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Performance and emission analysis of multicylinder common rail direct injection diesel engine powered with blends of tyre pyrolysis oil-ethanol diesel

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ABSTRACT

This paper mainly focuses on optimal replacement of diesel by Tyre Pyrolysis Oil-Ethanol blends to run a common rail direct injection (CRDI) multicylinder engine and compare the results with neat diesel fuel operation results at 1500 rpm. The engine was operated at different loads and speeds. From the experimental study, decrease in brake thermal efficiency (BTE) and oxides of nitrogen (NOx) emission with increase in carbon monoxide (CO), hydrocarbon (HC), and smoke emission was observed for blends as compared to diesel fuel. Further, it was also observed that increase in speed from 1200 rpm to 1800 rpm yielded higher BTE with decrease in CO, HC, and smoke emission. But a slight increase in NOx, peak pressure (PP), and heat release rate (HRR) was reported at higher speed as compared to lower speeds. Beyond engine speed of 1800 rpm, knocking was observed.

Keywords: Tyre pyrolysis oil (TPO); Common rail direct injection (CRDI) engine; Ethanol; Engine performance.

INTRODUCTION

Fast depletion of diesel besides stringent emissions norms being implemented by authorities of various countries around the globe has pushed researchers to look into feasible solutions to aforesaid problems of diesel engines (Dawodu et al., 2014; Afzal et al. 2018 & Khan et al., 2014). These two problems have boosted the research in the field of alternate fuels to replace diesel (Alenezi & Al-Anzi, 2013 & Avulapati et al., 2016). In recent years, a lot of research activity is being carried out to recover energy from waste materials such as rubber and plastics that have high energy density (Wang et al., 2010). The waste from tyres and tubes of automotive vehicles poses big environmental problem as it is not biodegradable (Jang et al., 1998). Pyrolysis has been regarded as a potential way towards sustainable economy for management of this type of waste (Martinez et al., 2020). Pyrolysis of tyres could minimize the dependency on fossil fuels besides addressing the concern related to climate change (Antoniou et al., 2013; Martínez et al., 2020). Pyrolysis is a thermochemical process, which converts waste to useful energy under oxygen free environment (Elbaba et al., 2012), and this process was demonstrated by many researchers (Murugan et al., 2008 & Sharma et al., 2013). The end product of tyre pyrolysis process yields three valuables: Tyre pyrolysis oil (TPO), pyro-gas, and carbon black (Arya et al., 2020). TPO alone cannot be used directly in diesel engine without its proper upgrading (Aydın et al., 2012, Ahoor et al., 2014 & Martínez 2014 et al.). The cetane number of TPO is below 30, which is an important fuel property for ignition of fuel (Hossain et al., 2013 & Van de Beld et al., 2013). TPO can be blended with diesel, which improves the cetane number, and then can be used in existing diesel engine (Murugan et al., 2008 & Frigo et al., 2014). Hariharan et al. (2013) studied the effect of diethyl ether on the performance of diesel engine run with TPO and reported 5% reduction in formation of oxides of nitrogen (NOx) as compared to diesel engine run with neat diesel. Different approaches have been reported on utilization of TPO in diesel engine like blending TPO with diesel (Frigo et al., 2014, Martínez et al., 2014; Aydın et al., 2015 & Seljak et al., 2015) or biodiesel (Sharma et al., 2015 (a); Sharma et al., 2015(b)), increasing the intake temperature of air (McNeil et al., 2012), increasing compression ratio (Van de Beld et al., 2013),varying injection timing and pressure (Sudershan et al., 2018a), artificial neural network (ANN) modeling (Khandal et al., 2020).

An experimental study on CRDI engine run with diesel blend fuel (5% ethanol–20% biodiesel–75% diesel) revealed that ethanol as an additive for the biodiesel–diesel blend does not need any modification of CI engine. A blend fuel 20% bioethanol, 10% rape seed methyl ester, and 70% diesel used in Euro 5 CI engine revealed reduction of smoke and NOx at all tested conditions and an increase of CO and HC, especially at partial loads. Blend of 10% ethanol, 10% microalgae oil, and 80% diesel was found to be homogenous and stable without using a surfactant with improved properties of the resulted blend.

An experimental work reported that the best operating parameters for maximum BTE besides lowered emissions were injection timing of 10^{o}bTDC and injection pressure of 900 bar with fuel combinations of 70% plastic pyrolysis oil (PPO) with 28% diesel and 2% ethanol (Sudarshan et al., 2018 b). Phase separation problem was encountered when percentage of ethanol was more than 15% in the blend (Gadwal et al., 2019). The presence of hydroxyl group in ethanol reduces particulate emissions of diesel engines. Blends of diesel-ethanol-fueled multicylinder diesel engine showed higher brake specific fuel consumption (BSFC) and improved brake thermal efficiency (BTE) for higher ethanol content in the blend. This work also revealed that carbon monoxide (CO) and smoke emission were reduced remarkably at higher loads (Armaset al., 2006 & Tomicet al., 2013). The usage of ethanol with diesel in farm equipment engine resulted in lower smoke and particulate emissions (Kwanchareonet al., 2007). A study with blend of ethanol and diesel on diesel engine showed lower BTE, CO, hydrocarbon (HC), and smoke emissions with higher NOx emission (Sayin, 2010). Influence of blends of ethanol and biodiesel showed higher BTE with lower NOx and particulate matter (PM) emission in comparison with diesel fuel. BSFC, CO, and HC emissions increased with increase in the proportion of ethanol in the blends (Banapurmath et al., 2010).

Motivation for the present study and objectives

The detailed literature review revealed that TPO has been blended with diesel or biodiesel. There are some studies that used TPO-ethanol blend with diesel and biodiesel as pilot injection fuels (Lee et al., 2013 & Lee et al., 2015). As scarce literature is available with blends of TPO-ethanol-diesel, an attempt is made to use blends of TPO, diesel, and ethanol as substitutes to neat diesel fuel to power diesel engine. The blends were prepared with different proportions of ethanol to study their effect on multicylinder engine performance, combustion, and tail pipe emissions.

The experiments were conducted on multicylinder CRDI diesel engine for varied ethanol-diesel proportions at different speeds, and engine performance was compared with diesel fuel operation to make critical inferences.

MATERIALS AND METHODS

The crude tyre oil obtained from pyrolysis process contains impurities such as carbon particles, sulfur and moisture. The steps followed to reduce the sulfur and moisture content are as follows: treatment of hydrosulfuric acid, treatment with activated Bentonite and calcium oxide (CaO), distillation by vacuum process, oxidative desulfurization, washing, and drying.

The first three steps reduce the impurities, and the last two steps reduce the sulfur content and moisture. 8% of sulfuric acid by weight was stirred well with 1000 ml of crude oil for 4 hours and heated till temperature reaches 50^{o}C. The mixture was then allowed to settle the sludge for 40 hours. Two layers were formed after specified period with top layer as viscous oil and lower layer sludge. The clear viscous oil was treated with activated Bentonite and CaO by mixing with stirrer and it is maintaining at 70^{o}C for 4 hours. After 24 hours, the whole content was filtered with cloth to obtain healed pyro oil. Distilled pyro oil was then heated in vacuum distillation. This distilled pyro oil was further treated with oxidative desulfurization process to reduce sulfur content. This was carried out using 100 gms of mixture containing 10 gms of 98% formic acid and 30% hydrogen peroxide mixed for 2 hours at 60^{o}C. This mixture was allowed to settle in one whole day. Clear pyrolysis oil got separated by separation funnel and further washed with distilled water. Finally, the clear pyrolysis oil was heated to a temperature of 110^{o}C for about 30 minutes to remove moisture contents if any. The final product obtained was tyre pyrolysis oil (TPO). The flow chart of the entire process is shown in Figure 1.

