. Variation of CO_2 at peak loads with respect to increase in brake power is shown in Fig. 10 for CR16 and CR20.

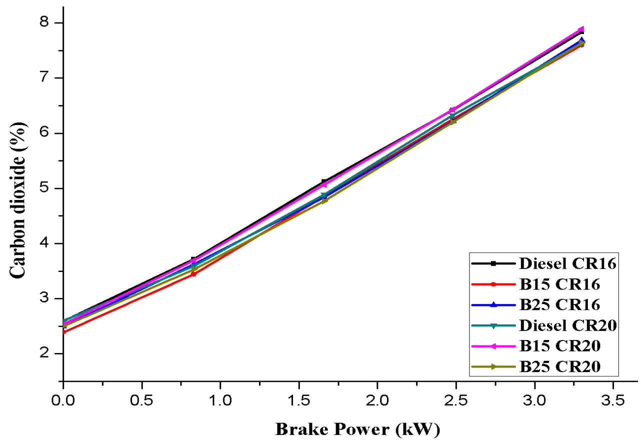

Fig. 10. Variation of CO_2 emission with BP

3.2.4. Analysis of nitrogen oxides

The mixing of oxygen with nitrogen at elevated operating temperatures is the primary cause for emission of NO and NO_2. This is because at higher operating temperatures the diatomic Nitrogen (N_2) is separated into monatomic Nitrogen (N), which is highly reactive with Oxygen and water vapors [28].

In this experimental analysis, initially NOx emissions were more at lower compression ratio 16 at peak load conditions for all blends along with diesel. Because of higher heating value, the diesel emits more NOx compared to B25. Variation of NOx emission with respect to brake power is shown in Fig. 11 for CR16 and 20. Intermediate results were obtained at CR18. The NOx emission at full load for CR16 was 2021ppm for diesel, and 1956ppm for B25 which is less by 3.21% compared to diesel. There was no considerable variation in NOX emission for B15 compared to diesel.

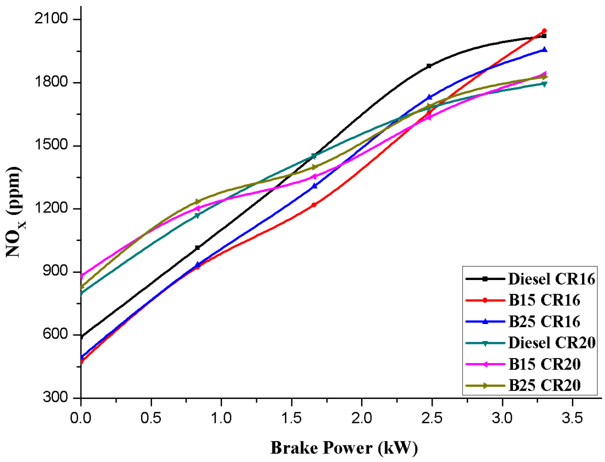

Fig. 11. Variation of NOx emission with BP

Lower heating values of B25 blend lead to lower NOx emission. But as the compression ratio increases to 18 and 20, the higher operating temperatures led to increasing trend of NOx emissions for both the blends compared to diesel. The NOx emissions for diesel, B15 and B25 at CR20 at full loads were 1796ppm, 1840ppm and 1828ppm respectively.

3.2.5. Analysis of smoke opacity

Generation of smoke emissions is due to insufficient mixing of air with fuel particles, and incomplete evaporation of fuel into air. Smoke emissions were in decreasing trend for blend B25 with increase in compression ratio because of increased operating temperatures led to increased evaporation rates. This decrease in trend of smoke opacity was also because of increased oxygen percentage with increased blend strength, resulting in the maximum carbon content to burn. The emission of smoke with respect to brake power is shown in Fig. 12.

It was observed that smoke emissions at CR20 for diesel, B15 and B25 were 63.7%, 62.4% and 54% respectively. It was reduced by 15.22% for B25 compared to diesel. It was observed as 56.7% at CR16 and 55.2% at CR18 for B25 blend.

