Potential for Using a Tyre Pyrolysis Oil-Biodiesel Blend in a Diesel Engine at Different Compression Ratios

Abhishek Sharma* and S. Murugan

Internal Combustion Engines Laboratory

Department of Mechanical Engineering

National Institute of Technology Rourkela

Rourkela-769008 (India)

Abstract

This study is aimed at investigating effects of varying the compression ratio at optimum injection timing and injection pressure on the behaviour of a diesel engine, using a non-petroleum fuel, i.e. a blend of 80% biodiesel, and 20% oil obtained from pyrolysis of waste tyres. The engine was subjected to one lower (16.5) and one higher (18.5) compression ratio in addition to the standard compression ratio of 17.5. At the higher compression ratio of 18.5 and full load, shorter ignition delay, maximum cylinder pressure and higher heat release rate were found for the blend, compared to those in case of the original compression ratio. The increase in the compression ratio from 17.5 to 18.5 for the blend improved the brake thermal efficiency by about 8% compared to that of the original compression ratio at full load. The experimental results indicated that for the blend at a higher compression ratio of 18.5, the brake specific carbon monoxide, brake specific hydrocarbon emissions and smoke opacity were reduced by about 10.5%, 32%, and 17.4% respectively, with respect to those of the original compression ratio at full load.

Keywords: Compression Ratio; Diesel Engine; Emission; Non Petroleum Fuel; Performance

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*Corresponding author

Email ID: drasharma58@gmail.com

Phone: +91-9861897954

1. Introduction

As petroleum based fuels are found only in limited reserves in the world, it has become imperative to explore alternative renewable fuels, which can be derived from other resources that are easily available in the country. The other issue is that the combustion of fossil fuels is the major source of global warming, ozone depletion and climate change and, it also has detrimental effects on human health harmfully [1].

Diesel engines are widely used in several applications, because of their lean operation, high thermal efficiency, lower fuel consumption and tendency to emit lower greenhouse gases compared to the spark ignition (SI) engines. Different pollutants emitted from compression ignition (CI) engines depend on many factors that include engine design parameters, operational conditions, fuel type, and exhaust emission after treatment employed [2]. In order to overcome these problems extensive research works were carried out in the last two decades. Several researchers suggested that the use of biofuels in small quantities with the conventional diesel fuel or as a sole fuel after a necessary fuel or engine modification would certainly help to solve these problems [3-4]. Biodiesel, in particular has a significant potential to be used as an alternative fuel for CI engines. It is the methyl or ethyl ester of fatty acids made from edible or non-edible vegetable oils, animal fats and algae. Many countries use different non-edible oils such as Jatropha curcas, Pongamia pinnata, Madhuca indica, Linseed and edible oils such as palm, soybean, sunflower oil etc. [5-9] for biodiesel production. The use of biodiesel in diesel engines results in significant reduction of unburned hydrocarbon (HC), carbon monoxide (CO), smoke opacity and particulate matter. As there is no sulphur in biodiesel, there is no or negligible oxides of sulphur emitted from the engine fueled with biodiesel. It contains about 10% oxygen in the molecules, which improves the ignition quality resulting in an enhanced combustion of the fuel inside the cylinder [10].

In spite of all these merits, the utilisation of biodiesel in CI engines is not proved to be a promising alternative fuel in many countries, because it is produced in less quantity. There are two possible options to solve this problem; one is replacing biodiesel by another non-conventional fuel which is derived from organic wastes and the other one is to increase the production rate. The former seems to be a better solution than the later, because it may reduce the disposal and environmental problems.

Waste automobile tyre is an organic waste from which useful energy in the form of liquid, gas or solid can be derived. Pyrolysis process, also termed as thermal distillation, is one of the methods employed for obtaining such fuels. The energy rich liquid, gas and solid products from pyrolysis of waste automobile tyres are referred to as viz., Tyre pyrolysis oil (TPO), pyro gas, and carbon black respectively [11]. Among these, the TPO can be a potential alternative fuel for CI engines but the main drawback is its lower cetane number, which is in the range of 25-30. In a preliminary research work that was carriedout in the past to study the effect of Jatropha methyl ester (JME)-TPO blend on diesel engine behaviour, five different blends (JMETPO10, JMETPO20, JMETPO30, JMETPO40 JMETPO50) were prepared. The numeric value indicates the volume percentage of TPO in the blend. Experiments were conducted in a constant speed, direct injection (DI) diesel engine, with a rated power of 4.4 kW at 1500 rpm. The test results confirmed that, the blend with 20% (by volume) TPO gave a better performance and lower emissions than those given by other blends. Addition of 20% TPO on to the blend resulted in substantial changes in the combustion, performance and emission characteristics of the engine. It has been observed earlier that the combustion commenced slightly later than in case of 100% biodiesel i.e. JME. However, it was found that brake thermal efficiency (BTE) improved by about 4.5% while brake specific energy consumption (BSEC) and brake specific nitric oxide emission (BSNO) reduced by about 1.9% and 7.9% respectively as compared to the case of using 100% JME [12].

Several researchers have reported on the optimum design parameters for diesel engines when fueled with alternative fuels, because the conventional diesel engine is designed only for diesel fuel. Most of the investigations have documented the form of blends with different alternative fuels, used in the existing diesel engine, or 100% biodiesel without any engine modifications [13-15]. Also, experimental investigations were carriedout to use higher percentage of biodiesel by altering the engine, and to find the optimum engine design parameters for a particular fuel [16-18]. Many researchers have reported results pertaining to the effects of varying injection timing, nozzle opening pressure and compression ratio on thermal efficiency, specific fuel consumption (SFC) and exhaust emissions of CI engines [19-22]. It has been reported that the engine performance may be improved by many ways, such as increasing the compression ratio, and nozzle opening pressure and advancing the fuel injection timing. The use of a higher compression ratio usually enhances the fuel-air mixture density, due to the increase in the pressure and temperature of the compressed mixture in the combustion chamber, leading to a rise in peak cylinder pressure and the burning speed of the fuel-air mixture. Experimental investigations were carriedout in the past to evaluate the CI engine characteristics at different compression ratios using various diesel-biodiesel blends [23-25].

Experiments were carriedout on a diesel engine using JMETPO20 (containing 80% JME + 20% TPO in volume basis) blend at varied injection timings and nozzle opening pressures. It was found by the authors that an advanced injection timing of 24.5 ˚CA bTDC and higher nozzle opening pressure of 220 bar improved the overall performance of the engine [26-27]. This experimental investigation was aimed to study the effects of operating the engine fueled with the JMETPO20 at different compression ratios, one higher (18.5) and one lower (16.5) in addition to the original compression ratio of 17.5 keeping the injection timing and nozzle opening pressure of the engine at optimum conditions.

2. Materials and methods

The TPO was blended with the JME on a 20/80% volume basis and the blend was kept under observation for 30 days, to ensure its stability. The details of preparation of JME and TPO have been described by the authors in elsewhere [12]. It was noticed that the TPO was not separated from the JME in the blend. Gas chromatography/Mass spectrometer (GC/MS) was used for analysing the composition of the blend, as shown in Fig.1. The GC-MS of the blend indicates that it contains compounds, like Pentadecanoic acid methyl ester, 10-Octadecenoic acid methyl ester, and Heptadecanoic acid methyl ester in large proportions. All the functional groups show the existence of oxygen, which is due to the presence of JME in the blend. Table 1 gives the comparison of the physico chemical properties of diesel, JME, TPO and the JMETPO20 blend.

