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RCCI COMBUSTION WITH A PARTIALLY PRE-MIXED CONCEPT IN A DIESEL ENGINE USING BIODIESEL, DI-ETHYL ETHER, AND ETHANOL BLENDS

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Abstract

This research aims to optimize combustion by using alternate fuel sources. It also studies biodiesel with an ignition improver blend at an increased compression ratio of 20 by supplying ethanol with preheated air. In order to obtain baseline data, the research process was first carried out with diesel. It was then analyzed using the blend of diesel, Prosopis juliflora methyl ester as biodiesel, and an ignition improver (di-ethylether) in the same conventional mode. This blend, known as B30DEE1, is composed of 30% biodiesel, 69% diesel, and 1% di-ethylether by volume. The experiment was repeated by using combustible mixture B30DEE1+ETH, which contains biodiesel 30%, diesel 69%, di-ethylether 1% and ethanol 10% on a volume basis. The ethanol mists were added by port injection in the proportion of 10% into the preheated air stream to attain partially premixed condition for reactivity controlled combustion. The elevated brake thermal efficiency and reduced HC and CO concentrations were recorded along with the increased heat release rate at reactivity controlled compression ignition (RCCI) mode.

Keywords: reactivity; combustion; biodiesel

1. Introduction

The octane number indicates the ability of the fuel to withstand the compression process. The elevated octane number indicates the capability to avoid detonation at the time of compression. The higher octane fuels are used in the gasoline engines with elevated compression. This may generate higher outlet power for these engines. The higher power

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output can be possible with the engine design that allows high compression pressures. The fuels with higher cenate index or with lower octane number are suitable for diesel engines. The fuels used for the diesel engines do not expose to the compression pressures, but they are injected directly into the highly compressed hot air. By utilizing advanced combustion modes, CI engines can operate on fuels with higher cetane and octane numbers.

The alternate fuels with these properties can be used in diesel engines to reduce conventional fuel usage. By adopting a combustion strategy like reactivity controlled compression ignition (RCCI), fuels with diverse physical and chemical qualities will be able to be used. The premixed charge compression ignition (PCCI) and homogeneous charge compression ignition (HCCI) are known for their elevated thermal efficiencies and reduced exhaust emissions like particulate matter. The limitations like mixture homogeneity, combustion phase control, and cylinder wall impingement draw attention towards RCCI from PCCI and HCCI modes of combustion. The RCCI is a duel fuel strategy of combustion with different auto-ignition temperatures. This phenomenon is mainly concerned with decreasing emissions like NOx and soot below the standard emission norms. And also to control the combustion phase, which was difficult to achieve in PCCI and HCCI modes at a wider range of load and speed conditions.

The combustible mixture reactivity varies across the combustion space in RCCI mode of combustion. The main injector can be used to directly inject fuel with a greater cetane number into the combustion chamber, while the port injection can be used to introduce fuel with a lower cetane number. The fuel's reaction rate can be affected by the injection process, such as early or late injections. The fuel properties, like the cetane index and boiling points, can also influence the reaction rate. The above parameters can influence the RCCI mode of combustion to a great extent. The flame temperature can be reduced by following these advanced combustion modes. Including RCCI, lower flame temperatures can be achieved by other combustion strategies like HCCI, PCCI, gasoline compression ignition (GCI), and partially premixed combustion (PPC). Emissions like NOx and soot strongly depend on combustion temperature and equivalence ratios. By controlling these parameters or by promoting low-temperature combustion, a reduction in these emissions will become possible. Particulate matter and NOx generation can be somewhat inhibited by lowering in-cylinder temperatures and encouraging appropriate air-fuel mixture formation. By using only neat diesel with a higher cetane index and lower volatility, it will become difficult to prepare a homogeneous mixture before the start of the combustion phase in the conventional mode of operation. It will be feasible to create a homogeneous charge prior to the commencement of burning by implementing sophisticated combustion techniques like RCCI, which combine the use of a low- and high-reactivity fuel. Figure 1 displays the schematic diagram for RCCI combustion.

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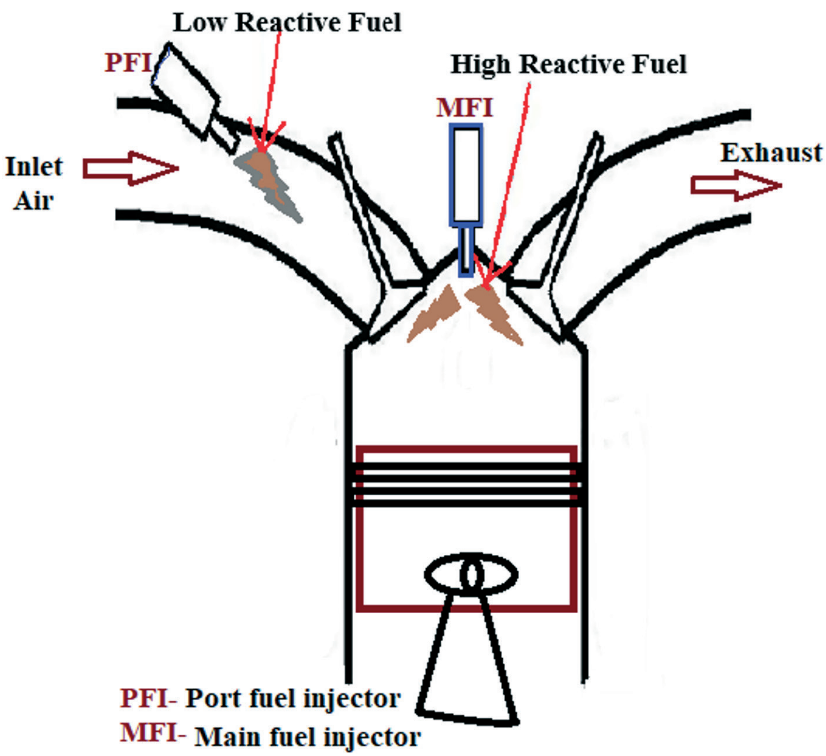

Fig. 1. Schematic diagram for RCCI enabled engine

The dual fuel combination was used by Kokjohn et al. [1] to investigate RCCI combustion. High-reactivity diesel fuel was pumped straight into the main chamber, whereas low-reactivity petrol was port-injected. The lack of combustion phase control along with higher pressure rise rates, generally arise in HCCI and PCCI modes at peak loads are addressed in this experiment analysis using RCCI. The aforementioned issues are fixed by altering the ratios of diesel and petrol at various load and speed conditions.

