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Water-containing *i*-propanol-*n*-butanol-ethanol (IBE) as a next-generation biofuel of *n*-butanol for diesel engine

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The high emission level of diesel engines has been an issue of global concern and the sophisticated means of controlling the emissions were not cost-effective. In this work, effects of water addition in a bio-derived fuel to mitigate engine emissions and enhances the brake thermal efficiency have been investigated. Four test samples including IBE10, IBE30, IBE29.5W0.5 and IBE29W1 have been prepared and tested in a diesel engine. The engine combustion characteristics, performance and emissions have been observed. It has been established that the water containing blends improve the BTE, BSFC and further reduces emissions at varying loads. In comparison with IBE30, IBE29W1 (29 vol. % IBE, 1 vol. % water and 90 vol. % diesel) has shown decreasing peak in-cylinder pressure and increases ignition delay and combustion duration by 0.13% – 4.8 %, 0.5% – 12.4 % and 0.26% – 3.8 % respectively. As for the engine performance, BTE has been increased by 2.6 % – 14.1% and BSFC decreased by 0.1% – 15 %, respectively, and the emissions of UHC, smoke, CO and NOx emissions was decreased by 21% – 42.6%, 0% – 21.7%, 5.4% –11%, and 0.64% – 9%, at varying loading conditions respectively.

Keywords: Diesel engine, Emissions, Water-containing *i*-propanol-*n*-butanol-ethanol/diesel

1 Introduction

Research on renewable fuels for internal combustion engines has become essential as a result of increasing concerns about the prospects of the availability of fossil fuels and various environmental issues[1,2]. Increasing human activities have also instilled fear over some uncertainties related to the fuel price hike, toxic emissions and energy security[2]. There is a steady rise in need for use of alternative fuels from non-edible oil. Biodiesel blends like Calophylluminophyllum[3], lemon peel oil[2], and pomegranate oil methyl ester[4] are being utilized for testing various engines. In C.I. engines applications, the significant improvement in physio-chemical properties of biodiesel blends has led to a promising trend to substitute alternative for conventional diesel fuel[4]. Selective catalytic reduction technique and advanced combustion technology such as homogenous charge compression ignition mode is seen as a viable means of reducing NOx emissions[5]. Also, mixing of oxygen concentrated substances to biodiesel has limited engine exhaust emissions like unburnt hydrocarbons (UHC), CO, particulate matter (PM) and smoke emissions to a larger extent [5].

Bio-alcohols such as ethanol, *n*-butanol and some fermentation intermediates like acetone-*n*-butanol-ethanol (ABE)[6,7], *i*-propanol-*n*-butanol-ethanol (IBE)[6,8], are being applied to both spark ignition (S.I.) as well as compression ignition (C.I.) engines. The IBE is an intermediate fermentation product of *n*-butanol with merit of saving recovery cost and energy. The IBE blends with gasoline[6,8] or diesel[9,10] are promising next-generation alternative fuels.

The addition of IBE is observed to improve the engine energy conversion efficiency and decrease the soot emissions. Li *et al*.[6] studied the performance, combustion and emissions of ABE and IBE at various equivalence ratios. Use of ABE10 shows lower brake power and emissions reduction. Li *et al*.[8] revealed the performance of IBE-gasoline blends in SI engines. The IBE30 provided better brake thermal efficiency (BTE) and pollutant reduction. Li *et al*.[9] reported

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effects of injection strategy on the emission characteristics of a common rail diesel engine fueled with IBE15 and IBE30. The findings revealed that pilot injection could reduce knocking and ringing intensity at higher blending ratios. Li *et al*.[10] studied the impact of dilute gas on the combustion characteristics of IBE15 and IBE30. The results showed that the BTE of pure diesel was higher than that of IBE15. All the blends investigated showed that the NOx discharges decreased, while CO and UHC emissions increased with an increase in proportion of diluent. Lee *et al*.[11] reported the thermal and emission behavior of IBE-gasoline and IBE-diesel blends used in gasoline and diesel engine respectively.

The impact of water addition to bio-derived fuels has been demonstrated by several researchers as a technique capable of improving combustion efficiency and reducing engine emissions. Adding a small proportion of water of less than 5% was reported to increase the engine output and decrease the emission levels[12,13]. The consequences of increasing fuel consumption and decreasing energy conversion efficiency has been attributed to the addition of a large proportion of water[12-14]. Vellaiyan *et al*.[15] attempted to emulsify diesel with water. Another notable effort was that of Senthilkumar and Jaikumar[16], who emulsified waste cooking oil with water. Similarly, Basha and Anand[17] considered the emulsification of Jatropha methyl ester oil blends with water. Their findings portrayed a decrease in NOx emissions, benefiting from the latent heat of absorption of water particles during the combustion process. Radhakrishnan *et al*.[18] investigated the effect of water addition to various blends of orange peel oil biodiesel. The findings reported a trade-off between the brake thermal efficiency and brake specific fuel consumption. A study of water addition to IBE/gasoline in SI engine was investigated by Li *et al*.[19]. It was observed that introducing a small proportion of water in lower IBE blends enhances energy efficiency and decreases various toxic emissions. Water-containing lower blends of IBE/gasoline were used for various engines[15-19]. However, there is less focus on the use of water-containing IBE/diesel blend. In order to fill this research gap, this study is aimed at investigating the combustion, performance, and emission characteristics of water-containing IBE/diesel blends with an emphasis on solving the tradeoff between the brake thermal efficiency and various emissions.

2 Materials and Methods

2.1 Preparation of the test fuel

Commercial form of diesel was procured and used as a baseline fuel. Conventionally in the fermentation stage, the mixing ratios of *i*-propanol, *n*-butanol and ethanol exist as 3:6:1 respectively. Analytical grade of *i*-propanol (99.8%), butanol (99.5%) and ethanol (99.8%) were sourced to prepare the IBE mixture. The three fuels were mixed by maintaining a constant volume ratio of 3:6:1 (I:B:E) using a temperature-controlled magnetic stirrer. The physicochemical properties of the IBE mixture were computed using the mixing rule as explained by Nithyanandan *et al*.[7]. Thereafter, the IBE mixture was blended with a mixture of mineral diesel and water. A typical diesel engine can take up to 40% blend of butanol fuel without any need for engine modification[10]. However, lower cetane number of IBE might pose a great challenge for ignitability of the IBE/diesel blend. More recent studies has suggested a blending ratio of not more than 30% (30 vol.% IBE, and 70 vol.% diesel)[6,8,9,10,19]. The samples of pure diesel (D100) blended with IBE or small proportion of water were stored in tubes at room temperature for two weeks to evaluate the fuel stability. These samples included IBE10, IBE30, IBE29.5W0.5, and IBE29W1 (29 vol.% IBE, 1 vol.% water and 70 vol.% diesel). No phase separation was observed after stability test. Various physicochemical properties of the test fuels obtained from literature[5,8,10,20,21,22] are summarized in Table 1.

