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Effect of *n*-Butanol Concentration on Combustion and Emission Phenomenon using Higher Blends of Cotton Seed Biodiesel in a Direct Injection CI Engine

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Abstract- This paper experimentally investigates the effect of cottonseed biodiesel at higher blend ratios with mineral diesel on the combustion aspects of a direct injection CI engine. The effect of *n*-butanol concentration at higher blend ratios were also analysed at all loading conditions. CBD was synthesized by base catalysed trans-esterification with methanol and KOH as catalyst with the molar ratio of 8:1. GCMS analysis exposed the presence of Octadecanoic acid methyl ester as major constituent. D50CBD50 (mineral diesel 50% and CBD 50%) a higher blend test fuel was used with D100 as base reference fuel. The experiments were carried out at constant speed of 1500 rpm and the result indicated higher in-cylinder pressure for D50CBD50*n*B10 at all loads. At low and part load, the rate of heat release was higher for D50CBD50 whereas, for D50CBD50*n*B20 it increased significantly at full load condition. The peak of D50CBD50nB10 was found to exhibit higher ROPR across all loading conditions. CHR of D100 was found to be highest at full load condition whereas D50CBD50*n*B20, exhibited peak values at part load condition. The ignition delay showed a decreasing trend across all fuel blends with biodiesel blends and it was lower than diesel. Emission analysis showcased a notable decrease in UBHC and CO emission with a significant increase in NO_x, CO_2 and Smoke emission. Among all the test fuels, D50CBD50*n*B10 exhibited better combustion phenomenon with reduced emission characteristics.

Keywords: Cotton seed biodiesel, Higher blend, Combustion, In-Cylinder pressure, Ignition delay, Emission

1. Introduction

Over ten decades compression ignition (CI) engine was playing a very critical role in the world. CI engines were widely used globally than any other type of engines due to its thermal efficiency and fuel economy. As the demand towards the diesel engine increased, the role of engineers, academicians and researchers got more vital towards the development process. As the production rate of CI engine increased, proportional rise was seen towards the depletion of fossil fuel which was its power source. Also the fossil fuel was exhausting harmful emissions. Emission norms of vehicles are more stringent these days than before. These notable reasons triggered the researchers to move on the pathway of using alternate fuel sources [1, 2].Biodiesel is a promising source of alternate energy for mineral diesel. It is a better alternate because of the reduction in emissions, biodegradable property, etc. Energy demand is also a reason to state that biodiesel is a promising fuel alternative. Initially, bio-oils like coconut, palm oil were used which was then trans-esterified to form esters. Later, the above mentioned oils were avoided as it was edible oils. Therefore the search was moved on to non-edible oils like Pongamia, Jatropha, Cottonseed oils etc, [3, 4].Cotton seed oil is a non-edible oil which acts as a good alternate biodiesel source with similar physio-chemical properties to diesel. Survey revealed that 38% of countries export alone was contributed by cotton vegetation. This information cleared the fact that cotton seed oil production will not acquire excess land for cultivation. The above said details provided the fact that cotton seed biodiesel will be a good alternate fuel for the diesel [5-7].

Research work was carried on for a while towards better combustion inside the cylinder. Studies from the previous works carried out gave the status that oxygenated biodiesels combustion starts later than the biodiesel at low and at higher loads. Similarity was seen for oxygenated biodiesels ROPR and in-cylinder pressure at lower load but the ROHR was slightly higher than biodiesel. At higher engine loads all three parameters were higher than biodiesel [8]. Qi *et al* stated that in-cylinder pressure, ROPR and ROHR during pre-mixed combustion phase whereas at higher load, peak in-cylinder pressure was similar but ROPR and ROHR for biodiesel was lower. Peak values for biodiesel occurred in advanced crank angles. For all engine loads, combustion started earlier for biodiesel due to advanced injection timing and shorter ignition delay based on higher bulk modulus and density [9]. Using biodiesel decreased the peak in-cylinder pressure slightly compared to mineral diesel which may be due to low calorific value. At all loads, ROHR was lower than diesel. It was to be noted that no direct correlation could be given between fuel blend ratio and heat release rate. Combustion duration was lower for biodiesel than diesel at high and part loading conditions [10]. Sahoo *et al* observed that higher peak cylinder pressure was given by neat biodiesel and biodiesel had shorter ignition delay. This type of in-cylinder pressure variations was observed in part and low load conditions. Peak pressure happened always after TDC favouring safe and efficient operation. Maximum heat release rate happened faster for diesel-biodiesel than diesel due to shorter ignition delay. As the delay period was low, the premixed combustion burning phase was less. Higher fuel viscosity leads to poor atomization extending the ignition delay for diesel fuel. Biodiesel blends marginally decreased the heat release rate than diesel [11, 38].

Experiment revealed that higher biodiesel blends provided high cylinder pressure. Viscosity reduced by the addition of oxygenate to the biodiesel blend thereby improving the volatility which atomized the fuel droplets better and prepared the mixture well with air during the ignition delay period. Peak pressure mainly depends on the Rapid combustion phase were fuel takes part for burning. As Cetane value was low, the volatile nature would be more which showed better vaporization [12]. Seung Hyun Yoon *et al* discussed that the heat release and combustion pressure was lower for biodiesel compared to mineral diesel at low load whereas at higher load, peak combustion pressure and peak heat release rate was higher for biodiesel than mineral diesel [13]. There was a less time for mixing when ignition delay got reduced which resulted in less energy release at premixed combustion phase [14]. Wojciech Ttak *et al* discussed about the oxygenate addition to the biodiesel and diesel at various proportions. Difference in heat release rate and pressure was found to be less at lower blends whereas at higher blend ratio, peak pressure was closer to TDC and pressure was little higher. Peak heat release rate was low as the oxygenate rate was increased [15].Oxygenated biodiesel revealed that SOC occurred later with less maximum pressure. It was observed that for biodiesel blends the ignition delay was higher than diesel. Sharp and higher premixed combustion peak was seen for biodiesel than neat oil which brought maximum heat release rate and pressure [16]. Butanol is an oxygenate of biomass based renewable fuel produced by alcoholic fermentation of biomass feed stock used for alcohol preparation. Its properties are listed as higher viscosity, higher heating value, less hydrophilic tendency, Cetane number and lower vapour pressure. It's mixing is better than other alcohols [17, 30]. Imdadul stated that there was a maximum pressure rise in diesel because of the fuel mass burned in the premixed combustion phase. Poor atomization was seen in biodiesel due to large atomic weight which reduced the fuel mass burned in the premixed combustion phase. But on addition of oxygenate to the biodiesel, it increased the pressure of the cylinder better than biodiesel but lesser than diesel. For biodiesel, SOC happened further late due to the difficulty in mixing and poor atomization [18]. Less combustion efficiency was seen because of the cooling effect due to higher latent heat of vaporization [19].

Finding new alternatives reduced the nation's dependency on petroleum-based fuels and possibility of economic crisis due to oil is lessened. Arul prakasajothi *et al.* stated that significant reduction in emissions can be done using alcohol as an oxygenated additive. In this study research carried out with octane has shown abilities to reduce the exhaustive emissions with various proportions [20]. To reduce the CO_2 emissions and to keep the soot level low, ethanol was chained with the biodiesel blends. The results showed that the particulates emitted were lower and the other emissions reduced to a significant level with better ignition delay. Alcohol addition to the biodiesel blends reduced the ignition delay and showed better combustion. Also lower emissions were noted for the oxygenated fuel blends than diesel and biodiesel.