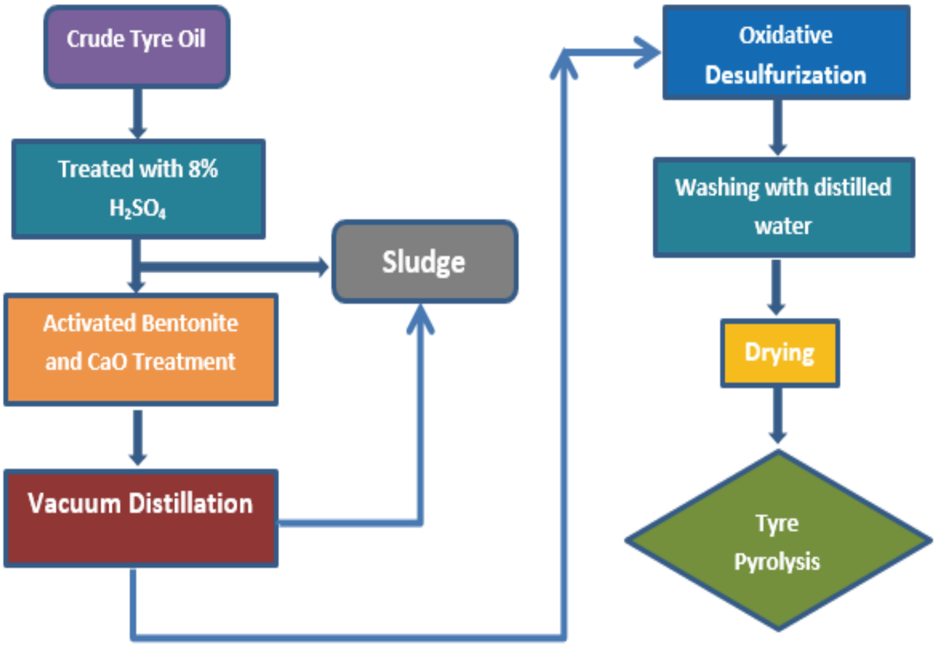


Figure 1. Flow diagram of TPO purification.

Table 1. Properties of base fuels.

Property	**Method**	**Ethanol (E)**	**TPO (T)**	**Diesel (D)**
Density (kg/m^3)	ASTM D-4052	789	890	819
Kinematic viscosity (cSt)	ASTM D-445	1.21	3.2	2.94
Flash point (^oC)	ASTM D-9390	14	48	57
Fire point (^oC)	ASTM D-9390	18	56	64
Calorific value (kJ/kg)	ASTM D-2015	26843	41430	44189
Cloud point(^oC)	ASTM D-4052	-7	4	2
Pour point (^oC)	ASTM D-2500	≤ -35	≤ -8	-16
Cetane number	ASTM D-613	8	47	52
Carbon residue %	ASTM D-524	-	0.02	0.35

Table 2. Properties of Blends.

Blends →	**Method**	**E: T: D 02: 60: 38**	**E: T: D 06: 60: 34**	**E: T: D 12: 60: 28**
Property				
Density (kg/m^3)	ASTM D-4052	848.4	839.3	832.2
Kinematic viscosity (cSt)	ASTM D-445	3.0	2.98	2.96
Flash point (^oC)	ASTM D-9390	48	40	35
Fire point (^oC)	ASTM D-9390	55	47	41
Calorific value (kJ/kg)	ASTM D-2015	43122	41150	39582
Cloud point(^oC)	ASTM D-4052	3	2	2
Pour point (^oC)	ASTM D-2500	-2	-4	-6
Cetane number	ASTM D-613	48.2	45.3	43.4
Carbon residue %	ASTM D-524	0.32	0.30	0.23

EXPERIMENTAL METHODOLOGY

Figure 2 depicts multicylinder CRDI engine test rig. Tests on CRDI diesel engine were carried out at different engine speeds of 1200 rpm, 1500 rpm, and 1800 rpm, and engine was operated from partial load to full load conditions. The blends were prepared with the combination of Ethanol-Pyrolysis oil-Diesel (E-T-D) in various proportions such as 02:60:38, 04:60:36, and 12:60:28, respectively, on volume basis. In blends, pyrolysis oil proportion was kept constant. For selected fuel combinations, experimentations were conducted in series for each combination. Six readings were recorded and averaged in order to analyze BTE and exhaust emissions such as CO, HC, and NOx. The results were compared with standard diesel fuel results. To record various gases namely HC, CO, and NOx an AVL DiGas444G gas analyzer was used, which works on a nondispersive infrared technology (NDIR). To measure the concentration of oxygen and NOx gas, electrochemical (Fuel Cell) sensors were utilized. Smoke density of exhaust gas was measured by Smoke Meter AVL 437C.

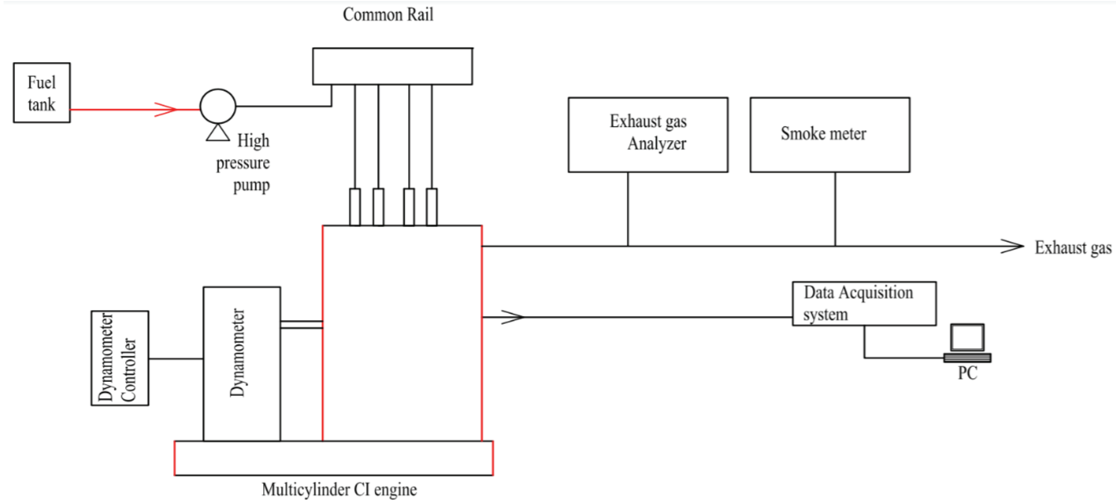


Figure 2. Schematic diagram of multicylinder CRDI experimental test rig.

Table 3. Specifications of CI Engine.

Sl. No	Parameter	Specifications
1	Make	Maruti Suzuki
2	Engine type	1.3 ltr CRDI Engine
3	Static injection timing	24^{o}bTDC
4	No. of cylinders	4
5	No. of strokes	4
6	Fuel	H. S. Diesel
7	Rated power	74 bhp@4000rpm
8	Maximum torque and engine speed	190 N-m@2000rpm
9	Cylinder diameter	0.0696 m
10	Stroke length	0.082 m
11	Compression ratio	17.5 : 1

Table 4. Specifications of Five gas Analyzer.

Type	AVL DIGAS 444G
Object of measurement	Carbon Monoxide (CO), Oxides of Nitrogen (NO_x) Hydrocarbon (HC)
Range of measurement	HC = 0 to 20,000 ppm as vol CO = 0 to 10% NO_X = 0 to 5000 ppm vol
Accuracy	HC = +/- 30 ppm HC CO = +/- 0.03% vol CO NO_X = +/- 10% ppm NO
Resolution	HC = 1 ppm CO = 0.01% Vol NO_X = 1 ppm
Warm up time	7 min
Speed of Response Time	Within 15 sec. for 90% response
Sampling	Directly sampled from tail pipe
Power source	100 to 240 V AC / 50 HZ
Weight	4.5 kg net weight without accessories
Size	270 mm x 320 mm x 85 mm

Table 5. Specifications of smoke meter.