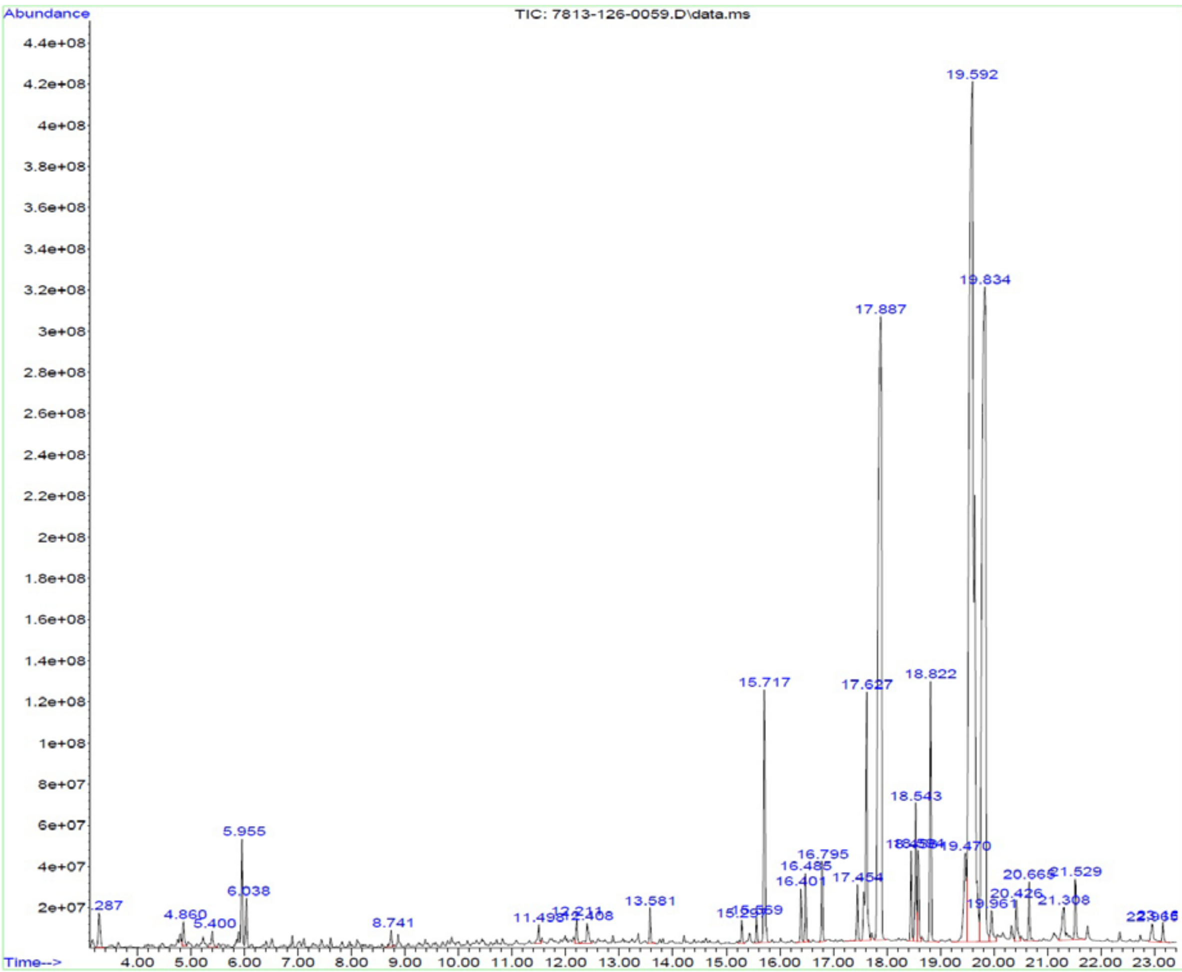

Fig. 1 GC-MS chromatogram of the JMETPO20 blend

Table 1 Physico-chemical properties of diesel, JME, TPO and the JMETPO20 blend

Properties	ASTM Test Method	Diesel	JME	TPO	JMETPO20
Specific gravity	D 4052	0.830	0.881	0.913	0.887
Viscosity (cSt)	D 445	2.6	5.6	3.35	5.2
Calorific Value	D 4809	43.8	39.4	38.1	38.82
Flash point (˚C)	D 93	50	156	49	132
Fire point (˚C)	D 93	56	171	58	145
Cetane number	D 613	50	55	28	52
Carbon (%)	D 3178	86.2	77.1	86.92	79.26
Hydrogen (%)	D 3178	13.2	11.81	10.46	11.31
Nitrogen (%)	D 3179	Nil	0.119	0.65	0.23
Sulphur (%)	D 3177	0.3	0.001	0.95	0.18
Oxygen by	E 385	Nil	10.97	1.02	9.02

3 Test details

The investigation was carriedout on a naturally aspirated, DI diesel engine, with a rated power of 4.4 kW at 1500 rpm. The technical specifications of the engine are listed in Table 2. Figure 2 shows the schematic layout of the engine experimental set up used in the investigation. The engine was coupled with an eddy current dynamometer for loading. The air consumption was measured using a sharp-edged orifice plate and U-tube manometer. A burette fitted with two optical sensors, one at a high level and, the other at a low level, was employed for measuring the fuel flow to the engine. The liquid flow through the high level optical sensor, gives a signal to the computer to start the time. Once the fuel reached the lower level optical sensor, the sensor would give the signal to the computer, to stop the time and refill the burette. The time taken for the consumption of fuel of a fixed volume was recorded. The engine exhaust gas temperature was measured using a K type (Chromel-

Aluminium) thermocouple connected to a digital indicator. The Kistler type piezoelectric pressure transducer was mounted on the cylinder head for the measurement of the cylinder pressure. A top dead centre (TDC) encoder was used to detect the engine crank angle. The engine setup was attached with a control panel, which had the capability to communicate with the pressure sensor, and to convert the signal from the pressure sensor to the analogue voltage signal, which was fed to the data acquisition system (DAS). The exhaust gas compounds such as CO, CO_2, HC, NO, and O_2 were measured with the help of an AVL DiGas 444 exhaust gas analyser. The smoke opacity of the exhaust gas was measured by an AVL 437 diesel smoke meter.

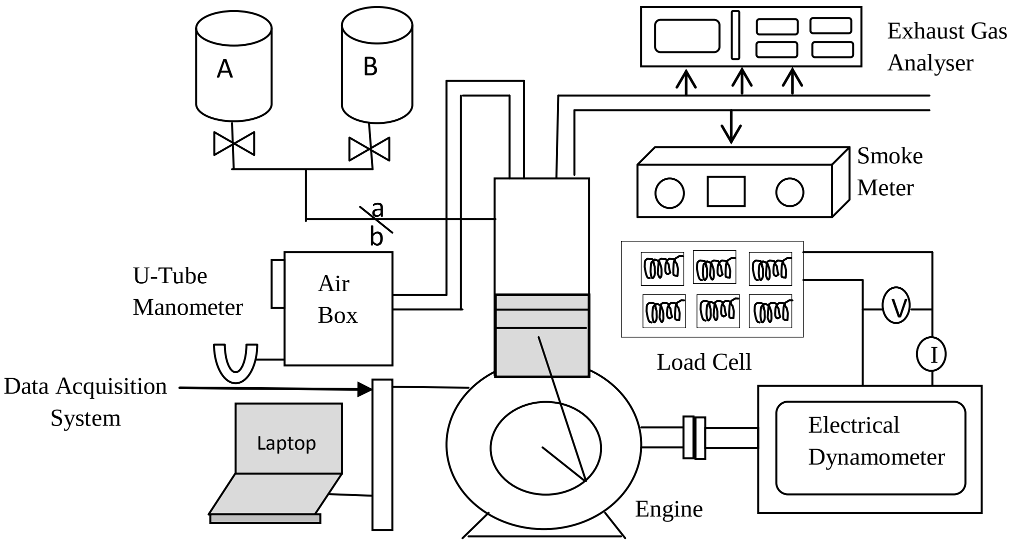

Fig. 2 Schematic layout of the experimental setup

The measurements of various parameters were recorded only after the engine attained the steady state. Each test was conducted for 3 times, ensuring the repeatability of the result. The values given in this study are the averages of these results. During the tests, the engine ran satisfactorily through the entire duration, and did not show any difficulty, when fueled with the JMETPO20 blend. Initially, the experiments were conducted using diesel and the JMETPO20 blend, under the original injection timing of 23°CA bTDC, nozzle opening pressure of 200 bar and compression ratio of 17.5, as set by the engine manufacturer for obtaining the reference data.