2.2 Experimental set-up

The experiment was carried out on a single-cylinder, four-stroke, Kirloskar TV1 diesel engine operating at a speed of 1500 RPM. The specifications of test engine and associated equipment are denoted in Table 2 and Table 3 respectively. Figure 1 depicts the illustrative of the experimental set-up used in this research. The experimental set-up comprises of the test engine, an eddy current dynamometer, fuel supply system, data acquisition system, emission analyzer, smoke opacimeter, etc. Various emissions like UHC, CO and NOx were measured using AVL emission analyzer. An opacimeter was used to measure the smoke opacity. An eddy current dynamometer was utilized to vary the engine load starting from 20% load and increasing in 20% steps. Kistler 6055B piezoelectric transducer amplified by Kistler 5015 charge amplifier was used to measure the in-cylinder

Table 1 — Fuel properties

Parameter	Fuel				
	Diesel	Ethanol	*i*-Propanol	*n*-Butanol	IBE
Chemical formula	C_{10} - C_{22}	C_2H_5OH	C_3H_7OH	C_4H_9OH	-
Octane number	-	100	112	87	95.8
Cetane number	52.65	8	12	15.92	13.952
Viscosity at 40 ºC (cSt)	3.00	1.13	1.74	2.27	1.997
Oxygen content (wt. %)	-	34.8	26.6	21.6	24.4
Density (kg/m^3)	820 - 860	795	786	813	803.1
Lower heating value (MJ/kg)	42.7	26.8	30.4	33.1	31.7
Boiling temperature (ºC)	282 - 338	78	84	118	-
Latent heat (kJ/kg)	260	904	758	582	667
Stoichiometric air-fuel ratio	14.3	9.0	10.4	11.2	10.7
Auto-ignition temperature(ºC)	250	420	399	343	-

Table 2 — Engine specifications

Item	Specifications
Make	Kirloskar TV1
Type	1-cylinder, direct injection
Rated power	3.5 kW
Engine speed	1500 rpm
Stroke × Bore	110×87.5 mm
Displaced volume	661 cm^3
Compression ratio	17.1:1
Rated power	3.5 kW
Number of nozzles	3
Number of valves	4
Diameter of nozzle holes	0.3 mm
Injection pressure	210 bar
Injection timing	23° BTDC

Table 3 — Range, accuracy and resolution of the measuring apparatus

Equipments	Measured variables	Range	Accuracy	Resolution
GE TCL-15, 4-35-1700	Torque	0 – 300 Nm	±0.5%	0.1 Nm
AVL DI Gas 444N	HC NOx CO	0 – 20000 ppm 0 – 6000 ppm 0 – 15.0 vol.%	±12 ppm Vol. ±3% ±0.06%	± 1 ppm Vol. ± 1 ppm 0.001 vol.%
AVL-437C model opacimeter	Smoke	0 – 100 %	±1%	± 0.1%
Piezoelectric transducer-Kistler 6055B	In-cylinder pressure	0 – 250 bar	±0.4 % FS	± 1%

pressure. The engine was started and made to run for 10 minutes to warm up and attain stable conditions. Each measured parameter was measured four times at an interval of 10 minutes to ensure accurate results and reduces experimental uncertainties. The variations observed for the test conditions were expressed as standard deviation and depicted as an error bar.

2.3 Engine combustion and performance parameters

In combustion phasing, the ignition delay, the start of combustion (SOC), end of combustion (EOC) are important parameters to be determined. In this context, the ignition delay was considered to be the interval of the crank angle between the start of injection and the SOC. The combustion duration was the time interval of the crank angle between the SOC and the EOC[10]. The heat release rate (HRR) was computed from Eq. (1).

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} V \frac{dV}{d\theta} + \frac{1}{\gamma - 1} P \frac{dV}{d\theta} \qquad \dots (1)$$

where, γ represents the ratio of specific heats capacities, P represents the in-cylinder pressure, V represents the volume at a crank angle θ. Integrating Eq. (1) yielded the cumulative HRR from the air-fuel mixture. The crank angle locations where 10% and 90% of the total heat released was established was defined as SOC and EOC respectively from the cumulative HRR curves. Similarly, the combustion center was defined as crank angle location where 50% of the total heat was released (CA50).

Based on the measured variables, the following parameters were calculated[8]. The brake power (W) was given by[23]:

$$P = T(2\pi N/60) \qquad \dots (2)$$

where, T is torque in (N-m) and N is engine speed in (RPM)