Type	AVL 437C
Object of measurement	Smoke
Measuring range opacity	0-100%
Accuracy	+/- 2% relative
Resolution	0.1%
Smoke length	0.430 m +/- 0.005 m
Ambient Temperature Range	-5 ^{o}C to +45^{o} C
Warm up time	10 min
Speed of Response Time	Within 11 sec. for 90% response
Sampling	Directly sampled from tail pipe
Power supply	100 to 240 V AC / 50 HZ

UNCERTAINTY ANALYSIS

A study revealed the performance of a novel control strategy for imperfect systems. This strategy takes care of unavoidable imperfections associated with physical systems accounting the hidden dynamics associated with them to improve overall system performance (Bucolo et al., 2019). The uncertainties for the current investigation are given in the Table 6. Average of six readings was taken to overcome the measurement errors and for further result analysis.

Table 6. Uncertainties for the present study.

Measured variable	**Accuracy (±)**
Load (N)	0.1
Engine speed (rpm)	1
Temperature, (^{o}C)	1
Fuel consumption (g)	0.1
Measured variable	**Uncertainty (%)**
HC	±1.4
CO	±2.7
NOx	±2.2
Smoke	±2.0
Calculated parameters	**Uncertainty (%)**
BTE (%)	±1.2
HRR (J/^{o}CA)	±1.4

RESULTS AND DISCUSSIONS

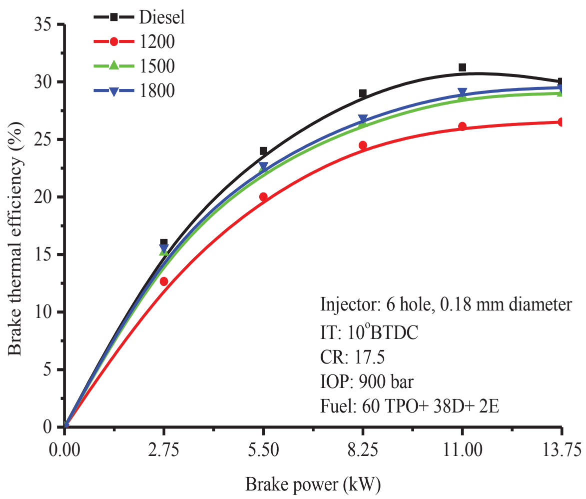
The performance, emission, and combustion characteristics of multicylinder CRDI engine powered with blends of TPO, diesel, and ethanol are discussed in this section. The test matrix is provided in Table 7.

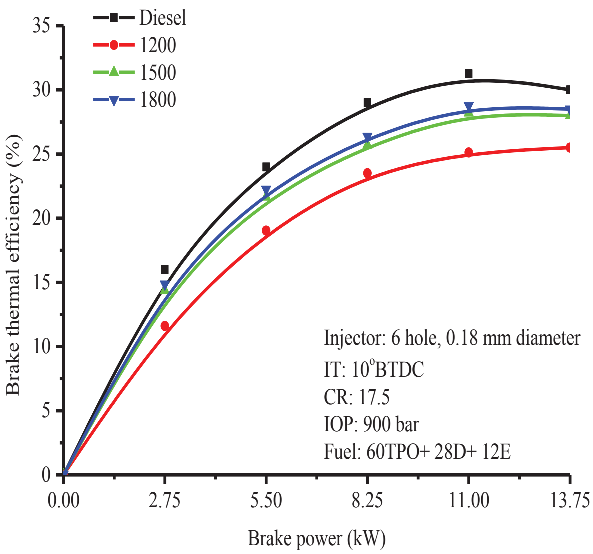
Table 7. Test Matrix.

Phase	Operating condition	Speed
Diesel Operation	CR=17.5, IP=900 bar, IT=10^{o}BTDC Fuel: Diesel	1500 RPM
First Phase	CR=17.5, IP=900 bar, IT=10^{o}BTDC Fuel blend: E-T-D = 02:60:38	1200, 1500, 1800 RPM
Second Phase	CR=17.5, IP=900 bar, IT=10^{o}BTDC Fuel blend: E-T-D = 02:60:38	1200, 1500, 1800 RPM
Third Phase	CR=17.5, IP=900 bar, IT=10^{o}BTDC Fuel blend: E-T-D = 02:60:38	1200, 1500, 1800 RPM

Influence of engine speed and fuel blends on brake thermal efficiency

Figure 3 shows the effect of engine speeds and selected blends on BTE of CRDI engine at different loads. For all fuel blends, BTE was lower as compared to diesel. This may be due to the blends having lower energy contents and higher viscosity (Montajir et al., 2000; Sudarshan, 2018b; Sayin, 2010& Manjunath et al., 2019). When concentration of ethanol was increased, marginal decrease in BTE was observed. Lower cetane number of ethanol could be another factor for low BTE (Gadwal, 2019). With increase in load and speed, BTE was increased for all combinations of fuel blends. For the selected combination, E02:T60: D38 showed better BTE as compared to other combinations. Among all speeds, 1800 rpm showed better results for all combinations of fuel blends (An et al., 2012). Knocking was observed when engine speed increased beyond 1800 rpm. Similar results were observed in literature (Sjoberg et al., 2005).





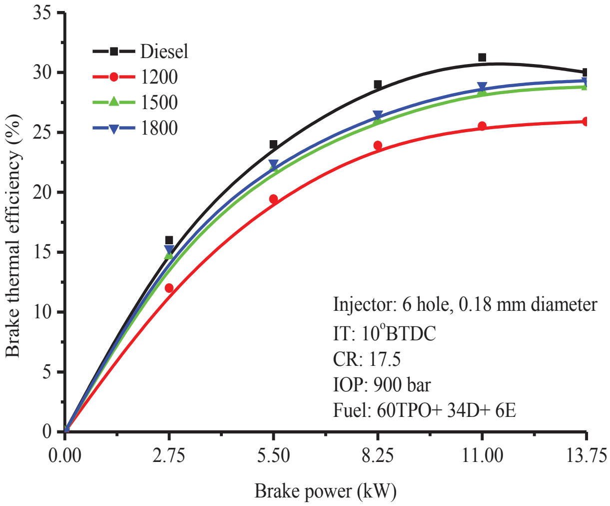


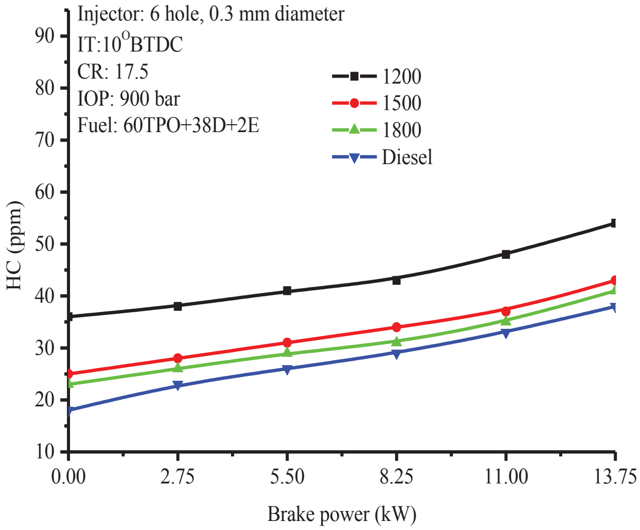
Figure 3. Effects of speed on BTE for different fuel blends at different loads.

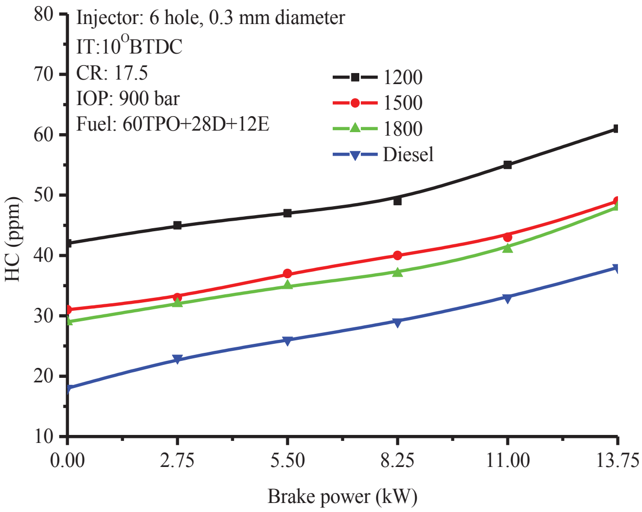
Emissions

In this section, exhaust emissions from the engine are discussed for the different fuel blends used.