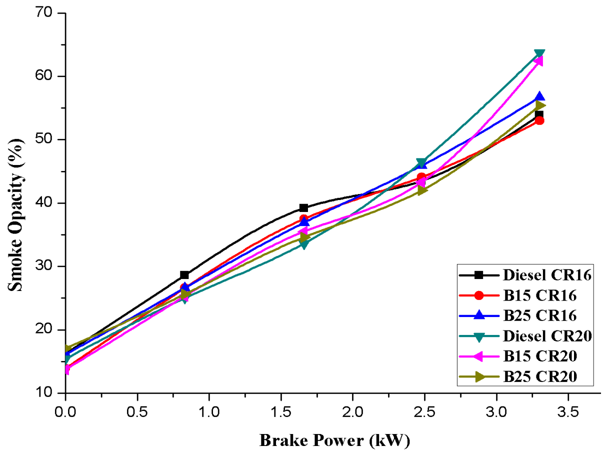

Fig. 12. Variation of Smoke Opacity with BP

3.3. Combustion Analysis

3.3.1. Analysis of blends cylinder pressure

The cylinder gas pressures depend upon the operating parameters of the particular engine and combustion characteristics. In-cylinder pressure directly related to power output. The in-cylinder pressure with respect to crank angle is shown in Fig. 13.

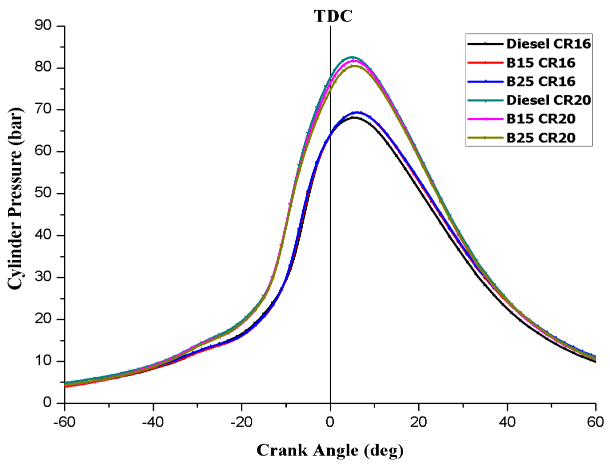

Fig. 13. Variation of Cylinder Pressure with Crank angle at CR16 and CR20

It had been observed that the peak pressures were slightly more for both the blends compared to diesel at compression ratio 16. The highest peak pressure was obtained for B25 followed by B15 and diesel. The value of peak pressure obtained for the blend B25 was 69.5 bar compared to 69.3 and 68 bars for B15 and Diesel respectively. The peak pressures were increased by 2.16% for B25 compared to Diesel. The maximum peak pressures were obtained by the result of uncontrolled combustion stage and availability of oxygen percentages in biodiesels.

The increased fuel supply with increased load conditions influenced the combustion rate with elevated temperature ranges was considered as one of the causes for improved cylinder pressures. Shorter ignition delay with increased blend strength oxygen content was also another cause for increase in peak pressures. In lateral stages of compression ratios of 18 and 20 the peak pressures were slightly reduced for the blends B15 and B25 because of improper protrusion of fuel droplets into compressed air with same injection pressure. Increase in viscosity and reduced cetane number can also influence spray atomization manner and ignition delay period in negative manner which can reduce the peak pressures in cylinder [29] for biodiesel blends.

3.3.2. Analysis of blends heat release rate

The heat release rate analysis is the better way to do in-depth analysis of combustion phenomenon. The net heat release rate depends upon the type of fuel used, injection pressure and compression ratio. It has been observed that for diesel HRR is more compared to other blends at all compression ratios, because of higher calorific value, evaporation rate and cetane number. The variation of HRR with respect to crank angle is shown in Figs. 14 and 15 at CR20 and CR16.

Negative HRR was observed before the start of combustion near 25^0 BTDC, because of evaporation of fuel caused heat to absorb from its surroundings in the combustion chamber space and also heat losses from engine cylinder walls [30, 31]. The Lower HRR for B15 and B25 blends has been observed as their increased viscosity with lower evaporation rate influences the physical delay period. The air temperature, pressure, air turbulence and velocity also can influence physical delay period [32]. The oxygen percentage can decrease the chemical delay period. The oxygen concentration of B25 blend resulted in more HRR compared to B15 blend. The B25 blend shown more HRR compared to B15 blend at all compression ratios because of increased ignition delay due to its decreased evaporation rate. This could result more quantity of fuel to accumulate in the combustion chamber, which was considered as another factor for increased HRR after the combustion process was initiated [33, 34].

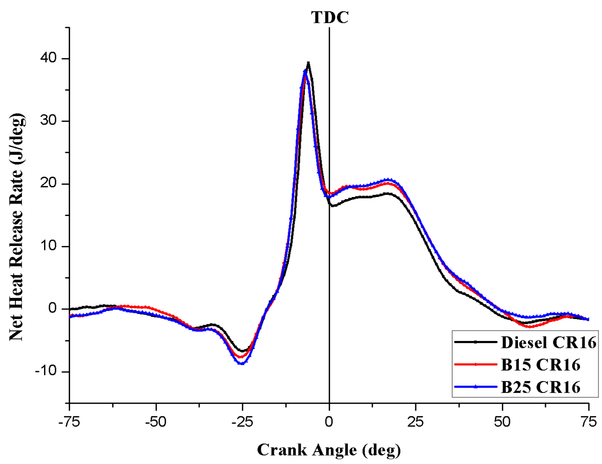

Fig. 14. Variation of HRR with Crank angle at CR16

The ignition delay further provides better fuel and air mixing time, which will reduce the heterogeneous mixture formations and allows better fuel atomization [35].

