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Effect of blending ratio and ınjection timing on combustion and emissions of a common-rail diesel engine fueled by ıso-propanol-butanol-ethanol (IBE) and conventional diesel

Geleneksel dizel ve IBE yakıtlı bir common-rail dizel motorun yanma ve emisyonları üzerinde karışım oranı ve püskürtme zamanının etkisi

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Effect of Blending Ratio and Injection Timing on Combustion and Emissions of a Common-Rail Diesel Engine Fueled by Iso-Propanol-Butanol-Ethanol (IBE) and Conventional Diesel

Highlights

- *Combustion of isopropanol-butanol-ethanol (IBE) were examined*
- *The effect of the injection timings and blending ratio was studied*
- *Using IBE delayed the combustion at low load compared with diesel fuel*
- *Smoke opacity reduced considerably by using IBE as additive to diesel fuel*
- *Increasing injection advance caused the NO_x emissions to be increased gradually up to 54%*

Graphical Abstract

The effect of using isopropanol-butanol-ethanol (IBE) in diesel fuel and the injection timings on combustion characteristics of a diesel engine were investigated

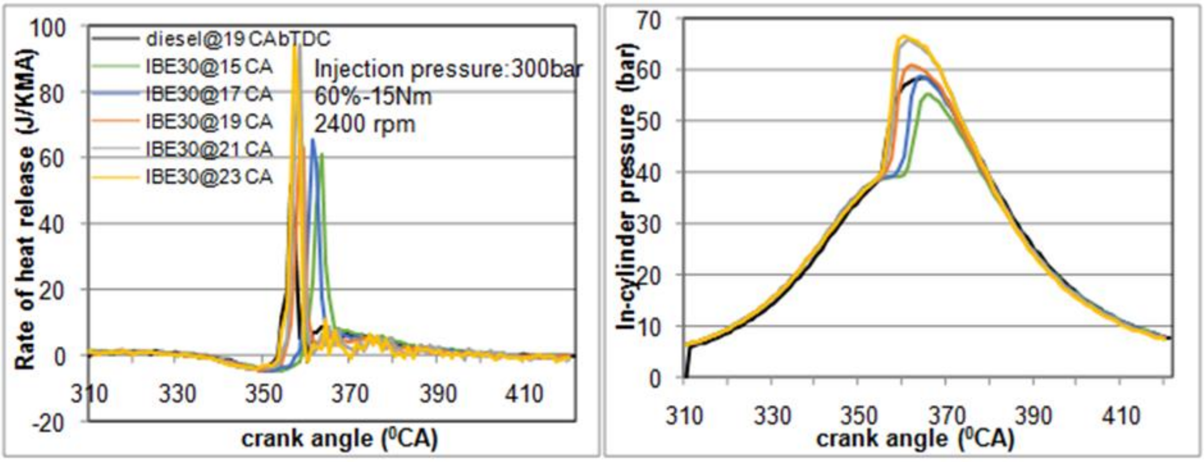

Figure. Graphical Abstract

Aim

The goal of the present study is to investigate the influence of the mixtures of IBE with diesel fuel and the injection timings on combustion and emissions of a CRDI Engine in order to specify the both optimum blending ratio and operating conditions.

Design & Methodology

IBE was added to diesel fuel at the rates of 0%, 10%, 20% and 30% by volume and afterwards tested on a single-cylinder CRDI engine under different loads, which correspond to 20%, 40% and 60% of maximum torque and at injection timings from 15°CA to 23°CA bTDC with 2°CA increments 60% load and constant speed of 2400 rpm

Originality

Despite an increasing popularity of IBE as a substitute for other alcohols, the studies about the use of IBE in diesel engine were rare. Therefore, in this study, a comprehensive investigation was performed to fill this research gap.

Findings

By using IBE blends, in-cylinder pressure and rate of heat release increased by 5.7% and 42% compared with diesel fuel. Unburnt HC emissions increased up to 35% at low load condition; while smoke opacity reduced considerably. NO_x emissions increased gradually up to 54% with increasing injection advance.