HC Emissions

Figure 4 represents the HC emissions of CRDI engine at varied engine speeds and loads using selected fuel blends. HC emissions were higher than the diesel for all fuel blends and speeds. It was observed that HC increases with increase in load. Increase in proportion of ethanol increased emissions of HC. Slow evaporation of ethanol blends due to high heat of vaporization results in poor mixing of air-fuel mixture. This leads to increase in spray dispersion, which causes undesirable fuel impingement on the walls of combustion chamber contributing to increased HC emission. Further higher viscosity and lower cetane number could be another reason for increased HC emissions (Indudhar et al., 2019; Sudarshan, 2018b; Sayin, 2010; & Gadwal, 2019). For higher engine speed, HC emission was lower for all combinations of fuels. This can be attributed to enhanced mixing of air and fuel mixtures at higher speeds (John, 1988).





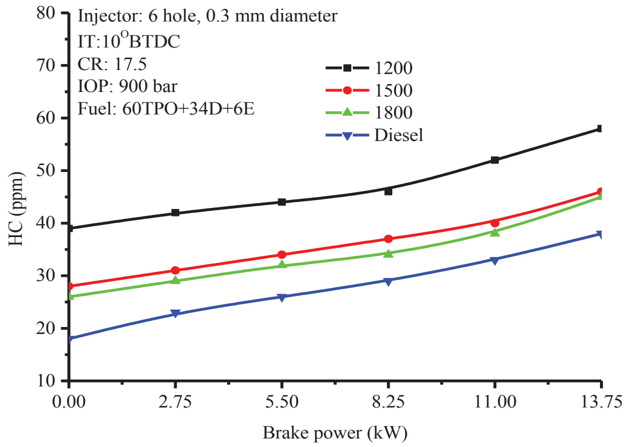
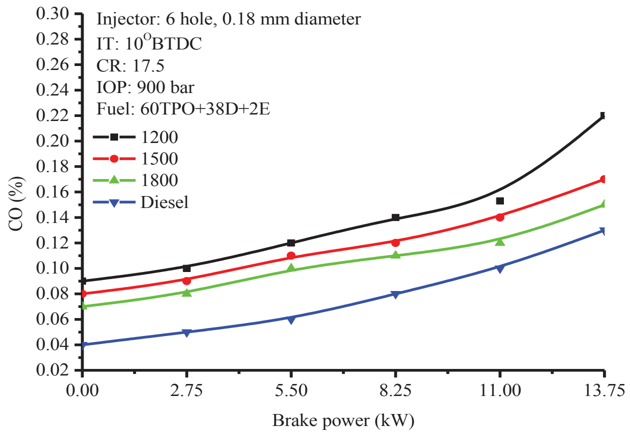
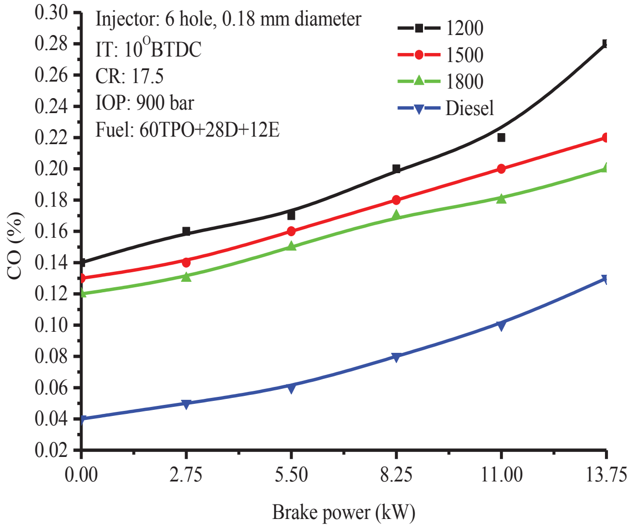


Figure 4. Effect of speed on HC emission of fuel blends at different loads.

CO Emissions

Figure 5 depicts the influence of engine speed and selected fuel blends on CO emissions at various engine loads. CO emissions of the blends were higher as compared to diesel, with an increase in load CO emissions increased. This may be due to high concentration of blends, which leads to inadequate combustion associated with smaller combustion periods when engine speed is high. Due to lower cetane number of the combinations of fuels, ignition delay (ID) increased, which is shown in Figure 8, resulting in the accumulation volume of fuel before the commencement of combustion. The CO emissions were increased with increase in ethanol proportion in the blends. The higher viscosity of blends causes the spray characteristics degraded, which results in improper mixture formation. Lower combustion temperature could also be the reason. Hence, poor combustion of selected combinations causes formation of more CO emissions. Similar results could be seen in literature (Sudarshan, 2018b; Gadwal, 2019; Sayin, 2010 & Jaichandar et al., 2013). This may be due to incomplete combustion of the blended fuels, which are having low cetane number with longer ID; moreover, the quantity of fuel burnt was more in expansion stroke due to more combustion duration at lower speed as shown in Figure 9. Further injected volume of blends was more due to low heating value, which leads to improper mixture formation.





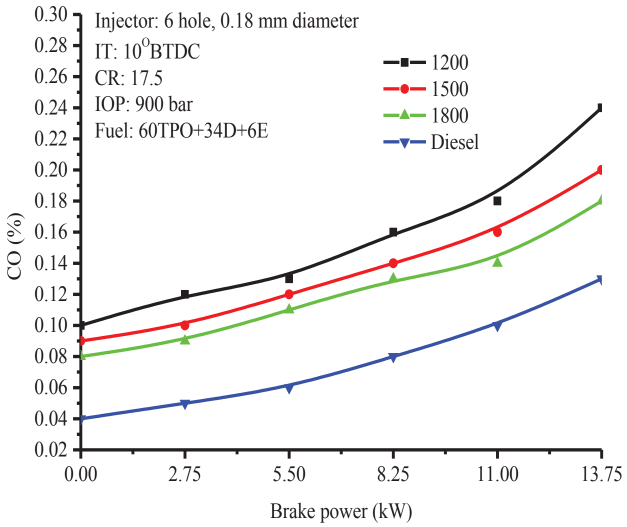
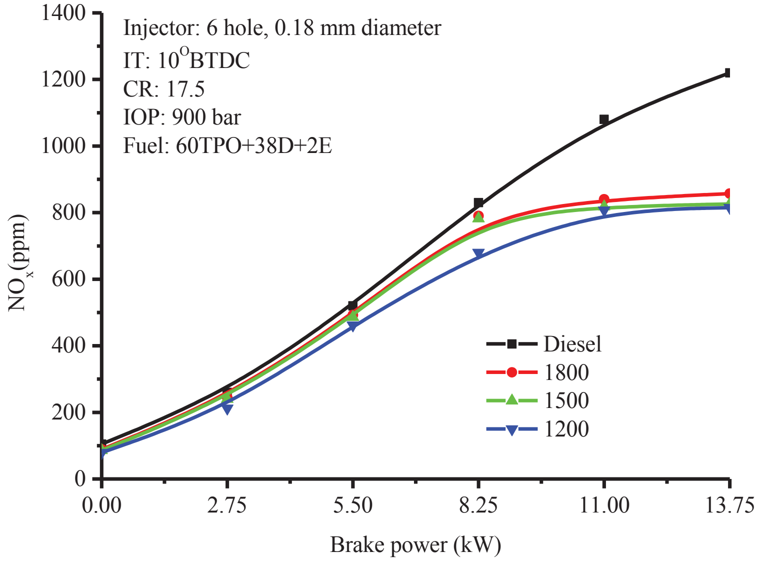


