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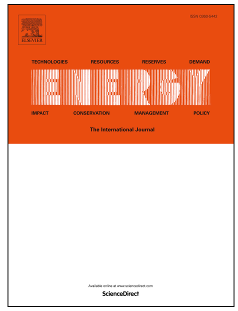

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Nagarajan G.: Writing – review and editing, supervision

Suraj Kumar: Software, methodology, resources

Basrin G: Investigation, Formal Analysis

Effects of waste vegetable oil biodiesel and hexanol on a reactivity controlled compression ignition engine combustion and emissions

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Abstract

In the present work, biofuels produced from agricultural waste have been proposed as a substitute for petroleum-based fuel. Biodiesel produced from waste vegetable oil has been used alongside hexanol in a reactivity controlled compression ignition engine. The waste vegetable oil biodiesel was prepared by trans-esterification and is directly injected inside the cylinder. Hexanol was injected into the port during early suction stroke. A modified 1-cylinder water-cooled diesel engine was used for the tests. The modified engine was tested at medium load and rated load for injection pressures of 400, 500, 600 bar. The proportion of hexanol to waste cooking oil biodiesel was also varied to find the optimal combination. The results were mapped and analyzed with diesel operation at similar conditions. A maximum increase of 1.5% in thermal efficiency was observed compared to Diesel. Oxides of nitrogen and smoke emissions reduced simultaneously for biodiesel-hexanol combinations compared to diesel. Injection pressure of 500 bar and hexanol proportion of 30% at medium load and 60% at rated load were found to be optimum concerning lowest emissions. This study proposes that waste vegetable oil biodiesel and hexanol combination in reactivity controlled compression ignition mode can be an effective replacement for conventional fossil fuel.

Keywords: Hexanol, Waste vegetable oil biodiesel, RCCI, Dual Fuel, waste to energy

Nomenclature

°CA	Degree Crank Angle	ID	Ignition Delay
BDH	Biodiesel Hexanol	KOH	Potassium Hydroxide
BMEP	Brake Mean Effective Pressure	LHoV	Latent Heat of Vaporisation
BTE	Brake Thermal Efficiency	LRF	Low reactivity Fuel
C	Carbon atom	LTC	Low-Temperature Combustion
CD	Combustion Duration	η_{ith}	Indicated Thermal Efficiency
CI	Compression Ignition	NO	Oxides of Nitrogen
CO	Carbon Monoxide	OH	Hydroxide
DI	Direct Injection	PFI	Port Fuel Injection
ECU	Electronic Control Unit	P_{inj}	Injection Pressure
EGR	Exhaust Gas Re-circulation	P_{max}	Maximum Combustion pressure
EoC	End of Combustion	RCCI	Reactivity controlled compression ignition
HC	Hydro Carbon	SoC	Start of Combustion
HRF	High reactivity Fuel	TGs	Triglycerides
HRR	Heat Release Rate	UBHC	Unburnt Hydrocarbons
HRR_{max}	Maximum Heat Release Rate	WVO	Waste Vegetable Oil
ICE	Internal Combustion Engines	WVOB	Waste Vegetable Oil Biodiesel

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1. INTRODUCTION

As the energy demand increases, so does the concern related to global warming. Steered by the growing energy demands in developing nations such as India, the worldwide utilization is forecasted to increase by 56% by 2040 [1]. Amid 2017 and 2018, carbon dioxide emissions surged by 6.3 % in India and by 4.7 % in China. In 2018, the global CO_2 emission was 1.8 % higher than compared to 2017[2]. The foreseen shortage of petroleum fuels, their ascending rates, and the growing concern for the emissions related to their combustion are driving global attention towards the study and production of biofuels [3].

A greener substitute that could lead to sustainable energy production and be a solution to the environmental complications associated to waste dumping would be attractive. A key obstacle in the extensive utilization of biodiesel is its higher price compared to petroleum-based fuels [4]. This cost could be reduced by decreasing the price of feedstock utilized for biodiesel production [5]. Waste vegetable oils (WVO) present themselves as such an alternative. WVO could be converted into biodiesel (WVOB) to be used as a liquid fuel. The conversion of WVO to liquid fuel also rectifies the dumping snags of cooking oil which is otherwise dumped into water sources and therefore pollutes drinking water and clog drainage. Biodiesels derived from WVO have the dual benefits to both the environment and the industries. They have lower emissions, are degradable, and also improve engine lubrication compared to mineral diesel [6]. Another added advantage of using WVO is that it doesn't contest with food consumption or agriculture.

WVO has to undergo pretreatment to make it viable for use in a diesel engine. Different pretreatment processes include preheating, dilution, or micro-emulsion with lower viscosity fuel, trans-esterification, and pyrolysis [7]. Although dilution and micro-emulsion improve the viscosity, they do not help in engine performance improvement [8]. Therefore chemical procedures viz. trans-esterification or pyrolysis are the preferred ones. Biodiesel is well-defined today, as the fatty acid methyl esters obtained through trans-esterification of vegetable oils [9]. Trans-esterification involves reacting triglycerides (TGs) of vegetable oil with lower alcohol viz. methanol in the presence of a catalyst.

Muralidharan and Vasudevan [10] studied the performance, emission, combustion attributes of waste vegetable oil biodiesel (WVOB) and diesel blends and reported increased thermal efficiency (BTE) in the case of biodiesel diesel blend B40 at half load conditions, lower carbon monoxide and unburnt hydrocarbon (UBHC) emissions and marginally increased NOx emissions. Lapuerta et al [11] explored the influence of methyl esters from WVO on particulate emissions from a diesel engine. A sharp decline was witnessed in the smoke and soot emission with the increase in WVO concentration. Fuel consumption was reportedly increased because of the inferior heating value of the WVO methyl esters. Kannan et al [12] studied the influence of ferric chloride additive on performance of WVO fuelled diesel engine and noticed an increase of 6.3% BTE. Yilmaz et al [13], reported a reduction in CO, HC, and NOx emissions for WVOB-pentanol blend at lower loads for the blend ratio B90P10 (90% biodiesel and 10 % pentanol), whereas at higher loads these emissions increased. Atmanli [14] studied the significance of mixing alcohol with WVOB on emissions of a CI engine and reported that all blends of higher alcohols reduced NOx emissions whereas CO emissions increased with the addition of alcohol. All these studies show that biodiesel from WVO is a viable alternative fuel for diesel engine applications.

Another viable and attractive choice for biofuel is higher alcohol (4 or more carbon atoms). Higher alcohols are produced from sugary, starchy, lignocellulosic biomass [15]. This lignocellulosic biomass includes agricultural waste like rice straw, corn stalks, or forest biomass waste like wood pulp, paper mill waste. Thus the production process doesn't solely dependent on food crops. Unlike lower alcohols such as methanol, ethanol which have inferior cetane index, higher resistance to autoignition, lower heating value and poor miscibility with mineral diesel oil which hinder their use in CI engines [16], higher alcohols have higher cetane number, higher heating value, and better blend stability. The rise in the number of carbon in the chain also enhances the ignition superiority of alcohol blends [17]. Higher alcohols exhibit lower hygroscopicity and therefore the ease of storage and lower corrosivity. Corrosivity also reduces by increasing molecular weight [18]. Flashpoint of higher alcohols is higher which makes handling and storage easier.

Kumar and Saravanan [15] also reported that the conjunction of higher alcohols with higher exhaust gas recirculation (EGR) and late combustion can simultaneously reduce NOx and soot in a diesel engine. Sathiyagnanam et al [19] studied the significance of hexanol addition on ethanol-diesel blend and reported that adding hexanol helps improve the stability of ethanol-diesel blend. The blends reduced the soot emissions whereas no effect was seen on NOx emissions. Sundar et al [20] explored the influence of hexanol blending with diesel in a 1-cylinder DI diesel engine. They reported increased BTE. Reduction in smoke and an increase in NOx was also reported. Damodharan et al [21] explored the influence of hexanol doping to waste plastic oil in a diesel engine. They reported a decrease in soot emissions and greater than before NOx emissions on the addition of hexanol. Babu and Anand [22] studied the significance of biodiesel-diesel-hexanol blend on performance, emission attributes of a CI engine and recounted that the lowest specific energy consumption was found for B90-D5-H5 (biodiesel 90 %, diesel 5%, and hexanol 5%). NOx emissions were lowest for B85-D5-H10. From the above literature, it can be realized that Hexanol could be utilized for blending in conventional mode owing to its energy density and cetane number. Hexanol as neat fuel hasn't been investigated previously.

From the above studies, it's perceived that when smoke reduces, NOx emissions increase. Concurrent decrease of NOx and smoke is tough to realize in a diesel engine, owing to the characteristic trade-off between them. Low temperature combustion (LTC) combustion strategy offers a concurrent drop in NOx and smoke emissions [23]. An important characteristic of LTC is a well-mixed air/fuel mix that undergoes combustion at a lower temperature by virtue of extended ignition delay [24]. Tornatore et al [25] utilized n-butanol to achieve LTC in a diesel engine. LTC was achieved with late injection and high EGR percentage. Considerable reduction in NOx and smoke emissions were reported with a marginal drop in BTE. Zhang et al [23] studied the performance, emission characteristics using different butanol-diesel blends in an HD-diesel engine. Smoke, CO, UBHC emissions were recounted to be reduced at higher EGR rates with the increasing proportion of butanol. Yang et al [26] tried to blend butanol and gasoline under LTC conditions. 30% addition of butanol was reported to reduce soot emission whereas NOx emission stayed unchanged.