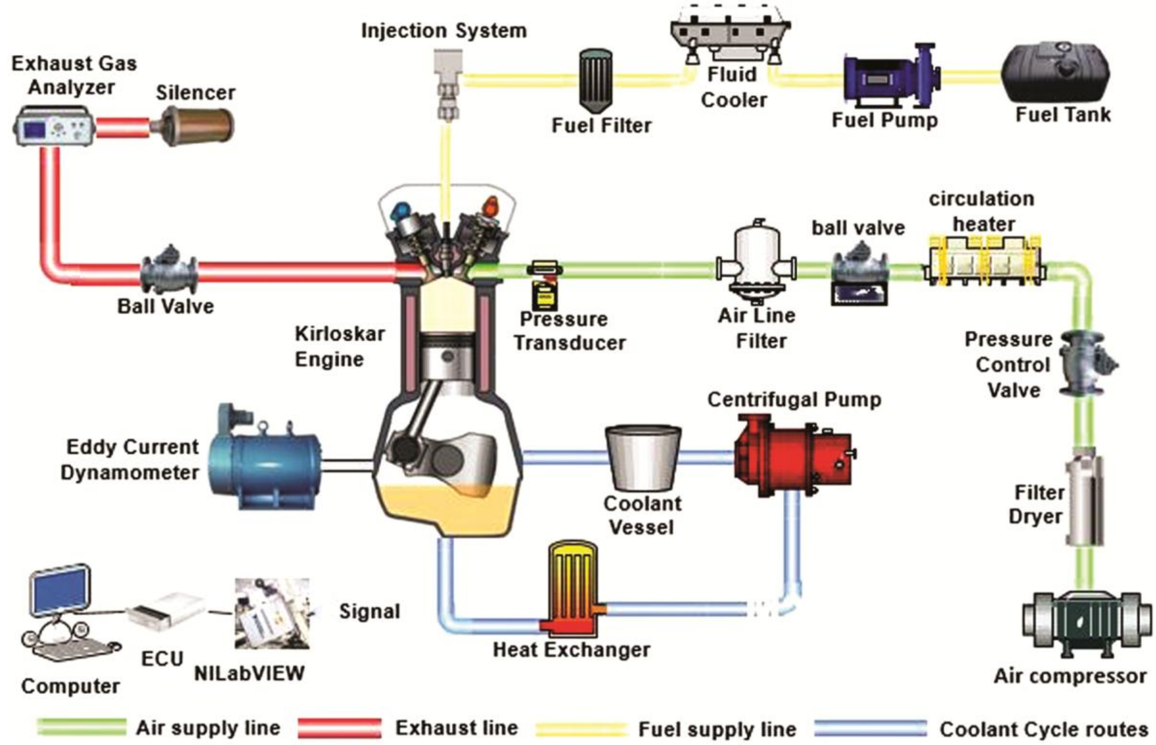

Fig. 1 — Schematic of the experimental set-up.

Diesel and IBE blends have different lower heating values. Thus, the fuel consumption cannot be calculated in the conventional way. The effective fuel consumption (m_{eff}) was given by[10,23] :

$$m_{\text{eff}} = \frac{m_{\text{fc}} \cdot \left(V_{\text{D}} \cdot \rho_{\text{D}} \cdot LHV_{\text{D}} + V_{\text{IBE}} \cdot \rho_{\text{IBE}} \cdot LHV_{\text{IBE}}\right)}{\left(V_{\text{D}} \cdot \rho_{\text{D}} + V_{\text{IBE}} \cdot \rho_{\text{IBE}}\right) / LHV_{\text{D}}} \quad \ldots (3)$$

where, m_{fc} stands for the measured fuel consumption in (g/h). LHV_{D}, ρ_{D} and V_{D} stands for lower heating value, density and volume of diesel.

LHV_{E}, ρ_{IBE}, and V_{IBE} stands for lower heating value, density, volume of IBE respectively.

The brake specific fuel consumption (kg/kWh) was calculated as:

$$BSFC = \frac{m_{\text{eff}}}{P} \times 10^{-3} \quad \ldots (4)$$

Subsequently, the BTE was calculated from Eq. (5)[12].

$$BTE = \left(\frac{3600}{BSFC \times LHV \times 100}\right) \quad \ldots (5)$$

where, LHV is the lower heating value of fuel (MJ/kg).

3 Results and Discussion

The variations in peak in-cylinder pressures with loads for diesel, IBE10, IBE30, IBE29.5W0.5 and IBE29W1 are depicted in Fig. 2(a and b). Under lower loading conditions, the peak in-cylinder pressure was lower. These values were higher for diesel fuel than the IBE as well as water-containing-IBE blends. This was attributed to the low-temperature environment that slows the combustion processes of fuels. Under medium and high loads, the peak in-cylinder pressure increased as a result of enhanced combustion temperature and more fuel admission. The higher calorific value of diesel indicates that diesel possessed greater energy levels that may lead to higher peak in-cylinder pressure[12]. These Figures showed an increase in in-cylinder pressure with an increase in engine load. As the engine load increases, more fuel was charged into combustion chamber that resulted in higher release of energy during combustion process[14]. The introduction of water to the IBE30 blend resulted in a decrease in the peak in-cylinder pressure. The cooling effect of water led to lower

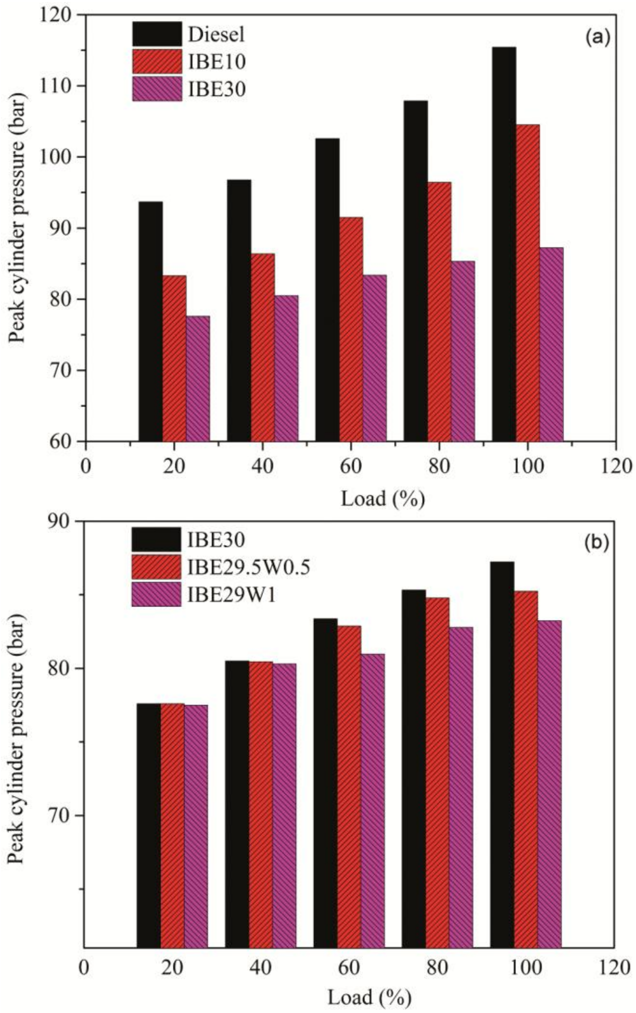

Fig. 2 — Variation of peak in-cylinder pressure with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