It was also demonstrated by Tang Q et al. and Kavuri C et al. [2, 3] that the combustion phasing and control could be regulated over a broader range of loads through raising the reactivity variance among the two fuels employed in the RCCI mode. Prikhodko VY et al [4] adopted the multi-stage injection methodology and EGR for increasing ignition delay period that can restrain formation of the particle. In the RCCI mode of operation, this may result in improved uniform mixture generation prior to the commencement of combustion. Diesel and natural gas were used in an experimental investigation with significant duty RCCI operation by Derek E. Nieman et al. [5]. It has been concluded that to make RCCI possible at moderate to high loads increased amounts of EGR along with reduced compression ratios are required. The

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obtained results also concluded that precise control over the injection pressures also required phase control for the combustion. Reed M. Hanson et al [6] utilised the gasoline and diesel combinations to attain reactivity control on US EPA 2010 heavy-duty CI engine. High-reactive diesel was fed straight into the combustion chamber, while gasoline was introduced into the port at an early cycle. At 9 bar IMEP condition or at medium engine load, decreased NOx and PM pollutants were observed without treatment, and an increased indicated thermal efficiency of 53% was also observed. Experimental work was done on a diesel engine by Splitter, D. A. et al. [7] at 1300 rev/min shaft speed and an unchanged net IMEP of 8.45 bar. The losses, like reduced thermal efficiency, can be minimised by regulating the intake pressures and temperatures, which are directly influenced by reactivity variations of the test fuels. It has also been concluded that RCCI allows flexible operating circumstances by changing the quantity of less-reactivity fuel and the main fuel injection rate. The maximum amount of energy has to be gained from less-reactive fuel for improved thermal efficiency and reduced NOx and soot formations [8]. Prior to high-reactive fuel direct injection, the primary goal of early port injection of low-reactive fuel is to allow for improved fuel mixing with air and re-circulated exhaust gases. This is meant to optimize the mixture uniformity in the RCCI mode [9]. The combustion and reduction in pollutant formations were improved using low-reactivity gasoline, inlet manifold injection, and directly injected high-reactive diesel and polyoxymethylene dimethyl ethers (PODE). The PODE fuel consists of an increased cetane number and is also highly volatile in nature. Notifications included reduced CO and smoke pollution, regulated mixed combustion, higher cylinder pressures, and increased indicated thermal efficiency [10]. The RCCI combustion using the combination of diesel with gasoline and 50% to 70% exhaust gas recirculation (EGR) reduced the peak pressures and heat released rates at lower loads [11]. The EGR percentage controlled the NOx formation, and elevated injection pressures minimized the ignition delay period, and elevated the atomization characteristics. The emissions of HC, and CO were increased by using the above operating parameters. The RCCI engine's efficiency and combustion attributes are significantly influenced by the spray direction, injection start, premixed proportion, and injection pressures. By reducing the angle of spray from 74° to 55°, the formation of pollutants (HC and CO) were reduced, resulting in improved combustion efficiency [12]. Temperatures and pressures inside the cylinder increased when the injection timing was advanced. As a result of the longer mixing time, NOx emissions increased but CO and HC emissions decreased. It was concluded that the optimum spray angle (55°) and injection timing, or start of injection, can play a crucial role in improved RCCI combustion by directly influencing the heat release rates and rate of pressure rise. The following Table 1 indicates the comparison of HCCI, PCCI and RCCI modes of the combustion.

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Tab 1. Comparison of HCCI, PCCI and RCCI modes of combustion [26]

No	HCCI	PCCI	RCCI
1	Low combustion temperatures with increased thermal efficiency	Low combustion temperature, the brake thermal efficiency depends upon the operating parameters and fuel type.	Low combustion temperature with increased brake thermal efficiency when compared to HCCI and PCCI.
2	It is a single or dual fuel combustion strategy	It is a single or dual fuel combustion strategy	It is a dual fuel combustion strategy using two fuels blends with different reactivity when compared to HCCI and PCCI.
3	Reduced NOx and PM emissions	Reduced NOx and PM emissions when compared to HCCI	Decreased NOx emissions even at elevated BMEP when compared to HCCI and PCCI
4	Increased CO and HC emissions	Elevated CO and HC pollutants when compared to HCCI	Reduced CO and HC emissions with alcohol as low reactive fuel when compared to HCCI and PCCI
5	Homogeneous combustible mixture	Ignition mode is not completely homogeneous when compared to HCCI	Ignition mode is not completely homogeneous in comparison to HCCI and PCCI.
6	Limited operating range	Enhanced operating performance range when compared to HCCI.	Enhanced operating performance range when compared to HCCI and PCCI.
7	The mixture preparation methods are not sensitive.	The mixture preparation methods are more sensitive when compared to HCCI.	The mixture preparation methods are more sensitive when compared to HCCI and PCCI.
8	The blends of diesel and fuels like ethanol and biodiesel are recommended for extending the load range	The blends of diesel and renewable fuels like ethanol and biodiesel are recommended for extending the load range	Low reactive fuel, is used up to 90%, plays an important role in extending the load range

Various researchers are also focusing on finding out the suitability of alcoholic fuels less reactivity for RCCI combustion. The alcohol group fuels like ethanol, methanol, and n-butanol are known for their high latent heat of operation, low pressure vaporisation, and reduced reactivity rates. These properties can perform long-range, stable RCCI combustion [13]. From the above literature survey, it was observed that biodiesels, along with alcohol fuel combinations, can be chosen to generate RCCI combustion. The low-reactive alcohol fuel was chosen as n-butanol, and the high-reactive biodiesel blended diesel fuel was chosen for main injection. In this present study, the selected qualities of Prosopis juliflora methyl ester (PJME) biodiesel are closer to those of diesel. Its seed oil is naturally non-edible. The study is unique in that no combustion evaluations on the use of PJME and ethanol are available for RCCI generation in an ordinary CI engine.