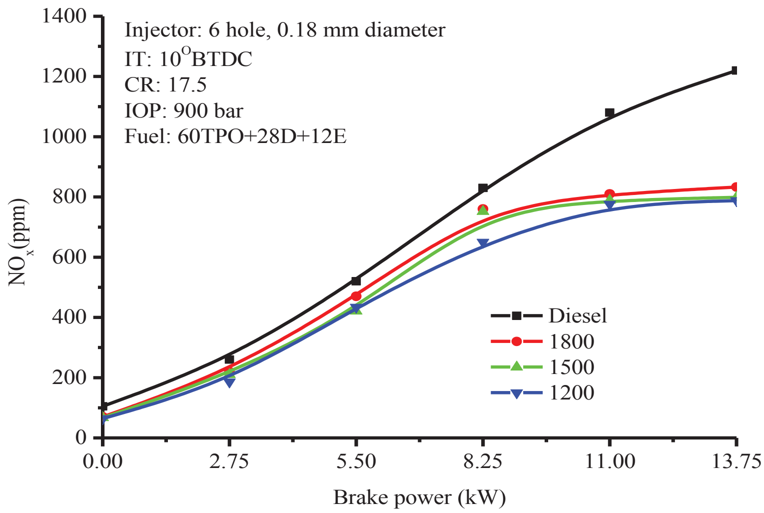
Figure 5. Effect of speed on Carbon monoxide emission of fuel blends at different loads.

NOx Emissions

Figure 6 represents the emissions of NOx of multicylinder CRDI diesel engine powered with blends of TPO with diesel and ethanol at different engine speeds and loads. NOx emissions for all blends were low compared to diesel. The rate of formation of NOx mainly depends on flame temperature, residence time, and availability of oxygen in combustion chamber.

At high engine load, more quantity of fuel was inducted resulting in high temperature of gas and in turn more formation of NOx. Compared to diesel, the peak in-cylinder temperature was lower for selected combinations of fuels. With increase in the proportion of ethanol in the blend, combustion temperature was lesser. This is due to poor vaporization rate of blended fuels and the high latent heat of evaporation of ethanol in the blends. Lower peak pressure (PP) and heat release rate (HRR) shown in figure 10 and figure 11, respectively, at higher ethanol proportion could also be the reason for lower combustion temperature. Hence, NOx emissions for the blends were lower as compared to diesel for all speeds. This trend could also be attributed to higher water and oxygen contents in TPO-diesel-ethanol blends. Similar results were reported in literature (Sudarshan, 2018b; Gadwal, 2019; Sayin, 2010 &Lee et al., 2013).





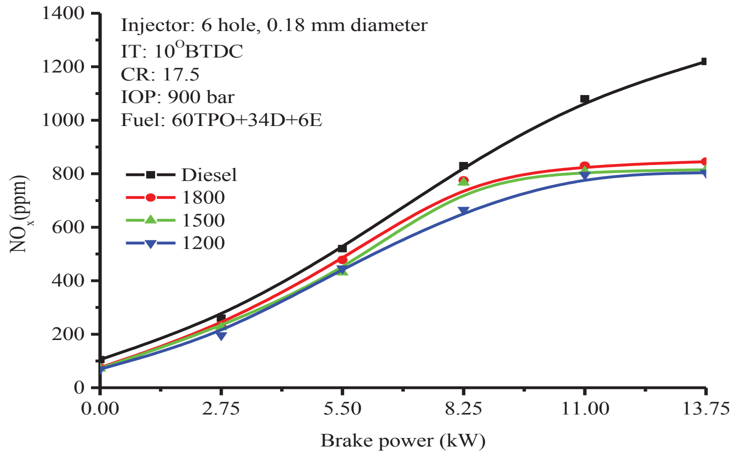
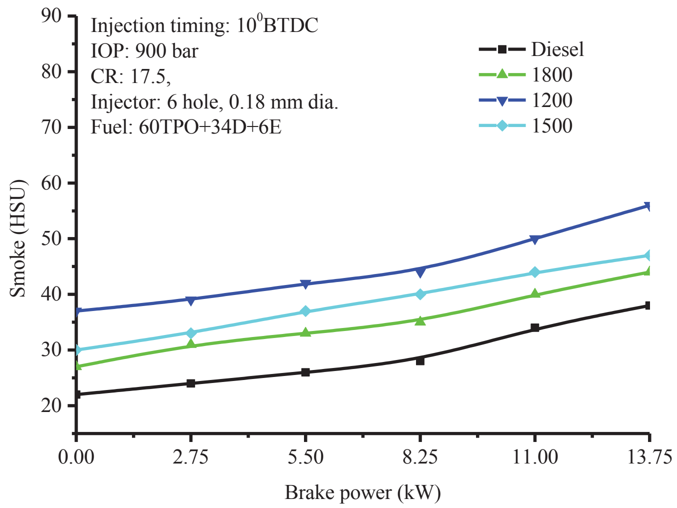
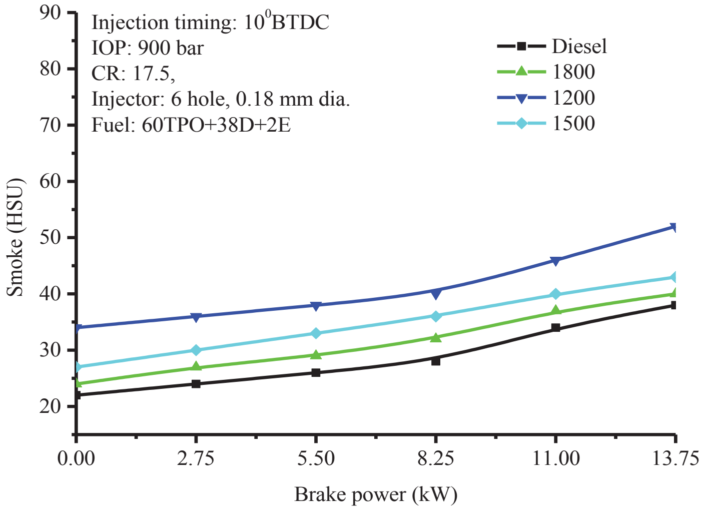


Figure 6. Effect of speed on oxides of nitrogen emission of fuel blends at different loads.

Smoke emissions

Figure 7 shows the influence of engine speed and load on smoke emissions of CRDI engine for the fuel blends. Increased smoke opacity was observed with increased loading operation for all the fuel blends for the three speeds of engine operation. Smoke formation is spontaneous at higher loads (Soudagar et al., 2019). Increased quantity of fuel injected during higher engine loads decreased air-fuel ratio that leads to higher smoke emissions. Smoke emissions were high when engine was operated with blends as compared to diesel. This is due to poor vaporization rate of blended fuels and the high latent heat of evaporation of ethanol in the blends. Smoke emissions decreased with increased speed, which may be attributed to earlier injection that leads to higher in-cylinder temperatures, facilitating more time for oxidation of the soot particles before opening of exhaust valve. Inbuilt fuel oxygen and improved mixing at higher speeds reduce the smoke formation. Similar results could be seen in literature (Sudarshan, 2018b; Gadwal, 2019; Banapurmath et al., 2008; Hulwan et al., 2011).