Reactivity controlled compression ignition (RCCI) refers to dual-fuelled LTC operation, where a fuel having a lower cetane index is inducted into the engine cylinder during early injection stroke in the port and the compressed lower cetane number fuel is then ignited with the injection of a secondary fuel having higher cetane number using direct injection (DI). The combustion takes place because of the reactivity gradient and the various combustion processes depend on local fuel reactivity [27]. Using this technique the fraction of inducted to injected fuel is varied [28]. The partially premixed charge reduces emissions. Tang et al [29] studied the effects of directly injected fuel's properties and the quantity of premixed fuel on a late injection RCCI engine. They observed that the rate of pressure rise can be controlled by controlling the ratio of auto-ignition to flame front propagation by adjusting the proportion of high reactivity fuel. Concurrent reduction of NOx and smoke was achieved by Soloiu et al [30] using a combination of butanol in the port and biodiesel direct injection (DI). Zheng et al [31] compared RCCI and blended dual-fuel combustion in a single-cylinder diesel engine using biodiesel and n-butanol at different EGR rates. RCCI mode of operation presented lower ignition delay compared to blended mode. It was also observed that the high load operation can be extended by the use of RCCI due to a lower rate of pressure rise. Zunqing et al [32] studied the effect of port injected biofuel property on RCCI combustion using n-butanol, ethanol, and 2, 5- dimethylfuran in a single-cylinder diesel engine. They observed that the latent heat of vaporization had a significant effect on the ignition delay and that ethanol/biodiesel combination had greater potential in the simultaneous reduction of soot and NOx. Liu et al [33] investigated the effect of hydrous ethanol on combustion and emission in dual-fuel RCCI engine using port injected hydrous ethanol and directly injected diesel. With the increase in the proportion of ethanol combustion efficiency and indicated thermal efficiency reduced. Similarly with a decrease in the purity of ethanol, combustion efficiency was observed to drop. Reduction in NO and increase in HC and CO emissions was also observed. Most of the studies using alcohol in RCCI use lower alcohol (up to butanol). The use of higher alcohol in RCCI mode has not been implored.

The present study focuses on the application of agriculture and food industry waste to power diesel engines, not only to operate a diesel engine but to reduce emissions associated with diesel mode operation. Most of the works on RCCI combustion uses lower alcohol and no work has been done using Hexanol-Waste vegetable oil and therefore this work would offer some insight into the use of these renewable fuels in the RCCI engine.

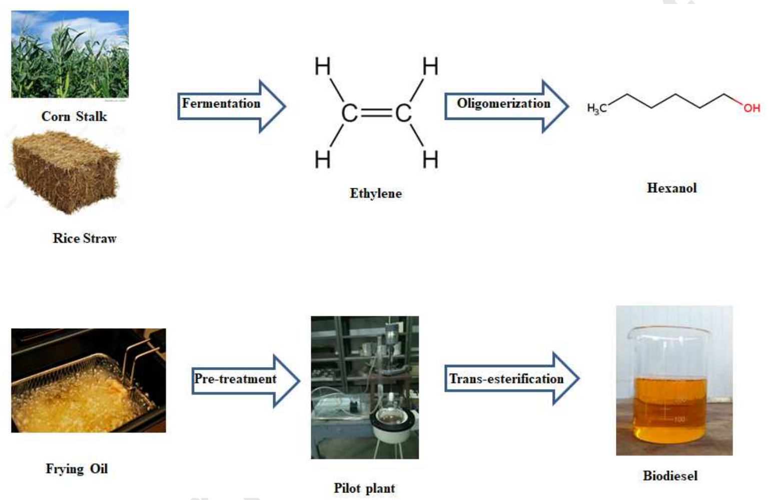


Fig.1: Fuel production, waste to energy

In the present study, waste vegetable oil biodiesel (WVOB) was prepared from the WVO through the trans-esterification process and used in advance RCCI mode with hexanol as low reactivity fuel (LRF). The engine research was carried out in a modified water-cooled, 1-cylinder RCCI engine. Hexanol was selected as LRF for port injection due to higher octane number and volatility, whereas WVOB was used as high reactivity fuel (HRF) for direct injection due to its higher cetane number. The injection pressure (P_{inj}) hexanol was maintained constant at 3 bar, whereas the P_{inj} of WVOB was varied from 400-600 bar. The percentage of LRF had also been varied. The outcome of these variables on engine combustion, emission attributes were investigated in present work. Conducting the same experiment with on-road vehicles is the future scope of the project.

2. MATERIALS AND METHODS

The current study aims to examine the influence of 1-Hexanol and WVOB, on the combustion, emission attributes of a modified water-cooled DI diesel engine. The prepared WVOB was used as HRF owing to its superior cetane index and the 1-Hexanol was used as LRF because of its inferior cetane index. WVOB was directly injected in-cylinder at P_{inj} of 400, 500, 600 bar using common rail and high-pressure fuel pump, whereas 1-Hexanol was injected at P_{inj} of 3 bar at the inlet port of the engine using the port injector. The direct injection fuel pressures were varied and so was the percentage of LRF to HRF. The outcome of these variables on engine combustion, emission was studied at half load, rated load operating conditions. Results were mapped and compared with diesel combustion in direct injection mode.

2.1 Test Fuels

The WVO used for the preparation of WVOB was collected from the hostel canteen of Anna University, Chennai. The collected oil was thoroughly filtered first using sieving cloth and then using filter paper. The oil was then trans-esterified.

The trans-esterification of WVO was done in the biodiesel lab of Anna University. A pilot plant with 10 liters capacity with a reflux condenser, heater and stirrer was used for the same. The oil was preheated to a temperature of 55 °C. Methanol was then added to the oil in a 4:1 molar ratio (methanol/oil), along with 1% (wt/wt) of KOH as a base catalyst [34]. The mixture was heated at 60 °C for about 120 minutes and stirred at a speed of 600 revolutions per minute. After completion of the reaction, the mixture was moved to separating flask and allowed to settle for 12 hours. The glycerol was removed from the flask and was further processed to produce soap. Hot distilled water was sprayed over the oil to eliminate traces of glycerol or soap. The treated oil was heated and maintained at 115 °C for about 10 minutes to eliminate water content.

The 1-Hexanol utilized for this study was procured from Research-Lab Chemical Corporation, Mumbai 02, India. The properties of WVOB and 1-Hexanol are summarized in Table 1 [15, 22].

Table 1: Test Fuel Properties

Properties	Diesel	Waste Vegetable Oil	1-Hexanol
Molecular weight	190-211.7	250-260	102.18
Density (kg/m^3 at 15° C) *	835	876	821.8
Kinematic Viscosity (cSt) at 40 °C *	2.39	4.76	3.32
Cetane Number	>47	57	23
Lower heating Value (MJ/kg) *	≈42.5	39.67	39.1
Latent heat of Vaporization(kJ/kg)	<300	-	603
Flash Point (°C) *	46	160	59
Fire Point (°C) *	54	165	64
Acid Value (mg KOH/g) *	-	0.45	-
Iodine value (gl2/ 100g)175	-	175	-

* = *Measured quantity*

2.2 Experimental Setup

Kirloskar AV1, naturally aspirated, water-cooled DI diesel engine was suitably modified for this investigation. Table 2 shows the engine specifications. The mechanical injector was replaced with a Delphi made solenoid injector. The solenoid injector was connected to a common rail and the high-pressure pump and was controlled by an open electronic engine control unit (EECU) from National Instruments. Provisions were made to mount a port injector in intake plenum. The Denso made port injector was also electronically controlled using the same EECU. An air preheater was provided along the intake air path to heat the incoming air. The engine was equipped with necessary sensors and transducers for controlling the operating parameters along with instruments for measurement as shown in Fig.2. AVL Di gas exhaust analyzer was used to measure engine-out emissions. An AVL 437c was used to measure smoke. Table 3 shows the range and accuracy of the measurement devices and Table 4 shows the uncertainty in measurement.

Table 2: Experimental engine specifications

Make and model	Kirloskar AV1
Bore x Stroke (mm)	80 x 110
Displacement (cc)	553
Compression ratio	16.5:1
Connecting rod length (mm)	235
Rated power kW at rpm	3.7 at 1500 rpm
Bowl Geometry	Hemispherical
Inlet valve opening (°crank angle bTDC)	5
Inlet valve closing (°crank angle bTDC)	145
Exhaust valve opening (°crank angle bTDC)	-145
Exhaust valve closing (°crank angle bTDC)	-5

Table 3: Measurement device range and accuracy

Measuring Device	Range	Accuracy
Speed Indicator	0–5000 rpm	±1 rpm
K type thermocouple	0–1000 °C	±1 °C
AVL Di gas analyzer		
CO_2	0–20% of Vol	±0.5% Vol
CO	0–10% Vol	±0.03% Vol
HC	0–20,000 ppm Vol	±10 ppm Vol
NO	0–5000 ppm Vol	±50 ppm Vol
AVL 437c Smoke meter	0–100% opacity	±0.1% opacity
Weighing balance for fuel measurement	0–10 kg	±0.1 g

Table 4: Uncertainty in Measurement

Measurement Parameter	Uncertainty (± %)
Speed	0.15
Flow Rate:	
Hexanol	1.05
Diesel	1.2
HC	0.66
CO	0.62
NO	0.5
Smoke	1.2
Total	2.25

2.3 Test Procedures

For this study, the base readings were taken by using Diesel in DI mode. The DI P_{inj} was varied between 400-500-600 bar. This was based on the literature available on RCCI [35, 36] as well as the engine design constraints. The engine was tested at rated load and medium load for all injection pressures. The temperature of inlet air was maintained constant at 40 °C at all conditions using air preheater. Combustion, emission data were recorded for each test run. Later tests were repeated for WVOB/Hexanol (BDH) combinations. The proportion of inducted (hexanol) to injected (WVOB) fuel was varied according to the load. For half load conditions, the inducted to injected fuel ratio was changed from 20:80 to 50:50(BDH 20 to BDH 50). For rated load conditions the same ratio was changed from 40:60 to 60:40(BDH 40 to BDH 60). The results were analyzed to find optimal conditions for RCCI operation. Table 5 summarizes the engine operating conditions.