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The seed oil of Prosopis juliflora is inedible by nature. The Prosopis juliflora plant is widely available, and it consists of 44 species throughout the world [14]. The higher the cetane number, oxygen percentage, and heating value of this fuel, the closer it is to diesel, making it a good alternative source for diesel engines. Conventional CI engines may run on an alcoholic fuel, such as ethanol, with little to no engine modifications. Diesel's characteristics are contrasted with ethanol's and Juliflora oil methyl ester's in Table 2. The fundamental reason for using ethanol is that it has a higher oxygen percentage, which improves the efficiency of combustion. It also has properties like reduced evaporation pressures, increased heating values, and a lower cetane number.

To ensure the biodiesel quality the American Society for Testing and Materials (ASTM) specification D6751, and for ethanol D4806 were proposed. This specification is based on inputs from researches, engine manufacturers and many other fuel related sources. The standards of ASTM will ensure optimum level of quality for the biodiesel. As per ASTM D6751 regulations the biodiesel should consist of "mono-alkyl esters of long-chain fatty acids". The biodiesel should contain one ester link connected to each molecule. The molecules containing three and above ester linkages will not be considered as biodiesel. The fuel which consists of long-chain fatty acid mono alkyl esters is designated as B100 (100% biodiesel) and it must satisfy the ASTM D6751 standards. Table 3 shows the characteristics of biodiesel in comparison to diesel according to ASTM and Bureau of Indian Standards (BIS) standards.

Tab. 2. Properties of the test fuels

Specifications	Diesel	Ethanol	PJME	ASTM D6751 Standard for biodiesel	ASTM D4806 Standard for ethanol
Cetane Index	50	8	49	47–65	2–12
Octane Number	–	108	–	–	108–109
Stochiometric A/F ratio	14.95	9	12.25	–	9
Oxygen (wt %)	0	34.7	12	11	–
Carbon (wt %)	86.12	52.1	77	77	–
Hydrogen (wt %)	13.87	13.5	12	12	–
Lower Calorific Value (kJ/kg)	43 390	26 700	39 500	~33 342	27 000
Kinematic Viscosity (cSt) at 40°C	3.9	1.056	4.90	4.0–6.0	1.525
Density (kg/m^3) at 15°C	838	810	920	874.73	794

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Tab. 3. ASTM & BIS standards for biodiesel in comparison to diesel [US Department of Energy 2016, and Bureau of Indian Standards, 2005]

Fuel Property	Diesel		Biodiesel	
Fuel standard	ASTM D975	IS 1460	ASTM D6751	IS 15607
Max. calorific value (kJ/kg)	~38624	–	~35687	–
Lower heating value (kJ/kg)	~36113	–	~33342	–
Kinematic viscosity (cSt) at 40°C	1.3–4.1	2–4.5	4.0–6.0	2.5–6
Density (kg/m^3) at 15°C	850.76	820–845	874.73	860–900
Carbon (wt %)	87	–	77	–
Hydrogen (wt %)	13	–	12	–
Oxygen (wt %)	0	0.6	11	–
Sulfur, wt % (ppm)	0.0015 (15 ppm max.)	50	0.0–0.0015 (0–15 ppm)	50
Carbon Residue (% mass)	0.35	0.30	0.05	0.05
Cetane index	40–55	46	47–65	51

2. Materials and Methods

The standards of American Society for Testing and Materials (ASTM) D6751 are applied for the preparation of biodiesel blends. By adhering to these guidelines, the test fuels were prepared by maintaining the blend density below 860 kg/m^3, the kinematic viscosity below 4.1 cSt, and the heating value of the blends above 35 MJ/kg.

Global reactivity and reactivity gradients are two ways to organise the RCCI mode. The fuel properties used and the volume of fuel injected into the combustion space determine the global reactivity. By using different injection strategies of higher octane and cetane fuels, the reactivity gradient can be adjusted. The timings of ignition in RCCI mode can be regulated by varying the high-cetane to high-octane fuel ratio. In this experimental analysis the fuel with a maximum cetane number is introduced directly into the combustion chamber and fuel with higher octane number is fed from the inlet manifold port to create reactivity control inside the combustion chamber. The Juliflora methyl ester was selected due to its greater cetane number, while the ethanol was selected due to its reduced reactivity and higher octane number. Di-ethyl ether, an ignition enhancer, was selected to combine with biodiesel. Both the cetane number and the rate of evaporation can be further raised by the di-ethyl ether.

The test fuel blend consists of 30% biodiesel and 1% ignition improver Diethyl Ether (DEE) and the remaining diesel on a volume basis and is named B30DEE1 (Juliflora Methyl Ester 30%, Diesel 69%, and DEE 1%). The 30% biodiesel blend was chosen because with the 25% biodiesel blend, the experimental results attained a higher brake thermal efficiency of 32% in the previous experimental analysis [17]. And for diesel, it was 31.1%. The ignition improver DEE is also blended due to its better heating value and cetane index [15].

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In the next stage, the ethanol was port injected into the preheated (35°C–40°C) inlet air stream in an accordance of 10% of the total injected fuel in the main combustion chamber. This combustible mixture was named B30DEE1+ETH (Juliflora Methyl Ester 30%, Diesel 59%, DEE 1% and ethanol 10%). The increased compression ratio of 20 was used with the diesel and test fuel blends from no load to maximum load conditions to increase the operating temperatures. The main objective of injecting ethanol into a preheated air stream is to supply ethanol in the vapor state into the engine cylinder. This can increase the reaction rate and decrease the physical delay period. This can also make the concept of reactivity controlled combustion more practicable. Ethanol in normal conditions can be used with SI engines because of its increased octane number. It is not suitable for diesel engines because of its poor auto ignition characteristics. For this reason Ethanol is restricted to be used only in small proportions in CI engines. The addition of ignition improvers also needs to be done to enhance cetane number to some extent. The compression ratios also can be increased to suppress its poor ignition characteristics. In this experimental analysis the test engine normal operating compression ratio is 18, and it was increased to 20 to make it suitable for utilizing ethanol blended test fuel. Figure 2 depicts the experimental setup's schematic view, and Table 4 provides its technical details. An eddy current dynamometer and a diesel engine form the experimental setup. In addition, a smoke meter and five gas analyzers are included. The experimental configuration is linked to the data acquisition system and includes features for measuring data relevant to combustion. By running a number of trials at the engine's stabilized condition with applied loads, an average value of all the test readings was recorded.