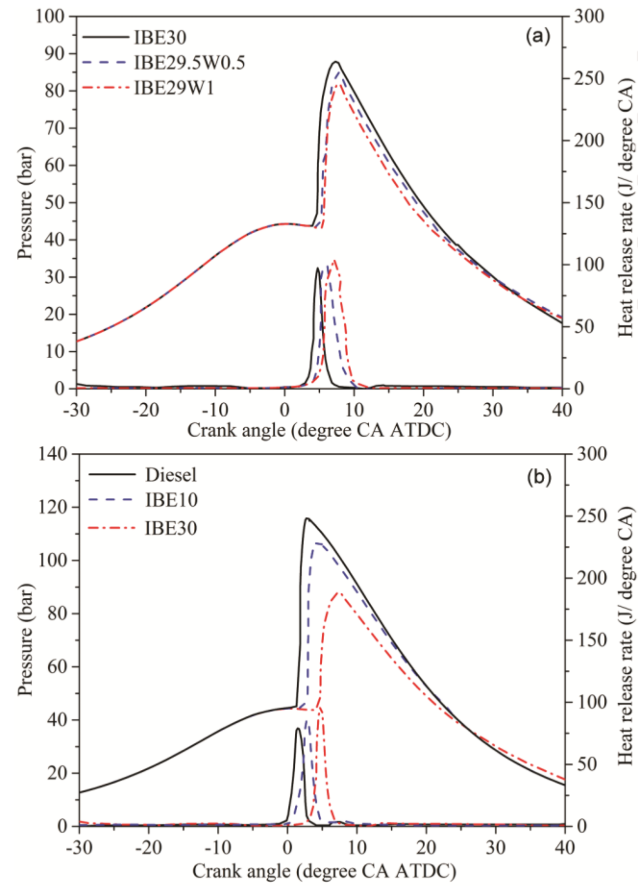

Fig. 3 — Variation of in-cylinder pressure and HRR for the test fuels at 100% load considering the effect of (a) water addition, and (b) blend ratio.

in-cylinder temperature which results in a lower peak in-cylinder pressure[12,13,24].

The variations of in-cylinder pressure development and HRR for the tested fuels at the maximum load was depicted in Fig. 3(a and b). Pure diesel had lower HRR than IBE10, IBE30, IBE29.5W5 and IBE29W1. The calorific value of diesel was higher than the other blends, this results in lesser utilization of the amount of diesel fuel during combustion and consequently lower heat release rate[25,26]. Also, the ignition delay of the fuel blends increases with increasing blends ratio as it allows perfect air-fuel mixing leading to fast spontaneous combustion in the premixed combustion chamber producing greater HRR[12,26]. Fuel blends had higher density and kinematic viscosity which resulted in increased fuel droplet size and consequently reduces the mass fraction burnt in the premixed combustion phase[27]. By emulsifying IBE, with 0.5 vol.% and 1 vol.% water, an increase in HRR was seen. Higher surface area to volume ratio of water particles and the catalytic activity of water vapor plays a significant role in enhancing the combustion process[28]. Apparently, for the particular blends considered, the HRR was proportional to increase in the amount of water added. The micro-emulsion phenomenon of water in IBE30 enhanced the air/fuel mixing, speeding up combustion process and resulting in a higher HRR[17,28,29].

Figure 4 shows the variations in the ignition delay period under various engine loads. As the engine load decreased, the ignition delay lengthens that results in a decrease in wall temperature and residual gas temperature. The ignition delay decreased at higher engine load as a result of higher in-cylinder gas temperature developed[30]. It was noticed from Fig. 4(a) that the combustion processes of IBE10 and IBE30 was delayed than that of pure diesel sample. This may be attributed to lower cetane number of the blends[10].

Another reason for the delay in the combustion process of the IBE/diesel blends was the need to absorb heat energy to vaporize. Also, fuel blends possessed poor ignitability as a result of lesser cetane number. This resulted in prolonged ignition delay and retarded combustion[9]. The water-containing blends exhibited longer ignition delay due to higher viscosity and lower cetane number than the baseline fuel and IBE30. This led to a shift in the SOC near TDC position. This prolonged ignition delay period as shown in Fig. 4(b) aided in perfect air-fuel mixing and hence the efficiency of process was enhanced[12]. The end of combustion was later for the water containing blends than the IBE30, thus the combustion duration was delayed as seen in Fig. 4(b). This was due to release of the latent heat of vaporization and emulsification of water with the blends which enhanced the fuel activity. It was noticed that the CA50 for diesel was earlier as compared to IBE10 and IBE30 blends[32]. This was due to the composition of the fuel blends which affected the start of the combustion (SOC)[10]. The addition of water showed a late completion of combustion completion due to gain in the density of the fuel as a result of water addition.

The variations in combustion duration with load was depicted in Fig. 5. The burning period reduced with an increase in engine load. At lower loads, the in-cylinder pressure developed was low for all fuel samples tested due to lower-temperature environment and the longer ignition delay period that resulted in delayed combustion process up to expansion stroke. But as the load increased, the combustion duration reduced and faster burning rate occurred as a result of higher temperatures in premixed phase phase[14,30]. The combustion duration of the blends was longer than the