1.Kirloskar AV1 test engine
2. Intake air preheater
3. Intake air surge tank
4. Crank angle encoder
5. Cam position sensor
6. 5hp ac motor
7. Electrical loading device
8. Exhaust gas recirculation
9. High pressure fuel pump
10. Common rail
11. Open ECU Controls
12. Data acquisition controls
13. Electrical loading device control
14. Direct injection assembly
15. AVL smoke meter
16. High reactivity fuel sump
17. Air preheater control and temperature display
18. PFI Injector assembly
19. Low reactivity fuel sump
20. NI DAQ

Fig.2: Modified RCCI engine test rig

Table 5: Engine operating conditions

Parameters	Values	Units
Brake Power	1.85, 3.7	kW
Crank speed	1500	rpm
Inlet air pressure	1	atm
Inlet air temperature	40	° C

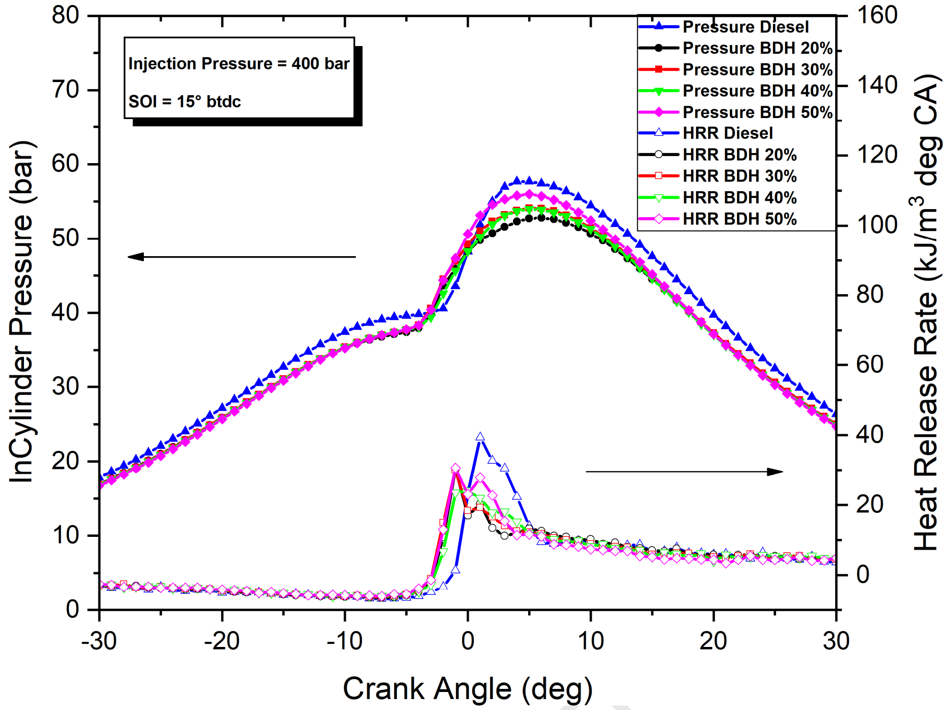
LRF injection		
Injection pressure	3	bar
Injection angle	5	° aTDC
LRF quantity	20-60	%
HRF injection		
Injection pressure	400-600	bar
Injection angle	-15	° aTDC
HRF quantity	80-40	%

3. RESULTS AND DISCUSSION

3.1 Combustion pressure and Heat Release Rate

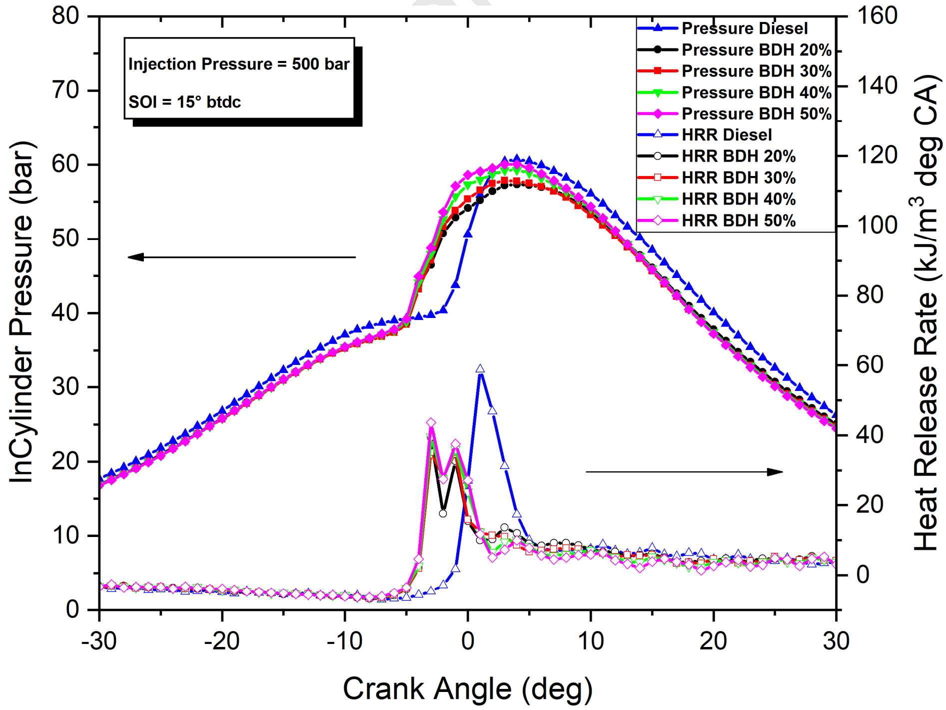
The combustion pressure of an engine gives an insight into how effectively the energy of the fuel is being converted into mechanical work. Fig.3 shows the combustion pressure vs. crank angle curves for the engine combustion of diesel and WVOB-hexanol combinations (BDH). Fig.3 (a), (b), (c) corresponds to the medium load operation whereas (d), (e), (f) corresponds to the rated load operation of the test engine. A noteworthy point is an advance in combustion for WVOB-hexanol; this is owed to the superior cetane index of WVO compared to diesel (Table 1) which leads to reduced ignition delay as is observed from Fig.9. As seen from Fig.3, the peak of pressure curve (P_{max}) increases with increasing P_{inj} of directly injected fuel at both loads. This is accredited to improved atomization of fuel and quicker mixing by virtue of smaller droplets, and as a consequence improved combustion [37]. It can also be observed that P_{max} increases with an increasing percentage of LRF for medium load and rated load, which is owed to the increased homogeneity of the charge available inside the cylinder during the premixed combustion phase. At medium load, as perceived from Fig.3 (a), (b), (c), although the P_{max} increases with increasing fraction of inducted fuel, hexanol at all injection pressures, still it is lower than diesel. This trend could be accredited to the lower global temperature at medium load and the high latent heat of vaporization (LHoV) of hexanol (Table 1) that further lowers the temperature compared to diesel combustion.

Whereas at rated load as perceived from Fig.3 (d), (e), (f) its seen that P_{max} is higher for WVOB-hexanol at P_{inj} of 400, 500 bar compared to diesel. Contrary to the medium load, global temperature is higher at rated load and overcomes the hurdle due to higher LHoV of hexanol. At P_{inj} of 600 bar, the diesel combustion is better than BDH40 and comparable to BDH50. With an increase in the proportion of hexanol the in-cylinder temperature also increases by virtue of improved combustion and therefore the pressure curve and the P_{max} advance [38]. This P_{max} further increases with increasing P_{inj} as discussed earlier. Fig.4 shows this variation in P_{max} with increased P_{inj} at medium load and rated load.