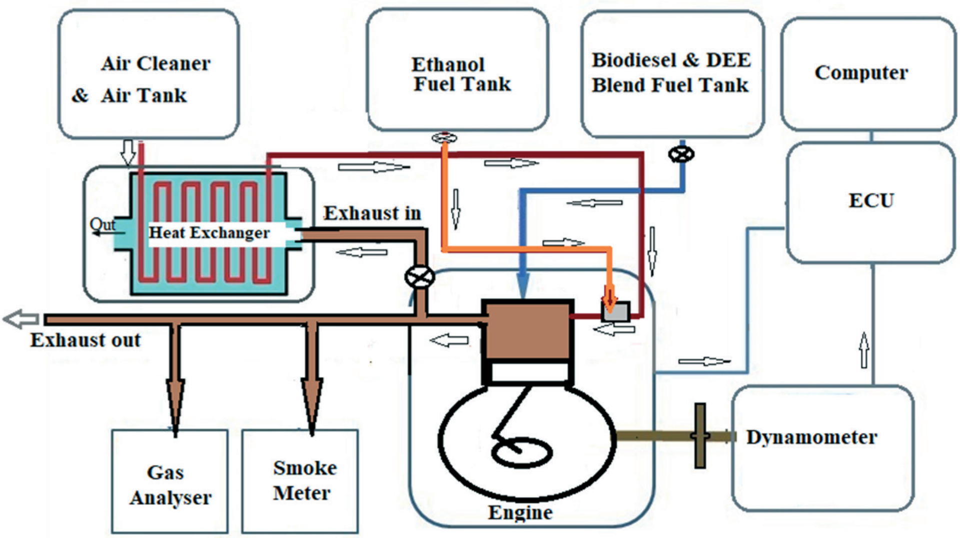

Fig. 2. Schematic view of the test rig along with ECU

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Tab. 4. The engine details

Parameters	Specification
Engine type	Kirloskar, single cylinder four stroke diesel engine
Energy / Speed	3.5 kW / 1500 rpm
Bore diameter x Stroke length	8.75 cm x 11 cm
Compression Ratio	20 (VCR 12 to 22)
Main injection pressure	300 bar
Main injection timing	25° BTDC
Stroke volume	661.45 (cc)
Dynamometer	Eddy current type
Port injection pressure	3 bar (Regulated)
Port injection method	Single, 355° BTDC

3. Results and discussion

3.1. Brake thermal efficiency

This efficiency can be expressed as the inverse product of the fuel's calorific value and the brake-specific fuel consumption, or as the ratio of the quantity of heat that actually turns to brake power to the total heat supplied [16]. Figure 3 illustrates the change in brake thermal efficiency (BTE) in relation to brake mean effective pressure (BMEP).

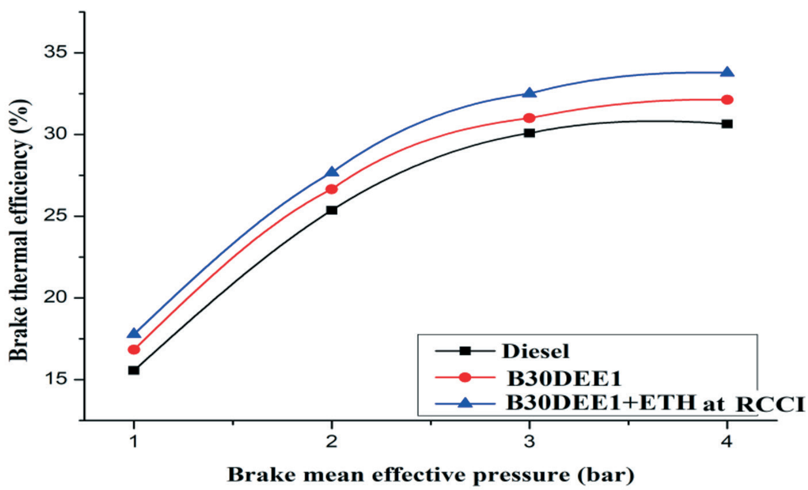

Fig. 3. Variance in brake thermal efficiency along BMEP

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The BTE for the B30DEE1 test fuel blend was more when compared to diesel. The increased cylinder pressures and operating temperatures with the increased compression ratio at maximum BMEP can cause enhanced evaporation and mixing with air [17]. The addition of DEE also improved and maintained the cetane number to the desired level and it also increased the oxygen percentage. The BTE was further increased in the next stage after introducing the ethanol vapors with air intake into the cylinder or with the combustible mixture B30DEE1. These vapours further improved the evaporation rate. Biodiesel and DEE increased the heating values and cetane numbers compatible with diesel. The cetane number and oxygen availability in the B30DEE1+ETH blend resulted in complete combustion, which in turn increased the BTE. The BTE values for diesel, B30DEE1, and B30DEE1+ETH observed from the figure are 30%, 32%, and 33.2%, respectively. The addition of ethanol and biodiesel results in an increase in the consumption of specific fuels. In consideration to diesel, higher fuel energy consumption was observed for both test fuel blend operations in both conventional and RCCI modes of operation. As a result of biodiesel's lower calorific value than diesel, there was more fuel to burn. The engine cylinder's air fuel mixture was not properly mixed due to the elevated viscosity levels of the biodiesel. Therefore, compared to diesel, biodiesel operations require more fuel and energy. A modest suppression of this problem occurs with an increased compression ratio of 18 to 20:1.