In this study, the cottonseed biodiesel is blended with diesel at the ratio of 50:50 and the main objective of this work is to analyse the combustion and emission parameters namely the in-cylinder pressure, ROPR and ROHR on addition of oxygenates to the blended biodiesel.

2. Materials and Methods

Kerawalla group, Mumbai, India supplied raw cottonseed oil. Mineral diesel was procured from a petroleum pump station in Chennai, India. About 99% purity industrial grade CH_3OH and KOH was acquired from ALPHA CHEMIKA, India. 99% pure Laboratory grade *n*-butanol was purchased from SUVCHEM laboratory chemicals, India.

Fig. 1. Cotton seed and its bio-oil

2.1 .Trans-esterification

By mixing CH_3OH to 1.8% KOH by weight CH_3OK solution was prepared with reaction temperature as 65°C and stirring speed as 450 rpm with the molar ratio of 8:1. CH_3OK and raw cotton seed oil were mixed in three necked round bottom flask. For 2 hours, the solution was heated at 75°C with stirrer being maintained at 450 rpm. After the reaction period, the solution was kept for separation of glycerol and biodiesel. This process gave maximum cottonseed biodiesel as shown in Figure 1.

2.2 .FAME Composition

The assorted fatty acid methyl esters present in cottonseed biodiesel was tested using JEOL GC-MATE II data system. GCMS analysis revealed the presence of Tricosanoic acid methyl ester, Decanoic acid methyl ester, Tetracosanoic acid methyl ester, Pentadecanoic acid methyl ester, Octadecanoic acid methyl ester, Docosanoic acid methyl ester, Heneicosanoic acid methyl ester, Eicosanoic acid methyl ester and Dodecanoic acid methyl ester. Later, FTIR test confirmed the existence of above said fatty acid methyl esters.

2.3 .Fuel Properties

Fuel properties were compared in the Table 1 which depicts the values of diesel, biodiesel, diesel-biodiesel blend and oxygenated blends. The cottonseed oil property showed vast variation from diesel fuel property. In-order to change the property closer to diesel, trans-esterification was performed. Trans-esterification converted the values mentioned in CSO 100 to CBD 100 which made the properties like calorific value, kinematic viscosity, cetane number, flash point, density closer to diesel values. Since the properties were seen closer, engine modification was not necessary yet these variations will affect the atomization and vaporization of the fuels.

Kinematic viscosity of diesel was 3.2 mm^2/s. Diesel-biodiesel blends had a Kinematic viscosity of3.42 mm^2/sand on adding oxygenate to the blend the viscosity reduced due to the thinning effect of *n*-butanol making the Kinematic viscosity of D50CBD50nB10 as 3.36 mm^2/s and D50CBD50nB20 as 3.28 mm^2/s. Density of diesel was 837 kg/m^3.Diesel-biodiesel blends had a density of 856 kg/m^3. On adding oxygenate to the blend, the density got reduced due to the thinning effect of *n*-butanol making the density of D50CBD50nB10 as 849 kg/m^{3}and D50CBD50nB20 as840 kg/m$^{3.}$ Flash point was little higher for diesel-biodiesel blend than diesel which also helps in better storage. Biodiesel and diesel-biodiesel blend showed less calorific value compared to diesel. Diesel showed the highest Cetane number while in other blends it was less but closer to diesel property [27, 37].The test fuels used in this experimental study are shown in Figure 2.

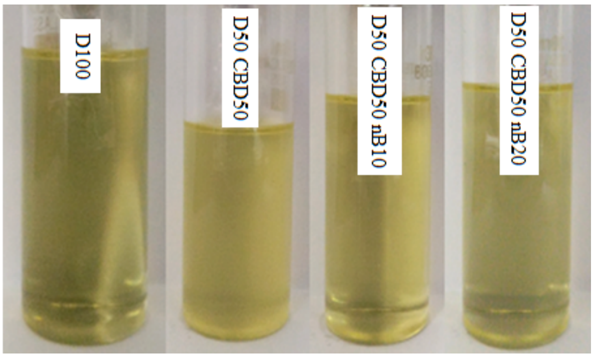

Fig. 2. Test fuel blends of Cotton seed biodiesel and Oxygenates

3. Experimental Setup

Kirloskar 240 PE model, single cylinder direct injection constant speed liquid cooled compression ignition engine was used for this investigation as shown in Figure 3. The specification of engine is described in the Table2. Bore was 87.5mmin length, stroke was 110mm in length, volume capacity was 66 cubic centimetre. Engine was operated at constant speed of 1500 rpm with compression ratio as industry set value 17.5:1 and injection timing being 23°bTDC. For loading purpose, water cooled eddy current dynamometer made by SAJ test plant private limited was used. To measure the loading the data, S beam type universal load sensor was used which had 0-50 kg capacity. Eureka model rotameter ranging 25-25lph of calorimeter, 40-400 lph of engine cooling was utilized for flow measurement. In order to monitor the combustion parameters PCB Piezotronics piezo sensor model SM111A22 was used which had a range of 5000 psi. A 422E5X series charge amplifier supported the pressure data converting high impedance charge signal from the transducer to low impedance voltage signal. Kubler Germany model 8.3700.1321.0360 crank angle encoder having supply voltage range of 5-30V DC was used to determine piston position an angle of rotation. Pressure inside the cylinder was plotted against crank angle which was acquired through NI-USB 6210 bus powered ‘M’ series DAC [i.e., data acquisition device] for an average of 100 working cycles to avoid errors. Pressures inside the cylinder, ROPR, ROHR were analysed using IC Engine Soft_9.0 version software. The emissions were recorded using AVL 437 smoke meter and AVL 444n Di-Gas analyser.

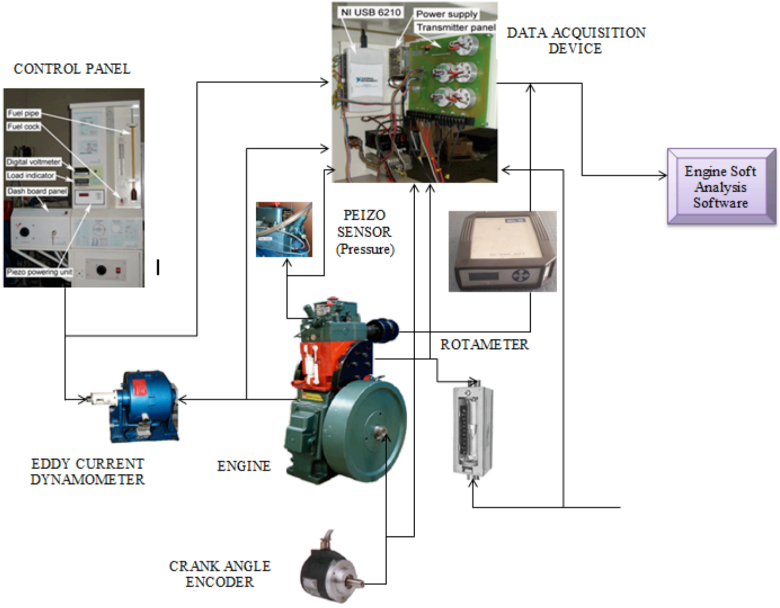

Fig. 3. Engine and Test bed – setup

Table 1. Properties of Test fuels

Properties	Diesel	CBD 100	*n*-butanol	D50CBD50	D50CBD50 *n*B10	D50CBD50 *n*B20
Cetane Number	54	52	25	53.4	50.2	48.4
Flash Point(°C)	70	123	28	87	82	79
Lower Calorific Value (kJ/kg)	43000	38500	33100	41250	40560	39870
Kinematic Viscosity (mm^2/s)	3.20	4	3.60	3.42	3.36	3.28
Density (kg/m^3)	837	874	810	856	849	840