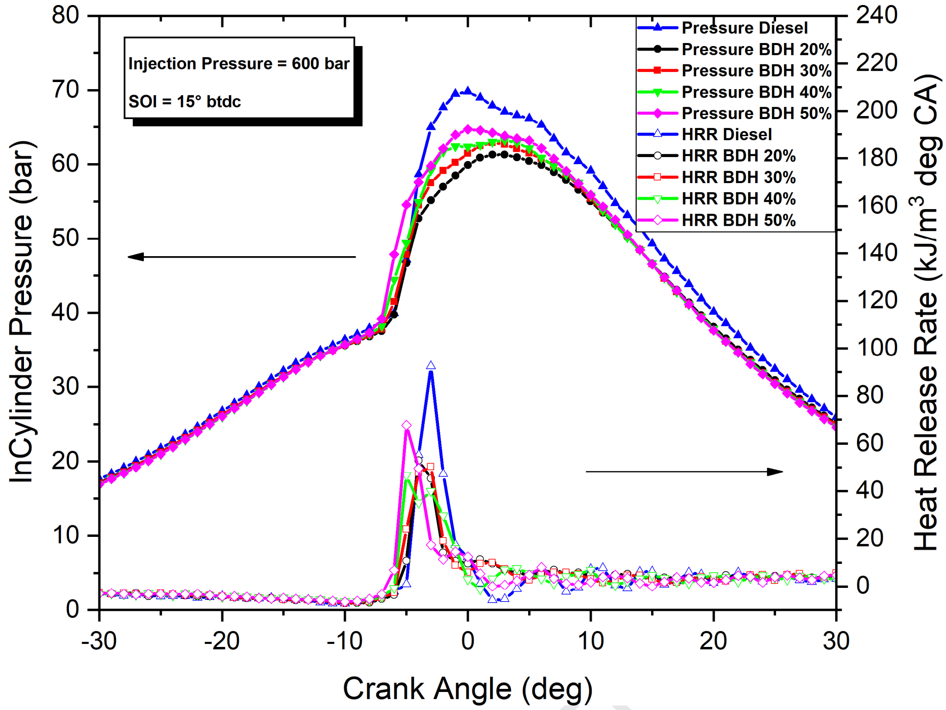
Injection Pressure = 400 bar
SOI = 15° btdc
Pressure Diesel
Pressure BDH 20%
Pressure BDH 30%
Pressure BDH 40%
Pressure BDH 50%
HRR Diesel
HRR BDH 20%
HRR BDH 30%
HRR BDH 40%
HRR BDH 50%
InCylinder Pressure (bar)
Heat Release Rate (kJ/m3 deg CA)
Crank Angle (deg)

(a)

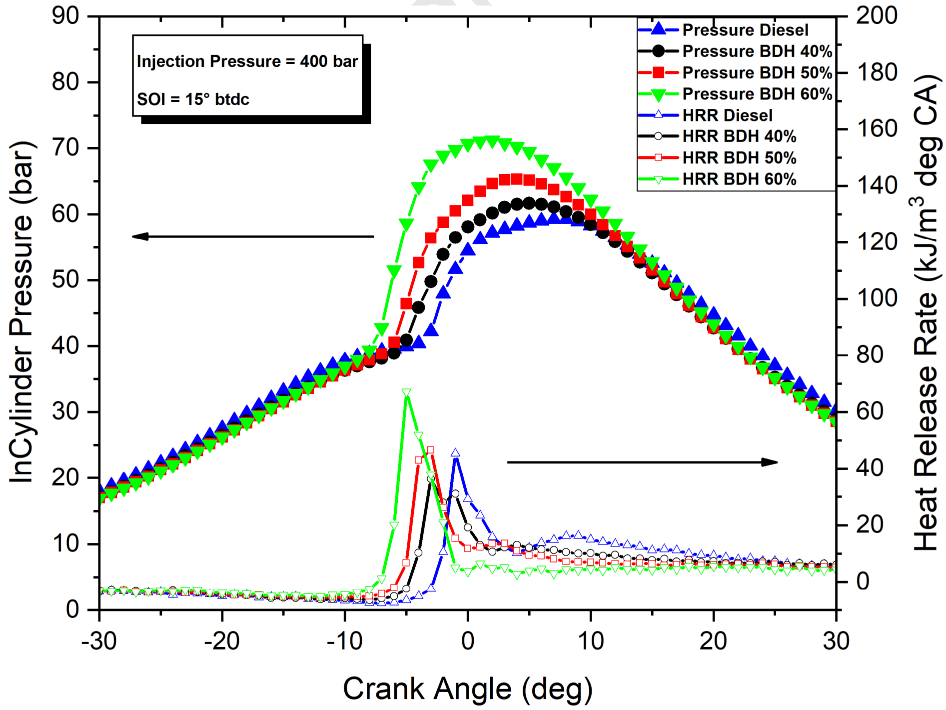


Injection Pressure = 500 bar
SOI = 15° btdc
Pressure Diesel
Pressure BDH 20%
Pressure BDH 30%
Pressure BDH 40%
Pressure BDH 50%
HRR Diesel
HRR BDH 20%
HRR BDH 30%
HRR BDH 40%
HRR BDH 50%
InCylinder Pressure (bar)
Heat Release Rate (kJ/m3 deg CA)
Crank Angle (deg)

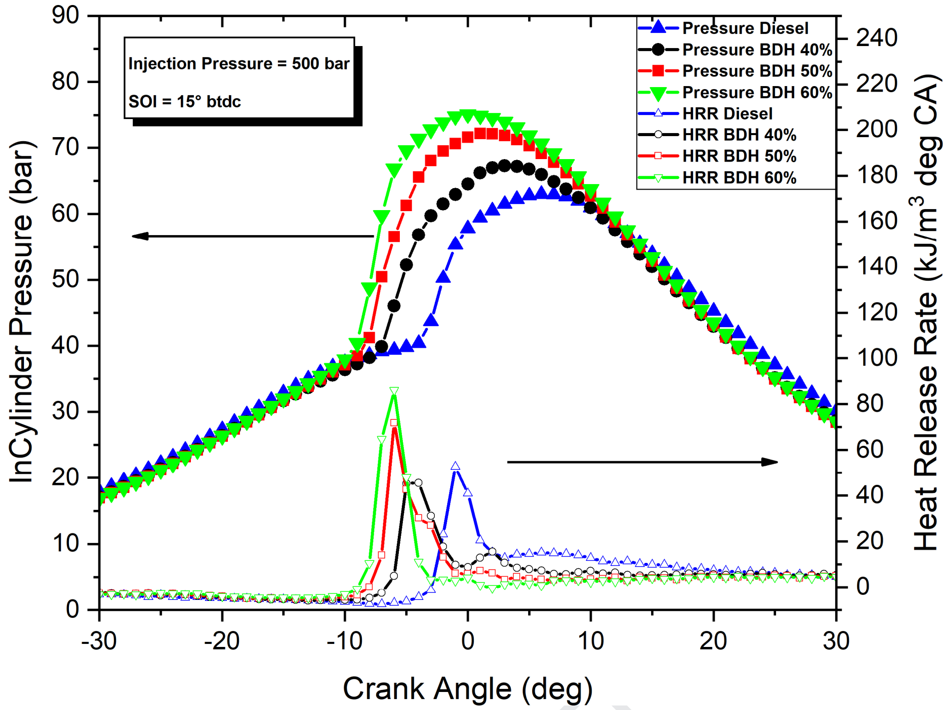
(b)


Injection Pressure = 600 bar
SOI = 15° btdc
Pressure Diesel
Pressure BDH 20%
Pressure BDH 30%
Pressure BDH 40%
Pressure BDH 50%
HRR Diesel
HRR BDH 20%
HRR BDH 30%
HRR BDH 40%
HRR BDH 50%
InCylinder Pressure (bar)
Heat Release Rate (kJ/m3 deg CA)
Crank Angle (deg)

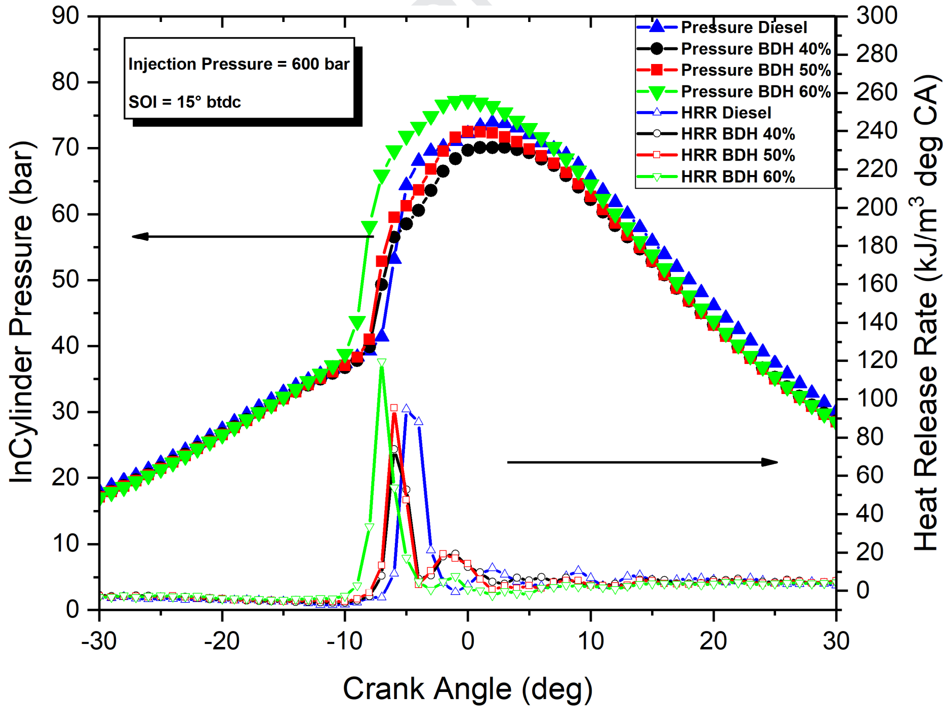
(c)