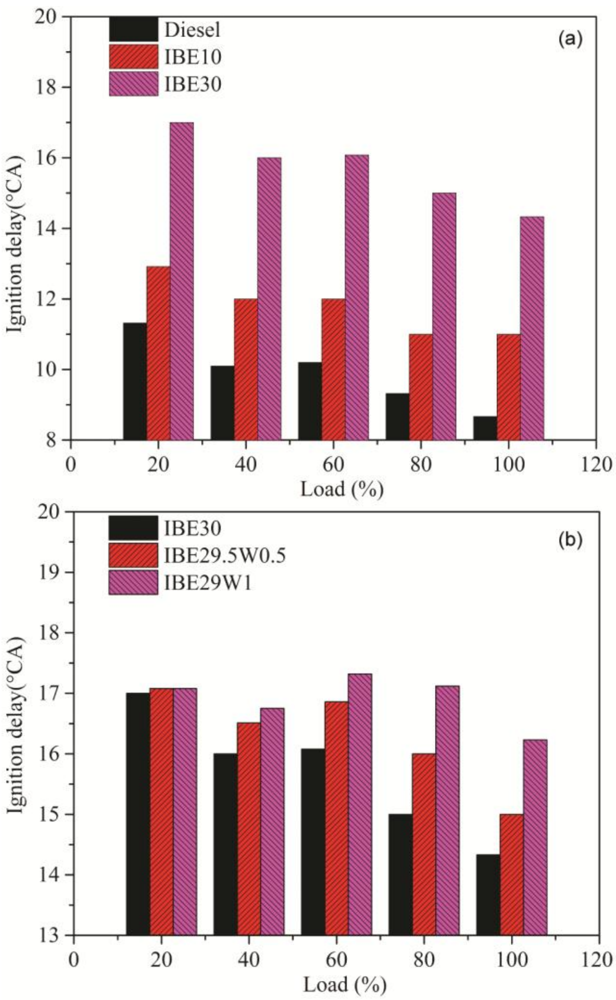

Fig. 4 — Variation of ignition delay with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

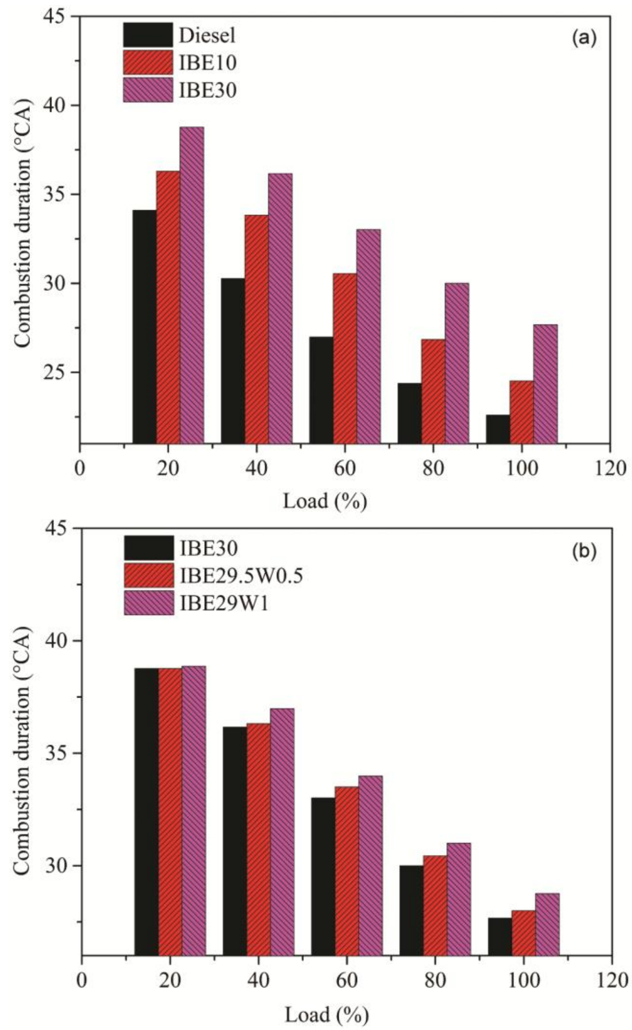

Fig. 5 — Variation of combustion duration with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

pure diesel as shown in Fig. 5(a). This was due to improvement in the premixed combustion of IBE blends, longer ignition delay and the effects of OH radicals produced by IBE blends[31].

IBE10 had 10.4 – 12.50% lower peak in-cylinder pressure, 12.4 – 21.2% higher ignition delay and 6.04 – 11.7% higher combustion duration as compared to pure diesel sample. IBE30 had 20.2 – 32.3% lower peak in-cylinder pressure, 8.4 – 59.1% higher ignition delay, and 12.01 – 18.7% higher combustion duration than that of pure diesel.

Similarly, IBE29.5W0.5 had 0 – 2.3% lower peak in-cylinder pressure, 0.5 – 4.6% higher ignition delay and 0 – 1.5% higher combustion duration than that of IBE30. IBE29W1 had 0.13 – 4.8% lower peak in-cylinder pressure, 0.5 – 12.4% higher ignition delay, and 0.26 – 3.8% higher combustion duration than that of IBE30.

The BTE of the IBE blends was lower than the pure diesel as shown in Fig. 6(a). The BTE was proportional to the engine load[14]. This may be attributed to an increase in the in-cylinder temperature developed and improved combustion process. Some physio-chemical properties of the IBE blends including lower density, viscosity, and calorific value were lesser than those of pure diesel fuel sample. These properties decreased with an increase in blending ratio which led to a decrease in BTE. Ideally, BTE decreased as a large percentage of water was added to a fuel sample[12]. However, a small quantity of water addition may increase the BTE of the diesel-IBE blends[19]. Interestingly, since the water-containing blends have higher density and viscosity, this affects fuel atomization which results in a decrease in BSFC. This trend was shown in Fig. 6(b). The emulsification of water in the IBE blends, also aided in air-fuel mixing and higher BTE[18,26].

The BSFC of the diesel-IBE blends was higher as compared to pure diesel sample as seen in Fig. 7(a). The blending of diesel with IBE fuel resulted in a decrease in lower calorific value, density and viscosity of the IBE blends. This necessitated the combustion of more amount of fuel. It was shown that BSFC had an inverse relationship with increase in engine load. The loss of energy to the combustion wall and friction was more pronounced at lower loads. As the engine load increased, the in-cylinder pressure and temperature raised, thus enhancing the combustion efficacy and leading to a decrease in the BSFC[33]. The addition of water decreased the BSFC as