Table 2 Technical specifications of the test engine

Manufacturer	Kirloskar
Model	TAF 1
Engine type	Single cylinder, four stroke, constant speed, air cooled, direct injection, CI engine
Rated power (kW)	4.4
Speed (rpm)	1500 (constant)
Bore (mm)	87.5
Stroke (mm)	110
Piston type	Bowl-in-piston
Displacement volume (cm^3)	661
Compression ratio	17.5
Nozzle opening pressure (bar)	200
Start of fuel injection	23 ˚CA bTDC
Dynamometer	Eddy current
Injection type	3- Hole pump-line-nozzle injection system
Nozzle type	Multi hole
No. of holes	3

Further, the experiments were conducted with the advanced injection timing of 24.5˚CA and injection pressure of 220 bar, using the JMETPO20 blend for the compression ratios of 16.5, 17.5 and 18.5. The compression ratio of the engine was altered by changing the clearance volume, by the replacement of gaskets of different thickness in between the cylinder and the cylinder head. Fig.3 shows the photographic view of the gasket fitted with cylinder block. The compression ratio below 16.5 resulted in a poor performance, and a compression ratio above 18.5 was not attainable, owing to the engine structural constraint.

The steps involved in the calculation of the compression ratio are as follows:

$$\text{Compression Ratio (CR)} = \frac{\text{Maximum cylinder volume (Vs+ Vc)}}{\text{Clearance volume (Vc)}}$$

Maximum cylinder volume = Swept volume (Vs) + Clearance volume (Vc)

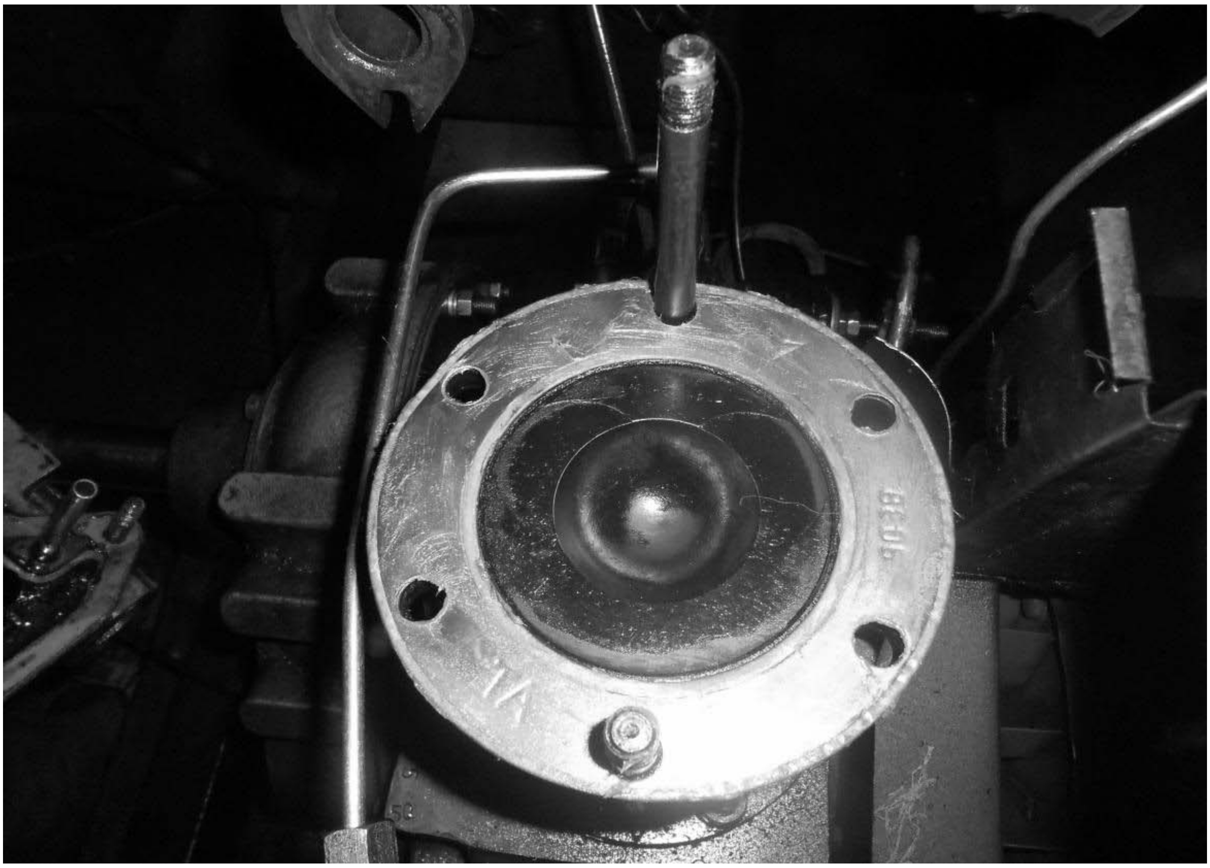

Fig. 3 Standard gasket fitted with cylinder block

$$Vs = \frac{\pi \ \mathrm{d}^2}{4} \times L$$ **(**Where, d = bore = 8.75 cm, L= Stroke = 11 cm)

$$CR = 17.5 = \frac{Vs + Vc}{Vc}$$

$$17.5 = \frac{Vs}{Vc} + 1$$

$$16.5 = \frac{Vs}{Vc}$$

$$V_c = \frac{V_s}{16.5} = \frac{661.45}{16.5} = 40.08\ cm^3$$

Gasket volume = 7.21 cm^3 (d= 8.75 cm, t= 0.12 cm)

For CR=18.5, Vc= $\frac{Vs}{17.5} = \frac{661.45}{17.5} = 37.79\ \mathrm{cm}^3$

Clearance Volume excluding gasket volume+ Gasket volume = 37.79 cm^3

32.87 + Gasket volume = 37.79 cm^3

Gasket volume required for compression ratio of 18.5 = 4.92 cm^3

Gasket thickness required = 0.08 cm

In the same manner, the gasket volume and thickness required for CR=16.5 was calculated. The calculated gasket volume and thickness corresponding to the different compression ratios are given below in Table 3.