Injection Pressure = 400 bar
SOI = 15° btdc
Pressure Diesel
Pressure BDH 40%
Pressure BDH 50%
Pressure BDH 60%
HRR Diesel
HRR BDH 40%
HRR BDH 50%
HRR BDH 60%
InCylinder Pressure (bar)
Heat Release Rate (kJ/m3 deg CA)
Crank Angle (deg)

(d)



(e)



(f)

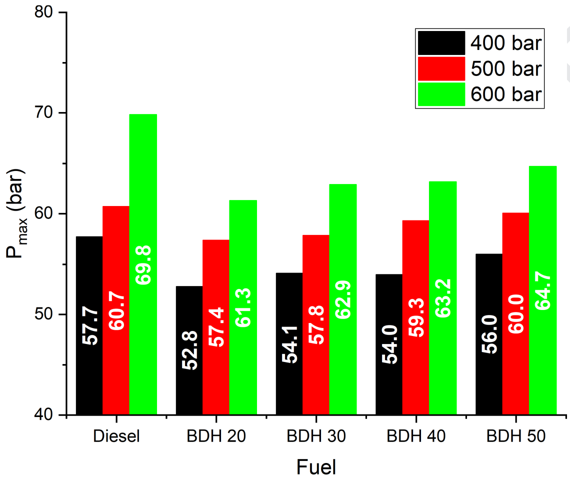
Fig.3: Pressure and Heat Release Rate (HRR) vs. crank angle for diesel and WVOB-hexanol fuel blends at different injection pressures varying with the percentage of LRF. (a)WVOB hexanol at medium load and 400 bar (b) medium load and 500 bar (c) medium load and 600 bar (d) WVOB hexanol at rated load and 400 bar (e) rated load and 500 bar (f) rated load and 600 bar.

HRR shows the speed at which the chemical energy of the fuel is being converted to mechanical work [39]. The HRR is plotted against the crank angle as shown in Fig.3. HRR depends on the energy density and physical properties viz. cetane number, flammability of the fuels considered [40]. At all P_{inj} and both the load conditions, it is comprehended that the HRR curve for WVOB-hexanol combination (BDH) is steeper in contrast to diesel fuel by virtue of its improved flammability [41]. At both the load conditions it is comprehended that the HRR curve for BDH is advanced in contrast to diesel fuel owing to the superior cetane index of WVOB. At 600 bar pressure, the HRR of diesel is closer to that of BDH which would indicate reduced physical delay by virtue of better atomization because of the increased P_{inj}.

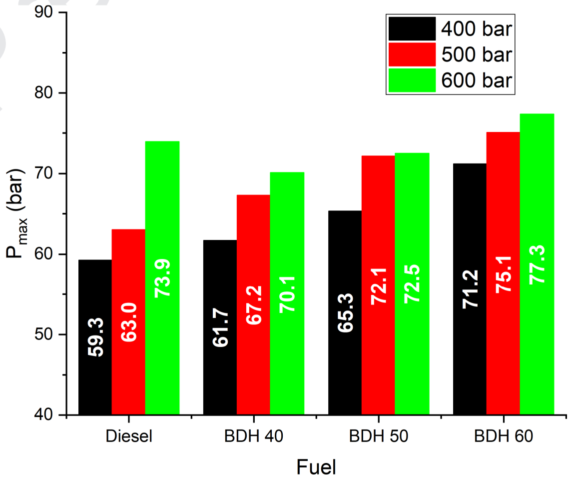
At medium load conditions the twin peaks in the HRR is clearly visible for BDH combinations as perceived from Fig.3 (a), (b), and (c), which is the characteristics of reactivity controlled CI [42]. Because of the low in-cylinder temperature at medium load conditions and the effect of higher LHoV of hexanol, the peak (HRR_{max}) for BDH combinations is lower in contrast to diesel for each P_{inj} as perceived from Fig.3 (a), (b) and (c).

Whilst, at rated load, global temperature is higher which overcame the LHoV issue of hexanol fuel. Better combustion at rated load condition owing to increased oxygen percentage in the BDH fuel combinations lead to increased peak as perceived from Fig.3 (d) and (e) (400 bar and 500 bar). At 600 bar pressure, the heat release for Diesel fuel was equivalent to that of BDH 50 and greater than BDH 40 by virtue of decreased physical delay because of better atomization at higher P_{inj} [43].

Fig.5 depicts a deviation of HRR_{max} with P_{inj} for different fuel permutations at medium load and rated load.

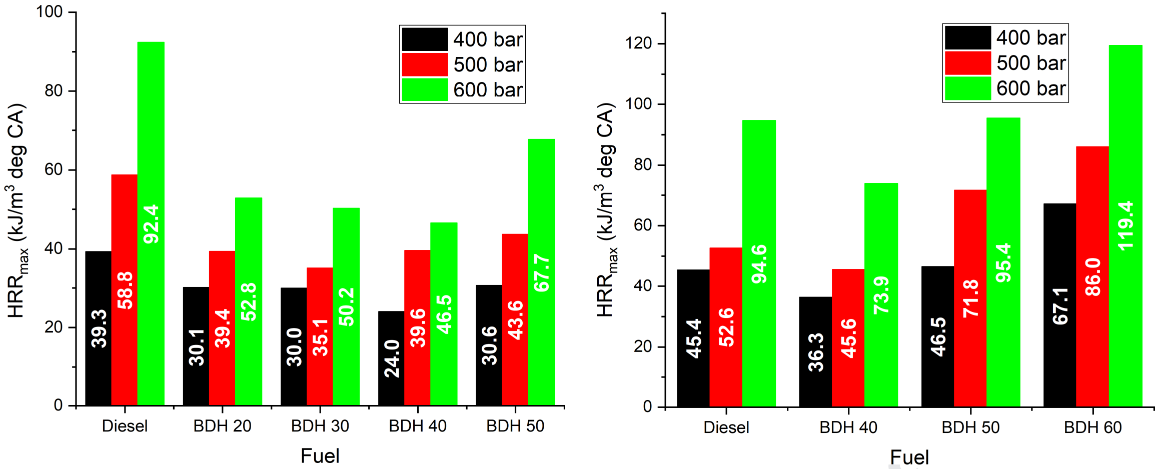


(a)



(b)

Fig.4: Variation of P_{max} with increasing P_{inj} for different fuel permutations at (a) medium load (b) rated load.

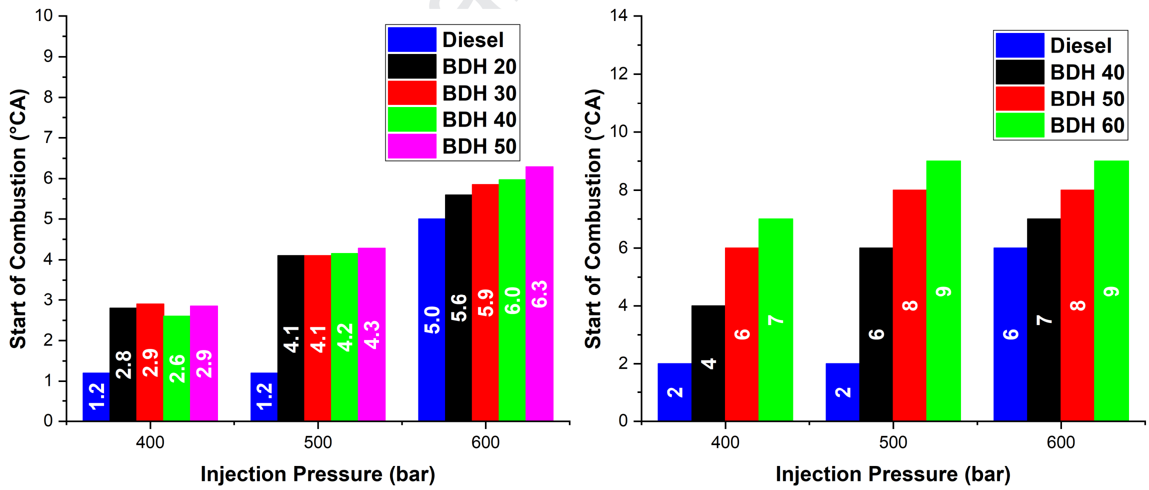


(a) (b)

Fig.5: Variation of HRR$_{max}$ with increasing P$_{inj}$ for different fuel permutations at (a) medium load (b) rated load.

3.2 Start of Combustion

The start of Combustion (SoC) is an important characteristic for studying combustion in an internal combustion engine (ICE). In CI engine, fuel is injected directly in-cylinder and then after a small delay, known as ignition delay (ID) the fuel ignites and rapid increase in temperature and pressure takes place by virtue of combustion. The point at which the HRR crosses over from negative to positive is considered as the SoC [40]. The SoC can be easily found out from the HRR vs. crank angle curve. Fig.6 shows the SoC for different fuel permutations at different P$_{inj}$ for both medium load and rated load conditions.