Table 2. Test Engine Specification

Engine type	Single Cylinder, 4 Stroke, Water Cooled
Cylinder diameter	88 mm
Stroke length	110 mm
Dynamometer arm length	185 mm
Displacement	661 cc
Rated engine power	3.5 kW
Rated engine speed	1500 rpm
Start of injection timing	23 ° bTDC
Data acquisition device	NI USB-6210, 16-bit, 250kS/s.
Software	Engine performance analysis software “Engine soft”
Emission analyzer details	
Make and Model	AVL444n Di-Gas analyzer
Measurement principle – HC and CO	Infra-red method
Air humidity	90% max., Non-condensing
Measurement principle of NO_x	Electro-chemical method
Smoke meter details	
Make and Model	AVL437 Smoke meter
Light source	Halogen bulb 12 V/ 5W
Measuring range	0-100% capacity in % , 0-∞ absorption m-1
Detector	Selenium photocell dia. 45 mm, Max. sensitivity in light
Measurement chamber	Effective length is 0.435 m ± 0.005 m
Frequency range	Between 550 to 570nm. Less than 430 nm and more

3.1 .*Experimentation*

Measurement was started with constant speed 1500 rpm, CR 17.5:1 and IT 23^{o}bTDC. 100 working cycles were averaged for each of the operating condition. Using piston TDC and above said mean pressure ROHR and ROPR was determined. The experiment was conducted for low, part and high load. First law of thermodynamics is utilized to evaluate the heat release rate using the following equation (1),

[illegible] (1)

Where [illegible] = gross heat release rate, [illegible] = net heat release rate and [illegible] = heat transfer rate to the walls.

3.2 .*Uncertainty Analysis*

Accuracy of the measurement were ensured by the instrument accuracy and the analysis performed for uncertainty. There were various factors due to which errors might occur like instrument accuracy, calibration, condition, selection of instrument, surroundings, observations, test execution, readings. To make sure about the accuracy of the measurement it was necessary to perform uncertainty analysis. Precession measuring was carried out by averaging a minimum of three time's continuous measured data's. Detailed values of the analysis were tabled regarding accuracy. Tabled values mentioned the every magnitude's standard deviation as averaged values percentage. Accuracy and uncertainty for the data studied in this work was discussed in Table 3.

Table 3. Error Analysis

Parameters	Accuracy	Uncertainty
Fuel flow rate	±0.03l/h	±0.4%
Time(s)	±0.5%	-
Temperature(^{o}C)	±1	-
Cylinder pressure (bar)	±6	-
Engine speed	±1%	-
Pressure transducer (bar)	±1	0.20%
Crank angle encoder(^{o}CA)	±1	0.20%
Load (N)	±0.25%	-
Intake air flow (l/min)	-	±1.2%
Speed sensor (rpm)	±10	1%
Measuring burette (cc)	±0.25	1%

4. Results and Discussion

The present study focussed on the combustion phenomena. The major combustion and emission parameters were studied to analyse the behaviour of the higher blended fuels and are tabulated in Table 4. The blends that compared with diesel were D50CBD50, D50CBD50nB10 and D50CBD50nB20. Analysis were carried out on these fuel blends in comparing the parameters like pressure rise, heat released under no, part and full load operating conditions. The discussed parameters under various operating condition were rate of pressure rise, in-cylinder pressure, ignition delay, rate of heat release, unburned hydrocarbon emission, carbon dioxide emission, carbon monoxide emission, oxides of nitrogen and smoke emission.

4.1 .*In-Cylinder Pressure*

In-cylinder pressure is majorly affected by the air flow, fuel flow, turbulence, pressure difference and temperature difference. Figure 4 represents the in-cylinder pressure variations of diesel and blended fuels at no load condition. Diesel showed on in-cylinder pressure as 42.8 bar at 9^{o} aTDC for no load condition.D50CBD50 showed pressure variation as 45.98 bar at 8^{o} aTDC which was lesser than oxygenated fuel. D50CBD50*n*B10 showed the maximum pressure as 46.63 bar at 8^{o} aTDC which may be due to the better mixing in the premixed combustion phase due to the addition of *n*-butanol which increased the volatility. Further increase in oxygenate upto 20% showed in-cylinder pressure of 45.55 bar at 8^{o} aTDC which was higher than diesel. Mineral diesel showed the maximum pressure of 52 bar at 9^{o} aTDC at part load conditions as shown in Figure 5, diesel showed as the combustion duration was higher than any other biodiesel blends. Diesel-biodiesel blend D50CBD50 showed 56.16 bar at 7^{o} aTDC which was 7.4% higher than diesel. This may be due to the shorter ignition delay. Further increase was seen at the addition of *n*-butanol by up to 10% as 56.7 bar at8^{o} aTDC which was 0.9% higher than D50CBD50. On adding oxygenate upto 20%,gradual decrease was seen upto 55.9 bar at 8^{o} aTDC which was lower than diesel-biodiesel blend. Diesel showed the maximum pressure as 59 bar at 8^{o} aTDC for the full load conditions as presented in Figure 6. D50CBD50 showed the pressure of 60.66 bar at 10^{o} aTDC.D50CBD50nB10 showed maximum pressure as 60.98 bar at 7^{o} aTDC which was 3.2% higher than diesel. This may be due to the O_2 presence in the fuel. Also it had lower ignition delay than diesel and diesel-biodiesel blend leading to better combustion which raised the in-cylinder pressure. On adding *n*-butanol by upto 20%, there was a marginal decrease in value as 60.82 bar at 7^{o} aTDC but higher than diesel. Because of the ignition delay effect, biodiesel had earlier start of burning at all loading conditions. Regardless of the fuel property, higher viscosity and unfavourable volatile nature of biodiesel, it had less ignition delay which may be due to the thermal cracking of higher molecules at high temperature during rapid chemical reaction which burns the fuel earlier thereby reducing the delay time. As the premixed combustion phase being shorter, the cylinder pressure rose as the combustion start before TDC avoiding the burning in diffusion phase which might result in loss of pressure. As the load increased, it showed even higher pressure than before which was due to the beginning of burning phase by few more crank angle before TDC. On adding 10% *n*-butanol to the diesel-biodiesel blends, the property of the blend got better by reducing the kinematic viscosity and higher volatility which made ignition delay lesser and thereby reducing the premixed combustion phase and increasing the in-cylinder pressure.

D50CBD50nB10 showed a better in-cylinder pressure for all the loads [22].