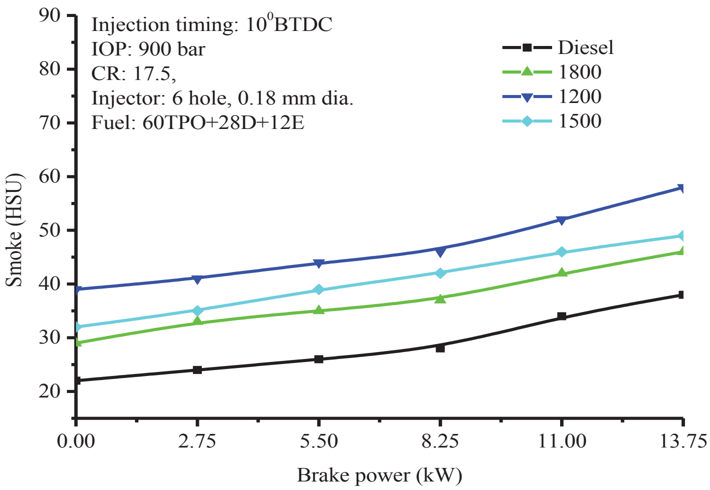


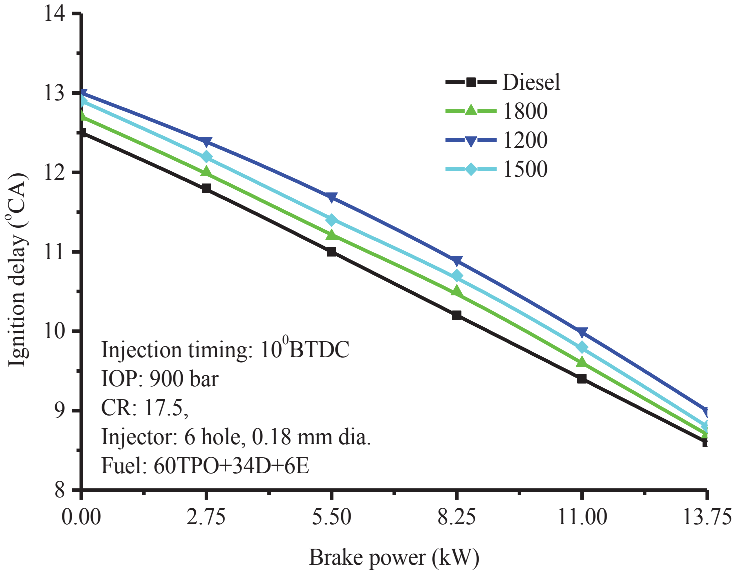
Figure 7. Effect of speed on smoke emission of fuel blends at different loads.

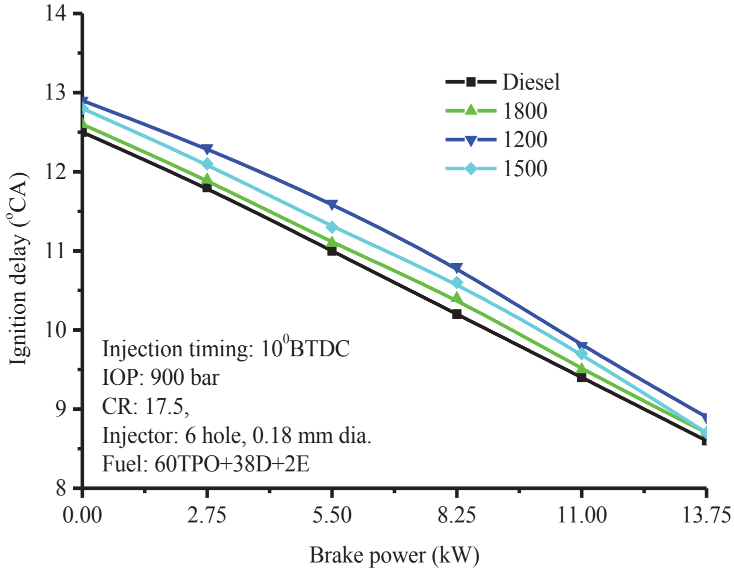
Combustion characteristics

In this section, combustion characteristics of the engine are discussed for the different fuel blends used.

Ignition delay (ID)

Figure 8 depicts the effect of engine speed and load on ignition delay of CRDI engine for the fuel blends. ID calculated based on the static IT. Decreasing trend of ID was found with increased BP. Temperature of combustion gases gets elevated at higher BP; thereby, the decreasing trend of ID was reported with all speeds, and, at the same time, lower ID was observed with higher speed. ID was high when engine was operated with blends in comparison to diesel. This is due to poor vaporization rate of blended fuels and the high latent heat of evaporation of ethanol in the blends. Decrease in cetane number and density could also be responsible for increase in ID at higher proportion of ethanol (Subbaiah, G.V. et. al., 2010; Clenci, A. et al., 2016). Also, higher self-ignition temperature of ethanol blends is responsible for the increase in ID. Similar results could be seen in literature (Banapurmath et al., 2008; Hulwan et al., 2011).





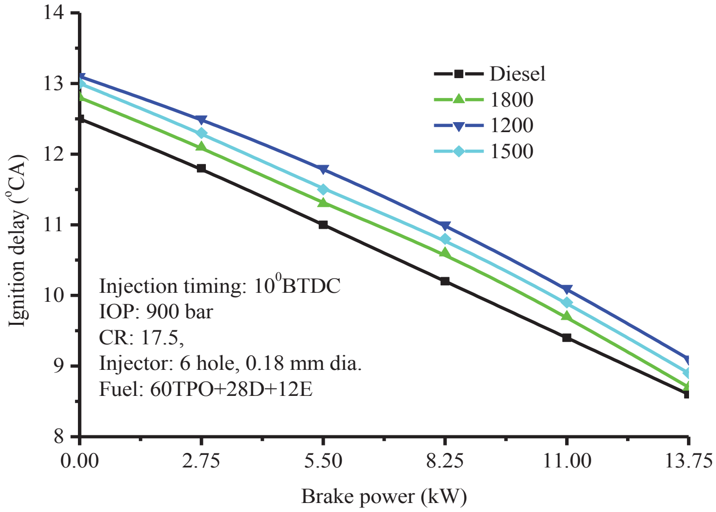
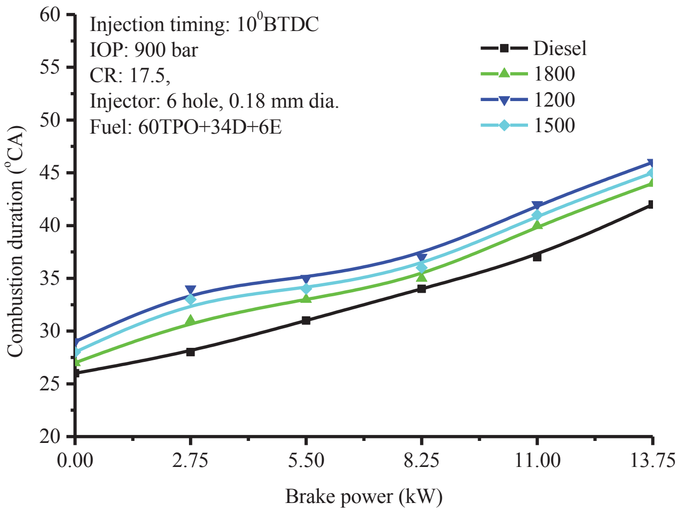
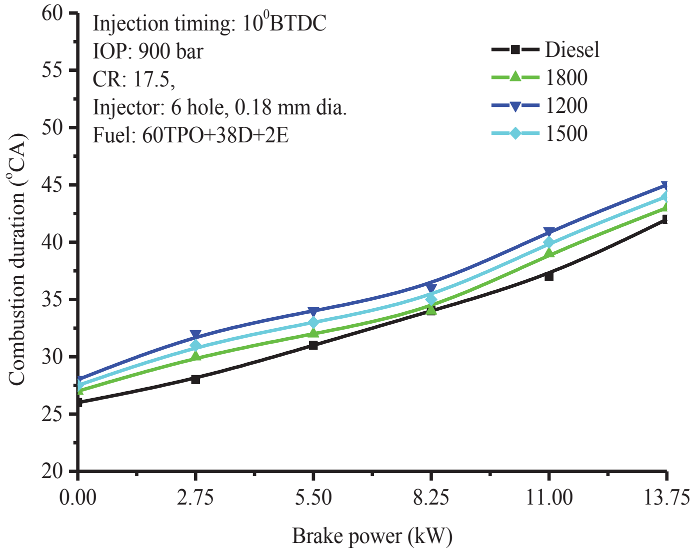


Figure 8. Effect of speed on ignition delay of fuel blends at different loads.