3.2. Temperature analysis for exhaust gases

For all of the test fuels, the exhaust gas temperature increases along BMEP. The combustion space temperature affects the NOx formation. The rate at which heat is released during combustion inside affects the exhaust gas temperature as well. This temperature also has some effect on the production of pollutants [18]. The increased exhaust gas temperature is caused by engine performance, diffusion combustion heat release rates, and oxygen availability during combustion.

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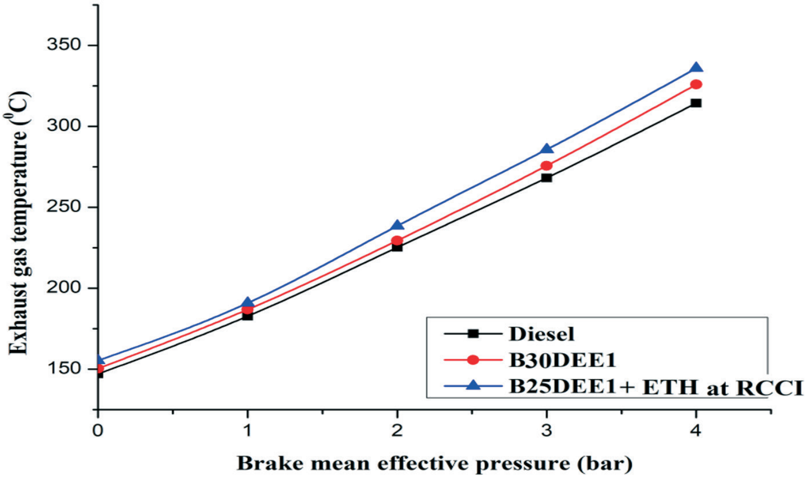

Fig. 4. Exhaust gas temperature with BMEP

For both test fuel mixes, the graphical representation in Figure 4 shows a small rise in exhaust gas temperatures when compared to diesel. Because of the biodiesel blend's slower rate of evaporation, there was an extended delay period, which was the cause of the increase in exhaust gas temperature. The temperature rise of the exhaust gas is also ascribed to the addition of ethanol. Exhaust gas temperatures for diesel, B30DEE1, and B30DEE1+ETH are 309°C, 312°C, and 316°C at maximum BMEP.

3.3. Carbon monoxide emission analysis

The influencing factors for this emission are the availability of oxygen and the reactive rates. In Figure 5, the CO concentration of exhaust gases in relation to BMEP is depicted. Incomplete combustion, along with a rich mixture and heterogeneous combustion with a decreased oxygen percentage can increase this emission [19]. It is observed that CO concentration in the pollutants are almost similar for the test blends in comparison to diesel. The negligible decrease in CO concentration for B30DEE1+ETH is due to a good oxygen percentage and a lower carbon percentage in biodiesel and fuel additives on a mass basis in consideration to diesel. But increased specific fuel consumption is also one of the reasons for CO concentration in the emission. The biodiesel consists of increased levels of unsaturated fatty acids, it can influence the reduced oxidation process, but the addition of DEE suppressed this condition to some extent. The addition of ethanol vapors decreased this emission as it allowed further improvement in flame propagation and the reaction rate. The operating temperatures are further increased with an increased load. This in turn reduced this emission and

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resulted in comparable emissions to diesel with biodiesel in conventional mode and biodiesel and ethanol in RCCI modes.

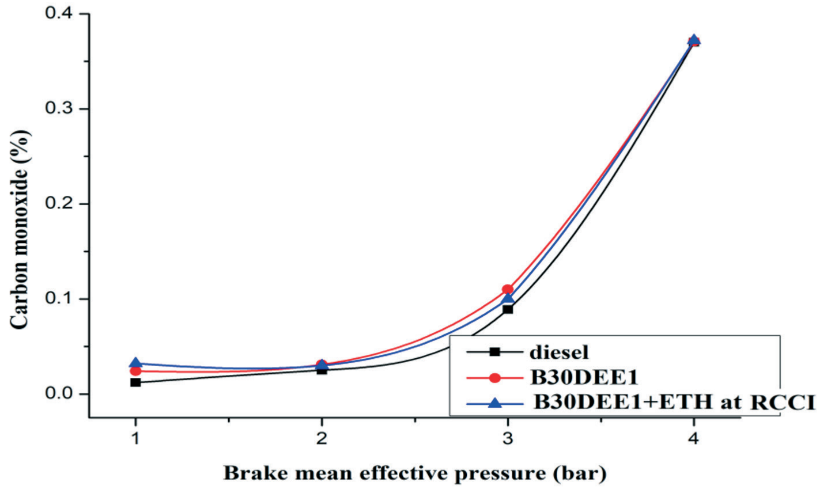

Fig. 5. Carbon monoxide emission versus BMEP

The CO concentration in the emissions at maximum BMEP conditions for diesel, B30DEE1 and B30DEE1+ETH is observed as 0.32%, 0.32% and 0.31% respectively from the figure.

3.4. Unburnt hydrocarbons emission analysis

The operating parameters, atomization of the fuel and combustion quality can influence this pollutant. The emission of HC with respect to BMEP is shown in Figure 6. The biodiesel blends are generally consists of increased viscosity levels and demands higher injection pressure. This condition is slightly suppressed by adding an ignition improver to the biodiesel blends. The oxygen availability and cetane number in the DEE can lead to effective combustion process, which in turn can lower the HC emissions [17]. When comparing B30DEE1 to B30DEE1+ETH, no significant distinction was found. The increased evaporation rates and oxygen availability in these blends leads to complete combustion in both conventional and RCCI modes. This in turn further reduced this emission when compared to diesel.