Table 4. Variation in Combustion Parameters

	Load	**Start of Injection (^{o}bTDC)**	**Start of Combustion (^{o}bTDC)**	**Ignition delay (in ^{o})**	**Combustion duration (in ^{o})**	**Peak Pressure (bar)**
Diesel	0	23	15.3	7.7	70.3	42.76
	2	23	15.6	7.4	83.6	45.74
	4	23	16.1	6.9	76.1	49.42
	6	23	16.4	6.6	79.4	51.97
	8	23	16.9	6.1	58.9	55.09
	10	23	17.6	5.4	63.6	56.81
	12	23	18.1	4.9	71.1	59.01
D50B50	0	23	15.6	7.4	85.6	46.63
	2	23	16.1	6.9	82.1	49.53
	4	23	16.5	6.5	61.5	53.26
	6	23	16.8	6.2	48.8	56.16
	8	23	17.2	5.8	6.2	58.89
	10	23	17.9	5.1	52.9	60.29
	12	23	18.4	4.6	55.4	60.98
D50B50 O10	0	23	15.7	7.3	62.7	45.98
	2	23	16.25	6.75	55.25	48.51
	4	23	16.66	6.34	51.66	52.18
	6	23	17.1	5.9	49.1	56.71
	8	23	17.6	5.4	49.6	57.81
	10	23	18.3	4.7	63.3	60.23
	12	23	18.6	4.4	52.6	60.66
D50B50 O20	0	23	15.9	7.1	60.9	45.55
	2	23	16.7	6.3	67.7	48.65
	4	23	16.9	6.1	48.9	53.1
	6	23	17.4	5.6	53.4	55.9
	8	23	17.8	5.2	47.8	59.19
	10	23	18.6	4.4	61.6	60.48
	12	23	18.9	4.1	55.9	60.82

4.2 .Rate of Heat Release

For the blends D50CBD50, D50CBD50*n*B10 and D50CBD50*n*B20 the variation of heat release rate was shown in the Figure 7, 8 and 9 with reference diesel value at no load, part load and full load conditions. Energy transfer at nearby cylinder walls and raised temperature of the gas combusted affects the rate of heat release. Negative rate of heat release were seen during the initial state due to vaporization of accumulated fuel due to delay period. Once combustion occurred, the rate of heat release became positive. Highest rate of heat release for no load condition was seen for D50CBD50 as 21.45 J/^{o} whereas the ignition delay was shorter than diesel. The peak of other blends and diesel were almost identical. D100 showed 20.32 J/^{o}, D50CBD50*n*B10 showed 20.83 J/^{o} and D50CBD50*n*B20 showed 20.9 J/^{o} at no load condition respectively. At part loads, the highest ROHR was shown by the blend D50CBD50 as 38.23 J/^{o} which had the shorter ignition delay compared to diesel fuel. Diesel showed value as 34.32 J/^{o} which was 6.6% lower than D50CBD50. On adding *n*-butanol to the biodiesel blended by upto 10% and 20%, it showed a significant decrease in the rate of heat release as 35.28J/^{o} and 35.7J/^{o} which was 7.7% and 6.6% lower than D50CBD50. This may be due to lesser calorific value than diesel biodiesel blends. At full load conditions, highest ROHR was for D50CBD50*n*B20 as 41 J/^{o} at 2^{o}aTDC. Highest value of ROHR for diesel was shown as 40.42 J/^{o} at TDC which was 6.6 % higher than D50CBD50 and 7.3%higher than D50CBD50*n*B10. This may be due to higher calorific value. D50CBD50 and D50CBD50*n*B10 showed the rate of heat release values as 37.72 and 37.43J/^{o} respectively. Combustion duration was found to be lower for diesel-biodiesel and oxygenated fuels at part and full load conditions compared to diesel [30]. It was noticed that burning initiated sooner for biodiesel at various load operations. It was clear that at low loads, diesel had a lower heat release value than biodiesel. At higher loads, diesel had a higher heat release compared to biodiesel which may be due to longer ignition delay.