Combustion duration (CD*)*

The variation in CD is provided in Figure 9, and CD was calculated with time duration between start of combustion (SOC) and 90% cumulative heat release. CD increased with higher BP for all speeds. Higher CD was observed for blends as compared to diesel, and higher proportion of ethanol in blends also showed higher CD. It might be because of higher viscosity that led to inhomogeneity in air–fuel mixing, lower gas temperature, and pressure. Also, due to delayed combustion after top dead center (aTDC). However, CD was reduced with speed due to better combustion as compared to lower speed. Similar results were reported in literature (Hulwan et al., 2011).





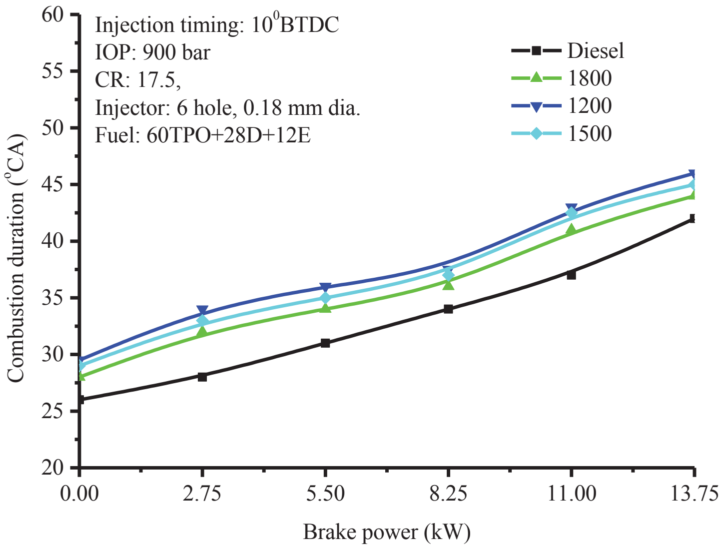


Figure 9. Effect of speed on combustion duration of fuel blends at different loads.

Peak Pressure (PP) and Heat Release Rate (HRR)

Figure 10 illustrates the effect of speed and load on peak pressure. The PP depends on the combustion speed and total amount of fuel burnt in rapid combustion phase period. Higher PP was found with higher BP. At all speed, slower burning nature of blends during ID period could be responsible for lower PP. It could be because of combined impact of slightly longer ID and slightly higher adiabatic flame temperature. Figure 11 shows variation in HRR with BP of the engine powered with blends used. All blends showed increasing trend of HRR with BP because of more fuel injected. Better air fuel mixture, better combustion, higher cylinder gas temperature, and pressure prevailed might be the reason for the higher HRR at higher speed. Blends showed lower HRR as compared to diesel due to poor combustion qualities. For higher ethanol proportion in blends, rapid premixed combustion and improved diffusive combustion were observed at high loads. Similar results were reported in literature (Hulwan et al., 2011, Tutak W et.al., 2016).

HRR was determined with in-cylinder gas pressure data. HRR at each CA was calculated using equation 1.

$$Q_{app} = \left(\frac{\gamma}{\gamma - 1}\right) p.dv + \left(\frac{1}{\gamma - 1}\right) v.dp + Q_{wall} \tag{1}$$

where

Q_{app} - Apparent heat release rate (J)
γ - Ratio of specific heats $C_p/(C_p - \bar{R})$
R - Gas constant in (J / kmol-K)
C_p - Specific heat at constant pressure (J / kmol – K)
V - Instantaneous volume of the cylinder (m^3)
P - Cylinder pressure (bar)
Q_{wall} - Heat transfer to the wall (J)

For this calculation, cylinder gas was assumed to behave as an ideal gas (air) with specific heats being dependent on temperature (Hayes et al., 1986). The specific heat was found using the equation 2.

$$Cp = [3.6359 - \frac{1.33736T}{1000} + \frac{3.29421T^2}{1 \times 10^6} - \frac{-1.91142T^3}{1 \times 10^9} + \frac{0.275462T^4}{1 \times 10^{12}}]\,R \tag{2}$$

for T < 1000 K

$$Cp = [3.04473 + \frac{1.338056T}{1000} - \frac{0.488256T^2}{1 \times 10^6} + \frac{0.0855475T^3}{1 \times 10^9} - \frac{0.00570127T^4}{1 \times 10^{12}}]\,R \tag{3}$$

for T > 1000 K

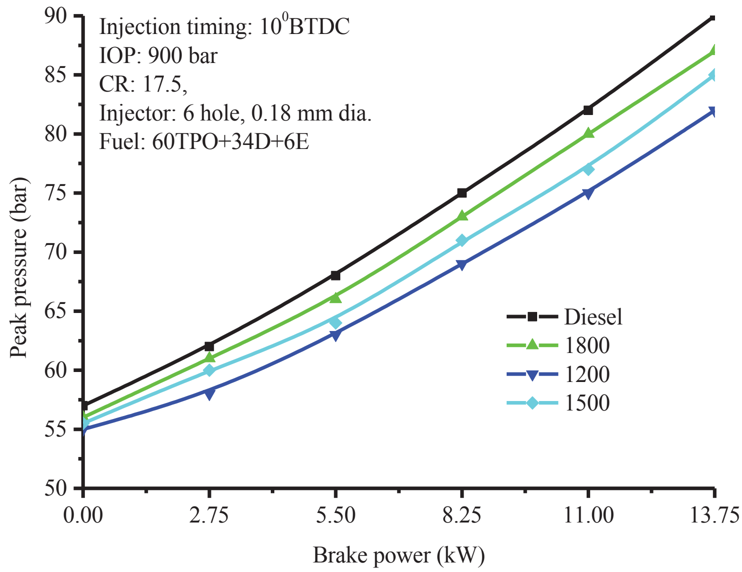
Heat transferred to the wall was determined with the Hohenberg equation (4) (Hohenberg, 1979) and assuming the wall temperature to be 723 (Hayes et al., 1986).

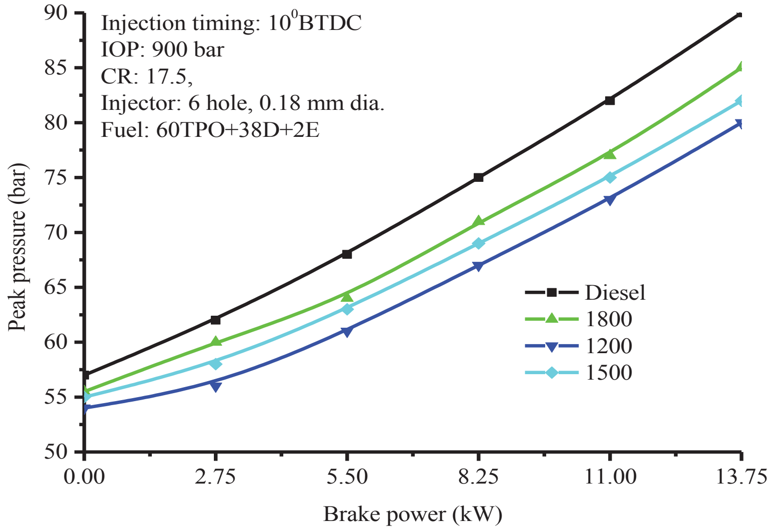
$$Q_{wall} = h \times A \times [\,Tg - Tw\,] \tag{4}$$

where $h = C1\,V^{-0.06}\,P^{0.8}\,T^{-0.4}\,(\,V_P + C_2\,)^{0.8}$ (5)

h - Heat transfer coefficient in W/m^2 K

C_1 & C_2 - Constants, 130 & 1.4
V - Cylinder volume in m^3
P - Cylinder pressure in bar
T - Cylinder gas temperature in K
V_P - Piston mean speed in m/s
A - Instantaneous Area (m^2)