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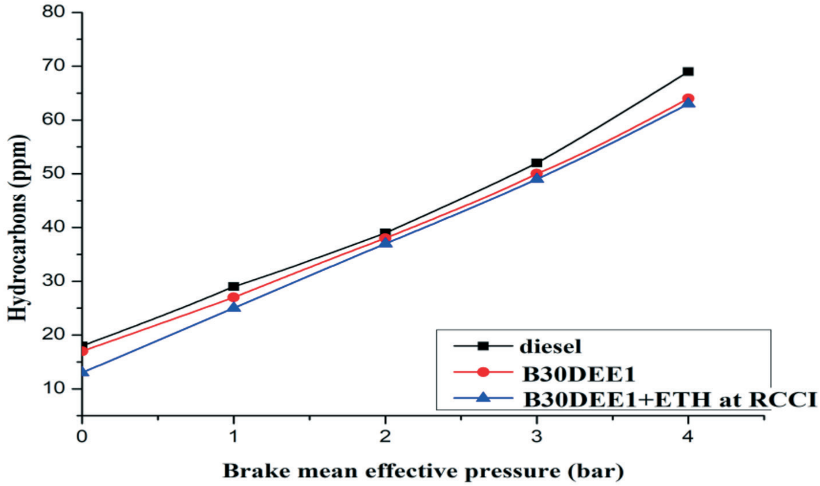

Fig. 6. Hydrocarbon emission versus BMEP

The HC emissions at maximum BMEP for diesel, B30DEE1 and B30DEE1+ETH are observed as 70 ppm, 65 ppm and 63 ppm respectively.

3.5. Nitrogen oxides emission analysis

This concentration of this emission will be influenced by the higher operating temperatures and resident times at those temperatures. The diatomic nitrogen will be encouraged to split into monatomic nitrogen by its elevated operating temperatures. Because of their strong oxygen reactivity, these mono-atoms will release NOx emissions. The greater cetane index and lower calorific value of the test blends may limit the formation of this emission. The reaction time, temperature, and oxygen concentration are the three variables that affect the formation of NOx. Changes in any one of these factors will impact NOx emissions. Because there is enough oxygen in the cylinder, the generation of NOx at low load is primarily dependent on the isolated high temperature zones rather than the oxygen content of the biodiesel.

The DEE addition affected the high latent heat of vaporization for both test blends [20]. This can reduce the combustion space temperature and decrease the formation of NOx for both test fuel blends and made it compatible to diesel. The availability of oxygen in both biodiesel and ethanol also can reduce the chemical delay period. This in turn avoids the accumulation of more quantity of the fuel to some extent during the delay period. The concentration of Nox emission in relation to BMEP is indicated in Figure 7.

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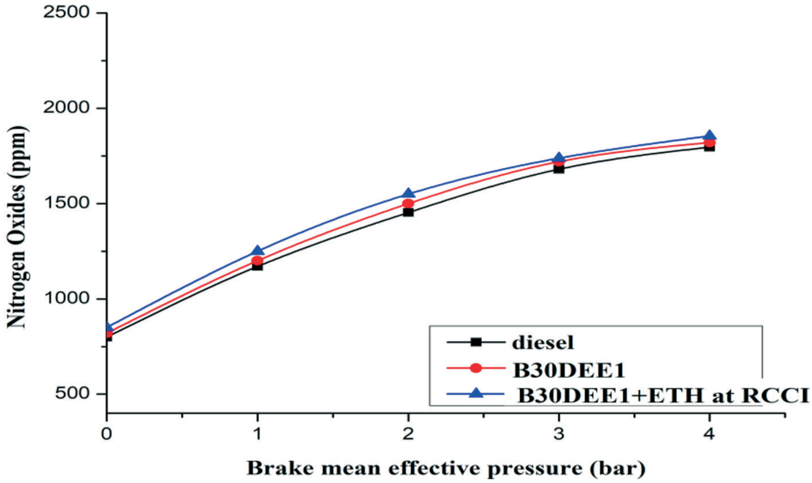

Fig. 7. Nitrogen oxide emission versus BMEP

There was no much variation between both blends for this emission concentration when compared to diesel. The concentration of NOx in the emission at peak load conditions or at maximum BMEP for diesel, N25DE1 and B25N15DE1 are observed as 1854 ppm, 1862 ppm and 1865 ppm respectively.

3.6. Smoke opacity analysis

Inadequate fuel-air mixing results in incomplete combustion and smoke production. Smoke is an exhaust release of unburned carbon. Figure 8 displays the test blends' smoke emissions in comparison to diesel. The method of mixing air with test blends was further enhanced by the addition of ethanol vapors and ignition improvers. The efficiency of combustion and the H/C ratio both affect this emission. For both test fuel blends, lower smoke emissions are encouraged by the increased H/C ratio for test blends in comparison to diesel [21]. Another way to reduce this emission is to encourage combustion that is reactivity-controlled before TDC. Ethanol vapors can also improve the phenomenon of reactivity-controlled combustion. The maximum carbon content to burn was partly a result of the higher oxygen percentages for both blends. For diesel, B30DEE1, and B30DEE1+ETH, the observed smoke emissions are 55.6%, 53.5%, and 52.2%, respectively.

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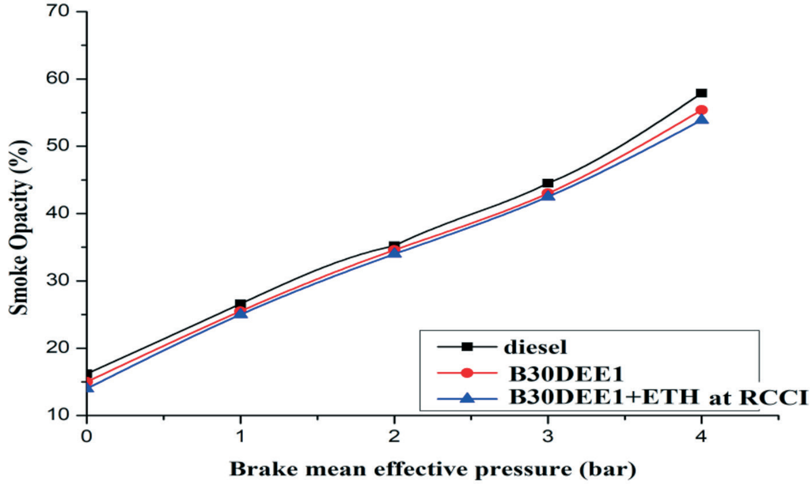