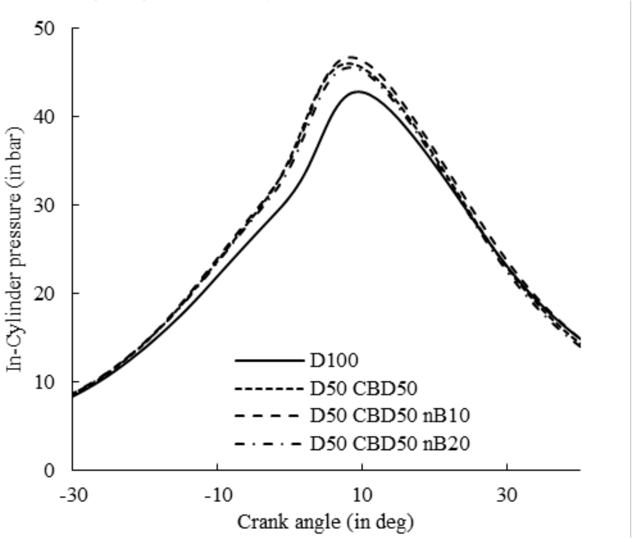

Fig.4. Variation in In-cylinder pressure at No load condition

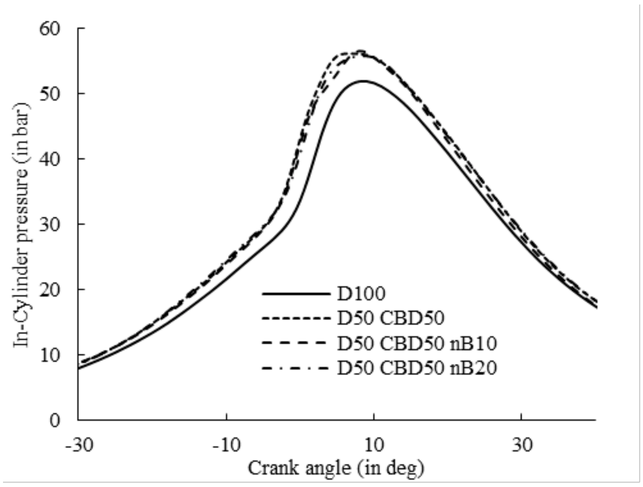

Fig 5. Variation in In-cylinder pressure at Part load condition

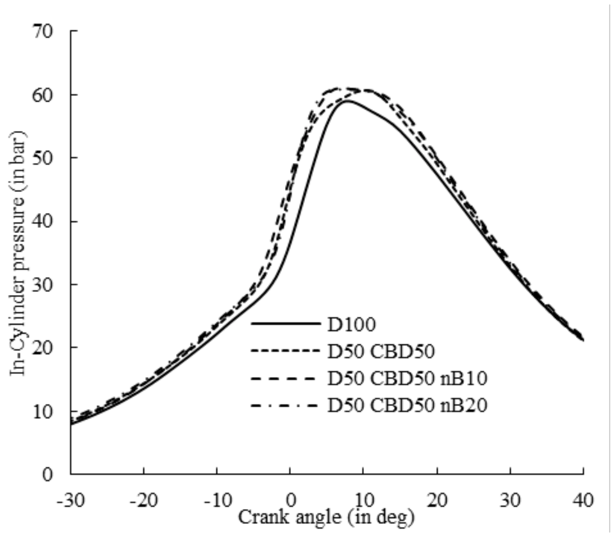

Fig 6. Variation in In-cylinder pressure at Full load condition

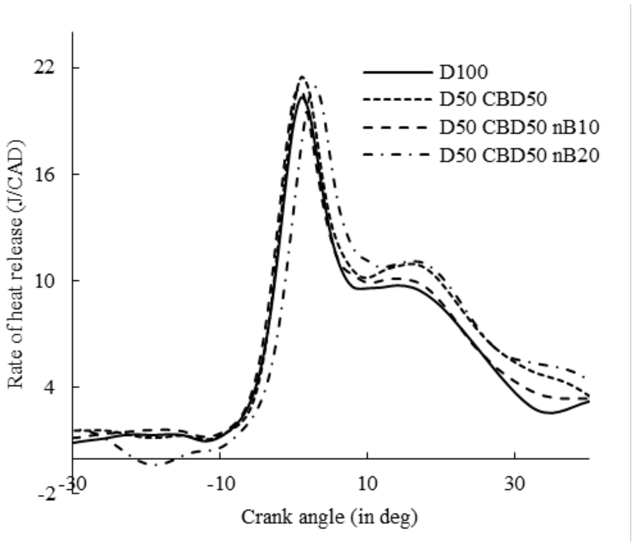

Fig 7. Variation in Rate of heat release at No load condition

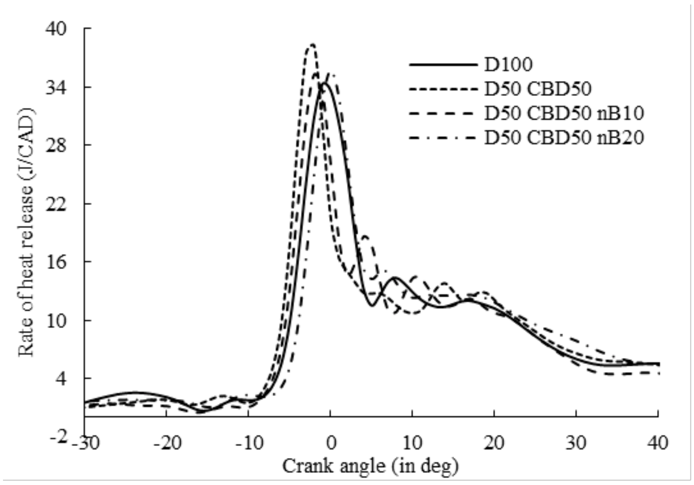

Fig 8. Variation in rate of heat release at part load conditions

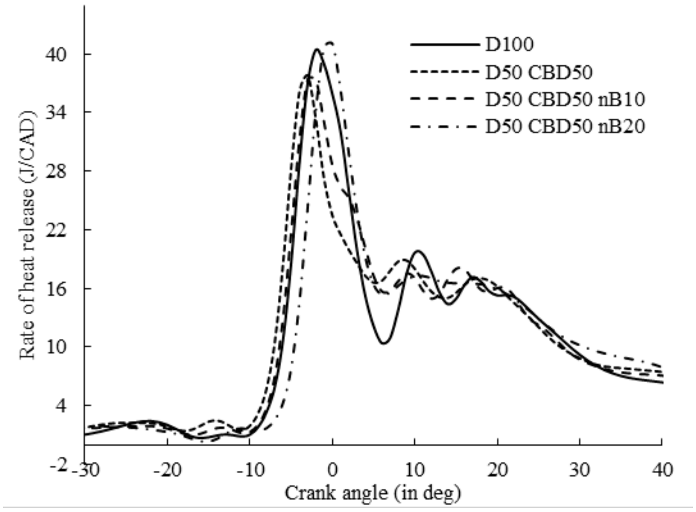

Fig 9. Variation in Rate of heat release at Full load condition

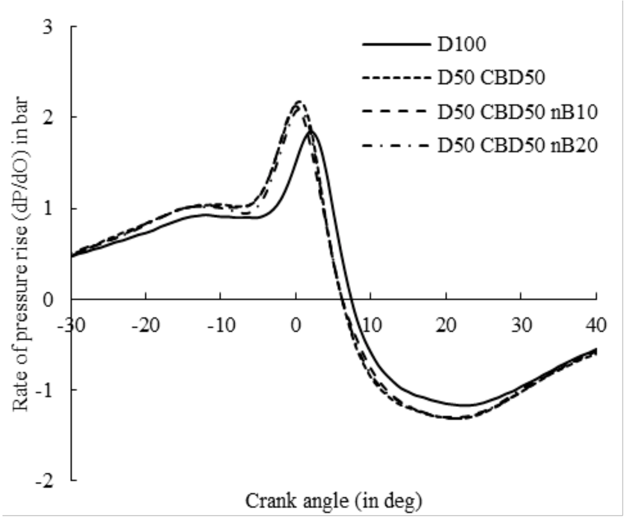

Fig 10. Variation in Rate of pressure rise at No load condition

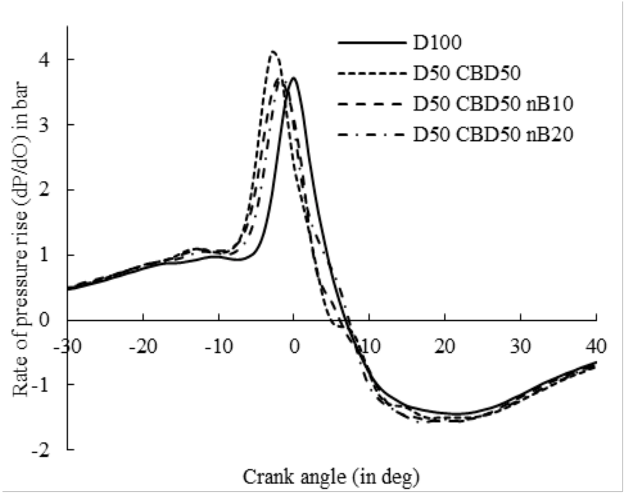

Fig 11. Variation in Rate of pressure rise at Part load condition

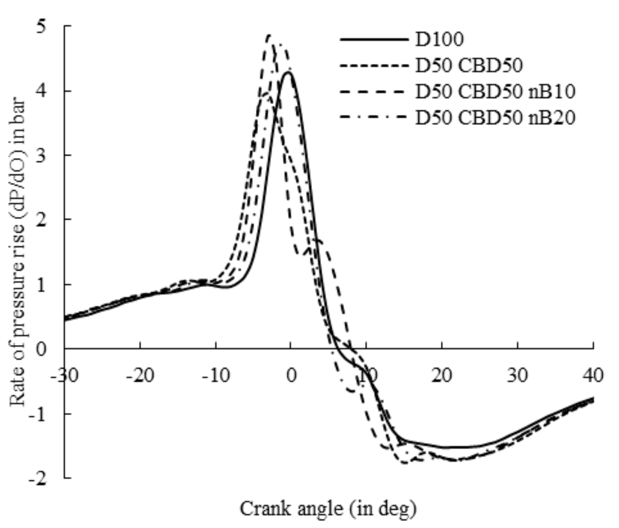

Fig 12. Variation in Rate of pressure rise at Full load condition

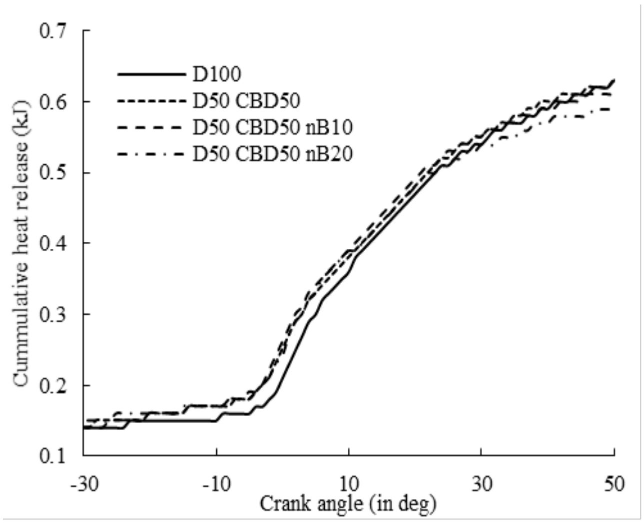

Fig 13. Variation in Cumulative heat release at No load condition

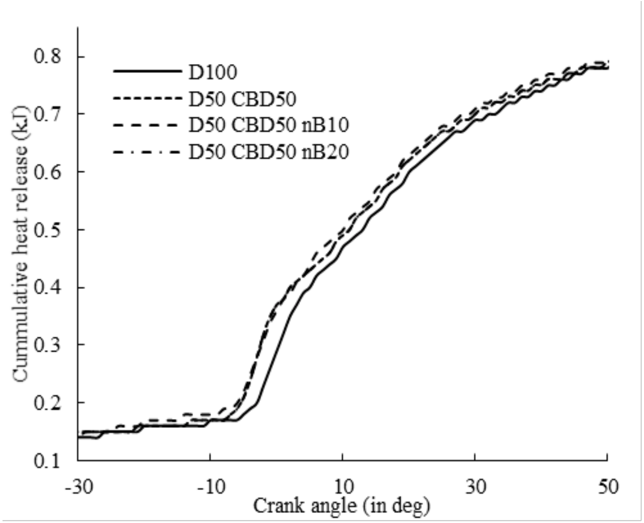

Fig 14. Variation in Cumulative heat release at part load condition

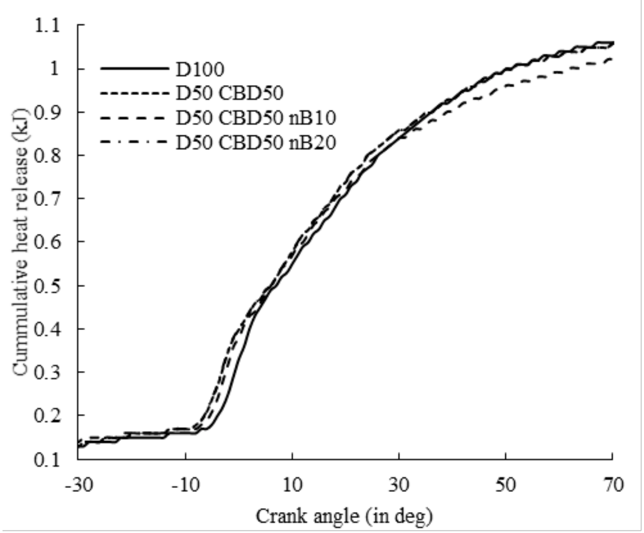

Fig 15. Variation in Cumulative heat release at Full load condition

This may be due to the SOC being later for diesel than biodiesel at lower engine loads. At no load condition, diesel-biodiesel blend showed a higher ROHR as 21.45 J/° whereas, diesel showed a lesser heat release where combustion starts later than biodiesel. On adding oxygenates heat release decreased than D50CBD50 which may be due to the viscosity index, low Cetane value, high latent heat of vaporization which decreased the cylinder temperature. Hence, the rate of heat release reduced as the combustion occurred inside the chamber were in-cylinder temperature was low. For part load condition, similar trend was followed whereas for full load condition, heat release got increased by up to 6.6% compared to D50CBD50 which might be due to higher temperature achieved, inside the cylinder which increased the in-cylinder temperature [34].

4.3 *.Rate of Pressure Rise*

Rate of pressure rise is denoted as dp/dθ i.e., increase in pressure with respect to crank angle degree. Oxygenated blends and diesel-biodiesel blends were compared with diesel as reference at no load, part load and full load condition. D50CB50 showed the highest value as 2.16 bar/° which was 14% higher than diesel whose value was 1.85 bar/°. This may be due to very small amount of fuel being injected at low loads. Oxygenated blends 10% and 20% showed a lesser pressure rise of 2.13 and 2.06 bar/° which was lower than D50CBD50 but higher than diesel at no load condition as shown in Figure 10. At part load condition as shown in Figure 11, higher rate of rise in pressure was delivered by D50CBD50 as 4.1 bar/° at 3° bTDC. This may be due to shorter ignition delay than diesel which had a good mixing of air and fuel making premixed combustion phase better with a good combustion