Table 3 Gasket volume and thickness required for different compression ratios

Compression ratio	Gasket volume (cm^3)	Gasket thickness (cm)
16.5	9.8	0.16
17.5	7.21	0.12
18.5	4.92	0.08

4. Results and discussion

Compression ratio is known to have substantial impact on the behaviour of a CI engine. Therefore, the effects of the compression ratio on the combustion, performance and exhaust emissions of a single cylinder CI engine fueled with the JMETPO20 blend investigated experimentally and the test results are presented in the subsequent sections. The experiments were conducted at the advanced injection timing of 24.5˚CA and higher injection pressure of 220 bar, for the compression ratios of 16.5, 17.5 and 18.5, and the results are compared with those of diesel operation under standard test conditions.

4.1 Combustion analysis

The cylinder pressure crank angle diagram is employed to analyse the engine combustion behaviour, as the cylinder pressure has an effect on the performance parameters and emission levels of the engine. The variations of the cylinder pressure with respect to the crank angle

(CA) at different compression ratio, for diesel and the blend at full load are shown in Fig.4. The peak pressure in a CI engine depends primarily on the combustion rate in the initial stages, and is influenced by the fuel taking part in the premixed combustion phase [28].

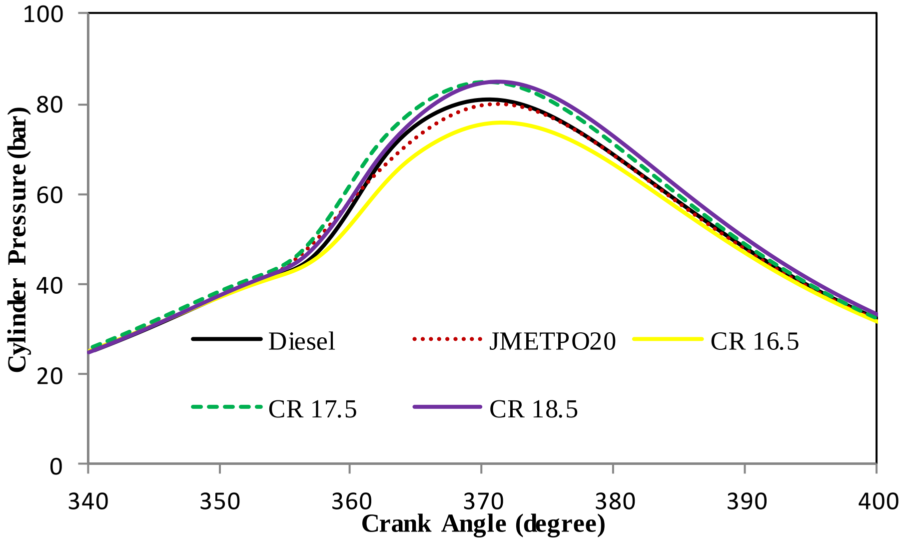

Fig.4 Cylinder pressure versus CA at different compression ratios

The cylinder pressures for diesel and the blend are obtained as 80.9 bar at 370.30 ˚CA and 79.9 bar at 371 ˚CA respectively, at the original compression ratio and full load. It is found that the combustion starts slightly earlier for the blend than for diesel at the original compression ratio. This is because of the higher cetane number and presence of oxygen in the blend, which results in improved combustion. The values of the maximum cylinder pressure for the blend at full load have been recorded as 75.8 bar at 371.7 ˚CA, 84.9 bar at 370.4 ˚CA and 85 bar at 371.4 ˚CA at compression ratios of 16.5, 17.5 and 18.5, respectively. It can be observed from the figure, that from a lower to a higher compression ratio, the maximum cylinder pressure is increased for the blend. The reason is that with the increase in the compression ratio, the intake air temperature increases, which provides better fuel atomization and mixture preparation with the air, and accelerates the complete combustion process [29]. The maximum cylinder pressure for the blend is found to be enhanced by about 6.2% at the compression ratio of 18.5 and full load, compared to that of the original compression ratio. For the blend at the lower compression ratio of 16.5, the maximum cylinder pressure is found to be lower compared to the original and the higher compression ratio, because of the relatively slower premixed combustion phase that ends up in a lower maximum cylinder pressure.

4.1.2 Ignition delay

Ignition delay (ID) is a period measured in terms of CA between the beginning of fuel injection and the beginning of combustion [30]. The ID depends on parameters, such as the fuel quality, atomization of fuel and duration of injection, air-fuel ratio, engine speed, cylinder gas pressure, intake-air temperature, injection pressure, and compression ratio [31]. Fig.5 compares the ignition delays of diesel and the blend with respect to brake power at three different compression ratios. As shown in the figure, as the load increases, the ID decreases for both the fuels at all compression ratios. This is because, as the engine load increases, the heat loss during compression decreases, resulting in higher temperature and pressure of the compressed air, and a shorter ID is obtained [32]. At the original compression ratio and full load, the values of the ID for diesel and the blend are about 11.5 and 11.4 ˚CA respectively. For the blend at full load, the values of ID are 10.1, 9.6 and 9.1 ˚CA at the compression ratios of 16.5, 17.5 and 18.5 respectively.

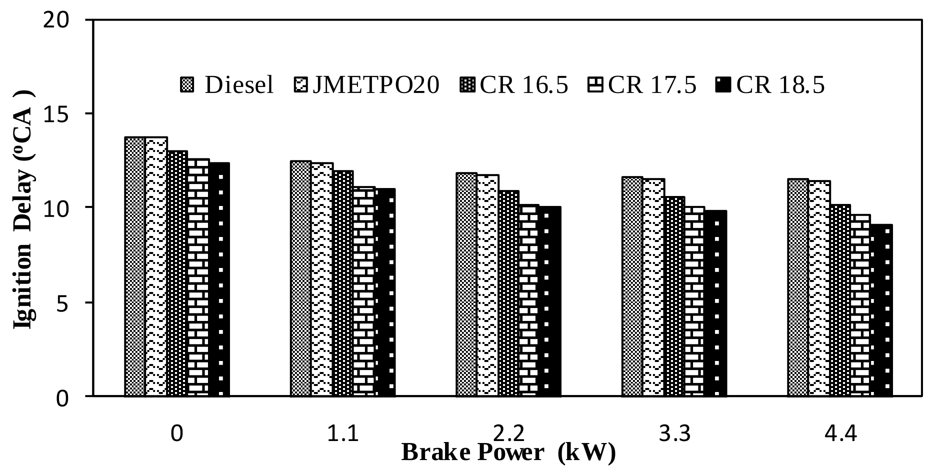

Fig.5 Ignition delay versus brake power at different compression ratios

The lowest value of ID is recorded as 9.1 ˚CA, at the compression ratio of 18.5 and full load for the blend and, it is found to be shorter by about 2.3 ˚CA, compared to the values at the original compression ratio. The increase in the compression ratio increases the compressed air temperature, which reduces the viscosity of the blend, by breaking down the intermolecular bonds, and decreasing the self ignition temperature of the fuel; and hence, the ID is shorter [33].

4.1.3 Heat release rate

The comparison of the heat release rate (HRR) curve at full load for diesel and the blend at different compression ratios is depicted in Fig.6. The HRR is an important parameter for the analysis of the combustion phenomenon in the engine cylinder, as the combustion duration and ignition delay can be easily estimated from the HRR-CA diagram. The HRR in this study was calculated, by using the cylinder pressure data [34]. The HRR at each ˚CA was determined by the following formula, which is governed by the first law of thermodynamics.

$$\frac{dQ}{d\theta} = P\frac{\gamma}{\gamma-1}\left(\frac{dV}{d\theta}\right) + \frac{1}{\gamma-1}V\frac{dP}{d\theta} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots Eq^{n}1$$

where dQ/dθ is the HRR (kJ/deg), P is the incylinder gas pressure (bar), V is incylinder volume (m^3), and γ is the ratio of specific heats.