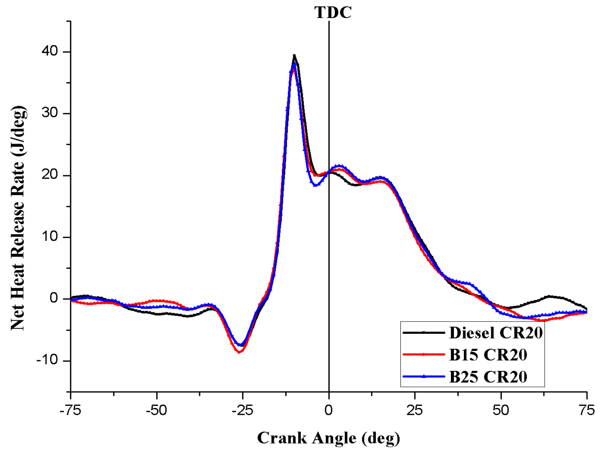

Fig. 15. Variation of HRR with Crank angle at CR20

The lower viscosity rates of biodiesel blends also influenced the atomization resulting in increased HRR [36]. At CR16, the maximum HRR for Diesel, B15 and B25 blends was found at 7^0 BTDC as 39.33 J/deg, 37.18 J/deg and 38.04 J/deg respectively. And at CR20, the maximum HRR for Diesel, B15 and B25 blends was found at 10^0 BTDC as 39.41 J/deg, 37.30 J/deg and 38.5 J/deg respectively at full load conditions.

3.3.3. Analysis of blends cumulative heat release

The cumulative heat release provides the information regarding progress of combustion and heat energy released by particular chemical nature of the fuel. Figure 16 shows the CHR with respect to crank angle for both the blends at CR16 and CR20 compared to diesel at full load conditions.

At the beginning of combustion both the blends followed the same trend like diesel, but as the combustion progresses the blend B25 showed higher CHR compared to diesel. This is because of B25's rapid rate of combustion due to its higher ignition delay compared to diesel. Presence of higher Oxygen percentage also considered as another cause for more CHR for B25 blend compared to diesel [37]. The similar trend has been observed for both the blends at CR20 compared to diesel.

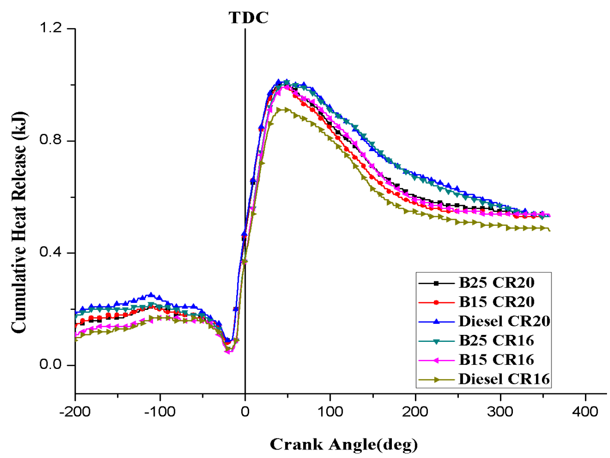

Fig. 16. Variation of CHR with Crank angle at CR16 and CR20

The primary factor influencing the CHR is ignition delay which is a combination of both physical and chemical delays, and can be defined as the time lag between initiation of injection and pre-ignition [38, 39].

The maximum CHR at CR16 was observed as 0.91kJ, 0.99kJ and 1.01kJ and with CR20 it was observed as 1.01kJ, 0.99kJ and 1.01kJ for diesel, B15 and B25 respectively at full load conditions. Intermediate results were obtained at CR18.

3.3.4. Analysis of blends mass fraction burned

The mass fraction burned is the amount of injected fuel burned to the total mass of fuel injected per cycle of the combustion process [40]. This is also used for to estimating time interval between the flame initiation and rapid rate of combustion [41].