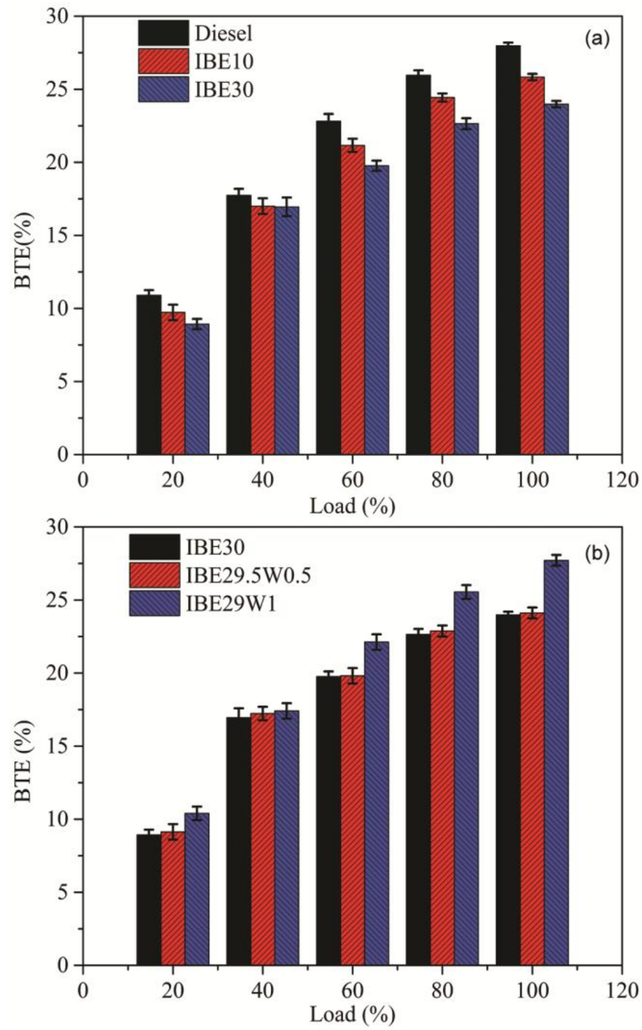

Fig. 6 — Variation of BTE with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

shown in Fig. 7(b). This was in line with available literature[12,18,34]. The emulsification of water with the blends enhanced the combustion process. A prolonged ignition delay was observed that aided in an increase in the quantity of fuel combusted during premixed stage of combustion[34]. Apparently, the water content might have been converted into superheated steam that resulted in an increase in engine power and a decrease the BSFC[12,18].

IBE10 demonstrated 6.3 – 12% lower BTE and 0 – 5% higher BSFC as compared to pure diesel sample. IBE30 demonstrated 4.7 – 22.1% lower BTE and 1.9 – 14.3% higher BSFC as compared to pure diesel sample. IBE29.5W0.5 had 0.3 – 2.19% higher BTE and 0 – 6.2% lower BSFC as compared to IBE30 sample. IBE29W1 had 2.6 – 14.1% higher BTE and 0 – 15% lower BSFC than that of IBE30.

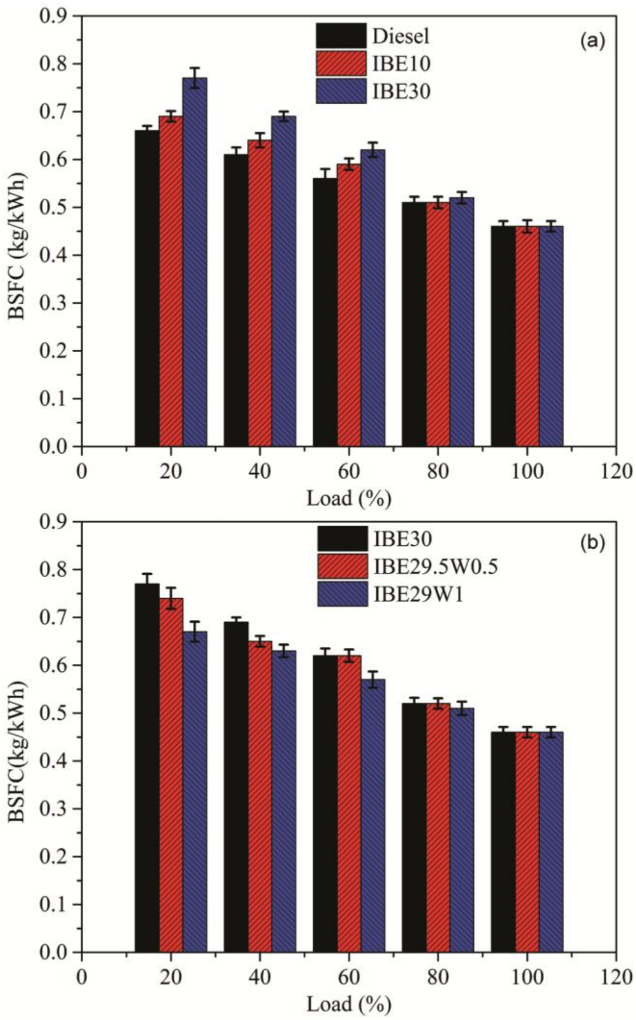

Fig. 7 — Variation of BSFC with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

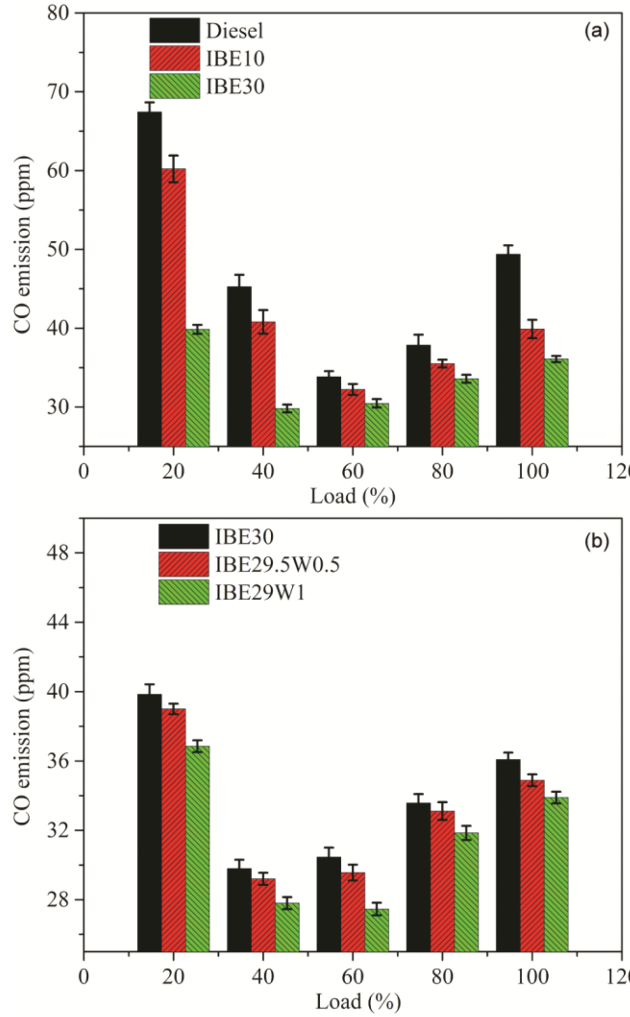

Fig. 8 — Variation of CO emission with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