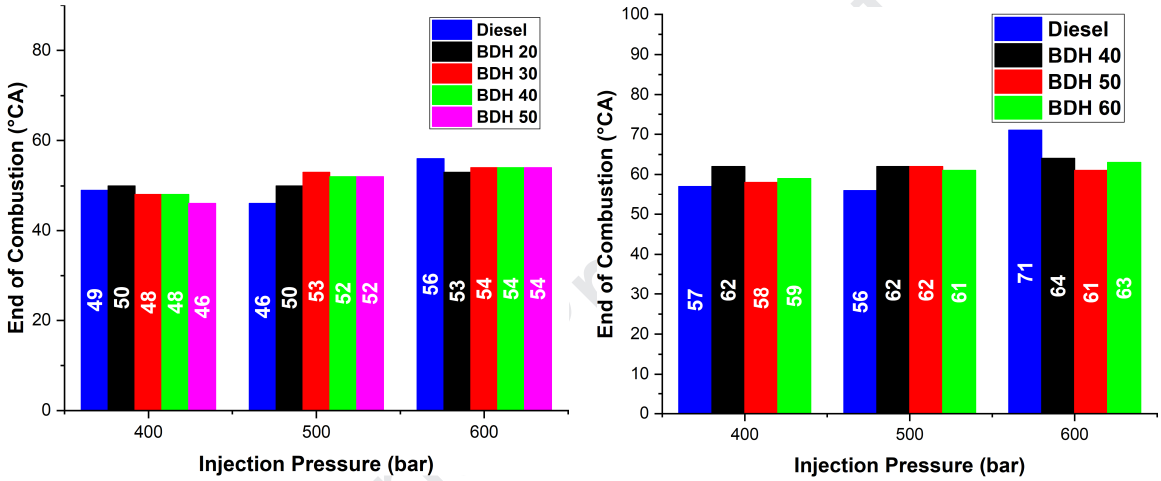
(a) (b)

Fig.6: SoC at (a) medium load and (b) rated load operating conditions at varying injection pressures for different fuel combinations.

It is seen that increasing the P$_{inj}$, will advance the SoC irrespective of fuel combinations or loads. This is owed to better atomization at higher P$_{inj}$, which reduces ignition delay, therefore the early onset of combustion [37]. It is also noticed that the SoC for BDH fuels is advanced compared to diesel because of the superior cetane index of WVOB and it further advances with the increasing proportion of hexanol for rated load and medium load. This is attributed to increased in-cylinder global temperature by virtue of better combustion because of the oxygen available in the fuel.

3.3 End of Combustion

The point at which ninety percent of the fuel had undergone combustion is considered as the end of the combustion (EoC) in this study. EoC was computed based on mass fraction burnt [40]. The EoC was plotted against different injection pressures for every combination at medium load and rated load as shown in Fig.7. By virtue of extra fuel needed at rated load operating conditions, the EoC is extended as seen in Fig.7(b). A small surge in EoC is witnessed with increasing P_{inj} at both loads by virtue of better atomization and mixing and therefore better combustion (more of the fuel to be utilized). There isn't much change in EoC for different fuel combinations. An exception is observed at 600 bar pressure where the EoC of diesel is extended over the BDH fuel combinations. This is owed to lower viscosity of diesel which results in better combustion at higher P_{inj} [44].

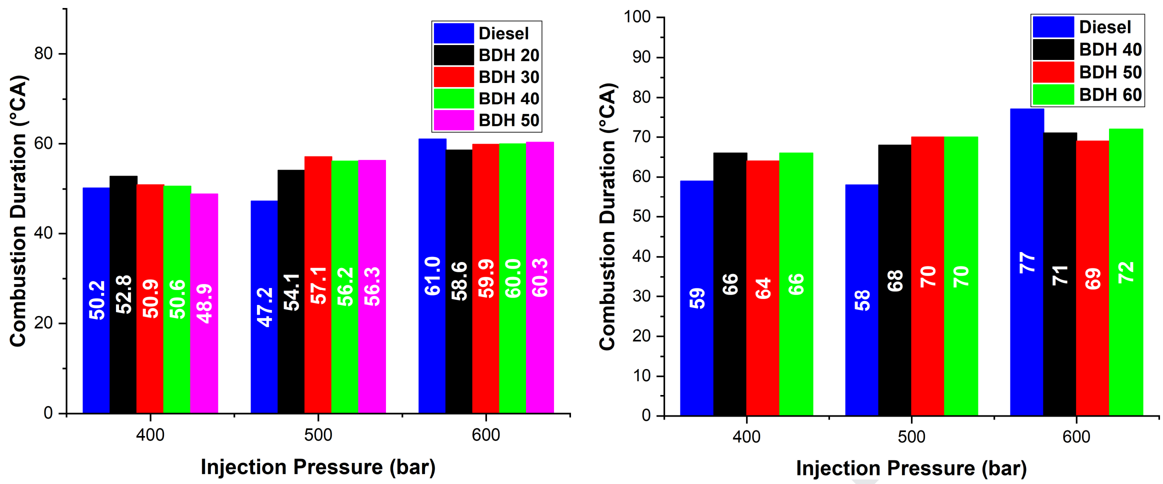


(a) (b)

Fig.7: EoC at (a) medium load and (b) rated load operating conditions at varying injection pressures for different fuel combinations.

3.4 Combustion Duration

Combustion duration (CD) is characterized as the period amid SoC and EoC in crank angle degrees [40]. CD for the different fuel combinations at different P_{inj} is plotted in Fig.8 for medium load and rated load operating conditions. Owing to advance in SoC and extended EoC at higher injection pressures, the CD is also longer. CD is comparable for different BDH combinations and is longer than diesel fuel at both medium load and rated load with an exemption for P_{inj} of 600 bar as seen in Fig.8. The reasons for which are described in the previous section.

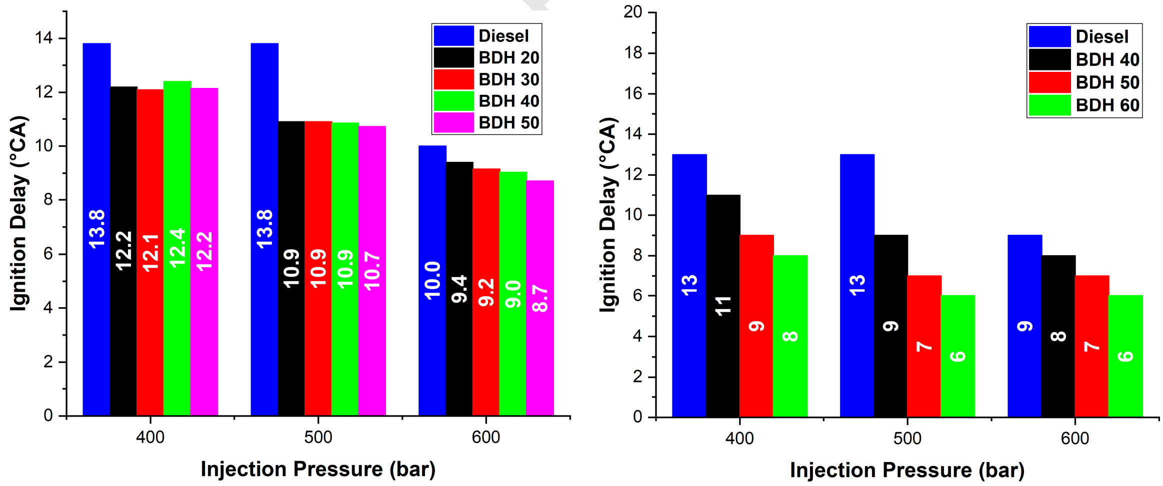


(a) (b)

Fig.8: CD at (a) medium load and (b) rated load operating conditions at varying injection pressures for different fuel combinations.

3.5 Ignition Delay

Ignition delay (ID) is the most important component when it comes to combustion study of fuels in an ICE. ID is termed as the interval amid the start of injection and SoC. ID relies on both, the properties of fuel such as viscosity, cetane number, and the in-cylinder conditions such as the temperature [45]. ID for different fuel permutations at the different P_{inj} at medium load and rated load is shown in Fig.9.



(a) (b)

Fig.9: ID at (a) medium load and (b) rated load operating conditions at varying injection pressures for different fuel combinations.

At each P_{inj} and both the loads, it is comprehended that ID for BDH combinations is less than diesel. This is owed to the superior cetane index of injected fuel (WVOB). With an increase in P_{inj} of directly injected fuel, there is a decrease in ID as observed in Fig.9 (a) and (b). This is primarily because of the reduced physical delay by virtue of better atomization and mixing at higher P_{inj} [37], along with increased in-cylinder temperature by virtue of better combustion. At medium load condition as seen in Fig.9 (a) the ID for BDH combinations is comparable at lower injection pressures (400 and 500 bar) whereas at higher pressure (600 bar) the ID reduces with increase in the proportion of hexanol in BDH combination by virtue of better combustion and increased in-cylinder temperature. At rated load operating conditions, the

ID shows a similar trend as in medium load with the reduction in ID with an increasing proportion of hexanol. The reduction in ID is distinguishable at rated load because of the increased in-cylinder temperature which aids the evaporation of atomized fuel.

3.6 Emissions

Exhaust emissions are paramount when it comes to the investigation of alternative fuels or advanced combustion techniques in an ICE. The exhaust emissions from an engine depend on several parameters which include the physicochemical properties of fuel such as viscosity, density, cetane number or the saturation, the unsaturation of hydrocarbons present in the fuel, etc. and the conditions within the engine cylinder viz. compression ratio, in-cylinder temperature, engine speed, load, fuel P_{inj}, injection angle and so on. In this study, CO, HC, NO, smoke emissions were taken for the BDH combinations in RCCI mode and are contrasted to diesel, in conventional direct injection mode for different P_{inj} at medium load and rated load.