Figure 7 indicates that, the value of the HRR is the maximum for diesel, compared to that of the blend at all compression ratios. This may be attributed to the higher calorific value of diesel; and more fuel accumulating owing to longer ID would increase the amount of fuel burnt during the premixed combustion phase, causing a higher HRR. At full load, the values of the HRR for diesel and the blend are found to be about 56.4 and 50.4 J/˚CA respectively, at the original compression ratio. At full load, the values of the maximum HRR in case of the blend are 42.3, 54.5 and 55.6 J/˚CA, for the compression ratios of 16.5, 17.5 and 18.5 respectively.

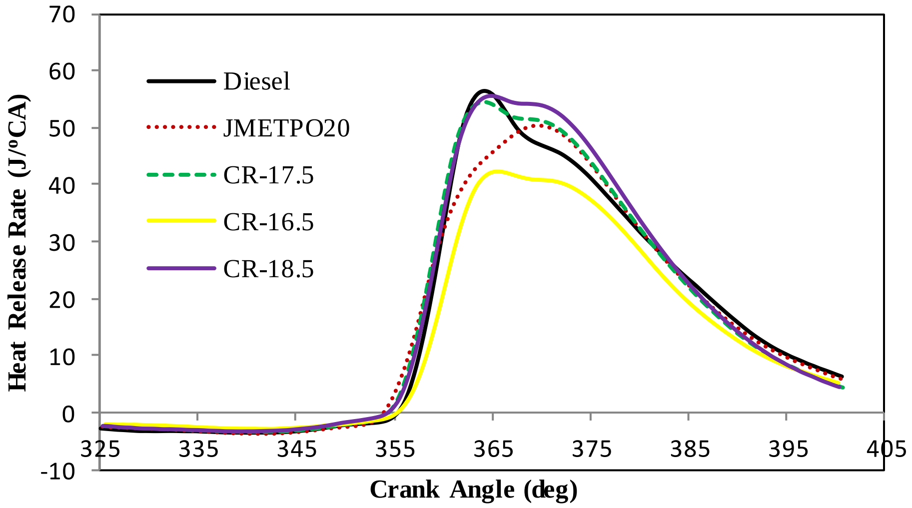

Fig.6 Heat release rate versus CA at different compression ratios

The HRR of 55.6 J/˚CA is obtained for the blend at the compression ratio of 18.5, which is 10.3% higher than that of the original compression ratio at full load. The higher compression ratio enhanced the HRR for the blend due to the reduction in viscosity, and this might promote a better spray formation as the intake air temperature increases with a higher compression ratio [35]. The lower HRR is observed for the blend at a lower compression ratio of 16.5, due to the slower air-fuel mixture formation, weak air entrainment and poorer combustion of the fuel.

4.1.4 Combustion duration

The combustion duration (CD) is described as the time duration required by the combustion process to reach 90% of its mass fractions burned [36]. Fig. 7 depicts the variation of the CD for diesel and the blend at different compression ratios. The CD becomes longer with the increase in the engine load for both the fuels, owing to the increase in the quantity of fuel injected. At the original compression ratio and full load, the values of the CD for diesel and the blend are found to be about 38.3 and 43.3 ˚CA respectively.

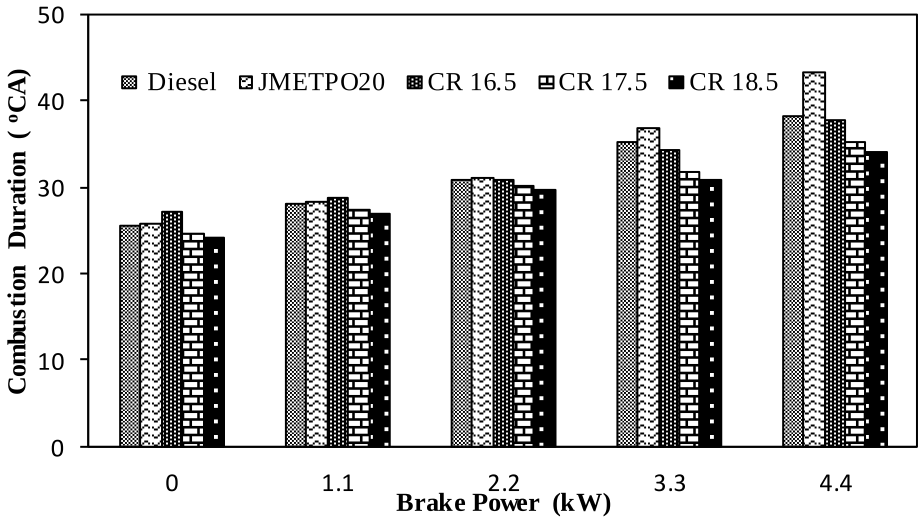

Fig. 7 Combustion duration versus brake power at different compression ratios

The longer CD obtained with the blend, at the original compression ratio is the result of an increase in the quantity of fuel consumed, to maintain the engine speed stable at different loads, as the calorific value of the blend is lower than that of diesel. The values of the CD are found to be about 37.9, 35.4 and 34.2 ˚CA, at the compression ratios of 16.5, 17.5 and 18.5 respectively, at full load. The lowest value of CD of 34.2 ˚CA is observed with the compression ratio of 18.5 for the blend, at full load. The increase in CD causes a rise in the in-cylinder air temperature and pressure that provides faster fuel vaporization, and accelerates the complete combustion process and a decrease in the CD [37].

4.1.5 Maximum rate of pressure rise

The variation of the maximum rate of pressure rise (MRPR) with brake power for diesel and the blend at different compression ratios, is shown in Fig.8. The MRPR $(dp/d\theta)_{max}$ in an engine combustion chamber has a substantial impact on the maximum cylinder pressure and smoothness of the engine operation. In general, it is considered that combustion is normal when $(dp/d\theta)_{max}$ is lower than 3 bar/ ˚CA whereas the engine is considered to be knocking, if is greater than 7 to 8 bar/ ˚CA [38]. The MRPR at the original compression ratio for diesel varies from 2.6 bar/˚CA at no load to 5.3 bar/˚CA at full load, and for the blend it varies from

2.7 bar/˚CA at no load to 5 bar/˚CA at full load**.** The MRPR is found to be rising with increase in compression ratio for the blend from no load to full load. The MRPR at full load is found to be the highest as 5.2 bar/˚CA for the blend at compression ratio of 18.5.

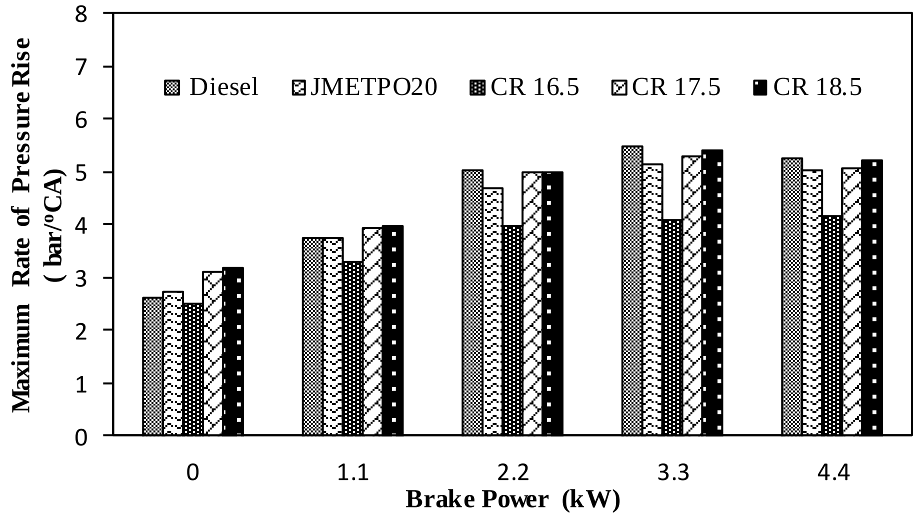

Fig. 8 Maximum rate of pressure rise versus brake power at different compression ratios

The oxygen enrichment in the blend due to the addition of JME accelerates the reactions, and this result in more complete combustion of fuels which is the cause for the increase in the maximum rate of pressure rise at a higher compression ratio [39]. The values of the MRPR at compression ratios of 16.5, 17.5 and 18.5, are found to be about 4.2, 5.1 and 5.2 bar/̊ CA respectively, for the blend at full load.