raising the pressure rise. Oxygenates addition to biodiesel blends by 10% and 20% gradually decreased the pressure rise upto 3.7 and 3.68 bar/° [24]. As the load was increased i.e., at full load conditions as shown in Figure 12, D50CBD50 showed lowest pressure rise as 3.95 bar/° which may be due to longer ignition delay period. Oxygenated blend D50CBD50nB10 showed highest pressure rise of 4.86 bar/°, where the ignition delay was less compared to diesel and diesel-biodiesel. Diesel had higher pressure rise than D50CBD50 as 4.26 bar/° which may be due to the heat release rate being high at the premixed burning phase. It was notable that ROPR for diesel was lower for no load condition which may be due to the lesser fuel amount being injected at low loads where accumulation of fuel will be lower thereby making the SOC at later crank angle. Hence, less heat release reduced the rate of pressure rise. At higher engine loads, ROPR was high for diesel which was due to the increased release of heat energy at premixed burning phase. On adding oxygenate to the diesel-biodiesel blend, the fuel property got increased which in turn gave better release of heat during the premixed combustion phase thereby raising the ROPR significantly. Oxygenate blend D50CBD50nB10 showed the better rate of pressure rise during all the operating modes. Also adding oxygenates up to 10% increases the blend property better.

4.4 *. Cumulative Heat Release*

Cumulative heat release rate explains the rate of heat release in premixed and diffused combustion duration. The details about CHR were described in the Figure 13, 14 and 15at no loads, part load and full load condition with the constant speed of 1500 rpm. It is seen clearly that the load had direct impact on combustion phase. Higher loads increased the heat release during the diffusion phase. For lower loads, premixed combustion phase had less variations in terms of CHR as well as pressure. The fuels studied here also showed similar trend. Highest CHR values were seen around 55° to 90° aTDC. For no load conditions, the peak value for diesel was 0.6kJ from 55° to 79° aTDC, for D50CBD50 it was 0.64kJ from 57° to 77° aTDC, for D50CBD50*n*B10 it was 0.63kJ from 58°to 75° aTDC and for D50CBD50*n*B20 it was0.66kJ ranged between 72° to 76° aTDC. For part load condition, the peak value for diesel was 0.84kJ from 73° to 88° aTDC, for D50CBD50 it was 0.83kJ from 69° to 89° aTDC, for D50CBD50*n*B10 it was 0.82kJ from 67° to 73° aTDC and for D50CBD50*n*B20 it was 0.83kJ which ranged between 70° to 90° aTDC. For full load conditions, the peak value for diesel was 1.05kJ from 73° to 77° aTDC, for D50CBD50 it was 1.06kJ from 70° to 81° aTDC, for D50CBD50*n*B10 it was 1.03kJ from 76° to 77° aTDC and for D50CBD50*n*B20 it was 1.07kJ which ranged between 72° to 81° aTDC. It clearly depicted that peak heat release for biodiesel was achieved earlier than diesel at all loads. This may be due to better combustion based on the presence of oxygen concentration which was more than diesel [25].