Fig. 8. Smoke Opacity versus BMEP

3.7. Analysis of cylinder pressure

The combustion quality and engine performance influences the in-cylinder pressure [22]. The RCCI combustion process, oxygen availability, and reaction rates affected the elevated cylinder pressures with rapid combustion. The variation of in-cylinder pressure with a maximum BMEP in regard to crank angle is demonstrated in Figure 9. The maximum cylinder pressures for test blends are just higher than diesel. The maximum cylinder pressure is observed for B30DEE1+ETH blend followed by B30DEE1 and diesel. The maximum in-cylinder pressures for both blends are high just after TDC, indicating the combustion did not deviate from its regular pattern like diesel. Uncontrolled combustion for B30DEE1, the amount of oxygen in the test blends, and the calorific value all are influenced by the peak pressures that were obtained. The heating value of the biodiesel compatible with diesel and the oxygen availability in both DEE and biodiesel were also taken into account as influencing factors for cylinder pressure for B30DEE1 blend. The combustible mixture increased these pressures after the introduction of ethanol vapors. When combustion initiated as a result of a maximum amount of fuel-burning, it resulted in maximum cylinder pressure.

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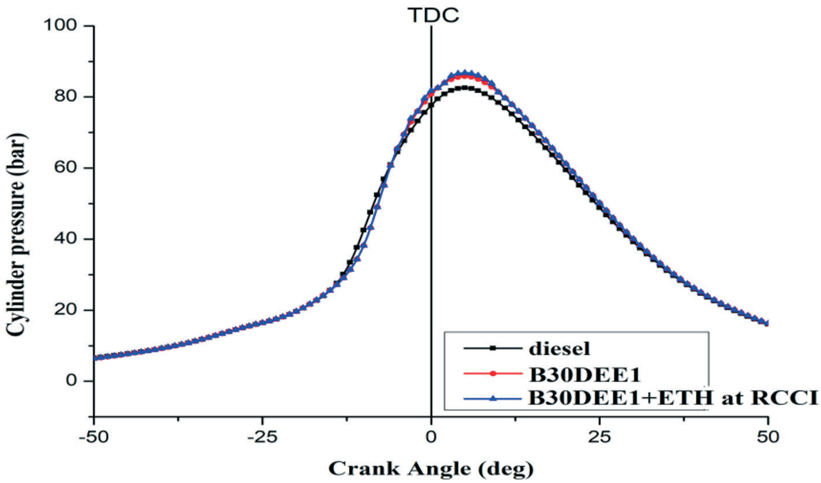

Fig. 9. Cylinder pressure versus crank angle

The increased fuel utilization at maximum BMEP increased the operating temperatures and evaporation rates. They could also be considered another influencing parameter for the increased cylinder pressures for both test fuel blends. The peak pressure obtained for diesel was 82 bar and for B30DEE1, B30DEE1+ETH, it was 84 bar and 84.9 bar respectively.

3.8. Net heat release rate analysis

The combustion methodology study is possible with net heat release rate analysis. The properties of the fuel and operating parameters can influence the net heat release rate. This parameter is further influenced by the heat energy released by the rapid combustion because of more quantity of fuel accumulated during the pre-mixed combustion duration.

Variation of NHRR in relation to the crank angle at maximum BMEP is shown in Figure 10. The increased NHRR is observed for both test fuel blends when compared to diesel. The accumulation of a large quantity of fuel during the ignition lag period is also considered as an influencing parameter for net heat release rate. The oxygen percentage of the both blends can influence more NHRR than diesel [23].

For the B30DEE1+ETH combustible mixture, the addition of ethanol vapors also increased the evaporation rate to some extent, and the NHRR was further increased. The premixed combustion can also utilize the maximum quantity of vaporized fuel at the end of the ignition lag period. The premixed combustion intensity depends upon the time accessible for the production of a combustible air-fuel mixture. This can promote more NHRR when compared to diesel.

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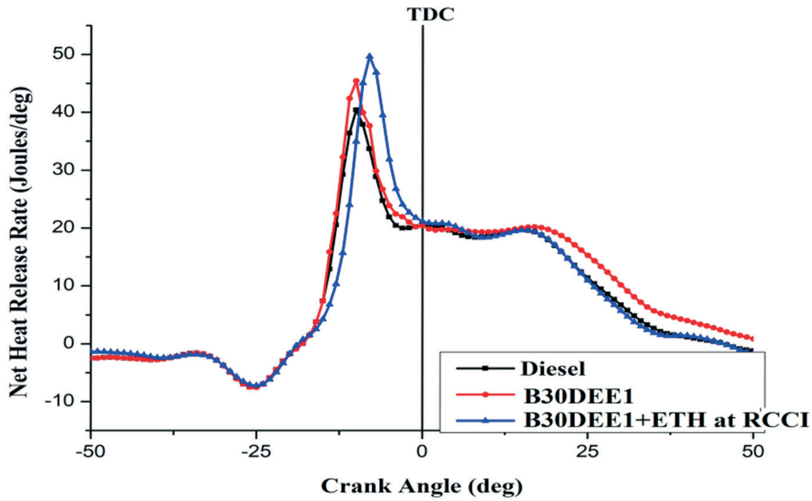

Fig. 10. Net heat release rate versus crank angle

The maximum NHRR was found for both blends were 48.2 J/°CA and 52.4 J/°CA for B30DEE1 and B30DEE1+ETH, respectively. The maximum NHRR for diesel found at 120 BTDC was 42 J/°CA.

3.9. Mass fraction of the fuel burned

This parameter can be used to analyse the time interval between the flame's initiation and uncontrolled combustion [24]. No considerable combustion delay was observed for both test fuel blends when compared to diesel. As the ethanol vapours were introduced, operating temperatures were also elevated. When compared to diesel and B30DEE1, this therefore resulted in higher volatility and flame speed for the B30DEE1+ETH combination. The MFB for both blends when compared to diesel is shown in Figure 11. At the same crank angle, the mass fraction of fuel burned is same for both blends. From the figure, the MFB observed at TDC for diesel, B30DEE1, and B30DEE1+ETH is 58%, 58.18%, and 61%, respectively.