4.2 Performance analysis

4.2.1 Brake thermal efficiency

Figure 9 shows the effect of the compression ratios on the brake BTE for diesel and the blend. The BTE is given by the ratio between the power output and the product of the fuel mass flow rate and lower heating value of the fuel [40]. It is clear from the figure, that the BTE is found to increase considerably with an increase in the load as a reduction in the heat loss and increase in power are encountered at higher loads. The values of BTE for diesel and the blend are very close to each other. The results indicate that the BTE of diesel and the blend at full load were acquired as 29.9% and 29.8% respectively at the original compression ratio. Generally, increasing the compression ratio improved the BTE of the engine. This is due to the fact that at a higher compression ratio, the compressed air temperature is higher, which ends up in better combustion of the fuel.

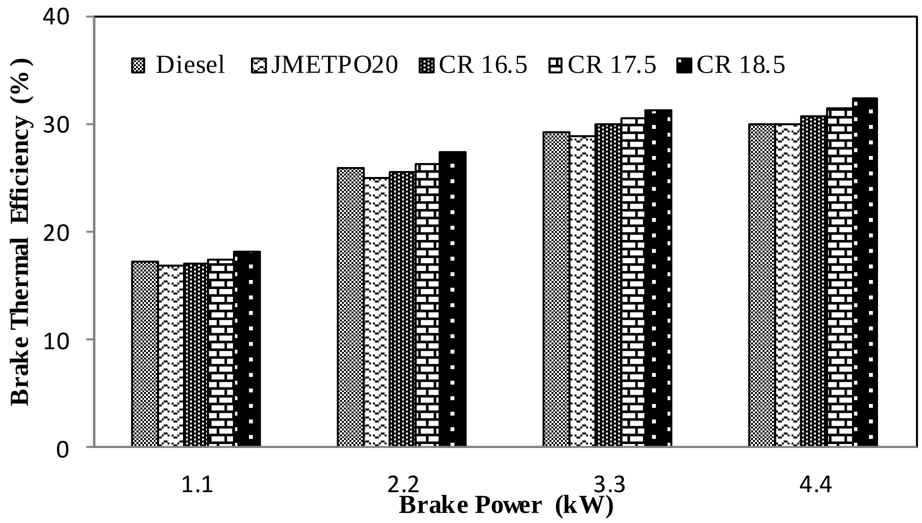

Fig. 9 Brake thermal efficiency versus brake power at different compression ratios

The BTE for the blend at full load was obtained as 30.7%, 31.4% and 32.3% at compression ratios of 16.5, 17.5 and 18.5, respectively. The blend has the highest BTE with 32.3% at the compression ratio of 18.5, and it is about 8% higher than that of diesel. The possible reason for this may be the proper mixing of the fuel-air, and improved fuel spray characteristics that occurred with the higher compression ratio of 18.5. At full load, the BTE for the blend at the compression ratio of 16.5 was decreased by about 2.2% and 4.8% compared to that of the compression ratios of 17.5 and 18.5, respectively. The mechanical efficiency of diesel and the blend were acquired as 84% and 83.2% respectively at the original compression ratio and full load. The values of mechanical efficiency for the blend were about 83.4%, 84% and 85% at compression ratios of 16.5, 17.5 and 18.5 and full load, respectively.

4.2.2 Brake specific energy consumption

The brake specific fuel consumption (BSFC) is not always a reliable factor when two fuels of different calorific values and densities are blended together [41]. The BSEC is described as the multiplication of the BSFC and lower calorific value of the fuel.

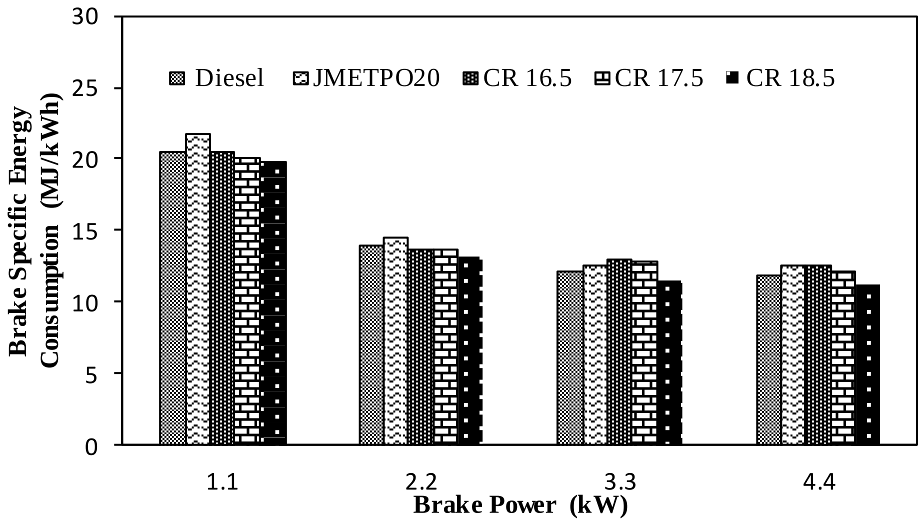

Fig. 10 Brake specific energy consumption versus brake power at different compression ratios

Fig. 10 illustrates the variation of the BSEC for diesel and the blend with the brake power at different compression ratios. At the original compression ratio and full load, the values of BSEC for diesel and the blend were recorded as 11.9 and 12.6 MJ/kWh respectively. The BSEC for the blend at full load was obtained as 12.5, 12.1 and 11.2 MJ/kWh at compression ratios of 16.5, 17.5 and 18.5, respectively. From the figure it is clear that while increasing the compression ratio of the engine the BSEC will be reduced for the blend. The lowest value of BSEC for the blend is found to be about 11.2 MJ/kWh, at full load and original compression ratio. The possible reason may be that higher compression ratio enhances the extent of evaporation and subsequently the combustion process. But, with the lower compression ratio, the BSEC is increased owing to incomplete combustion, resulting in a lowered power output and decreased BTE.

4.2.3 Exhaust gas temperature

The analysis of the exhaust gas temperature (EGT) gives qualitative information on the combustion of the fuel [42]. The variations of the EGT for different compression ratios are shown in Fig. 11. With the increase in engine load, the EGT is found to increase due to higher combustion temperature inside the cylinder as more fuel is burnt with increasing load. At the original compression ratio, the values of EGT are found to be about 303 and 335 ˚C for diesel, and the blend respectively, at full load. For the blend as the compression ratio increases, the EGT decreases. At full load the values of EGT for the blend are recorded as 310, 290 and 285 ˚C for the compression ratios of 16.5, 17.5 and 18.5 respectively. The blend has the lowest value of EGT at compression ratio of 18.5 and it is lower by about 18 ˚C as compared to that for diesel at full load.