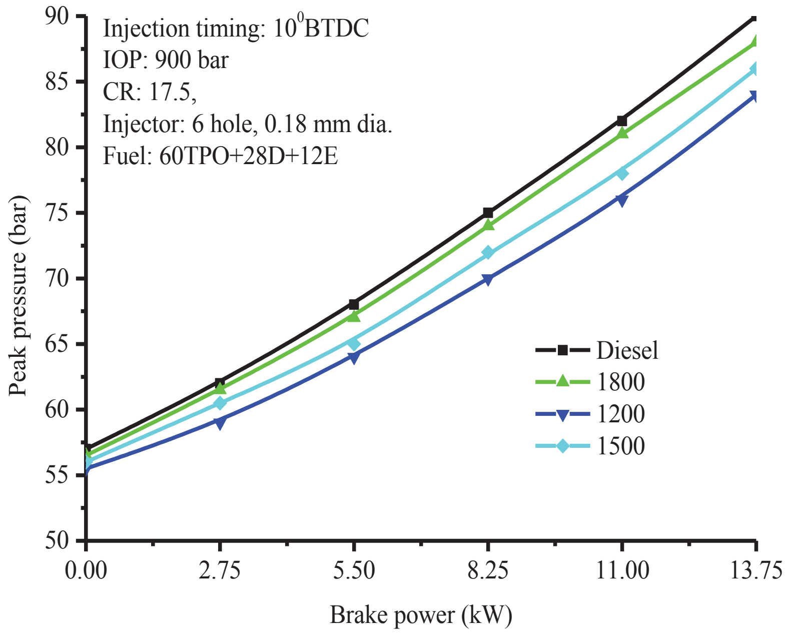
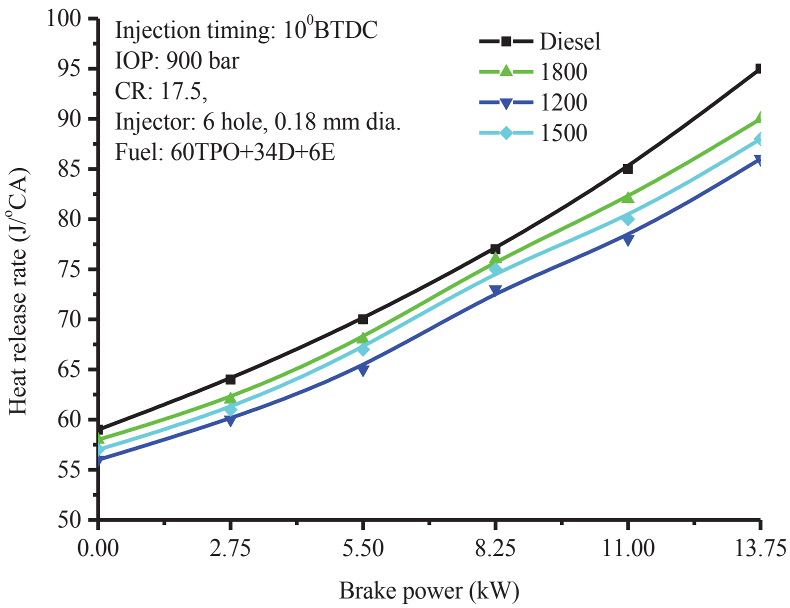
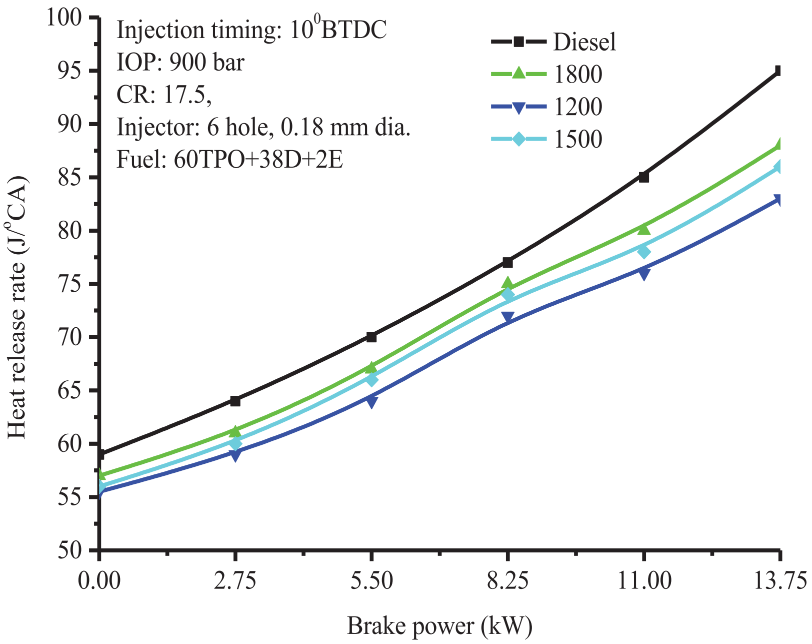


Figure 10. Effect of speed on peak pressure of fuel blends at different loads.





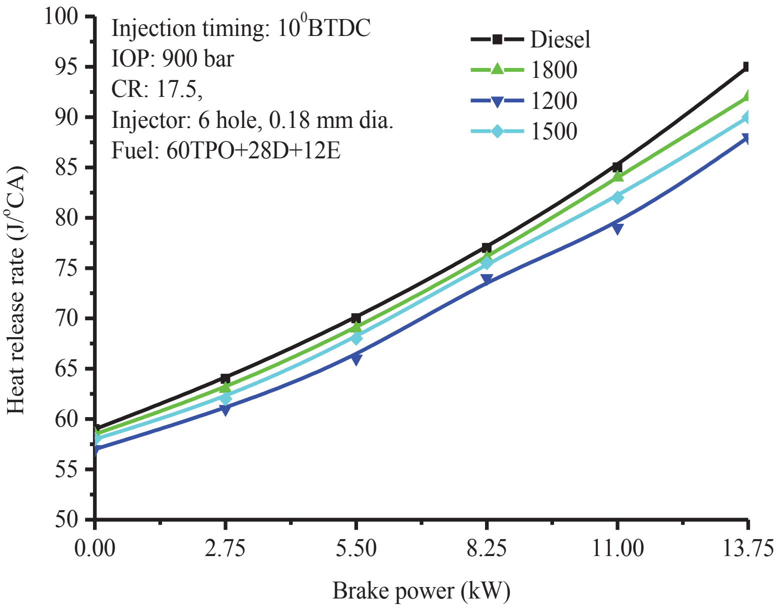


Figure 11. Effect of speed on Heat release rate of fuel blends at different loads.

CONCLUSIONS

Experimental investigations were carried out on a multicylinder CRDI engine fueled with ethanol- pyrolysis oil and diesel blends using fixed volume of tyre pyrolysis oil in each combination selected. CRDI Engine performance, emission, and combustion characteristics were examined at different engine speeds and loads. The results achieved were compared with base diesel fuel operation results. The following conclusions are drawn from the results:

- TPO obtained from waste tyre through pyrolysis could be used as an alternative fuel along with ethanol and diesel.
- BTE increased with increase in load and speed but engine operation with blends selected showed lower performance in comparison to diesel. Further, BTE decreased with increase in ethanol proportion in blends.
- Engine yielded more CO, HC, and smoke emissions with all blends at different loads as compared to diesel but lowered to some extent with speed. However, NOx emissions with blends were less as compared to diesel.
- Combustion parameters like ID, PP, and HRR decreased with increase in ethanol content in the blend. However, CD increased with ethanol proportion in the blends.

Overall, it could be concluded that 2% of ethanol with 38% diesel and 60% TPO blend gave better results as compared to other combinations selected in the study with little penalty on BTE.

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