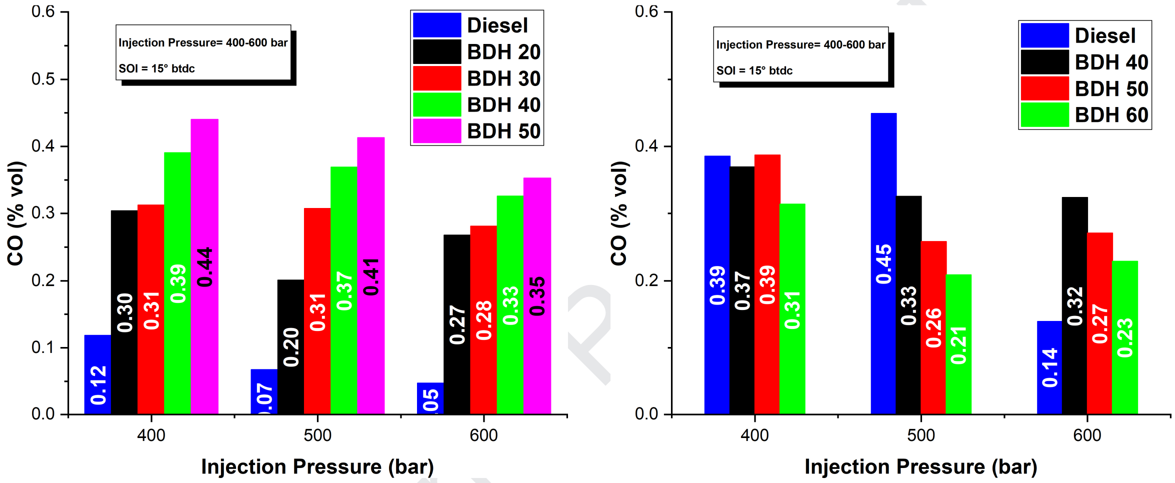
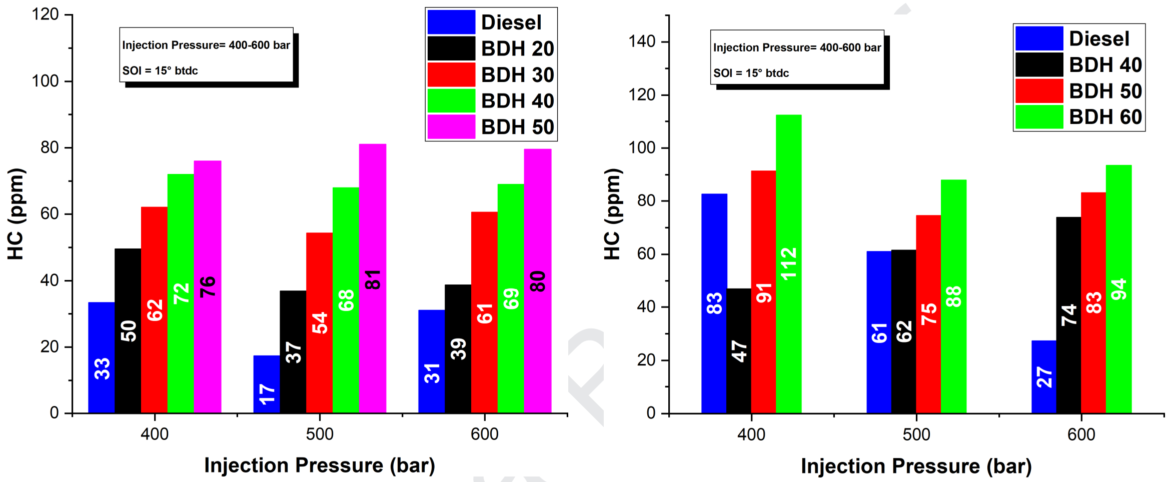


Fig.10: CO emissions at (a) medium load and (b) rated load operating conditions corresponding to varying injection pressures for different fuel combinations.

CO emissions for the BDH combinations operating in RCCI mode are plotted against conventional mode diesel fuel in Fig.10 for both medium load and rated load operating conditions. At medium load, it is comprehended that CO emissions increased for BDH combinations compared to diesel. The reason behind the increased CO emissions is the LTC that takes place which reduces the in-cylinder temperature compared to pure diesel operation added to the lower temperature at medium load operating conditions [46]. Therefore the CO is unable to oxidize into CO_2. It can be further perceived that CO emissions increased with the proportion of hexanol from 20% to 50%. This is owed to the higher LHoV of hexanol which induces a cooling effect that is dominant at lower load operation [38]. With an increase in P_{inj} a decline in CO emissions is observed owing to better atomization and mixing leading to better combustion at higher pressure which ultimately increases the in-cylinder temperature (Fig.5). At rated load, it's found that CO emission is reduced with the increasing proportion of hexanol in BDH combination. This is because at rated load the engine temperature is higher and availability of oxygen with the increasing proportion of hexanol leads to enhanced combustion and therefore lower CO. With increasing P_{inj} from 400 to 500 bar, there is a decline in CO emission but for 600 bar it almost remains the same. A surge in CO emission is seen for diesel at 500 bar.

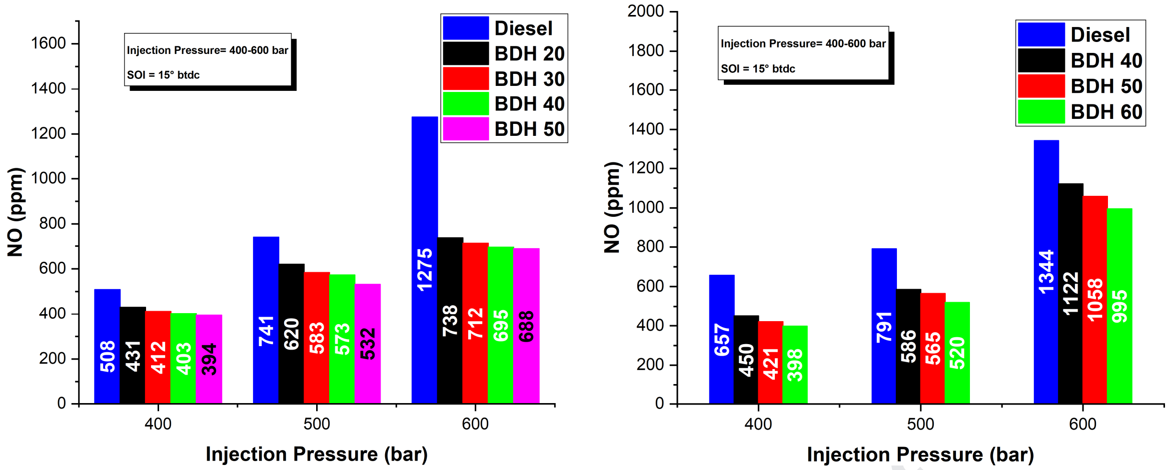
Hydrocarbon (HC) emissions for the different fuel samples at different P_{inj} and engine loads is plotted in Fig.11 in comparison to diesel. It is comprehended that HC emissions increase for BDH combinations as seen in Fig.11 (a) and (b). This is owed to LTC which reduces the in-cylinder temperature compared to diesel operation [46]. Therefore the HC is unable to oxidize into hydrogen and CO_2. As discussed earlier the cooling effect of hexanol would be dominant at lower loads, therefore the HC emissions increase with the increasing proportion of hexanol in BDH as seen in Fig.11 (a). With an increase in P_{inj}, a drop in HC is seen at 500 bar P_{inj} (Fig.11 (a)) which could mean better combustion compared to P_{inj} of 400 and 600 bar. At rated load operation the HC emissions reduced by increasing P_{inj} for diesel fuel which is a result of better atomization and therefore better combustion [37]. The inferior cetane index of hexanol compared to diesel or WVOB depreciates the auto-ignition attribute and aids the quenching effect in lean regions which tends to increase the HC emissions [47]. Also by virtue of the cooling effect of hexanol, HC emissions increase compared to diesel [48]. At the same time, oxygen present in the fuel favors better oxidation and therefore lower HC [49]. For BDH combinations HC emissions increase with increasing hexanol proportion by virtue of the combination of all these factors neutralizing each other.



(a) (b)

Fig.11: HC emissions at (a) medium load and (b) rated load operating conditions corresponding to varying injection pressures for different fuel combinations.

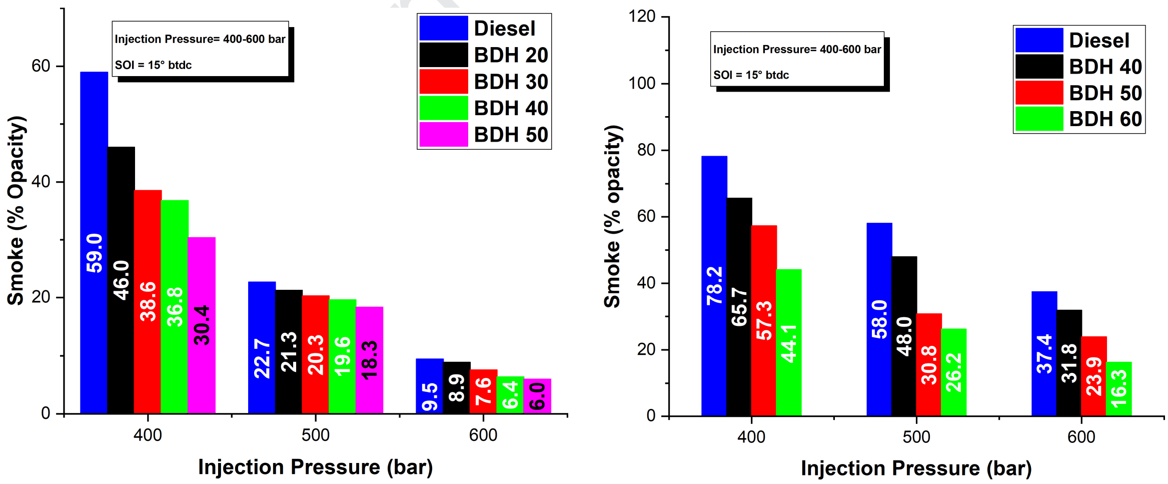
As seen in Fig.12 the NO emissions for BDH combinations are less than diesel at each operating loads and injection pressures. This is owed to the LTC. Although fuel bound oxygen is available, the lower temperature inhibits the production of NO. With increasing P_{inj} there is a slight increase in NO emissions by virtue of improved combustion because of better atomization. At both the operating loads, NO emissions decrease with the increasing proportion of hexanol in BDH combination by virtue of lower calorific value and higher LHoV of hexanol [48] which reduces the in-cylinder temperature. Lower pre-mixed combustion duration as perceive from Fig.3 is another reason for lower NO emissions [40].