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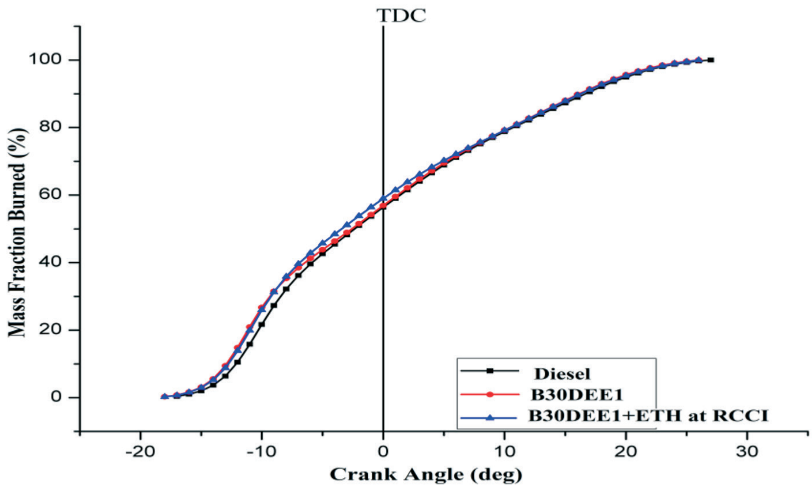

Fig. 11. Mass fraction of the fuel burned versus crank angle

3.10. Cumulative heat release analysis

The combustion quality and the heat energy obtained by the combustion of the fuel can be obtained by this analysis. Figure 12 shows an illustration of the cumulative heat release along crank angle. From the initiation of the combustion onwards, the test blends followed the same pattern as a standard curve for diesel, but the blend B30DEE1+ETH and B30DEE1 shown increased CHR, because of rapid rate of combustion. This was due to the oxygen availability in these blends [25]. The ignition delay period can also be considered another parameter for determining the elevated CHR for the test blends when compared to diesel. The maximum CHR is observed as 0.8 kJ, 0.93 kJ, and 0.94 kJ for diesel, B30DEE1, and B30DEE1+ETH respectively, from the Figure 12.

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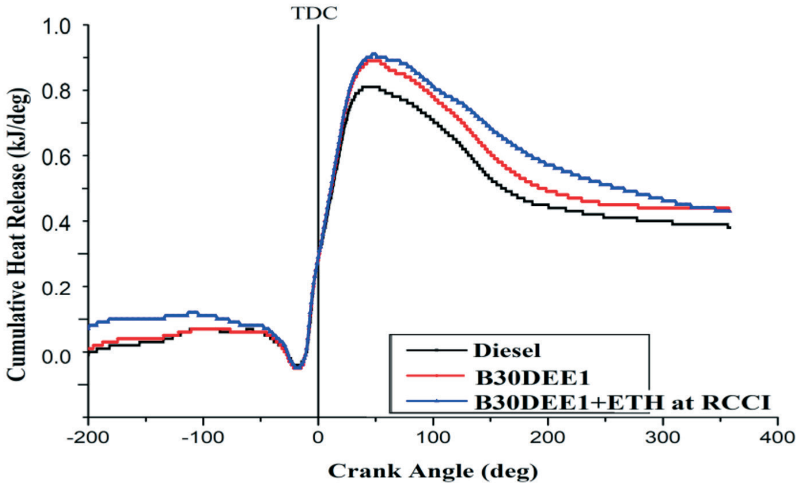

Fig. 12. Cumulative heat release versus crank angle

4. Conclusions

The experimental work for both conventional and RCCI modes of operations has concluded the following statements when compared to diesel at maximum BMEP conditions:

- Since the peak cylinder pressures were reached shortly after TDC, neither of the test fuel blends showed signs of inappropriate combustion. There was no abnormal pressure rise path observed in comparison to diesel with a crank angle. No noticeable abnormalities were detected for both test fuel blends during the combustion process.
- The introduction of ethanol vapours in the RCCI mode resulted in higher peak pressures when compared to utilising diesel. Its burning process was not affected due to its higher octane number and negligible cetane number. The addition of DEE maintained the optimum cetane levels for the combustible mixture B30DEE1+ETH.
- Because of the enhanced combustion process, the test fuel blends showed improved NHRR and cylinder pressures than diesel.
- An increase in brake thermal efficiency was recorded for the test fuel blend B30DEE1. The addition of DEE to the biodiesel blend has influenced the evaporation rate, cetane index, and oxygen percentages in a positive manner. The brake thermal efficiency was further improved by adopting the RCCI mode of operation for the combustible mixture B30DEE1+ETH due to improved pre-mixed combustion characteristics.
- The concentrations of CO and HC in the emissions are comparable to diesel for both test fuel blends, due to the improved combustion characteristics achieved by adopting RCCI mode. The properties of biodiesel blends as per the ASTM standards also lead to better operating conditions.

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- This research also recommends the increased injection pressures with respect to increased compression ratio for biodiesel blends because of its increased viscosity levels considering diesel.

All of the preceding has demonstrated that ethanol, along with biodiesel and DEE, can be blended with diesel in modest amounts to conserve conventional fuel, reduce pollutants, and optimize the combustion process to yield RCCI for CI engines.

5. Nomenclature

ASTM	American Society for Testing and Materials
BIS	Bureau of Indian Standards
BMEP	brake mean effective pressure
BTDC	before top dead center
B30DEE1	30% biodiesel, 69% diesel, and 1% DEE by volume
B30DEE1+ETH	biodiesel 30%, Diesel 69%, DEE 1% and ethanol 10% on a volume basis
CI	compression ignition
CO	carbon monoxide
CHR	cumulative heat release
DEE	di ethyl ether
HC	hydrocarbons
NHRR	net heat release rate
NOx	nitrogen oxides
PJME	Prosopis juliflora oil methyl ester
ppm	parts per million
RCCI	reactivity controlled compression ignition
PCCI	premixed charge compression ignition
HCCI	homogeneous charge compression ignition
GCI	gasoline compression ignition
PPC	partially premixed combustion

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