Figs. 17 and 18 show the initiation of combustion from zero position by MFB curve and up to 100% means the end of combustion. The difference of zero to 100% indicates the period of combustion. The comparison of MFB at CR16 for both the blends with diesel is shown in Fig. 17 and at CR20 shown in Fig. 18. It is observed that from graphs, the initiation point of combustion for both the blends is same as that of diesel at CR16 and CR20. Increased oxygen levels for both the blends and cetane number near to diesel are considered as primary factors for initiation of combustion even at lower compression ratio 16. Even though the combustion duration of diesel was same as of the two blends, the burning process exceeded the combustion process of blends because of its higher combustible nature compared to the blends. As the CR value increased from 16 to 18 and 20, the blend B25 showed similar trend like diesel at higher loads because of increased operating temperatures and evaporation rates. This condition further influenced the mixing of fuel with air leading to reduction in the ignition lag period.

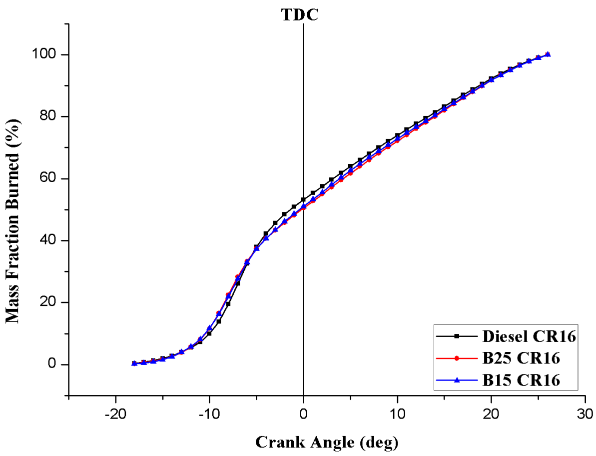

Fig. 17. Variation of MFB with Crank angle at CR16

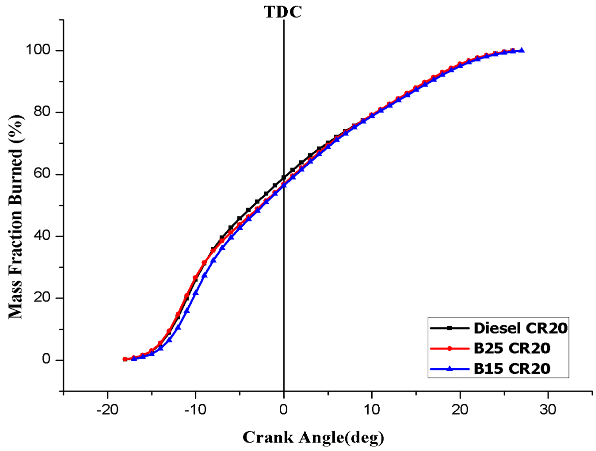

Fig. 18. Variation of MFB with Crank angle at CR20

4. Conclusions

The experimental work using PJME blends at compression ratios 16, 18 and 20 has drawn the following conclusions compared to diesel

- There is no considerable variation in combustion process using both the blends compared to diesel.
- The slight improvement in peak pressure occurred for B25 blend compared to diesel at lower compression ratio and with increase in compression ratios the peak pressures obtained were near to diesel for both the blends.
- The pressure rising trend was followed like diesel with variation in crank angle and no unusual behavior was observed.
- All the blends shown the HRR near to diesel at all compression ratios.
- The rate of combustion was faster with increased mass fraction burned for all the blends. The blend B25 followed the similar nature as of diesel with increased loads.
- The emissions for both the blends were slightly increased compared to diesel at lower compression ratios and they were in decreasing trend with increase in compression ratios.
- This analysis also concluding to concern over suitable injection pressure (especially for biodiesel blends having higher viscosity), which was maintained constant even compression ratios were increased.
- From all above it is proved that the PJME could be added to diesel in small proportions to save conventional fuel usage for diesel engines for some extent.

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Nomenclature

B15	15% Biodiesel, 85% Diesel	CR18	Compression ratio 18
B25	25% Biodiesel, 75% Diesel	CR20	Compression ratio 20
PJME	Prosopis Julyflora oil methyl ester	SFA	Saturated Fatty Acids
VCR	Variable Compression Ratio	BSFC	Brake Specific Fuel Consumption
CRDI	Common Rail Direct Injection	BTE	Brake Thermal Efficiency
CHR	Cumulative Heat Release	CO_2	Carbon dioxide
CR16	Compression ratio 16	NOx	Nitrogen Oxides
deg	Degree	HC	Hydro Carbons
SFC	Specific Fuel Consumption	ppm	Parts per million
BP	Brake power	BTDC	Before Top Dead Centre
UHC	Unburnt Hydro Carbons	MFB	Mass Fraction Burned
cc	cubic centimetres		
HRR	Heat Release Rates		

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