Figure 8 shows the variations in CO emissions for different fuels tested at different loading conditions. The formation of CO emissions was aided by lower fuel/air ratios or lower in-cylinder temperature[13]. At lower load, the CO emission were higher as a result of a deficit in fuel/air ratio and lower combustion temperatures. By increasing the engine load, the quantity of fuel admitted inside the cylinder was raised, leading to insufficient combustion and consequently higher CO emissions. The CO emissions for IBE blends was lower than pure diesel sample. This was as a result of oxygen-enriched contents of IBE blends which aided in combustion process[8,10,18]. It was also shown in Fig. 8(b) that the addition of water further reduced the CO emissions. This was related to ability of water to reduce the combustion temperature due to it higher heat absorption capacity[13]. However, as the quantity of the water added was small, decreased in the CO emissions were reported that was in line with other literature[13,18,21,35].

From Fig. 9(a) it was seen that the diesel-IBE blends showed higher NOx emissions as compared to diesel. By increasing the engine load, the formation of the NOx also increased due to the increasing amount of fuel supply and higher combustion temperature[36,37]. The excess oxygen content which was a beneficial in decreasing CO emissions was now a detriment in NOx emission. The excess oxygen content of oxygenated fuel coupled with elevated combustion temperature provoked NOx formation in the fuel blends. This was in line with previous reported works[8,36,37]. However, a decrease in NOx emission was seen in Fig. 9(b) as a result of water addition at all load conditions. The water addition reduced the

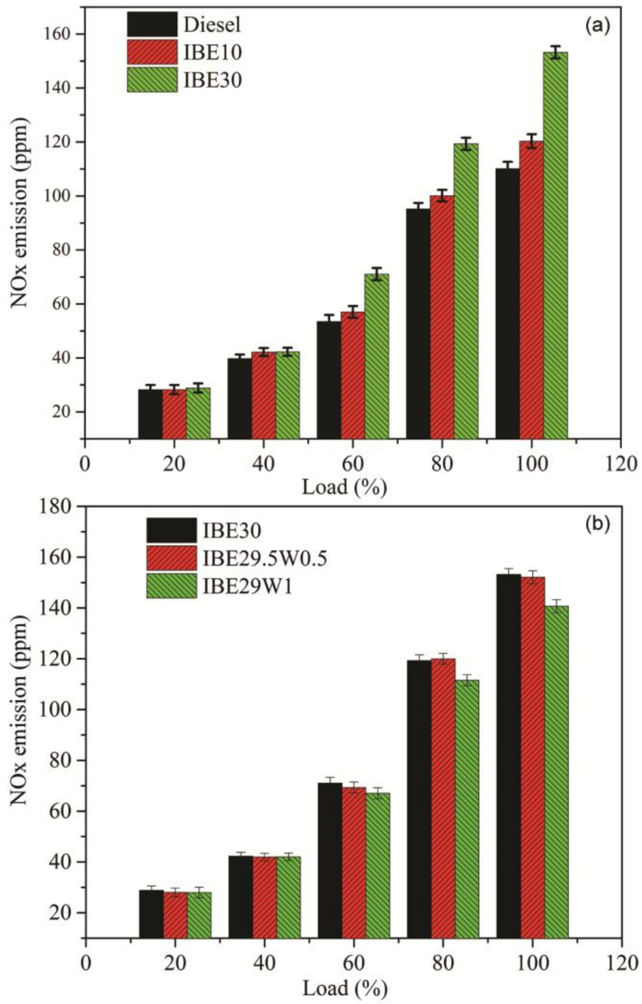

Fig. 9 — Variation of NOx emissions with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

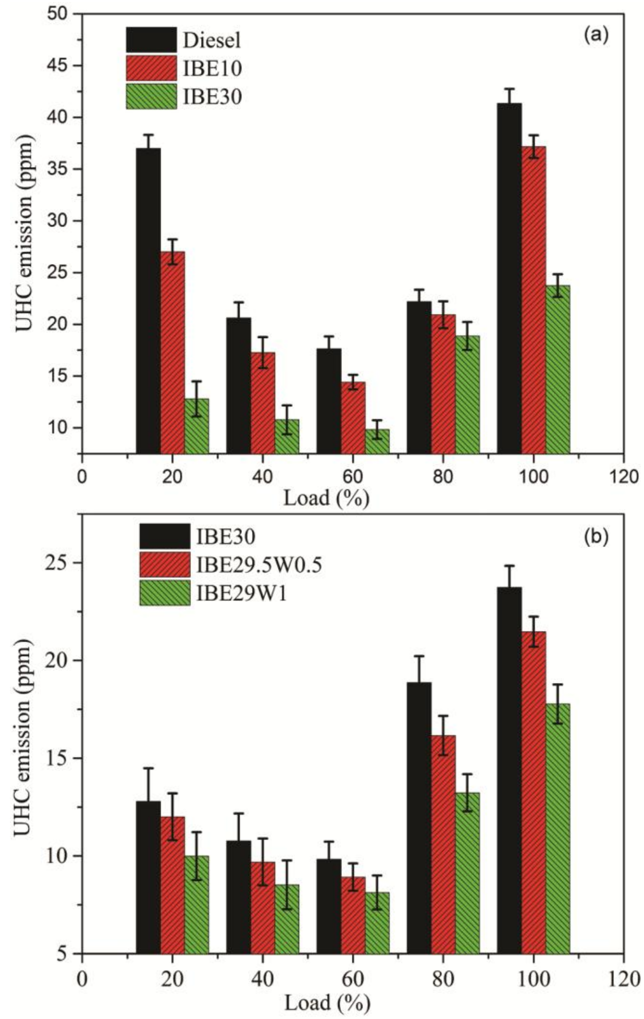

Fig. 10 — Variation of UHC emissions with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

adiabatic flame temperature that decreased the in-cylinder temperature developed resulting in lower NOx emissions[12,26]. Another logic behind the decrease in NOx emissions was that with a increase in water addition, the higher latent heat of inner phase water particles results in a drop in local combustion temperature[18,38]. Increasing the water content decreased the NOx emissions. This was as a result of heat absorbed by the water droplets that inhibits the reaction between N_2 and O_2 to form NOx[21,39].

Figure 10(a) shows the variation of UHC emissions of diesel and IBE/diesel blends at varying load. The excess amount of oxygen in IBE enhanced the combustion process and decreased UHC emissions[12,40,41]. The influence of water addition can be seen in Fig. 10(b). Water addition decreased the UHC emission from the IBE blends. The UHC decreased due to the addition of water in the IBE30 which was contrary to the assertion that the high latent heat of vaporization of water leads to a decrease in the in-cylinder temperature by quenching. However, the quantity of the water added was just enough to improve the combustion process and reduce the UHC efflux[8,12,14,18].

The concentration of smoke opacity is shown in Fig. 11(a). It can be seen that smoke emission decreased with increasing blend ratios and increased as the engine load increases[42,43,44]. This was because by increasing the engine load, more fuel was admitted into the cylinder which resulted in higher smoke opacity[45]. Similarly, the rich oxygen content of fuel blends promoted the lesser emission levels. The impact of water addition can be noticed in Fig. 11(b) which showed a further decrease in smoke emissions. The addition of water enhanced the mixing of the

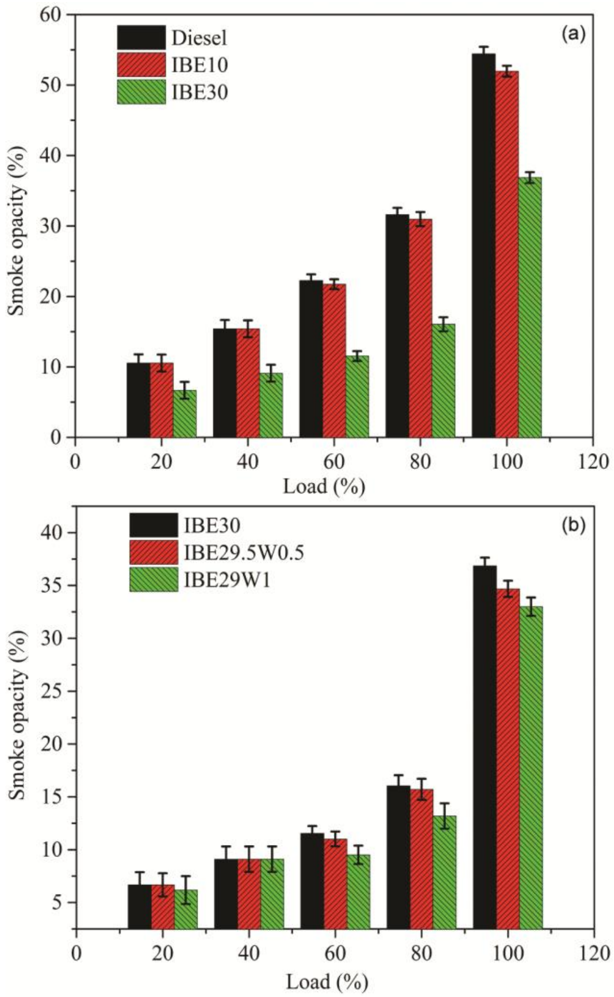

Fig. 11 — Variation of smoke emission with loads for the test fuels considering the effect of (a) blend ratio, and (b) water addition.

IBE/diesel blend as a result of micro explosion phenomenon. Other reasons could be due to enhancement of spray volume and an increase in OH radical concentration in the combustion enclosure[26,46].

IBE10 had 5 – 23.8% lower CO emissions, 0.01 – 8.5% higher NOx, 6 – 37% lower HC and 0.01 – 4.7% lesser smoke emissions as compared to pure diesel sample. IBE30 had 11 – 69.3% lower CO, 2 – 28.1% higher NOx, 17.6 – 89% lower HC, and 48 – 97% lower smoke than that of pure diesel. Similarly, IBE29.5W0.5 had 1.4 – 3.4% lower CO, 0.5 – 3% lower NOx, 6.6 – 16.8% lower HC and 0 – 6.2% lower smoke emissions as compared to IBE30 sample. IBE29W1 had and 5.4 – 11% lower CO, 0.64 – 9% lower NOx, 21 – 42.6% lower HC, and 0 – 21.7% lower smoke emissions than that of IBE30.

4 Conclusion

The combustion, performance and emissions characteristics of diesel, IBE/diesel blends and water-containing IBE/diesel blends were investigated for a single-cylinder direct injection diesel engine. Some vital conclusions have been drawn as follows:

(i) Comparing D100, IBE10 and IBE30, it was realized that the addition of IBE resulted in enhancing the combustion processes. The diesel fuel had higher in-cylinder pressure developed as compared to IBE/diesel blends. It was shown start of combustion process for D100 sample was earlier than IBE/diesel blends. It was noticed that the combustion duration for the IBE/diesel was longer than the D100 sample. Lower engine emissions were noted for various blends. The brake thermal efficiency decreased, whereas BSFC increased respectively.

(ii) The addition of water increased the ignition delay, lengthen the combustion duration, decreasing the in-cylinder pressure and increasing the HRR which resulted in better fuel-air mixing and more amount of fuel accumulation inside combustion chamber. The emissions levels of the water-containing blends especially and thermal efficiency was observed to decrease and increases respectively as compared to IBE30.

(iii) IBE29W1 (29 vol.% IBE, 1 vol.% water and 90 vol.% diesel) showed a decreased peak in-cylinder pressure developed, increased ignition delay and combustion period by 0.13% – 4.8%, 0.5% – 12.4% and 0.26% – 3.8% respectively as compared to IBE30. The BTE enhanced by 2.6% – 14.1% and BSFC dropped by 0.1% – 15%, respectively. IBE29W1 was recommended as an ideal fuel for application in diesel engines.

(iv) The small proportion of water addition in the blends may be used as a potential technique in a diesel engine fueled with IBE/diesel blends to simultaneously preserve the engine thermal efficiency and decrease the engine emissions levels as other methods are not cost-effective.

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