4.5 *. Effect of Engine Load on Ignition Delay*

Ignition delay is the time period between the start of injection timing and the time at which the combustion starts. Fuels derived from petroleum products have longer ignition delay which may be due to lower oxygen concentration besides having poor atomization which slowed down the mixing process. Biodiesel had higher O_2 concentration which ensured better atomization showing a shorter ignition delay.

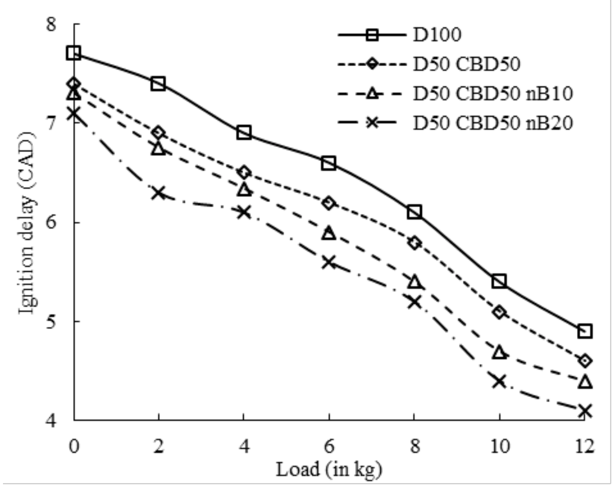

Fig 16. Variation in Ignition delay at all loads

Figure 16 explains the details of delay period for the test fuels diesel, D50CBD50, D50CBD*n*B10 and D50CBD*n*B20. Diesel showed the higher ignition delay for all the loads ranging from 7.7° crank angle to 4.9° crank angle. As the biodiesel was blended with diesel there was reduction in premixed combustion phase thereby reducing the ignition delay which may be due to the existence of oxygen content in the biodiesel [25]. For D50CBD50, the ignition delay was shorter than diesel ranging from 7.4° crank angle to 4.6° crank angle. On adding *n*-butanol (oxygenate) which had higher concentration of oxygen, thereby providing better combustion with shorter ignition delay than diesel-biodiesel blend D50CBD50. Henceforth, further decrease in ignition delay was seen in oxygenated fuels. D50CBD*n*B10 showed an ignition delay ranging from 7.3° crank angle to 4.4° crank angle whereas for D50CBD*n*B20, the range was from 7.1° crank angle to 4.1°crank angle. The ignition delay decreased as the load was increased and oxygen concentration is increased.

4.6 .Unburned Hydrocarbon Emission

Fuel droplets staying in the crevice side lead to the unburned hydrocarbon emission. Figure 17 shows the UBHC emission with respect to the loads. UBHC emission is the exhaust emission due to incomplete combustion of fuel parcels inside the cylinder. It can be noticed that as the load was increased, UBHC emission reduced. At lower loads, UBHC emission was higher which may be due to the improper mixing of fuel since the fewer amounts of fuel was injected inside the cylinder and prevailing lower temperature [26]. Increasing the blend up to 50% with diesel showed a lesser UBHC emission at lower loads which may be due to relatively larger oxygen content which ensured better combustion, whereas at higher loads, UBHC was higher which might be due to higher viscosity of D50CBD50 fuel blend. On adding *n*-butanol up to 10% reduced the viscosity and by which the mixing of air fuel occurred faster enhancing a better combustion. This may be the reason for lower emission of oxygenated fuels compared to diesel (D100) and biodiesel blend (D50CBD50). D50CBD50*n*B10 showed the lowest HC emission during full, part and low load conditions as 31.8%, 22.7% and 10% on comparison with conventional diesel fuel respectively.

4.7 .Carbon Monoxide Emission

Fuel air ratio affects greatly the CO emission. Improper flame propagation towards the mixture, unavailability of required oxygen content and prolonged ignition delay is the reasons for the formation of CO emissions. Figure 18 shows the variation of CO emission with respect to the loads. It can be noted from the graph that diesel fuel depicted the highest CO formation at all loads. This may be due to the non-availability of oxygen in the chemical structure of diesel fuel. On the other hand, in the biodiesel blend and oxygenated blend, molecular oxygen was present which enhanced complete combustion. The lowest CO emission was noted with D50CBD50*n*B10 blend. This may be due to the lower viscosity. Also, it can be noted that D50CBD50*n*B20 blend showed a higher value than D50CBD50*n*B10 which may be due to lower Cetane number which increased the ignition delay leading to CO emission due to insufficient time for fuel burning [28].

4.8 .Carbon dioxide Emission

CO_2 emission in the exhaust gas denotes the complete burning of the fuel inside the cylinder. Figure 19 illustrates the CO_2 emission with respect to the loads. CO_2 emission was lesser for all the blends compared to diesel fuel. This may be due to higher conversion of CO to CO_2 emission as the complete combustion has occurred. As the load was increased, better combustion was expected as the ignition delay was reduced. It is to be noted that CO_2 emission increased up to 30%. The highest carbon dioxide emission was emitted by the oxygenated blend D50CBD50*n*B10 [27].