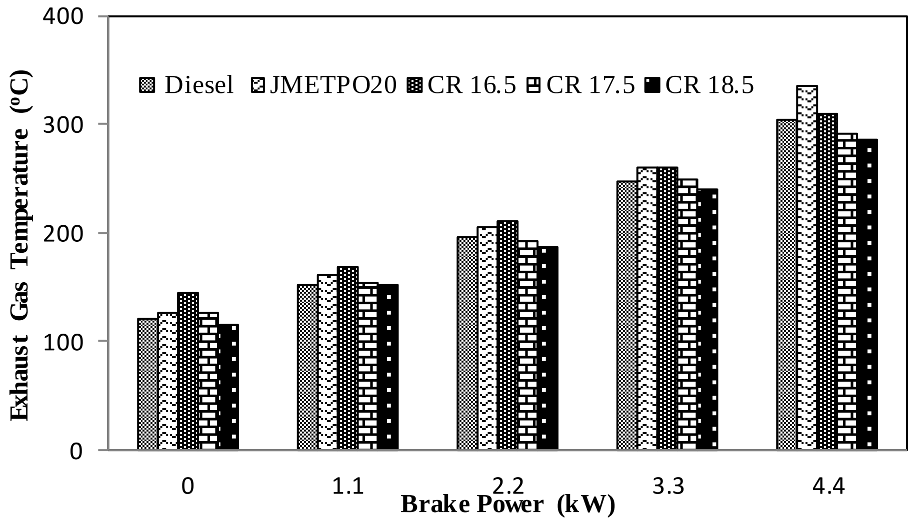

Fig. 11 Exhaust gas temperature versus brake power at different compression ratios

This may be due to the fact that air entered during the suction stroke at higher compression ratio is compressed, which increases the air temperature. The increased air temperature helps for better atomization of fuel which contributes in complete combustion and resulting reduction in the EGT. At lower compression ratio of 16.5 the EGT for the blend is higher as to more amount of heat is released during diffusion phase resulting in more amount of heat going along with exhaust gas.

4.3 Emission analysis

4.3.1 Brake specific carbon monoxide emission

It is known that the rate of CO emission is a function of the unburned fuel availability and mixture temperature, which controls the rate of fuel decomposition and oxidation. In the presence of sufficient oxygen, the CO emission is converted into CO_2 [43].

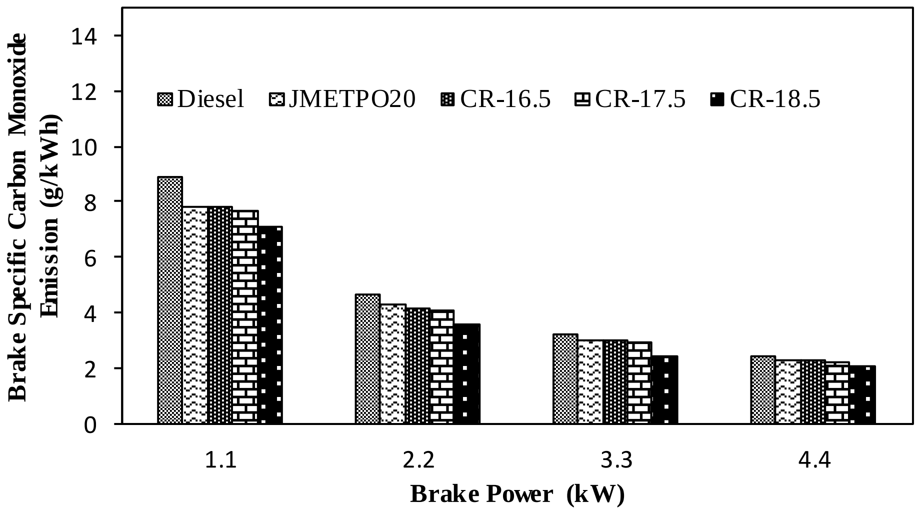

Fig. 12 Brake specific carbon monoxide emission versus brake power at different compression ratios

The brake specific carbon monoxide (BSCO) emission results are illustrated in Fig.12 at different compression ratios of the blend, in comparison to those for diesel. The BSCO emissions for diesel and for the blend at the original compression ratio and full load are about 2.5 and 2.3 g/kWh respectively. The BSCO for the blend at full load was obtained as 2.3, 2.2 and 2 g/kWh at compression ratios of 16.5, 17.5 and 18.5, respectively. The BSCO emission for the blend is marginally lower on increasing the compression ratio to 18.5, compared to the original compression ratio. This could be due to the fact that that the increased compression ratio increases the air temperature inside the cylinder subsequently reducing the delay period leading to better and more complete burning of the fuel and so lower BSCO emission [44]. The lower value of BSCO emission at full load for the blend was found to be 2 g/kWh, at the compression ratio of 18.5.

4.3.2 Brake specific hydrocarbon emission

The HC emission consists of fuel that is completely unburned or alone partially burned. The HC emission is influenced by the fuel-air mixing, and is abundantly affected by the overall air-fuel equivalence ratio, as the equivalence ratio varies broadly from very rich at the core of the spray to very lean at the spray boundaries [45]. The brake specific hydrocarbon (BSHC) emission results at different compression ratios for diesel and the blend are illustrated in Fig.13. The BSHC emissions for diesel and the blend at the original compression ratio are about 0.059 and 0.055 g/kWh respectively at full load.

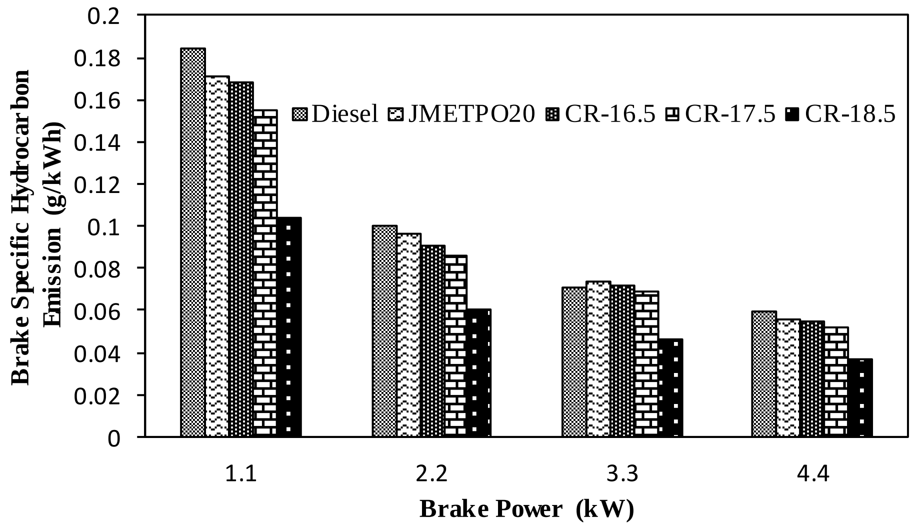

Fig. 13 Brake specific hydrocarbon emission versus brake power at different compression ratios

The figure shows that the BSHC emission of the blend, is lower at the higher compression ratio of 18.5. At the compression ratio of 18.5, the minimum BSHC emission of about 0.037 g/kWh is obtained with the blend, which is lower by about 32%, compared to the original compression ratio at full load. The increase in compression ratio enhances the air density and temperature in the cylinder, resulting in better fuel-air mixing in the combustion chamber, which contributes to the more complete combustion of the fuel. The BSHC emission for the blend was measured to be 0.055, 0.052, and 0.037 g/kWh, at the compression ratios of 16.5, 17.5 and 18.5, respectively and full load.