(a) (b)

Fig.12: Oxides of nitrogen (NO) emissions at (a) medium load and (b) rated load operating conditions corresponding to varying injection pressures for different fuel combinations.

Smoke emissions for different fuel permutations at varying P_{inj} for both the operating loads are plotted in Fig.13. It is comprehended that the smoke emissions decrease for both medium load and rated load for diesel and BDH combinations with increasing P_{inj}. This could be accredited to improved combustion with increasing P_{inj} by virtue of better atomization, therefore less fuel-rich regions. The earlier inducted hexanol forms a homogenous mixture by the time WVOB is injected and therefore there are very less fuel-rich regions. Also, the increasing proportion of oxygen with increasing hexanol proportion in BDH combinations helps the unsaturated hydrocarbons to get oxidized as an alternative to participating in soot progression reactions and as a result, the smoke opacity reduces [38].

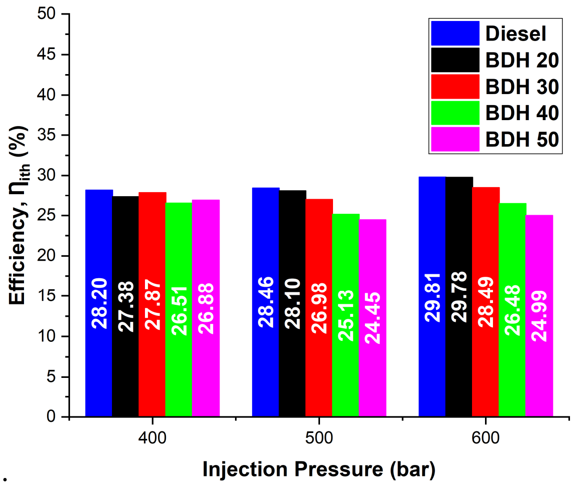


(a) (b)

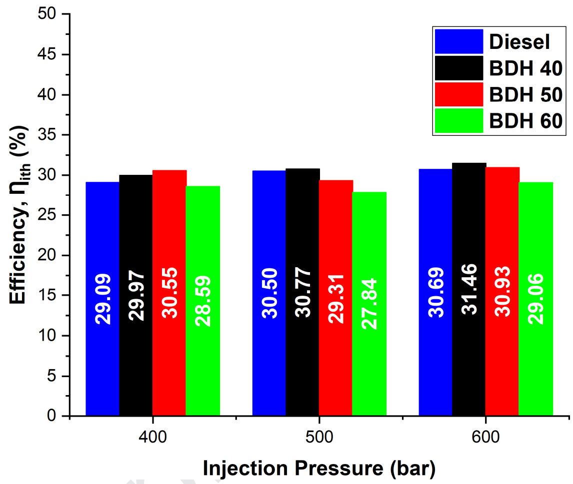
Fig.13: Smoke emissions at (a) medium load and (b) rated load operating conditions corresponding to varying injection pressures for different fuel combinations.

3.7 Efficiency

The indicated thermal efficiency (η_{ith}) is plotted against varying P_{inj} and operating load conditions for different fuel permutations as seen in Fig.14.



(a)



(b)

Fig.14: Indicated thermal efficiency at varying P_{inj} for different fuel permutations at (a) medium load and (b) rated load conditions.

At rated load, the efficiency is only marginally higher than at medium load. At medium load conditions, the η_{ith} for diesel fuel is higher than BDH combinations. This is owed to the oxygenated fuels and the cooling effect of hexanol which is dominant at lower load [38]. With the increasing proportion of hexanol in BDH combinations, the η_{ith} further reduces as seen in Fig.14 (a) due to the reasons stated earlier. There is a slight rise in η_{ith} with an increase in P_{inj} for diesel fuel owing to better atomization and therefore better combustion. For BDH combinations this is opposed by the cooling effect. At rated load operating condition, it is comprehended that η_{ith} of the BDH combinations is higher than diesel for BDH 40 and BDH 50 combinations. This is because, at rated load conditions, global temperature is higher which aids in better combustion in the presence of excess oxygen. With further addition in the proportion of hexanol, the η_{ith} reduces owing to the cooling effect of hexanol which counteracts the increase in the efficiency due to better oxidation. It is observed from Fig.14 (b) that at 400 bar pressure, BDH 50 gives the best η_{ith}. At P_{inj} of 500, 600 bar BDH 40 gives the best η_{ith} due to the reasons discussed earlier.

4. ENVIRONMENTAL AND ECONOMIC ANALYSIS

The use of biodiesel, as well as alcohol derived from plants/trees, has a lower impact on the environment in terms of emission [50]. This is mostly because such biofuel is considered to be carbon neutral since they use up more carbon dioxide during photosynthesis than they produce during combustion in an engine [51]. Because of lower NO emission compared to petroleum-based fuel [50] as also can be seen from the results and discussion section, their contribution to acid rain is also considerably less.

The costs associated with the production of fuel are also considerably reduced by the use of waste resources. For example, 1 liter of sunflower oil costs ₹ 150 whereas waste cooking oil collected costs only about ₹ 10. Including the cost of chemicals used for trans-esterification (KOH, Methanol), the cost of final biodiesel would be around ₹ 12-15 (methanol can be extracted from the glycerol and reused) compared to ₹ 70 for Diesel. 1 liter of Hexanol costs ₹ 350, but it can be brought down if produced in bulk (~₹ 100 or lower). Considering the positive impact it has on the environment concerning emission, renewability, it is a price to be paid. Moreover, the promotion of these biofuels would enable the agricultural economy and help the farmer as they can run the farm equipment and transports using this fuel generated from their waste. It would also promote small scale industries and business owners in the rural area as they can set up their production plant and sell the fuel conveniently.

5. CONCLUSION

Waste vegetable oil biodiesel and hexanol fuels were operated in RCCI mode in a modified 1-cylinder diesel engine. The objective was to use a biofuel derived from agricultural waste to run a diesel engine efficiently and to simultaneously reduce the NO-smoke emissions associated with diesel engine operation. Being renewable, both biodiesel and hexanol ease the load on consumers due to depleting reserves of fossil fuels. Also, the effective utilization of waste to create energy solves the existing waste disposal problems. The engine was tested on 2-fuels, WVOB, and hexanol at injection pressures varying from 400 to 600 bar, the varying proportion of port injected to directly injected fuel at medium load and rated load. The combustion, performance, emission data were collected for RCCI operation of BDH combinations and analyzed with diesel fuel. The following conclusions are drawn from the comparison;

(a) The P_{max} and HRR_{max} increased with increasing concentration of hexanol for BDH combinations. At medium load, these peaks are lower than diesel whereas at rated load the peak is higher.
(b) ID for BDH combinations is less than diesel in all the cases. A maximum reduction in ID by 23% for 500 bar at medium load and 56% for 500 bar at rated load was observed. As a result, the SoC advances for BDH combinations. With almost similar EoC, this results in longer CD for BDH combinations in contrast to diesel fuel.
(c) NO and smoke emissions reduces significantly for BDH combinations in contrast to diesel fuel at both medium and rated load operation. On the contrary, HC and CO emissions increased, as observed for LTC typically. Considering exhaust emissions the P_{inj} of 500 bar and hexanol proportion of 30% (BDH 30) at medium load and 60% (BDH 60) at rated load are recommended.
(d) There is a noticeable increase in indicated thermal efficiency at rated load (maximum 1.5% for BDH 50 at 400 bar pressure) when using BDH combinations. 600 bar P_{inj} and hexanol proportion of 40% (BDH 40) gives the best efficiency (~31.5%).

From this study, it can be concluded that WVOB and hexanol combination is a substitute for conventional diesel fuel in a modified diesel engine. The proportion of induced hexanol beyond 60% was restricted because of the port injector flow rate. Redesigning the combustion chamber and injector position could reduce the HC and CO emissions. The NO and smoke emissions from the BDH combinations can be further reduced with the use of EGR, and multiple injections which are the future scope of the work.

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Highlights

- Hexanol-Waste cooking oil biodiesel in an advanced combustion engine.
- Simultaneous reduction of smoke and oxides of nitrogen.
- Increased thermal efficiency compared to Diesel.

Declaration of interests

☒ The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

☐The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: