



# Combustion, performance and emission analysis of diesel engine fuelled by higher alcohols (butanol, octanol and heptanol)/diesel blends

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## ABSTRACT

As there were limited efforts towards the substitution of diesel fuel with higher alcohols to operate diesel engine, the current work aims to evaluate diesel engine performance at wide operating conditions fuelled by blends of higher alcohols; including heptanol and octanol. To increase the trust of the received results, blends of lower alcohols (butanol) was studied where the attained results showed good agreement with those cited in the literature. Single-cylinder diesel engine setup equipped with the necessary sensors for measuring cylinder pressure, engine speed, output power, air flow rate, fuel flow rate and temperature at various locations in engine was used to realize the current objective. The study limits the blending ratios to be 10% and 20% v/v to keep the fuel system without any modifications. These blends showed stable and homogenous features for period of four months without phase separation. The steady state experiments were carried out at various engine loads (0%, 25%, 50% and 75% of the full load) and engine speeds (900 rpm and 1500 rpm). Results revealed that bsfc and BTE increased almost for all tested higher alcohol/diesel blends. The ignition delay increased with higher alcohols/diesel blends and the longest value was achieved by But10 and But20. The premixed combustion was enhanced for all tested higher alcohol/diesel blends. The highest cumulative net heat released was achieved for Hept20. The emissions of NO<sub>x</sub> and opacity were reduced while those of CO and HC emissions were increased for all tested higher alcohol/diesel blends compared to the corresponding values for D100.

## 1. Introduction

The production and usage of alcohols to substitute conventional fuels for engine applications have been studied over the last decades to promote energy sustainability and the reduce air pollution [1]. Alcohols are liquid biofuels produced biologically through fermentation of biomass to producing various alcohols such as methanol, ethanol, butanol and other higher alcohols [2]. The alcohols production through fermentation process found to be more cost effective and environmental friendly compared to other chemical processes [3]. The chemical and physical characteristics of long chained alcohols (commonly called as higher alcohols) found to be compatible with diesel engines [4]. Therefore, higher alcohols are recommended to substitute the conventional fuels used for compression ignition (CI) engines [4].

Alcohols such as ethanol (C<sub>2</sub>H<sub>5</sub>OH) and methanol (CH<sub>3</sub>OH), known as lower alcohols, are the most widely tested alcohols as blends with conventional diesel fuel in CI engines. However, they suffer from several drawbacks such as their very low calorific value and high enthalpy

of evaporation that deteriorate the engine performance and increase engine emissions of CO and HC [5]. Additionally, ethanol and methanol have very high resistance to autoignition (causing long ignition delay periods and less combustion controllability) because of their low cetane numbers of 8 and 5, respectively [6]. They also require more strict safety precautions in fuel handling and storage due to their high vapor pressures and low flash points [7]. Furthermore, the poor lubricating properties and poor miscibility of ethanol and methanol with conventional diesel fuel prevent their usage in CI engines [7].

To overcome the previously discussed drawbacks of lower alcohols, researchers studied various ways, other than direct blending with diesel fuel, to increase the appropriateness of these lower alcohols for diesel engine operation; including emulsion [8] and fumigation [9,10]. Also, researchers examined the injection of lower alcohols either into exhaust manifold [11] or into intake port utilizing low temperature combustion strategies [12,13]. The use of ternary blend of alcohol/diesel/biodiesel or vegetable oil is another examined technique to overcome the previously discussed problems. Pinzi et al. [14] suggested the usage of

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Nomenclature	
AHRR	apparent rate of net heat release [J/deg]
ASTM	American Society for Testing and Materials
aTDC	degrees after TDC
bmep	brake mean effective pressure [MPa]
bsfc	brake specific fuel consumption [g/kWh]
bTDC	degrees before TDC
BTE	brake thermal efficiency [%]
But10	10% butanol + 90% diesel (by vol.)
But20	20% butanol + 80% diesel (by vol.)
CA	crankshaft rotation angle
CA05	angle at which 5% of cumulative net heat is released [° aTDC]
CA50	angle at which 50% of cumulative net heat is released [° aTDC]
CA90	angle at which 90% of cumulative net heat is released [° aTDC]
CI	compression ignition
CO	carbon monoxide
CO <sub>2</sub>	carbon dioxide
COV	coefficient of variation
D100	pure diesel oil
DAQ	data acquisition system
dP/dθ	rate of pressure rise [bar/deg]
dQ <sub>net</sub> /dθ	apparent rate of net heat release [J/deg]
dV/dθ	rate of change of cylinder displacement with crankshaft angle [m <sup>3</sup> /deg]
P(dV/dθ)	rate of work transfer
EGT	temperature of exhaust gases [°C]
EOC	end of combustion
HC	unburned hydrocarbons
Hept10	10% heptanol + 90% diesel (by vol.)
Hept20	20% heptanol + 80% diesel (by vol.)
imep	indicated mean effective pressure [MPa]
NO <sub>x</sub>	nitrogen oxides
O <sub>2</sub>	oxygen
Oct10	10% octanol + 90% diesel (by vol.)
Oct20	20% octanol + 80% diesel (by vol.)
OH	hydroxyl group
ppm	particle per million
Q <sub>net</sub>	cumulative net heat released [J/cycle]
SOC	start of fuel combustion process in the combustion chamber
SOI	start of fuel injection into the engine cylinder
TDC	top dead center
V	instantaneous cylinder volume [m <sup>3</sup> ]
γ	specific heat ratio
θ	crank angle degree
σ <sub>imep</sub>	standard deviation in imep

castor oil as a co-solvent to improve the stability and miscibility of ethanol and diesel blends. They found that, the addition of castor oil to ethanol and diesel blends enhanced their physical, chemical and combustion characteristics.

On the other hand, there are many advantages of using lower alcohols (ethanol and methanol) in diesel engines. The high oxygen contents of ethanol and methanol and the existence of hydroxyl (OH) group enhance the soot oxidation during the mixing controlled and late combustion phases leading to lower opacity levels particularly at high engine operating loads [15–17].

Recently, researchers got attention towards the usage of higher alcohols (C<sub>4</sub>–C<sub>20</sub>) to substitute diesel fuel as they have closure physical properties approaching those of diesel in comparison with the corresponding properties of lower alcohols. The higher alcohols defined by Kumar and Saravanan [4] as a long chain alcohols with high molecular weight and carbon atoms more than three atoms (C<sub>4</sub>–C<sub>20</sub>). Higher alcohols includes butanol (C<sub>4</sub>H<sub>9</sub>OH), pentanol (C<sub>5</sub>H<sub>11</sub>OH), hexanol (C<sub>6</sub>H<sub>13</sub>OH), heptanol (C<sub>7</sub>H<sub>15</sub>OH), octanol (C<sub>8</sub>H<sub>17</sub>OH), decanol (C<sub>10</sub>H<sub>21</sub>OH) and up to phytol (C<sub>20</sub>H<sub>39</sub>OH) [4]. Compared to lower alcohols, the higher alcohols showed good potential for diesel engine operations because of their higher calorific value, better blend stability with diesel fuel (no phase separation occurred), higher density, higher viscosity, better lubricity and higher cetane numbers and so better ignition quality [4,18,19]. Additionally, the corrosion effect of higher alcohols on fuel injector, fuel pump and delivery lines are very low as compared to lower alcohols because of their low water contents. The longer carbon chained higher alcohol requires less safety precautions in fuel handling and storage due to its high range of flash points [20]. Moreover, higher alcohols with longer carbon chains consume less energy during its production compared to lower alcohols, the breaking down process is shorter [21]. Considering all the above features, higher alcohols have very good potential for their usage in diesel engines.

Many researchers studied the combustion, performance and exhaust emissions of CI engines fueled by several types of higher alcohols. Butanol (C<sub>4</sub>H<sub>9</sub>OH) is the most studied type of higher alcohols in diesel engines [20–27]. The blends of butanol with diesel up to 40% by volume lead to higher engine brake thermal efficiency (BTE) and lower exhaust gases concentration [28,29]. For the combustion analysis, the

ignition delay period increased for higher butanol/diesel blending ratio due to the decrease in the overall cetane number of the blended fuel; cetane number of butanol is approximately 17 [16]. The longer ignition delay periods corresponding to lower cetane number allows longer mixing time between air and fuel before autoignition that results in higher rate of heat release in the premixed combustion phase. This results in more homogenous fuel/air mixture is formed and more fuel is ready to be burned in the premixed mode [28]. Additionally, butanol has high oxygenated contents that improves the fuel burning rate during the mixing-controlled burning phase. The laminar flame speed of butanol is 45 cm/s [30], which is higher than that of diesel fuel (33 cm/s). The high laminar flame speed leads to high combustion efficiency. For engine performance, the brake specific fuel consumption (bsfc) reported to be increased with increasing butanol/diesel blending ratio [31]. The increase in bsfc is due to the lower heating value of butanol compared to that of pure diesel fuel (D100) by 21%. Therefore, the engine requires additional input energy which means more injected fuel quantity to substitute this reduction in the fuel calorific value to maintain engine brake power. The engine brake thermal efficiency (BTE) increases with increasing butanol/diesel blending ratio [31]. The higher thermal efficiency could be owing to the high oxygen contents which improves the fuel burning process and fuel conversion efficiency.

Considering the engine emissions, the emissions of nitrogen oxides (NO<sub>x</sub>) decreased with increasing butanol/diesel blending ratio because of its high enthalpy of evaporation [4]. The enthalpy of evaporation of butanol is 581 kJ/kg compared to 280 kJ/kg for diesel fuel [16]. The high enthalpy of evaporation for butanol results in decreasing the local combustion temperature and consequently lower NO<sub>x</sub> emissions [32]. However, the lower cetane number (longer ignition delay) and higher oxygen contents in butanol enhances the premixed combustion with higher flame temperature which results in higher NO<sub>x</sub> emission. Also, the NO<sub>x</sub> emissions may remain unchanged compared to neat diesel because of these two opposite effects. The balance between these two tendencies in addition to engine specification and operating conditions will determine whether the NO<sub>x</sub> emissions will increase or decrease. However, generally the NO<sub>x</sub> emissions reported to be decreased with butanol [4,32]. Also, it was reported that soot concentration decreased with higher butanol/diesel blending ratios. This returns to the existence

of the hydroxyl (OH) group in butanol chemical structure. The OH group accelerates the oxidation process of soot formed during mixing controlling combustion process. The reduced soot concentration is also attributed to the existence of oxygen molecule in butanol chemical structure with lower carbon contents [20]. The HC and CO emissions increased with increasing butanol/diesel blending ratio [33]. The increase was attributed to the decreased local flame temperature caused by the high enthalpy of evaporation, slow fuel vaporization and fuel-impingement to the walls [4].

On the vehicle test level, Lapuerta et al. [34] tested the effect of butanol/diesel blends with maximum of 16% volumetric ratio on the performance and emissions of Euro 6 car at cold and warm ambient temperature. They found that the addition of butanol enhanced the vehicle performance with some limitations of operation at cold start and reduced the PM emission. They also reported increase in emissions of  $\text{NO}_x$  and CO. Atmanli et al. [35] examined the butanol as a co-solvent between diesel and several types of vegetable oils. They reported that butanol addition improves the stability of the ternary blend without observing phase separation. They concluded that butanol/vegetable oils/diesel blends reduced the engine output power, bmep, BTE,  $\text{CO}_2$  and HC emissions while increased the bsfc, CO and  $\text{NO}_x$  emissions.

Pentanol ( $\text{C}_5\text{H}_{11}\text{OH}$ ) is a longer straight chain alcohol with more potential to be used in CI engines because of its high calorific value, higher density, higher viscosity and better blending stability with diesel compared to methanol, ethanol, and butanol [36]. The effect of pentanol on CI engine combustion, performance and emission has been studied with a blending ratio up to 45% by volume [37]. The results found that ignition delay period was increased with increasing pentanol/diesel blending ratio owing to the reduction in cetane number of the fuel blend; as pentanol has cetane number of 20 [36]. The longer ignition delay periods for pentanol/diesel blends resulted in more fuel burned in the premixed phase leading to higher rate of net heat released during premixed combustion phase and consequently higher combustion pressure rise rate. This is also because of the fact that pentanol has higher laminar flame velocity (50 – 60 cm/s) compared to butanol (45–50 cm/s) [38,39]. The BTE was reported to be increased for diesel engines fuelled by pentanol/diesel blends compared to that for neat diesel fuel. This increase is attributed to the high oxygen contents of pentanol/diesel blends [18]. The bsfc of pentanol/diesel blends is higher than that for pure diesel fuel owing to the lower heating value of pentanol (19% lower than that of pure diesel). Consequently, more fuel consumed to generate the same brake power [40]. For exhaust emissions,  $\text{NO}_x$  emissions were reduced by using pentanol/diesel blends. However, at high engine loads, high  $\text{NO}_x$  emissions were reported with the addition of pentanol [41]. The reported high  $\text{NO}_x$  emissions can be attributed to the high local flame temperature due to the existing of oxygen contents and the long ignition delay periods of pentanol/diesel blends that tend to increase thermal  $\text{NO}_x$  as per Zeldovich mechanism. The unburned hydrocarbon and carbon monoxide emissions were reported to be increased with the addition of pentanol [42]. However, pentanol/diesel blends showed a reduction in smoke emissions from diesel engine [43]. The experiment performed on the constant volume combustion chamber also proved that pentanol is strong oxidant for soot at all tested conditions [44]. However, comparing to butanol, pentanol had less efficient in the reduction of PM emissions [45].

Heptanol ( $\text{C}_7\text{H}_{15}\text{OH}$ ) and octanol ( $\text{C}_8\text{H}_{17}\text{OH}$ ) are longer chain higher alcohols compared to pentanol. Their physical and chemical characteristics indicate that they are very close to the characteristics of conventional diesel fuel. However, to the authors knowledge, limited research work considered the effect of heptanol/diesel blends on the emissions of CI engine [46,47]. The  $\text{NO}_x$  emissions reported to be reduced for heptanol/diesel blends compared to that of neat diesel. The heptanol/diesel blends had no significant effect on smoke emissions. The low oxygen contents of heptanol is the main reason of producing the insignificant effect on smoke and unburned hydrocarbon emissions [46]. To the authors' knowledge, a lot of aspects concerning the effect

of heptanol/diesel blends on the combustion, performance and exhaust emissions of diesel engine have not been studied yet.

Few researches work that considered the effect of octanol ( $\text{C}_8\text{H}_{17}\text{OH}$ ) blends with diesel on the combustion, performance and exhaust emissions of diesel engines are available in the literature. Deep et al. [48] studied the influence of blending octanol with diesel up to 30% blending ratio on diesel engine performance. They found that all emissions increased with the increase of octanol/diesel blending ratio without significant effect on engine efficiency. Zhang et al. [49] tested the effect of octanol/diesel blend of 30% by volume on mechanical performance and exhaust gases emissions of CI engine under the European stationary test cycle. The BTE was not affected while the bsfc was slightly increased. Soot and CO emissions were reduced, and  $\text{NO}_x$  emissions were increased as compared to those for pure diesel fuel. Kerschgens et al. [50] analyzed the effects of octanol/diesel blends on the combustion and emissions of the CI engine. Very low soot and  $\text{NO}_x$  emissions were reported, within the limits of Euro 6. However, an increase of CO and HC were reported. It was also reported that the ignition delay increased with octanol blends compared to pure diesel.

From the above reviewed literature, it can be concluded that few studies were considered the influence of octanol/diesel blends on the combustion characteristics of diesel engines with the relevant information, especially the optimum blending ratios, being incomplete and limited for specific operating conditions. Additionally, obvious gap exists since many aspects remain untested especially concerning the effect of heptanol/diesel blends on the diesel engine combustion, performance and emissions. The novelty of the current work is to close this gap by studying the effects of butanol/diesel, octanol/diesel, and heptanol/diesel blends of 10% and 20% v/v on the diesel engine combustion characteristics, mechanical performance and exhaust gases emissions. The main reason of testing butanol/diesel blends in the present study is to check the quality of the current work by comparing the main findings of the present work for butanol/diesel with those cited in the literature. The agreement in this comparison would confirm the accuracy of results and applied procedure that will be received for heptanol/diesel and octanol/diesel.

## 2. Experimental setup

The test setup is comprised of a single cylinder, air cooled, four stroke, and direct injection CI engine without any technical modifications. The engine is a naturally aspirated compression ignition engine with a compression ratio of 17. The engine capacity is 824 cc with 105 mm stroke and 100 mm bore. The combustion chamber is flat with a flat piston crown. The fuel injection is started at  $24^\circ$  before top dead center (bTDC) using four holes injector. The fuel pump is a diaphragm feed pump type and delivers the tested fuel to the combustion chamber with a pressure of 175 bar. The engine technical characteristics are presented in Table 1. Fig. 1 shows a full schematic diagram of the test setup. Table 2 indicates the technical characteristics of the used devices and sensors including model, measuring range, and relative uncertainty.

A 450 L air damping box (approximately 500 times the engine displacement) is connected to the intake manifold as shown in Fig. 1 to reduce the intake air pressure fluctuations. The air volume flow rate was determined by measuring the pressure drop across the laminar flow element model MERIAM 50MC2 that installed upstream the air damping box. The pressure drop was measured using differential pressure transducer of Setra model 239. The laminar flow element was calibrated to compensate the effect of ambient temperature and air density. The tested blend volume flow rate was determined by measuring the time duration required to drain 80 mL of the fuel out of the graduated cylindrical beaker to the fuel injection system. This measurement repeated five times at each tested condition to ensure the reading accuracy.

The combustion analysis conducted by recording the in-cylinder

**Table 1**  
The diesel engine main technical characteristics.

Items	Specifications
Model	DEUTZ F1L511
Compression ratio	17/1
Engine Type	Direct injection, air cooled, naturally aspirated CI engine
Displacement [L]	0.824 L
Maximum torque	44 Nm/900 rpm
Rated power at 1500 rpm rated speed	5.7 kW
Cylinder number	1
Bore × Stroke [mm × mm]	100 mm × 105 mm
Number of nozzle holes	4
Fuel delivery advance [° bTDC]	24
Fuel delivery pressure [MPa]	17.5
Fuel delivery pump type	diaphragm feed pump

pressure each 0.3° crank angle by piezoelectric pressure transducer of Kistler 6061B type installed inside the combustion chamber. The pressure transducer output signal wire is connected to charge amplifier of Kistler 5018A type. The position of the top dead center (TDC) in each cycle was picked up by inductive proximity sensor type LM12-3004NA that adjusted at 4 mm in front of the flywheel. Signals from the inductive proximity sensor, the charge amplifier and the differential pressure transducer were sampled using DAQ system (Model NI PCI6251 with terminal block SCB-68) and data were recorded using LabVIEW software in Excel sheet for later analysis. This acquired data was used to analyze the combustion process, calculate the rate of heat release and estimate the combustion phasing.

To ensure the steady-state operation of the engine during each test,

the temperatures of the exhaust gases, intake air, cylinder wall, lubricant oil, and ambient air were measured using thermocouples type-K and digital thermometer and the corresponding data of each test were recorded if these temperatures were steady. The engine is coupled with a DC generator (MODEL, MEZ-BURNO with generated power up to 10.5 kW) that used for measuring the engine brake power as well as for regulating the engine load. The generated power was consumed by six electric heaters installed inside a flowing water tank. The electrical current and voltage were measured using ammeter and voltmeter as shown in Fig. 1 to determine the generated output power. More details about the currently used experimental setup are also presented by El-Seesy et al. [51].

The emission analyzer Ecom J2KNPro used to determine the concentrations of NO<sub>x</sub>, CO, CO<sub>2</sub>, O<sub>2</sub> and HC (in form of CH<sub>4</sub>); the overall technical characteristics are given in Table 3. The AVL DiSmoke 4000 with technical characteristics shown in Table 2 was used for measuring the opacity level and soot concentration.

### 3. Methodology

#### 3.1. Data analysis and experimental program

The fuel burning process in the current study was analyzed by calculating the apparent rate of net heat release using the measured in-cylinder pressure variation with crankshaft rotation angle following Eq. (1) [52].

$$\frac{dQ_{net}}{d\theta} = \frac{1}{\gamma - 1} V \frac{dP}{d\theta} + \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} \quad (1)$$

At each tested condition, the in-cylinder pressure data were recorded for 50 consecutive cycles and averaged to minimize the cyclic

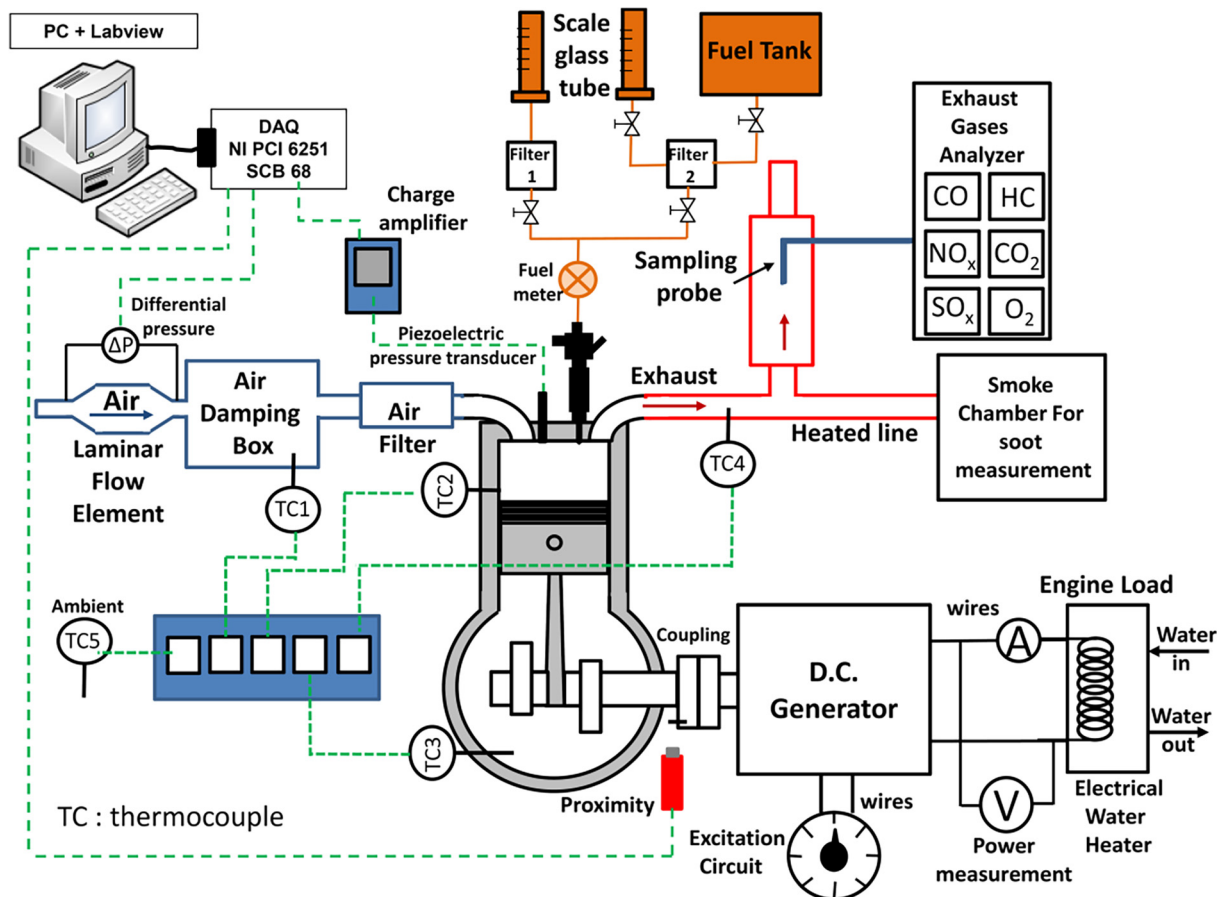


Fig. 1. Overview of the test system including setup components, sensors, devices and measurements.

**Table 2**  
Technical characteristics of the used sensors and devices.

Physical parameters	Sensor/device model	Measurement range	Least reading	Relative uncertainty
Cylinder pressure	Piezoelectric pressure transducer Kistler 6061B connected to charge amplifier model Kistler 5018A	0–250 bar sensitivity – 27.5 pc/bar	–	± 1% FS
Air volume flow rate	Laminar flow element (MERIAM 50MC2)	11,300 LPM	–	± 1% FS
Pressure drop through the laminar flow element	Setra differential pressure transducer Model 239	0–1245 Pa	0.01 Pa	± 0.14% FS
Temperatures of exhaust gases, intake air, lubricant oil, ambient air, cylinder wall	Thermocouple type-K with OMEGA Model 650 type-K Thermometer	– 200–1370 °C	0.01 °C	± 1 °C
Data Sampling	Data acquisition system type NI PCI6251 with terminal block model SCB-68 and LabView 7.1	–	–	–
Smoke opacity	AVL DiSmoke 4000	0–100%	0.1%	0.1%

**Table 3**  
Main technical characteristics of Ecom J2KNPro emission analyzer.

Measured gas	Measurement range	Sensor principle	Least reading	Error
O <sub>2</sub>	0:21 vol%	Electrochemical	0.01 vol%	0.2%
CO	0:63,000 ppm (6.3 vol%)	NDIR	1 ppm	0.1%
CH <sub>4</sub>	0:4 vol%	NDIR	0.001 vol%	0.025%
NO	0: 5000 ppm	Electrochemical	1 ppm	0.2%
NO <sub>2</sub>	0:1000 ppm	Electrochemical	1 ppm	0.1%
CO <sub>2</sub>	0:20 vol%	NDIR	0.1 vol%	0.5%
SO <sub>2</sub>	0: 5000 ppm	Electrochemical	1 ppm	0.2%

variation and provide appropriate determination of the cylinder pressure. During the current study, the cyclic variations were evaluated and the engine shows acceptable stability since the coefficient of variation ( $COV_{imep}$ ) for all tested condition was below 4% [53].

The analysis of combustion process conducted in the present study followed the methodology presented by Nour et al. [11,17]. A sample analysis of the measured data is shown in Fig. 2 for neat diesel combustion including the combustion phases and combustion timing. Additionally, the coincident variation of the measured in-cylinder pressure, apparent rate of net heat released and cumulative heat released per cycle are also shown in Fig. 2. The phases of the combustion process are an important parameter that affects the engine performance and formation of emissions. The main four phases of the CI combustion process are the ignition delay, premixed combustion, diffusion combustion phase (mixing controlled combustion phase) and the late

combustion phase. The duration and time limitation of each phase is estimated and shown in Fig. 2. The definitions of CA05, CA50 and CA90 which determines the end of one phase and the start of another phase are also presented in Fig. 2. The sample analysis shown in Fig. 2 is performed for each tested condition and each tested higher alcohol/diesel blend. Additionally, the NO<sub>x</sub> emissions index was calculated in g/kg fuel using Eq.(2).

$$C_{NOx} [g/kg \text{ fuel}] = C_{NOx} [ppm] \times 10^{-6} \times \frac{M_w [NOx]}{M_w [Exhaust \text{ gases}]} \times \frac{m_{exhaust \text{ gases}} [g/hr]}{m_{fuel} [kg/hr]} \quad (2)$$

where  $C$  is the NO<sub>x</sub> concentration,  $M_w$  is the molecular weight and  $m$  is the mass flow rate.

Table 4 shows the experimental conditions, the test program and the tested fuel blends. The experimental procedures followed in the current work summarized in Fig. 3. The fuel blends were tested while the engine speed was controlled at 900 and 1500 rpm. The engine speeds of 900 and 1500 rpm were selected to cover the low and high engine operation speeds. At each engine speed, the engine load was also controlled at various values of no-load, 25%, 50%, 75%, and engine full load. These loads were selected to cover broad band of engine operation loads and conditions.

The errors accompanied with the measured parameters are listed for each sensor in Table 2 and Table 3. To minimize this error, the value of the measured parameter recorded five times to ensure the repeatability of measured values, the averaged value was calculated and plotted to

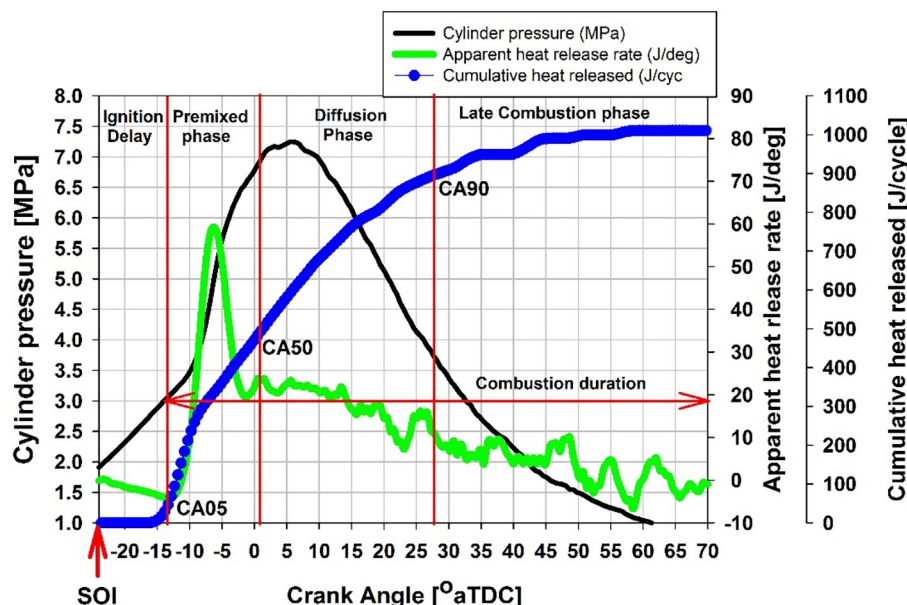
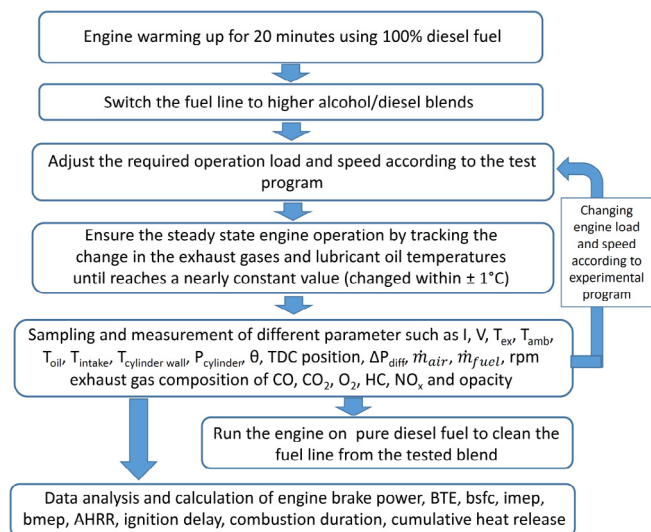


Fig. 2. combustion analysis and combustion phasing.

**Table 4**  
Engine test conditions and tested blends.

Engine speed [rpm]	900, 1500				
Tested loads [%]	0%	25%	50%	75%	100% at 1500 rpm
Equivalent imep for each load [MPa]	0	0.25	0.4	0.6	0.74
Fuel injection pressure [bar]	175				
Fuel injection timing [° bTDC]	24				
Tested fuels	Pure diesel				D100
	10% butanol + 90% diesel (by vol.)				But10
	20% butanol + 80% diesel (by vol.)				But20
	10% heptanol + 90% diesel (by vol.)				Hept10
	20% heptanol + 80% diesel (by vol.)				Hept20
	10% octanol + 90% diesel (by vol.)				Oct10
	20% octanol + 80% diesel (by vol.)				Oct20



**Fig. 3.** Experimental procedures.

represent the test condition. The error in derived values such as engine speed, bsfc, indicated mean effective pressure (imep), and brake power determined to be 0.5% ( ± 5 rpm), 2.3%, 3.8%, and 1% according to the procedure described by Kline [54] and Holman [55].

**3.2. Fuel blending and characteristics**

The current study tested the influence of various types of higher alcohols such as butanol, heptanol and octanol on engine performance which is strongly affected by blending fuel physical and chemical

**Table 5**  
Physical and chemical characteristics of higher alcohols and diesel.

Characteristics	Test method/reference	Diesel	Butanol	Heptanol	Octanol
Molecular formula	NA	NA	C <sub>4</sub> H <sub>9</sub> OH	C <sub>7</sub> H <sub>15</sub> OH	C <sub>8</sub> H <sub>17</sub> OH
Molecular structure	NA	–	<chem>CCCCO</chem>	<chem>CCCCCO</chem>	<chem>CCCCCO</chem>
C (%wt)	CHNSO Mass Analysis (Eurovector EA3000)	86.21	66.82	72.16	73.68
H (%wt)		13.79	13.64	13.71	13.91
O (%wt)		0	19.54	14.13	12.41
Molecular mass (kg/kmol)	NA	190–211.7	74.12	116.2013	130.23
Density, Specific gravity (kg/m <sup>3</sup> ) at 15 °C	ASTM D4052	0.835	0.811	0.818	0.825
Viscosity at 40 °C (mm <sup>2</sup> /s)	ASTM D445	2.72	2.28	5.75	7.59
Lower heating value (MJ/kg)	ASTM D240	42.54	33.63	39.92	38.4
Enthalpy of evaporation (kJ/kg)	[4,50,58]	270	584.19	574.95	315.1
Self-ignition temperature (°C)	[4]	254–300	345	275	270
Cetane number	ASTM D613/[16]	48	17	29.5	39.1
Stoichiometric air to fuel ratio	Calculated	14.5:1	14.2:1	14.41:1	14.45:1

characteristics. Table 5 presents the measured and reported properties of the tested fuels according to ASTM standard. The current study used absolute butanol, heptanol and octanol of purity 99%, while the diesel fuel was purchased from local gas station. Two blends with ratio of 10% and 20% by volume with diesel were prepared for each tested higher alcohol. Firstly, the blends were prepared by using mechanical mixing device of rotor stator type (IKA Dispersers T18 digital ULTRA-TURRAX). Then, the mixing between higher alcohol and diesel enhanced by ultrasonic dispersion for 20 min using ultrasonic processors (Hielscher model UP200S). After mixing, the homogeneity for the tested blends were noted for one week before experiment and three months after experiment, no phase splitting occurred. The tested butanol and octanol found to have better miscibility characteristics with diesel, no emulsifying agents were required. Also, there is no phase separation even after four months. The available literature also confirmed that the higher alcohols forms homogeneous mixtures with diesel. Kumar and Saravanan [4] reported that higher alcohols have better blending stability with diesel and no co-solvent would be required to maintain the blending stability. This returns to that higher alcohols are less hygroscopic nature compared to lower alcohols, i.e. methanol and ethanol [36]. Lapuerta et al. [16] found that butanol/diesel blends were stable at wide ranges of volume fractions and ambient temperatures as shown in Fig. 4. The unstable region detected by the area under the curve. The stability of butanol/diesel is also maintained in case of water existence [56]. Liu et al. [57] showed that the addition of octanol to hydrous ethanol/diesel blends promote the blending stability and prevent phase separation. However, the behavior of heptanol/diesel blends and their blending stability was not reported in the literature.

Thermogravimetric analysis (TGA) is also performed to characterize the behavior of fuel homogeneity, droplet vaporization, decomposition, and fuel-air mixing [59]. The TGA is conducted for all the tested fuels including diesel (D100), But10, But20, Hept10, Hept20, Oct10, and Oct20. The homogeneity can be checked from the vaporization and decomposition indicated from the weight loss curve. However, to predict the behavior of a certain blend, the TGA should run with the same temperature evolution that fuel experiences in the in-cylinder pre-combustion process. The analysis was conducted using Setaram LABSYS evo thermal analyzer that can operate in temperature range 25–1600 °C with temperature scanning rate ranged from 0.01 to 100 K/min and TG resolution of 0.02 µg. The tested fuel blends were mixed long time before the TG test. The weight of each samples was approximately 100 mg. The test procedure started with raising the furnace temperature from room temperature and fixed at 30 °C for 10 min to achieve thermal stability. Then, the temperature raised to a limit that ensures the complete fuel evaporation. This limit was set to be 300 °C in case of butanol, heptanol and octanol. In the case of diesel, But10, But20, Hept10, Hept20, Oct10, and Oct20 the temperature limit raised to be 500 °C. The temperature scanning rate used for both cases was set to be

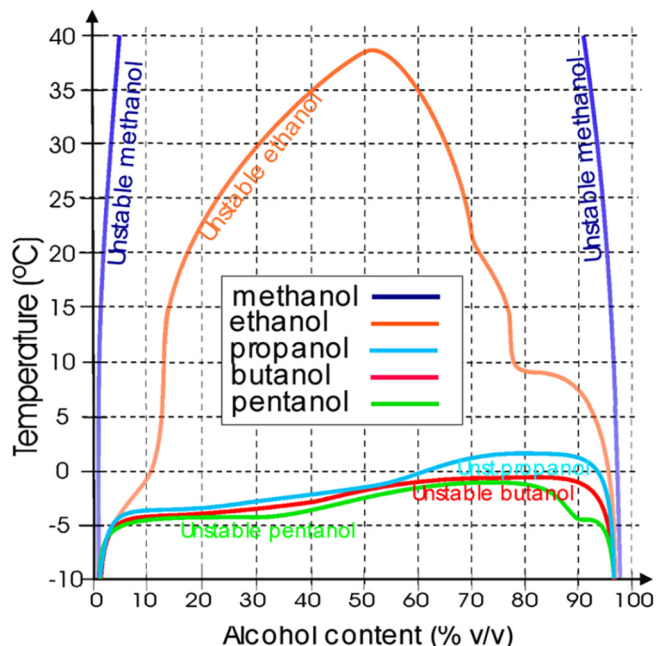


Fig. 4. Blending stability of several alcohols with diesel fuel [16].

20 K/min. The flow rate of the carrier gas, which is a high purity Argon, was 200 mL/min to prevent oxidation of the vaporized components.

Fig. 5 illustrates the thermogravimetric analysis of the tested fuels and blends. Fig. 5(a) indicated the weight losses as a percentage versus the temperature and Fig. 5(b) presented the rate of mass losses (dTG). The weight losses with temperature are different from fuel to another depending on fuel density, viscosity, boiling point, and volatility. It can be observed that butanol starts to evaporate at lower temperature compared to other fuels results in an enhanced fuel-air mixture, but on the other hand it leads to combustion cooling effect due to its higher latent heat of vaporization. Heptanol and octanol evaporated at higher temperatures compared to butanol. This is due to that heptanol and octanol have relatively high density, viscosity, boiling point, and low volatility compared to butanol results in poor atomization and improper in-cylinder mixing with air. Additionally, due to the longer carbon chain and more massive chemical structure compared to butanol. However, the vaporization characteristics of heptanol/diesel blends and octanol/diesel blends are much better than that of D100 which leads to enhanced fuel-air mixing process and reduces the physical delay in ignition due to fuel vaporization. The rate of mass losses for fuel blends (multi-component fuel) of But10, But20, Hept10,

Hept20, Oct10, and Oct20 covers a wide range of temperatures as illustrated in Fig. 5. This leads to that fuel vaporization starts earlier as noted for But20, Hept20 and Oct20 enhancing the fuel-air mixing compared to D100. Therefore, the higher rate of fuel vaporization of higher alcohol/diesel blends may result in a shorter delay compared to D100 with enhanced fuel-air mixture. Additionally, this test is performed after long time of fuel storage and the behavior of higher alcohol/diesel blends evaporation and decompositions proves the fuel homogeneity even after very long fuel storage time.

#### 4. Results and discussion

##### 4.1. Combustion analysis

Fig. 6 presents the effect of tested blended fuels on the in-cylinder pressure, the apparent rate of net heat release (AHRR), and in-cylinder average gas temperature at 25% low engine operation load and speeds of 900 and 1500 rpm. Fig. 7 shows also the same variations at high engine load of 75%. The major part from pressure history inside the cylinder where the combustion events are occurred is only shown in these figures. The events start from fuel injection up to the moment where the whole injected fuel is almost burned. Thus, only the duration from 25° bTDC to 25° aTDC is shown to concentrate on the combustion process and to improve the readability of these figures.

It can be observed that Hept10 and Hept20 blends achieved the highest peak cylinder pressure as indicated in Figs. 6(a) and 7(a). While blends of butanol (But10 and But20) show slightly lower peak pressure than that of D100. These behaviors are mainly due to the different in blended fuels energy contents; as in Table 5. The position of the in-cylinder peak pressure indicates the speed to release this energy. It is important to mention that, the rate of energy release depends on existing of oxygen in fuel structure (that accelerates combustion), viscosity (that affects the fuel atomization and vaporization) and latent heat (that directly affects the ignition delay and combustion cooling). Combining these factors can explain the behavior of in-cylinder pressure for Hept10, Hept20, But10 and But20 compared to that for D100 and octanol blends (Oct10 and Oct20). The retarded peak cylinder pressure for Hept10, Hept20, But10 and But20 is attributed to the prolonged ignition delay for heptanol/diesel and butanol/diesel blends. Also, the increase in the amount of heptanol and butanol in the blend results in longer delay in ignition. High self-ignition temperature, and the relatively low cetane number are the main reasons of the long ignition delay. Table 5 shows the self-ignition temperature and cetane number of heptanol, butanol and D100. Additionally, the high enthalpy of evaporation of butanol (584.19 kJ/kg) and heptanol (574.95 kJ/kg) as compared to that of D100 (270 kJ/kg) and octanol (315 kJ/kg) leads to longer ignition delay. Also, the high enthalpy of evaporation leads to

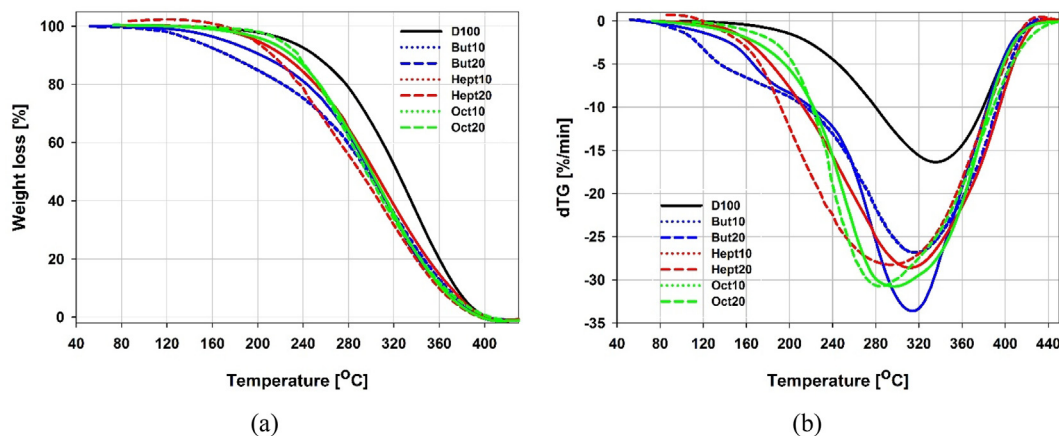


Fig. 5. TGA and dTG for the tested fuels.

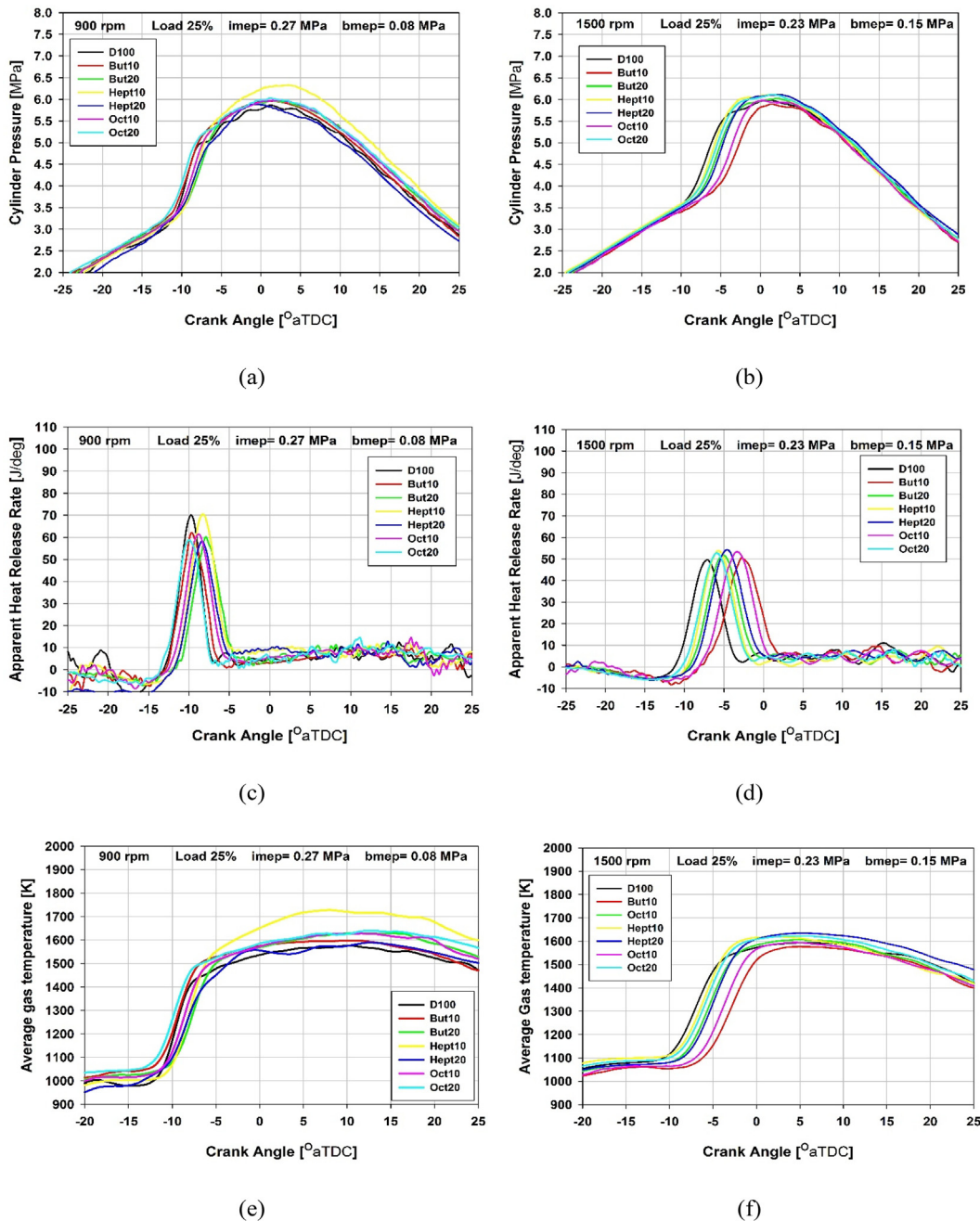


Fig. 6. The effect of various tested fuel blends on the variations in-cylinder pressure, apparent rate of net heat release, and in-cylinder average gas temperature at 25% low engine load and engine speeds of 900 and 1500 rpm.

combustion cooling effect which leads to lower in-cylinder pressure and consequently the in-cylinder average gas temperature as for But10 in Figs. 6(f) and 7(f). The increase in the butanol content in the fuel blend with diesel enhanced and accelerate the fuel vaporization and improves the fuel-air mixing in addition to the long ignition delay that results in improvement in premixed combustion and higher in-cylinder average gas temperature and pressure as shown in Figs. 6(f) and 7(f).

Unlike butanol, heptanol blends shows a shorter ignition delay period with higher in-cylinder average gas temperature and pressure and improved premixed combustion as shown in Fig. 6. The enthalpy of evaporation of heptanol (574.95 kJ/kg) is high enough to lead to a longer ignition delay but the cetane number is higher than that of butanol (29.5) results in shorter ignition delay. The premixed combustion for Hept10 and Hept20 improved due to the enhanced fuel-air mixing properties indicated by TGA analysis.

Octanol/diesel blends show slightly lower in-cylinder average gas temperature and pressure with sometimes short ignition delay. This is attributed to the poor fuel atomization due to octanol high viscosity in addition to the engine low injection pressure. The higher viscosity of octanol ( $7.59 \text{ mm}^2/\text{s}$ ) compared to that of diesel ( $2.72 \text{ mm}^2/\text{s}$ ) adversely affects the combustion efficiency due to the expected worsening in fuel atomization and consequently the inadequate fuel/air mixing. Therefore, less fuel fraction is burned in premixed phase and more fuel burned in mixing-control phase, so complete oxidation of the burned hydrocarbons may not be accomplished before opening of the exhaust valves.

The AHRR at 900 rpm and 25% and 75% loads as well as at 1500 rpm and 75% load also confirms that But20 has the longest ignition delay with the most retarded peak of AHRR during premixed phase. At 1500 rpm and load of 25%, But10 shows the longest ignition



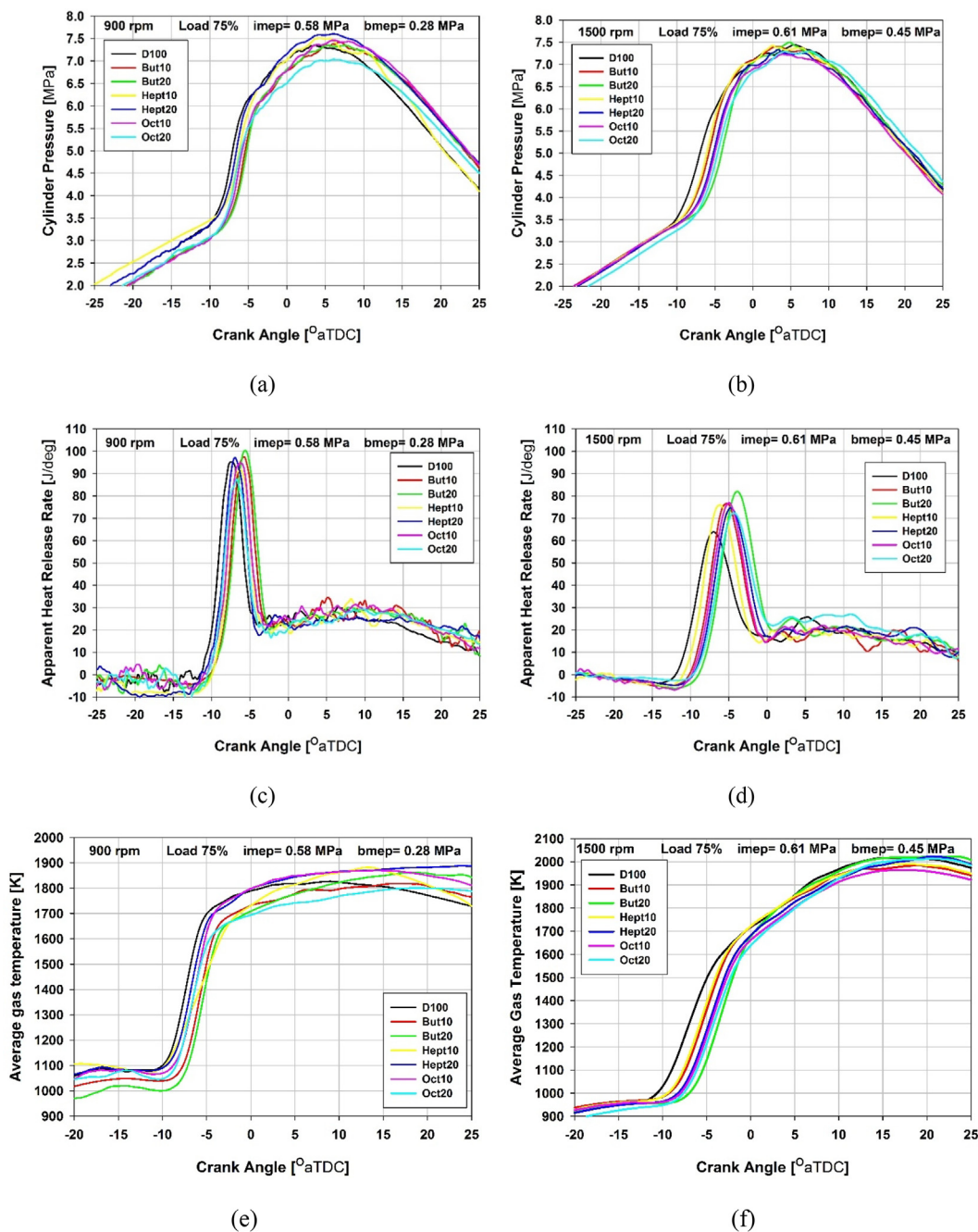


Fig. 7. The effect of various tested fuel blends on the variations of in-cylinder pressure, apparent rate of net heat release, and in-cylinder average gas temperature at 75% high engine load and engine speeds of 900 and 1500 rpm.

delay with the most retarded AHRR peak as clearly shown in Fig. 6. At high engine load of 75% for both tested speeds, the high AHRR peak at premixed combustion phase for But20 and But10 is attributed to the long ignition delay period that provides longer time for fuel and air to be vaporized and mixed. This longer mixing time allowing more fuel to be burned in premixed phase leading to higher AHRR; see Fig. 7. Additionally, butanol has more oxygen content than heptanol and octanol and this also enhances the fuel burning in the premixed phase. The high oxygen contents and the existence of hydroxyl group enhances the fuel burning not only during premixed phase but also in the mixing-controlled burning phase. This enhanced premixed combustion process is only shown at higher engine load of (75%) while it is not clear for lower engine loads as the quality of diffusion flame is improved at high temperature attained at high engine loads rather than low engine loads.

Despite the fact that longer chain alcohols such as heptanol and octanol have lower oxygen contents, compared to shorter chain alcohol such as butanol as shown in Table 5, the combustion process during premixed phase is improved for their blends compared to that for D100 as a result of this low oxygen contents and the long ignition delays due to their higher viscosities and their lower cetane numbers; as given in Table 5.

Fig. 6(b) reflects the increase occurred in the ignition delay due to the addition of heptanol and octanol where the peak cylinder pressure is slightly increased due to the improvement in premixed combustion AHRR. The longer ignition delay is expected due to the high enthalpy of evaporation for butanol (581.4 kJ/kg), heptanol (574.95 kJ/kg) and octanol (367 kJ/kg) compared to (270 kJ/kg) for D100. On the other hand, the high calorific value of heptanol and octanol increases the

heating value for tested blends and accordingly increases the cylinder pressure and the amount of heat liberated per cycle.

The higher viscosity of heptanol (5.75 mm<sup>2</sup>/s) and octanol (7.59 mm<sup>2</sup>/s) compared to that of diesel (2.72 mm<sup>2</sup>/s) adversely affect the combustion efficiency due to the expected worsening in fuel atomization and consequently the inadequate fuel/air mixing. Therefore, less fuel fraction is burned in premixed phase and more fuel burned in mixing-control phase, so complete oxidation of the burned hydrocarbons may not be accomplished before opening of the exhaust valves.

Fig. 8(a) presents the variation of the ignition delay (duration between SOI to CA05) and combustion duration (duration between CA05 to EOC) for butanol, heptanol and octanol blends with diesel fuel under various engine loads 25–75% at 1500 rpm. The error bars were added to show the minimum and maximum deviation from the average point. The ID period includes the physical delay period and chemical delay period. During the physical delay period the fuel atomization, vaporization, decomposition and mixing with air occurs. The chemical delay represents the time taken to start the combustion reaction. TGA analysis was performed to characterize the behavior of fuel vaporization, decomposition, and fuel-air mixing for the tested higher alcohol/diesel blends relative to D100 as indicated in Fig. 5. The higher alcohol/diesel blends enhanced the fuel-air mixing compared to D100. Additionally, the cetane number for all tested fuels are presented in Table 5. Butanol has the lowest cetane number (17) compared to other examined fuels, therefore the corresponding ignition delay is the longest among tested fuel blends.

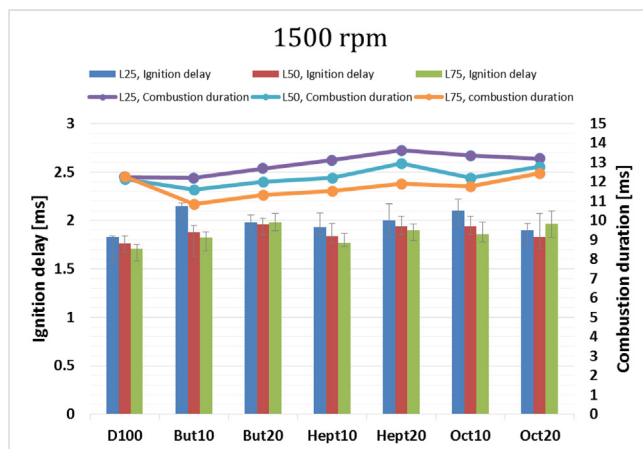
The combustion duration decreased with increasing the engine load. At higher engine speed, higher amount of fuel injected that required a higher amount of air enters with high turbulence intensity leads to fast fuel-air mixing and shorter burn duration. The combustion duration for But10 and But20 is lower than that of D100 at 50% and 75% engine loads. This could be due the enhancement of the premixed combustion process where more fuel is burned in premixed mode. Additionally, the laminar flame speed of butanol is 45 cm/s [39] which is higher than that of D100 (33 cm/s) [30]. Heptanol/diesel and octanol/diesel blends show longer ignition delays compared to that for D100, but shorter than that for butanol except at low engine loads (where ignition delay for But10 was the longest), see Fig. 8(a). This is due to their relatively low cetane number. The combustion duration for heptanol and octanol is longer than that of butanol blends and that of pure diesel fuel. As peak pressure location is changed to occur during the piston expansion, there is a greater tendency to extend the combustion duration as the in-cylinder temperature is lowering during the piston expansion. This would be compensated by regulating the fuel injection timing.

Fig. 8(b) illustrates the variation of SOI, CA05 (SOC), CA50 and EOC for different tested fuel at 75% engine load and 1500 rpm. Data presented in Fig. 8(b) reflects the results presented in Fig. 8(a). CA50 is a very important parameter for the performance and combustion of engine. CA50 present the end of the premixed combustion phase and the start of diffusion combustion phase. The premixed combustion duration (CA05 to CA50) for tested higher alcohols/diesel blends is shorter than that of D100.

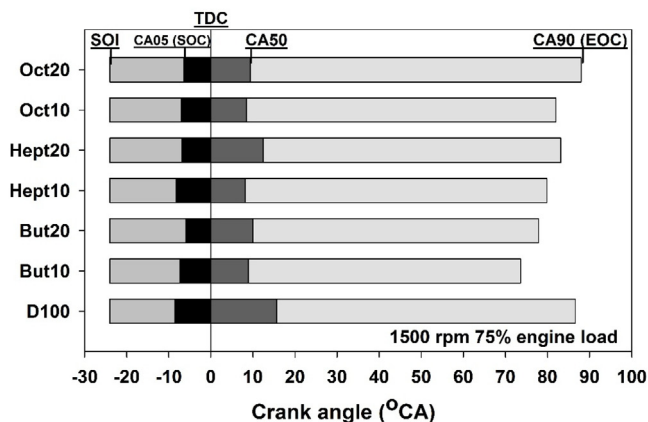
Fig. 9 presents the data of cumulative net heat released and imep for different tested higher alcohol/diesel blends under various engine loads 25%–75% at 900 and 1500 rpm. It can be observed that the imep (the average pressure that would act on piston per cycle generated from fuel combustion to provide the indicated power) is increased with the increase of the engine load to withstand the engine higher loads. At high engine speeds where turbulence of incoming air enhances the fuel distribution and mixing, the values of imep are approximately the same for all tested fuel blends. However, at the engine idling speed of 900 rpm, the variation in the physical properties of tested fuel blends has better tendency to motivate the cyclic variation at 900 rpm than at 1500 rpm. From Fig. 9, it can be concluded that, Hept20 shows the highest imep at 900 rpm while But20 shows the highest imep at 1500 rpm.

Moreover, from Fig. 9 it is noted that, the total heat released per cycle is increased with increasing the engine load to provide the specified engine power. The total heat released is slightly decreased for But10 and But20 compared to D100 because of its lower calorific value. Another reason is because part of combustion energy is consumed to withstand the higher enthalpy of evaporation for butanol than that for D100. For tested heptanol/diesel blends the total heat released is increased as a result of its high heating value which is close to that of D100 and much higher than that of butanol. The increase of the heptanol percentage in the fuel blend leads to an increase in the cumulative heat release as indicated in Fig. 9 for all tested engine conditions. For octanol/diesel blends, the total heat released is also increased due to that the higher calorific value of octanol. Therefore, the addition of octanol results in an increase in the calorific value for the overall blend and consequently the total heat released per cycle is increased. Also, the enthalpy of evaporation for octanol is lower than other tested higher alcohols which eliminates the flame cooling effect.

The cumulative net heat released shown in Fig. 9 changed with the fuel type due to the change of the fuel energy contents. The ratio between the cumulative net heat released per cycle and the energy available in the fuel delivered to the combustion chamber represent the efficiency of the combustion process. The combustion efficiency will give indication of how well the energy available in the fuel will be converted to heat release during the combustion process. Fig. 10 shows the combustion efficiency for different tested higher alcohol/diesel blends under various engine loads 25%–75% at rated speed of



(a)



(b)

Fig. 8. Ignition delay, combustion duration, SOI, CA05 (SOC), CA50 and EOC analysis for different tested fuel blends under various engine test conditions.

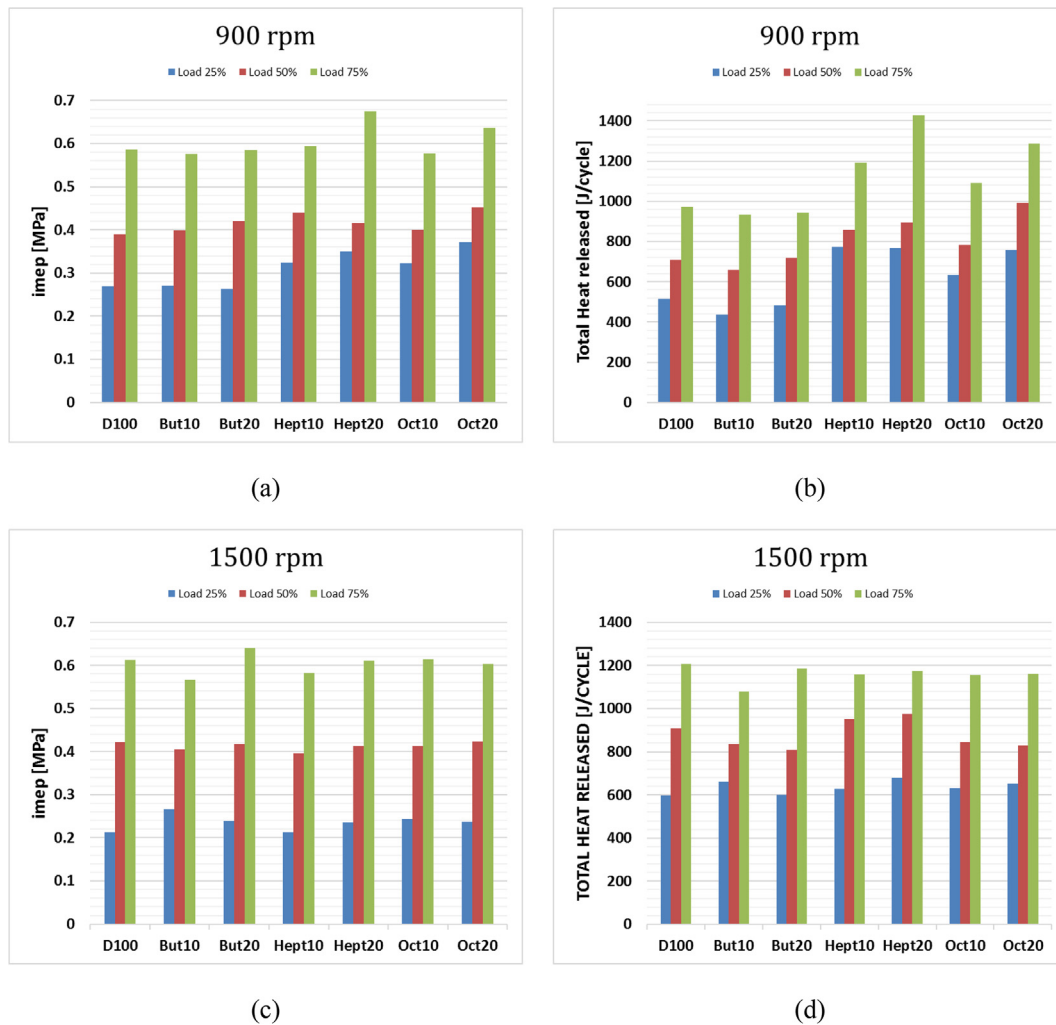


Fig. 9. Total heat released and imep for different tested higher alcohol/diesel blends under various engine loads 25–75% at 900 and 1500 rpm.

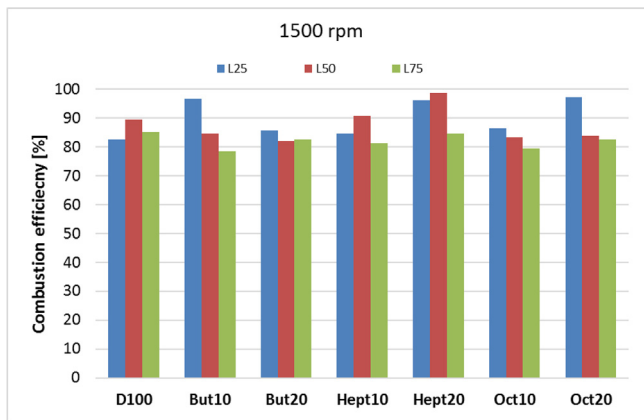


Fig. 10. Combustion efficiency for different tested higher alcohol/diesel blends under various engine loads 25–75% at 1500 rpm.

1500 rpm. The combustion efficiency of D100 is 82.5%, 89.37%, and 85% at 25%, 50%, and 75% engine loads, respectively. The combustion efficiency is slightly low, and the heat losses ranged from 10% to 17% which are quite high. This returns to that the tested engine is a single cylinder naturally aspirated conventional diesel engine with very low injection pressure, poor fuel atomization, and high heat losses. At 25% of engine load, all the tested higher alcohols/diesel blends show a higher combustion efficiency compared to D100. For example, But10,

Hept20 and Oct20 have a combustion efficiency of 96.6%, 96% and 97%, respectively. This is due to the enhancement of the combustion process by the oxygen contents in the fuel although the poor fuel atomization at low load conditions. At 50% engine loads, the combustion efficiency for But10, But20, Oct10, and Oct20 are lower than D100 by 4%, 6%, 5%, and 5%, respectively. However, Hept10 and Hept20 have combustion efficiencies of 90.6% and 98.7% which is higher than that of D100 and other higher alcohols/diesel blends. At 75% load, all the tested higher alcohols have lower combustion efficiency compared to D100 except Hept10. This is attributed to the poor atomization, lower heating value, high enthalpy of evaporation which result in a lower total heat release as illustrated in Fig. 9(d).

The comparable received results compared with those cited in the literature especially for butanol blends confirm the accuracy of the current setup and confidence in the procedures used to perform combustion analysis [30–35]. This is an important step to increase the reliability of results represented for other higher alcohols, i.e. heptanol and octanol, for which there were limited data in the literature.

#### 4.2. Engine performance

Diesel engine performance can be evaluated in terms of engine brake specific fuel consumption (bsfc), brakes specific energy consumption (BSEC), brake mean effective pressure (bmep), brake thermal efficiency (BTE), and exhaust gas temperature (EGT). Fig. 11 presents the change in fuel consumption and bmep at different engine brake

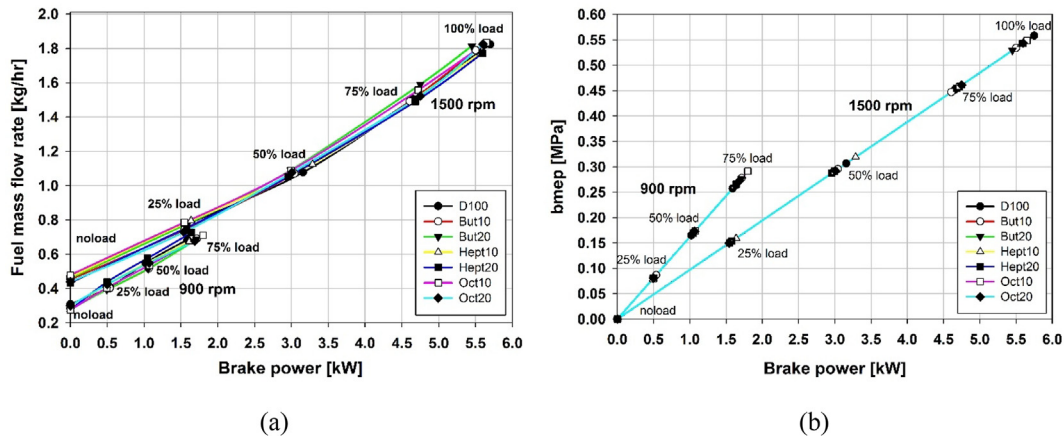


Fig. 11. (a) The variation of fuel mass flow rate with brake power (b) the variation of bmep with brake power for different tested higher alcohol/diesel blends under various engine loads 0–100% at 900 and 1500 rpm.

power at 900 and 1500 rpm for the tested higher alcohols/diesel blends. From Fig. 11(a) it is observed a linear relationship showing the increase in fuel flow rate with engine brake power no matter the tested fuel (with almost identical slopes as the engine frictions are constant for specific speed) or engine speed (with lower slopes or lower frictions for lower engine speeds). The increase in the higher alcohol percentage in the blend results in a higher fuel consumption. At 900 rpm, the fuel consumption for higher alcohol/diesel blends are comparable or less than that of diesel fuel. Fig. 11(b) shows a perfect linearity between the produced power and the bmep. The identical values of bmep for the various tested fuels at different engine speed show the good repeatability and accuracy in measurements to maintain test conditions of engine load and engine speed. As the bmep and brake power provide the same significant meaning, the bmep substitutes the brake power in the following sections to describe the engine useful power.

The variation of bsfc under different engine loads and speeds for tested higher alcohol/diesel blends are presented in Fig. 12. The bsfc measures the fuel conversion efficiency to provide the engine useful power and is commonly to measure the engine fuel economy. It is noticed that for low engine loads where the temperature of engine cylinder is somewhat low leading to poor fuel atomization and weak fuel/air mixture formation, larger fuel quantity is needed to generate the engine specified brake power. Then, the bsfc values decreases gradually as the engine load increases due to the received better fuel atomization and better fuel/air mixing. This leads to better effectiveness of energy release in the combustion process especially at full engine load where the values of bsfc are the lowest. As the engine speed is increased, the

turbulence intensity is increased and so better fuel atomization, evaporation, and mixing with air, and so better fuel conversion into useful power, see Fig. 12. At low engine power and low engine speed, the bsfc for heptanol and octanol blends with diesel fuel is higher than that of D100 by 15% because of their higher viscosity (5.75 mm<sup>2</sup>/s for heptanol and 7.59 mm<sup>2</sup>/s for octanol) compared to that of diesel (2.72 mm<sup>2</sup>/s). This higher viscosity is responsible for worsening the fuel atomization and inadequate the fuel/air mixing. Therefore, lower fuel fraction is burned in the premixed phase and more fuel would be burned in mixing control phase and so lower fuel conversion efficiency leading to an increase in the bsfc specially at lower engine loads. Even though octanol has a higher heating value than that of diesel which should results in a lower bsfc for Oct10 and Oct20 blends, the higher viscosity and poor fuel atomization overcomes the effect of the high heating value. Furthermore, the slightly higher enthalpy of evaporation for heptanol could be another reason for achieving the highest bsfc than that for D100. For But10 and But20 blends, the bsfc is slightly lower than or equal that for D100 as a result of its lower viscosity compared to that of diesel as shown in Table 5. For 50% and 75% of engine loads at 900 rpm, the bsfc for all the tested alcohols is lower than that of D100 by 9.3% owing to the enhanced fuel atomization and improved mixing between fuel and air except for Hept20 which is slightly higher than D100 by 2% (within the error limits) may be owing to its higher heating value.

At 1500 rpm, the effect of fuel calorific value and enthalpy of evaporation become more dominant due to the higher operation speed and load. At 25% load and bmep of 0.15 MPa, as indicated in Fig. 12(b), the

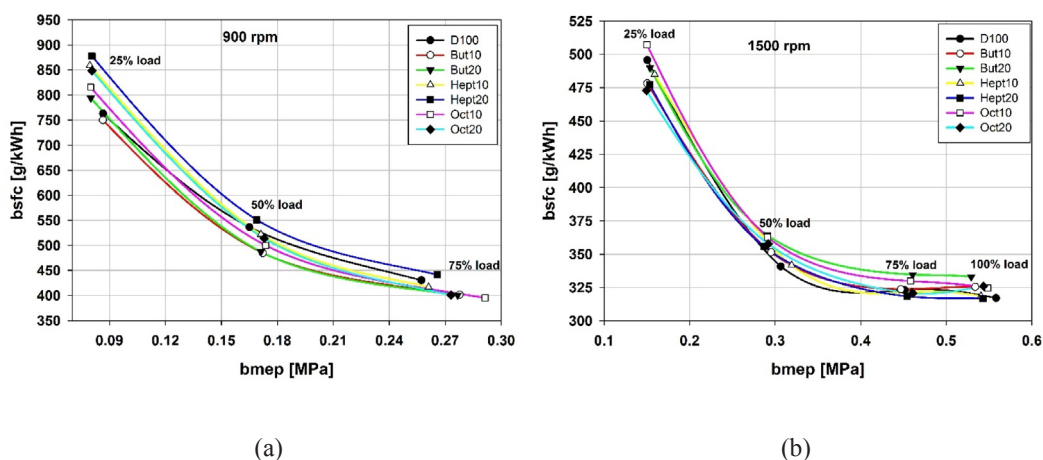


Fig. 12. The change of bsfc for various tested higher alcohol/diesel blends under different engine loads 25–100% and speeds (a) 900 rpm and (b) 1500 rpm.

bsfc for all higher alcohol/diesel blends is lower than that of D100. Oct20 achieved the lowest bsfc which is lower than that of D100 by 5%. This result was expected due to the high calorific value of octanol compared to that of D100. At 50% and 75% engine loads, the bsfc increases for all tested blends to become higher than that of D100. But20 has the highest bsfc at 50% and 75% engine loads as butanol has high enthalpy of evaporation that increases the heat absorbed from in-cylinder during fuel vaporization and so reduces fuel conversion efficiency. At 1500 rpm/100% load, D100 and Hept20 show the lowest bsfc and But10 and But20 achieved the highest bsfc.

The bsfc is not only affected by the fuel calorific value, however the blend viscosity, enthalpy of evaporation, fuel/air mixing, and fuel reaction mechanism are other main parameters. In conclusion, at low operation speed and load, the bsfc for higher alcohol/diesel blends is higher than that for D100 with maximum of 15%. However, at higher operation speed and loads the bsfc for all tested fuel blends are comparable and change within  $\pm 5\%$  of the corresponding values of D100.

The brake specific energy consumption (BSEC) is defined as the ratio of the power delivered by the engine (brake power) and the energy supplied in the fuel. The BSEC also describe the energy conversion efficiency to a useful output power in MJ/kWh and it has the same trend of bsfc. The tested higher alcohol/diesel blends have different energy contents which reflects the different values of fuel consumption and the delivered power. The change of BSEC for various tested higher alcohol/diesel blends under different engine loads 25%–75% and speeds is shown in Fig. 13. At 900 rpm, Octanol/diesel blends and heptanol/diesel blends shows higher BSEC compared to D100 while butanol/diesel blends have lower BSEC compared to D100 at all tested loads. At 1500 rated speed, Oct10 has higher BSEC compared to D100 at all tested loads while the BSEC for butanol/diesel and heptanol/diesel are lower than D100.

The variation of BTE with engine load at speeds of 1500 rpm and 900 rpm is shown in Fig. 14. From Fig. 14 (a) it can be noticed that, the addition of butanol increases the BTE by 15.4% compared to that for D100 and other fuel blends where But20 provides higher BTE than But10 by 4.5%. This behavior could be owing to the highest oxygen contents in butanol among other tested fuels, so the increase of butanol fraction in the fuel blend means an increase in the oxygen contents that improves the combustion process and increases the fuel conversion efficiency. Additionally, butanol has lower molecular weight compared to all tested fuels which consumes less energy during fuel degradation and so improved the fuel conversion efficiency. The BTE for heptanol/diesel and octanol/diesel blends is similar to that of D100 for most conditions. The BTE of Hept20 at 75% load and 900 rpm is lower than that of D100 by 7.7%, while at load of 25%, the BTE of Oct20 is lower than that of D100 by 17.4%.

Even higher alcohols have lower heating value than that of D100, there is no a significant change in the BTE of butanol/diesel, heptanol/diesel, and octanol/diesel from that of D100 at 1500 rpm; see Fig. 14(b). However, the high oxygen content achieves good combustion efficiency that improves the BTE. On the other side, oct/diesel blends have slightly lower BTE than that of D100. This reflects the worsening in fuel burning process and so the reduction in fuel conversion efficiency that results from the poor fuel atomization of the high viscous alcohols. On the opposite side, the longer chain alcohols such as octanol have lower oxygen contents compared to that of butanol, but it would enhance the combustion process during the premixed phase and results in a high combustion efficiency. At 1500 rpm/100% load, hept20 shows the highest BTE compared to other tested fuel.

The EGT variation is shown in Fig. 15 for various tested higher alcohol/diesel blends under engine tested operating conditions. At engine speed of 900 rpm, the exhaust gases temperature for all higher alcohol/diesel blends is lower than that of D100 owing to the influence of alcohols high enthalpy of evaporation. The lowest EGT is for But10 and But20 which have the highest enthalpy of evaporation. At engine speed of 1500 rpm, the EGT is comparable for all tested blends at engine part load, but at medium and high engine loads of 50%, 75% and 100%, the EGT for higher alcohol/diesel blends is lower than D100 owing to the effect of higher viscosity and the higher latent heat of vaporization.

#### 4.3. Exhaust emissions

The exhaust emissions of the diesel engine were evaluated by the measurement of  $\text{NO}_x$ , Opacity, HC, CO,  $\text{O}_2$  and  $\text{CO}_2$ . Fig. 16 presents  $\text{NO}_x$  variations for different tested higher alcohol/diesel blends under engine tested conditions. Formation of  $\text{NO}_x$  emissions in CI engines is mostly according to thermal  $\text{NO}_x$  mechanism (Zeldovich mechanism) at the high flame temperatures particularly at high operation loads and partly due to prompt  $\text{NO}_x$  formation mechanism during the droplet burning. Fig. 16(a) shows the variation of  $\text{NO}_x$  at 900 rpm and indicates that the highest  $\text{NO}_x$  concentration is for D100 at all tested engine loads. Oct10 and Oct20 have relatively high  $\text{NO}_x$  concentration at load 50% since octanol has a high calorific value and low enthalpy of evaporation compared to heptanol and butanol. Also, at load 25%/900 rpm, Hept10 and Oct20 have high  $\text{NO}_x$  emissions compared to other higher alcohol/diesel blends due to their high average gas temperature as illustrated in Fig. 6(e). For 1500 rpm, D100 shows the highest  $\text{NO}_x$  concentration compared to other higher alcohols/diesel blends.

Fig. 17 shows the soot concentration in exhaust gases for the different engine tested conditions. The smoke opacity provides important information about soot emitted in the exhaust gaseous as its main

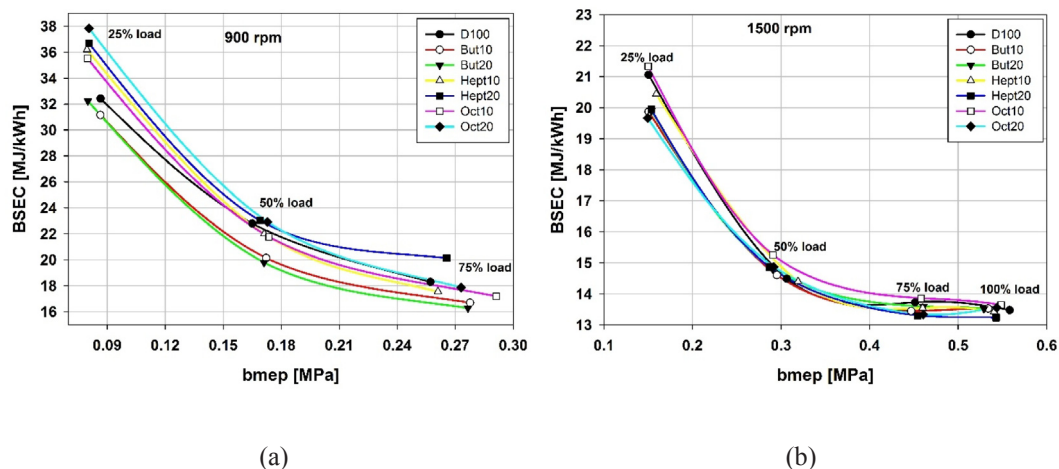


Fig. 13. The change of BSEC for various tested higher alcohol/diesel blends under different engine loads 25–100% and speeds (a) 900 rpm and (b) 1500 rpm.

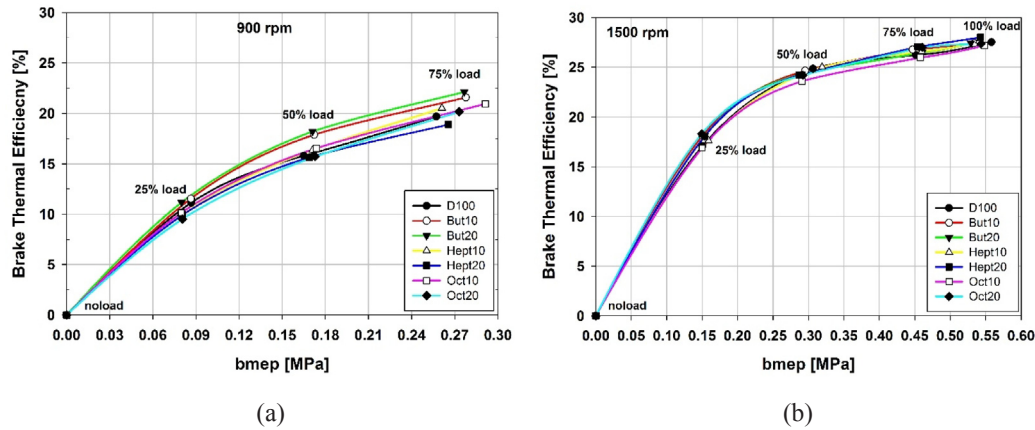


Fig. 14. The brake thermal efficiency for various tested higher alcohol/diesel blends under different engine loads 0–100% and speeds (a) 900 rpm and (b) 1500 rpm.

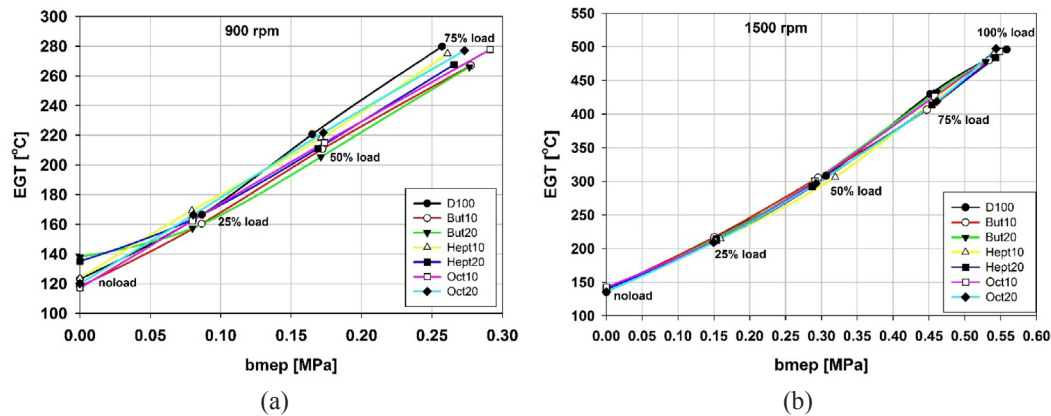


Fig. 15. The exhaust gases temperature for different tested higher alcohol/diesel blends under various engine loads 0–100% and speeds (a) 900 rpm and (b) 1500 rpm.

compound is carbon. The OH group exists in alcohols molecular structure promotes soot oxidation and results in less soot and smoke emissions. The additional generated OH radicals during the combustion chemical reaction enhance the combustion process and accelerate the soot oxidation in flame zone [60]. However, smoke opacity is also affected by other several parameters including viscosity, cetane number and fuel carbon contents. Lower cetane number leads to longer delay in fuel ignition which means lower time allowed for the formed soot to be oxidized and so higher smoke opacity [61,62]. The high viscosities of the fuel leads to poor fuel atomization resulting in the formation of

larger fuel droplet sizes and so worsening the fuel and air mixing and so higher opacity level is emitted. Also, the increase of the carbon contents in fuel structure, like octanol and heptanol, will increase the soot emissions and opacity levels. On the other hand, the existing of oxygen in fuel structure has a remarkable influence on accelerating the soot oxidation. Generally, soot oxidation is improved as the reaction temperature is high (affects by heating value), oxygen is available (and so quality of reactant mixing), time is enough for this reaction. The net effect of these complex competitive parameters affects the net emitted soot.

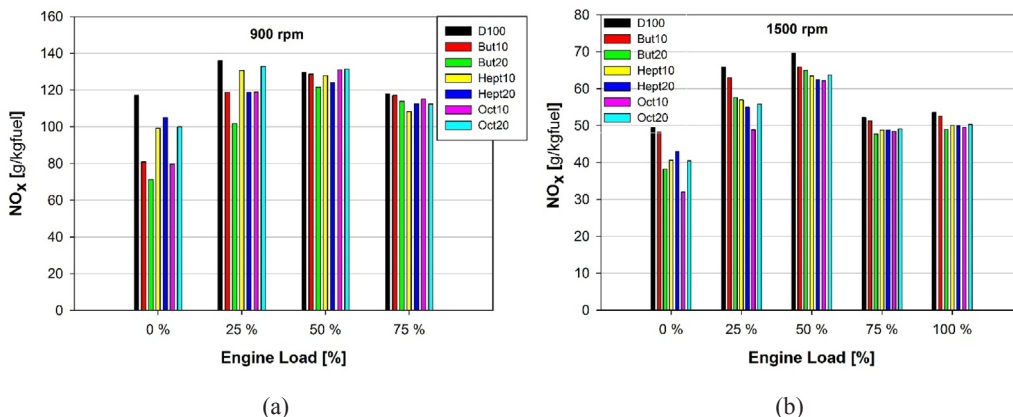


Fig. 16. NO<sub>x</sub> emissions for different tested higher alcohol/diesel blends under various engine loads 0–100% and speeds (a) 900 rpm and (b) 1500 rpm.

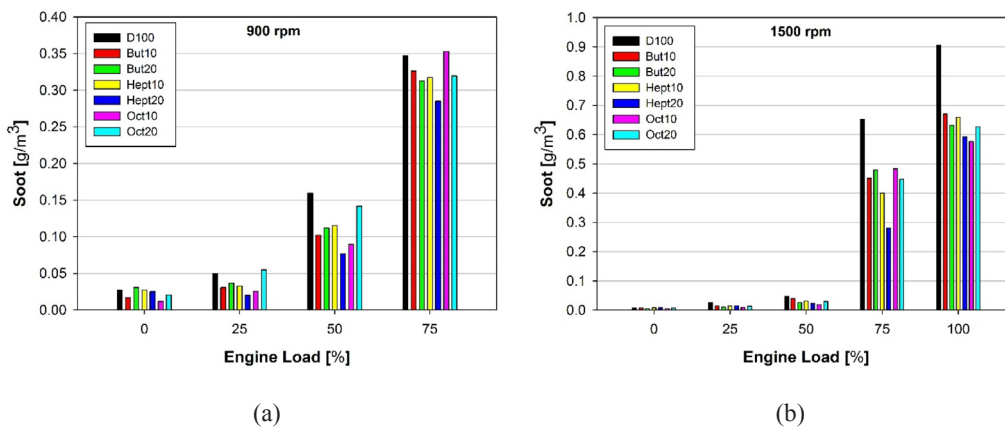


Fig. 17. Soot emissions variation for various tested higher alcohol/diesel blends under different engine loads 0–100% and speeds (a) 900 rpm and (b) 1500 rpm.

At engine speed of 900 rpm, D100 provides the highest opacity levels compared to other tested blends no matter engine load. Also, octanol/diesel blends show high soot concentration like that of D100 mainly due to its high viscosity and high carbon contents. Other tested blends of heptanol and butanol have lower soot concentration than that of D100 and Oct20 due to the higher oxygen contents, and lower carbon atoms. The higher operation speed of 1500 rpm where inside turbulence motion is improved leads to the same trend with opacity level for all tested higher alcohols blends lower than that for D100 at all examined loads. Hept20 has the lowest opacity level with a reduction in the opacity level by 34.9% and 35.5% at 1500 rpm and 900 rpm, respectively, compared to the corresponding value of D100.

The unburned hydrocarbon (HC) emissions for higher alcohol/diesel blends at different tested condition are presented in Fig. 18. At speed of 900 rpm, the HC emission for But10, But20, Oct10 and Oct20 are lower than those for D100 as shown in Fig. 18(a). This is attributed to the fuel oxygen content that improves the combustion efficiency and achieved better BTE. Also, the longer combustion duration that gives enough time for more complete fuel conversion. However, heptanol has higher HC emission compared to other tested fuels due to its lower BTE shown in Fig. 14 and due to the negative effect of higher viscosity and higher number of carbon atoms.

On the other hand, at 1500 rpm, Fig. 18(b) shows that the HC emission is increased for all examined higher alcohol/diesel blends compared to D100. This is attributed to the fact that prolonged ignition delay, short burning duration and the retarded combustion phasing towards later in the expansion stroke will lead to higher HC, less time at high temperatures to give complete conversion. Additionally, the enthalpy of evaporation for all the tested alcohols is higher than that for D100 leading to combustion quenching and reduced flame temperature

and consequently higher HC. Also, Fig. 18(b) shows that octanol and heptanol and their blends with diesel leads to an increase in HC emissions compared to butanol/diesel blends.

Fig. 19 indicates CO emissions variations for different tested higher alcohol/diesel blends under various engine loads 0–100% and speeds of 900 and 1500 rpm. CO emission shows no fixed trend for various operation load and speeds. At low engine load the CO emission is high owing to the worsening in fuel atomization, formation of large droplet size, and the inadequate fuel-air mixing. At medium engine load, the CO emission decreases and then increases again for high loads due to the injection of a large fuel quantity. Generally, the oxygen content of alcohol fuel should enhance CO oxidation and reduces CO emission. However, the undesirable effect of other parameters such as high enthalpy of evaporation, high viscosity, and retarded combustion phasing lead to an increase in CO emissions.

At both tested speeds, higher alcohol/diesel blends show a higher CO concentration as compared to D100 at 0% and 25% loads. However, at 50%, 75% and 100% loads the CO emission for higher alcohol/diesel blends is comparable or lower than that for D100.

### 5. Conclusions

In the present study, the effect of butanol/diesel, heptanol/diesel and octanol/diesel blends on diesel engine combustion, performance and emissions were investigated using direct-injection single-cylinder air-cooled CI engine. Experiments were carried out using alcohol/diesel blending ratios of 10% and 20% by volume in addition to the base diesel fuel. The stability and homogeneity of all tested blends were checked along four months and no phase separation was observed. The blends were examined under various engine loads ranged from no-load

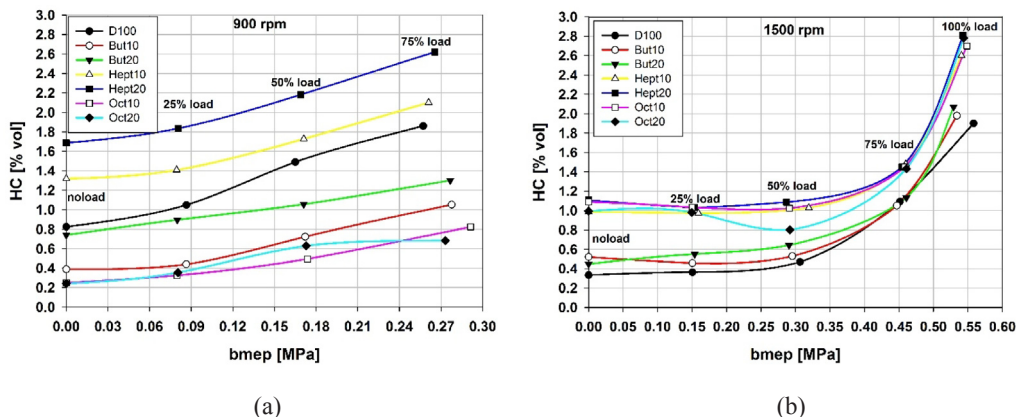


Fig. 18. Hydrocarbon emissions for different tested higher alcohol/diesel blends under various engine loads 0–100% and speeds (a) 900 rpm and (b) 1500 rpm.

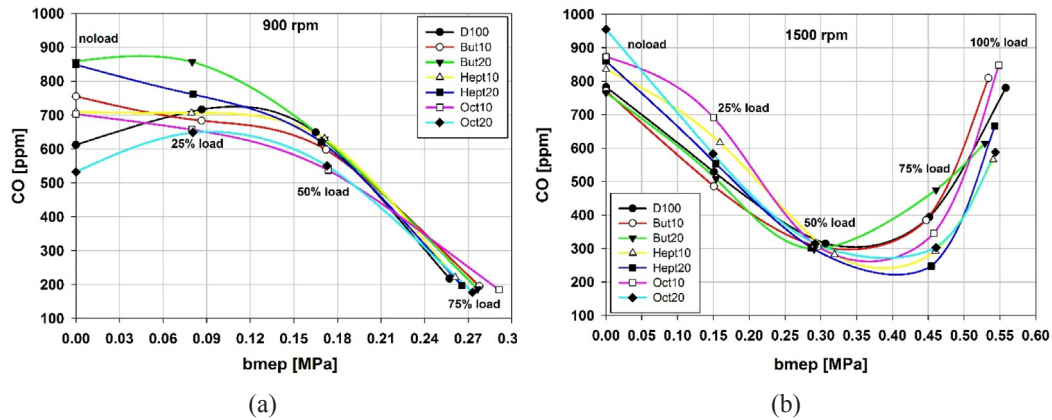


Fig. 19. CO emissions for different tested higher alcohol/diesel blends under various engine loads 0–100% and speeds (a) 900 rpm and (b) 1500 rpm.

to 100% of the engine full load with step of 25%. Also, the tested engine speeds were 900 rpm (idling speed) and 1500 rpm (engine rated speed) to cover a wide range of engine operating conditions. The main results of the present study can be summarized as following:

1. Heptanol/diesel blends and octanol/diesel blends showed longer ignition delay compared to that of pure diesel fuel and butanol/diesel blends. The combustion duration for heptanol and octanol was longer than that of butanol blends and that of pure diesel fuel.
2. Heptanol/diesel blends provided the highest heat released per cycle at all tested conditions compared to that of butanol and octanol fuel blends. Also, Heptanol/diesel blends shows the highest combustion efficiency among the tested fuels.
3. The bsfc for all the tested alcohols at 50% and 75% of engine load and 900 rpm was lower than that of D100 by 9.3% (except for Hept20 which is slightly higher than D100 by 2%) due to the lower heating value and the worsening in fuel atomization and poor mixing of fuel with air.
4. At high engine speed, Oct20 achieved the lowest bsfc which was lower than that of D100 by 5%. At 50% and 75% engine loads, the bsfc for all the tested blends exceeded that of D100. But20 provided the highest bsfc at 50% and 75% load due to its high enthalpy of evaporation that results in reducing the local flame temperature and lowering the fuel conversion efficiency.
5. The BTE for heptanol and octanol blends at 900 rpm is similar to that for D100 at most studied engine conditions. At high engine load of 75%, the BTE of Hept20 was lower than that of D100 by 7.7%. Also, at load 25% the BTE of Oct20 was lower than that of D100 by 17.4%. At higher engine speed of 1500, the BTE of heptanol and butanol were similar to that of diesel however octanol blends showed a decrease in BTE by 10%.
6. The lowest EGT was attained for But10 and But20 which have the highest latent heat of vaporization. At 1500 rpm, the EGT was comparable for all tested blends at no load and 25% load. For higher engine loads of 50% and 75%, the EGT for alcohol/diesel blends was lower than that for D100 owing to the high enthalpy of evaporation and the worsening fuel atomization due to their high viscosity.
7. Opacity levels decreased for alcohol/diesel blends compared to that for D100 at all tested conditions. Hept20 provided the lowest opacity level; reduced by 34.9% and 35.5% at 1500 rpm and 900 rpm, respectively compared to the corresponding values of D100.
8.  $\text{NO}_x$  emissions reduced for all tested blends at all the tested conditions. The Lowest  $\text{NO}_x$  concentration achieved by But20 and Oct10.
9. CO and HC emissions increased for all the tested alcohol/diesel blends.

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