4.3.3 Brake specific nitric oxide emission

The formation of nitric oxide (NO) emission is highly dependent on the maximum temperature of burned gases during the premixed combustion phase, oxygen concentration, and the time available for the reactions to take place [46]. Fig.14 presents the BSNO emission values for diesel and the blend at different compression ratios. The BSNO emissions for the blend at full load are found to be about 3.9, 4.3, and 4.5 g/kWh, at the compression ratios of 16.5, 17.5 and 18.5, respectively.

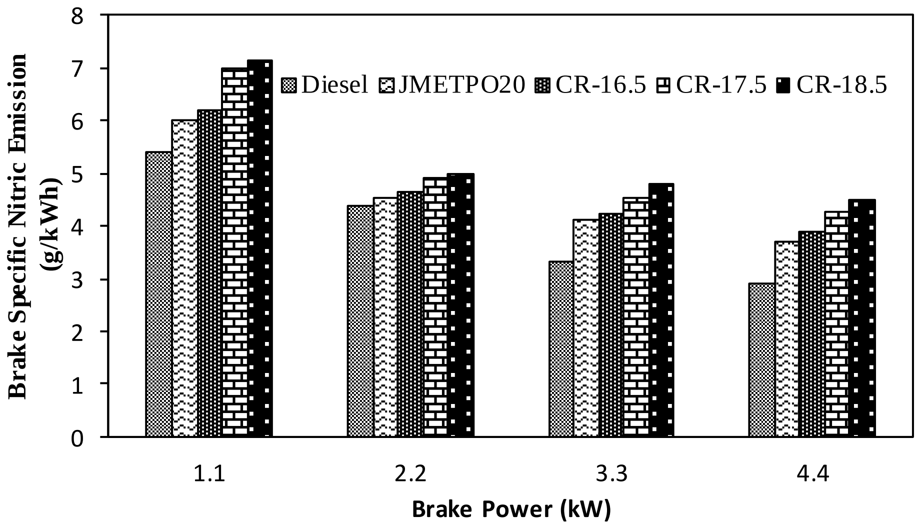

Fig. 14 Brake specific nitric oxide emission versus brake power at different compression ratios

As observed in this figure, for the blend, at the higher compression ratio of 18.5, the BSNO emission is boosted compared to the original and lower compression ratio. It is quite obvious that at the higher compression ratio, the temperature in the combustion chamber is expected to be higher due to improved combustion, and also the amount of oxygen present in the blend, results in higher amount of NO formation. The increase in the air intake temperature due to the rise in the compression ratio generates faster combustion rates, resulting in higher burned gas temperatures. The BSNO emission is decreased for the blend, while decreasing the compression ratio to 16.5, compared to that of the original compression ratio, because of lower premixed heat release rates, which cause a lower combustion temperature. In India, as per emission norms of central pollution control board, the acceptable range for BSCO, BSHC and BSNO emissions is 3.5, 1.3 and 9 g/kWh respectively for stationary diesel engine.

4.3.4 Smoke opacity

The smoke formation depends mainly on the incomplete burning of the hydrocarbon fuel, and partially reacted carbon content in the liquid fuel [47]. The results of smoke opacity are depicted in Fig.15 at different compression ratios. It is apparent from the figure that the smoke opacity grows with rise in the engine load due to the overall richer combustion, longer duration of the diffusion phase and reduced oxygen concentration [48].

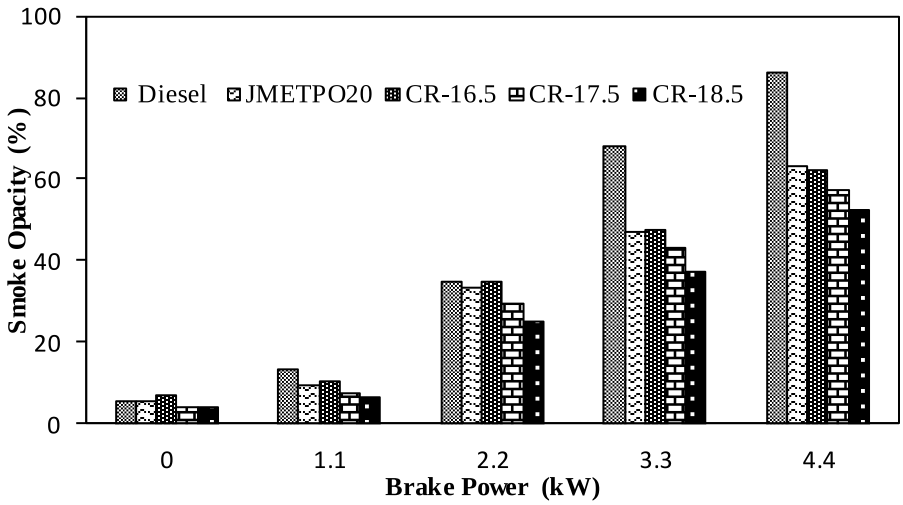

Fig. 15 Smoke opacity versus brake power at different compression ratios

At the original compression ratio, the smoke opacity for diesel and the blend is about 86.3% and 63.1% respectively, at full load. At the original compression ratio for the blend, the smoke opacity is relatively less in comparison with diesel, due to the presence of oxygen in the blend that contributes to a complete fuel oxidation. This actually leads to a significant drop in smoke opacity. For the blend at the compression ratio of 18.5, the smoke opacity is lower by about 17.4%, compared to that of the original compression ratio at full load. At full load, the values of smoke opacity for the blend are about 62.3%, 57.1%, and 52.1%, at the compression ratios of 16.5, 17.5 and 18.5, respectively. The smoke opacity reduced at the higher compression ratio of 18.5 compared to the original and lower compression ratio, because as the compression ratio increases, the combustion temperature increases due to improved fuel atomization, and this leads to the reduction in smoke opacity.

5. Conclusions

Experimental investigations were carried out to study the behaviour of a single cylinder, four stroke, air cooled, constant speed, DI diesel engine running on JMETPO20 blend, at varying compression ratio from 16.5 to 18.5, has been made. The conclusions of the experimental investigation are as follows:

- The maximum cylinder pressure and heat release rate at the compression ratio of 18.5 were higher by about 6.2% and 10.3% respectively, than those of the original compression ratio.
- The ignition delay period decreased by about 2.25 ˚CA at the compression ratio of 18.5 than that of the original compression ratio.
- It is found that increasing the compression ratio of the engine, the brake thermal efficiency is enhanced irrespective of the engine load. The maximum brake thermal efficiency obtained at the compression ratio of 18.5 is higher by about 8% than that of the original compression ratio. Also it was found that at the compression ratio of 18.5 the BSEC of the engine running with the blend was reduced by about 11% compared to original compression ratio.

- The reduction in the BSCO, BSHC emissions and smoke opacity by about 10.5%, 32%, and 17.4% respectively, is obtained at the higher compression ratio of 18.5, compared to that in case of the original compression ratio.
- The brake specific nitric oxide emission is greater by about 20% at the compression ratio of 18.5 compared to that of the original operating condition.

The above experimental findings suggest that the combustion, performance and emission characteristics for the JMEPTO20 blend are relatively better at the higher compression ratio of 18.5 as compared to those at standard operating conditions. Although there is a small increase in the BSNO emission, it still lies within the acceptable range and is quite comparable with that of diesel.

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