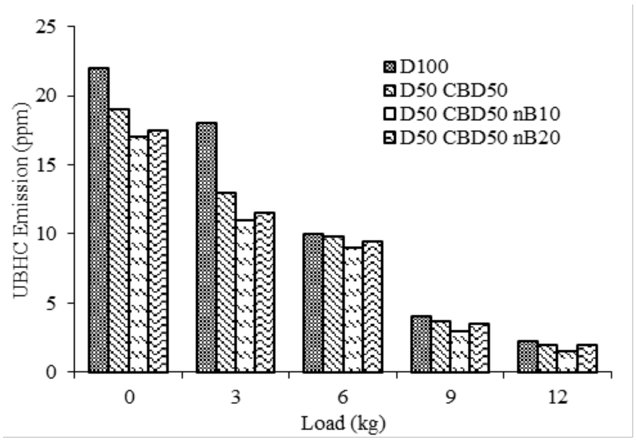

Fig. 17. UBHC Emission

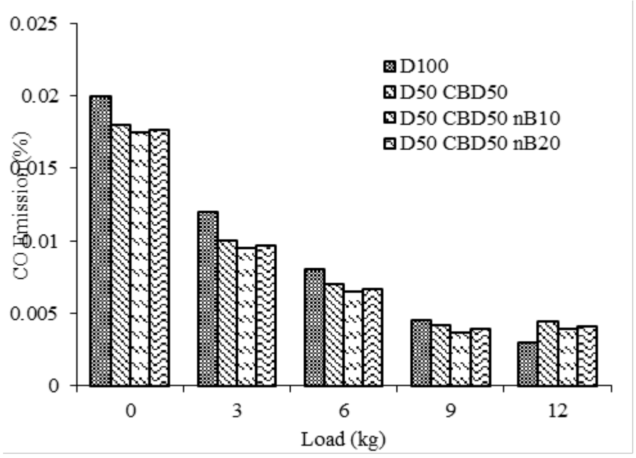

Fig. 18. CO Emission

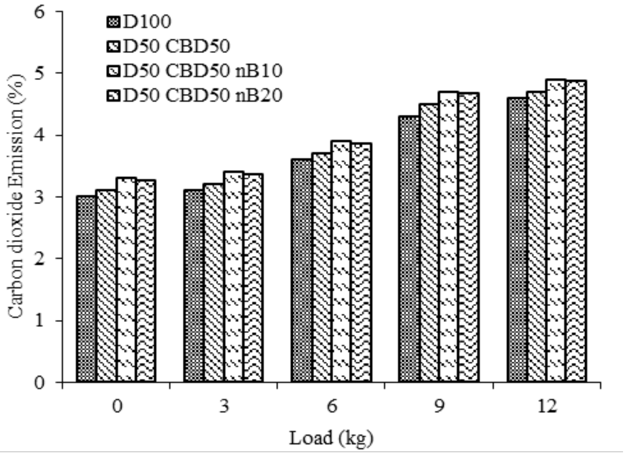

Fig. 19. CO_2 Emission

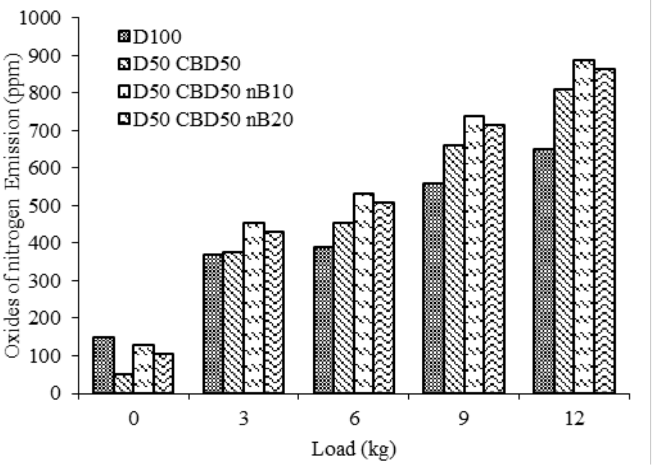

Fig. 20. NO_x Emission

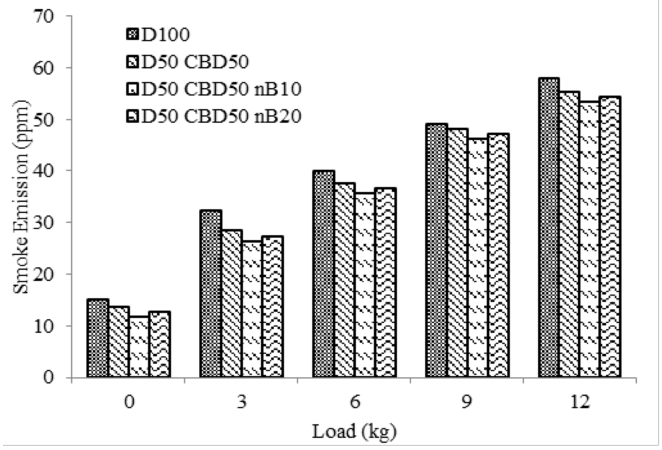

Fig. 21. Smoke Emission

4.9 Oxides of Nitrogen Emission

Figure 20 shows the comparison of NO_x emission with respect to the loads for D50CBD50, D50CBD50*n*B10 and D50CBD50*n*B20 along with mineral diesel as conventional fuel. As can be seen in the plot, lowest NO_x emission was showed by diesel fuel in all loads which may be due to absence of molecular oxygen. D50CBD50 blend showed NO_x emission as 50 ppm to 812 ppm with an increasing load. This may be due to the complete combustion of fuel which enhanced the inbound temperature which led to increased NO_x emission. Adding 10% *n*-butanol increased the NO_x emission from 129 ppm to 889 ppm. Further addition of oxygenate showed a reduced NO_x emission which may be due to lower calorific value and Cetane number [29-32].

4.10 Smoke Emission

Incomplete combustion results in carbon enriched particulates in the exhaust as emission called smoke. Figure 21 shows the smoke emissions with respect to the loads. It can be noted that all the blends emitted a lower smoke than diesel fuel. Adding oxygenate to the fuel must have decreased the viscosity which led to better atomization thereby enhancing better combustion leading to reduced smoke emissions. Adding oxygenate to the fuel blend decreased the smoke emission from 8 to 20% on comparing it with the diesel fuel [33].

5. Conclusion

The experimental analysis of test fuels derived from cottonseed oil D50CBD50, D50CBD*n*B10, D50CBD*n*B20 and reference fuel mineral diesel were studied in this work. Analysis were done for in-cylinder pressure, rate of heat release, rate of pressure rise and cumulative heat release rate at no load, part load and full load conditions. The conclusions were made based on the work carried out during analysis.

1) D50CBD*n*B10 showed the highest in-cylinder pressure at all loading conditions as 46.63 bar at 8^oaTDC for no load, 56.7 bar at 8^oaTDC for part load and 60.98 bar at 7^oaTDC for full load. The higher pressure maybe due to the premixed combustion phase being smaller, lesser ignition delay and on adding oxygenate, it reduced the viscous nature of blend thereby making the combustion better by raising the cylinder pressure.
2) Rate of pressure rise showed the similar trend of in-cylinder pressure ie., D50CBD50*n*B10 showed highest ROPR at all load conditions as 2.16 bar/^{o} at 1^oaTDC for no load, 4.1 bar/^{o} at 3^obTDC for part load and 4.86 bar/^{o} at 3^obTDC for full load. This peak of ROPR may be due to the highest heat release in the premixed combustion phase of the test fuel D50CBD50*n*B10.
3) Heat release rate was dependent on ignition delay, start of combustion and diffused combustion phase which had effect on the in-cylinder temperature. These variations may also change based on loading conditions. Here the results showed highest ROHR for D50CBD50 at no load as 21.45 J/^{o} at 10^obTDC and 38.23 J/^{o} at 2^obTDC for part load. For full load condition, highest ROHR was for D50CBD50*n*B10 which is 41 J/^{o} at TDC.
4) Highest cumulative heat release for the loading conditions differed for the test fuels. This may be due to the duration of combustion in premixed and diffused phase. For no load condition, diesel showed the highest CHR as 0.66kJ ranging from 72^o to 76^o aTDC, for part load condition D50CBD50*n*B20 showed the highest CHR as 0.84kJ ranging from 73^o to 88^o aTDC and for full load

condition diesel fuel showed the highest CHR as 1.07kJ ranging from 72° to 81° aTDC.

5) Ignition delay is the detailed combination of physical and chemical delay. Here in the result, decreasing trend was followed in diesel fuel, D50CBD50, D50CBD50*n*B10 and D50CBD50*n*B20. Diesel fuel had the ID ranging from 7.7° crank angle to 4. 9° crank angle. Also it was the highest value at all loading conditions compared to all other test fuels. D50CBD50 had the ID range from 7.4° crank angle to 4. 6° crank angle which was lower than diesel. D50CBD50 *n*B10 had the ID range from 7.3° crank angle to4.4° crank angle which was lower than D50CBD50. D50CBD50*n*B20 had the ID range from 7.1° crank angle to 4.1° crank angle which also was lower than D50CBD50*n*B10.
6) UBHC and CO emissions decreased as the load increased, whereas smoke, CO_2, NO_x emission had significantly increased with the rise in engine load.

Nomenclature & abbreviations	
CI	Compression ignition
TDC	Top dead centre
SOC	Start of combustion
ROPR	Rate of pressure rise
ROHR	Rate of heat release
CH_3OH	Methanol
KOH	Potassium hydroxide
CH_3OK	Potassium methoxide
GCMS	Gas chromatogram mass spectrometer
FTIR	Fourier transform infrared
bTDC	Before top dead centre
DC	Direct current
CR	Compression ratio
IT	Injection timing
CAD	Crank angle degree
CBD	Cotton seed biodiesel
CHR	Cumulative heat release

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