Conclusion

Using IBE with diesel fuel led to a significant decrease in smoke opacity, CO and unburned HC emission, which are beneficial for diesel engines in many respects such as meeting the stringent emission standards and avoiding after treatment systems.

Declaration of Ethical Standards

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Geleneksel Dizel ve IBE Yakıtlı Bir Common-Rail Dizel Motorun Yanma ve Emisyonları Üzerinde Karışım Oranı ve Püskürtme Zamanının Etkisi

Araştırma Makalesi / Research Article

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ÖZ

Bu çalışmada, İzopropanol-bütanol-etanol (IBE) ile petrol kökenli dizel yakıtının hacimsel olarak %10, %20 ve %30 oranlarında karışımlarının kullanıldığı tek silindirli common-rail dizel motorunun yanma ve emisyon karakteristikleri maksimum torkun %20, %40 ve %60'ına karşılık gelen farklı yükler altında incelenmiştir. Püskürtme zamanlamasının etkisi de üst ölü noktadan önce (ÜÖNö) 15 °KA (Krank Mili Açısı) ve 23 °KA arasında 2 °KA'lık artışlarla %60 yükte ve 2400 rpm sabit hızda incelenmiştir. Deneysel sonuçlar, dizel yakıtı ile karşılaştırıldığında, IBE karışımlarının düşük yük koşullarında yanmayı geciktirdiğini ancak yükteki artışın bu gecikmeyi azalttığını göstermiştir. IBE kullanımında silindir gaz basıncında %5.7'ye kadar ve ısı salınım oranında ise %42'ye kadar bir artış gözlemlenmiştir. Emisyonlarla ilgili olarak, yük ve IBE oranı ile CO emisyonlarında önemli bir değişiklik gözlenmemesine rağmen; yanmamış HC'lar düşük yük koşullarında %35'e kadar artmıştır; ancak, %60 yükte azalmıştır. Duman opaklığında konvansiyonel dizel yakıtı ile karşılaştırıldığında IBE30 ile %64'e kadar bir azalma olurken, NO_x emisyonları %10 civarında artmıştır. Püskürtme avansındaki azalma, basınç ve ısı salınım oranında azalmaya neden olurken, NO_x emisyonları kademeli olarak artan püskürtme avansı ile %54'e kadar bir artış göstermiştir.

Anahtar Kelimeler: İzopropanol-bütanol-etanol (IBE); emisyon; püskürtme zamanı; common-rail.

Effect of Blending Ratio and Injection Timing on Combustion and Emissions of a Common-Rail Diesel Engine Fueled by Iso-Propanol-Butanol-Ethanol (IBE) and Conventional Diesel

ABSTRACT

In present study, combustion characteristics of mixtures of isopropanol-butanol-ethanol (IBE) with petroleum-based diesel fuel at the rates of 10%, 20% and 30% volumetrically in a single-cylinder CRDI engine were examined under different loads which correspond to 20%, 40% and 60% of maximum torque. The effect of the injection timings from 15 °CA to 23 °CA (bTDC) with 2 °CA increments was also studied at 60% load and constant speed of 2400 rpm. Experimental results showed that IBE blends with diesel fuel delayed the combustion at low load while increasing load reduced this delay when compared with diesel fuel. An increase in cylinder gas pressure up to 5.7% and rate of heat release up to 42% in using IBE was observed. Regarding emissions, despite no significant change in CO emission was observed with the load and IBE ratio; UHCs increased up to 35% at low load conditions; however, they reduced at 60% load. While smoke opacity reduced up to 64% with IBE30 when compared with diesel fuel, NO_x emissions increased about 10%. The decrease in injection advance caused a decrease in pressure and heat release rate, while NO_x emissions increased gradually up to 54% with increasing injection advance.

Keywords: Isopropanol-butanol-ethanol (IBE); emissions; injection timing; common-rail.

1. INTRODUCTION

Compression ignition (diesel) engines have been widely used in a number of fields such as transportation, agriculture, construction and electric production and despite their advantages such as high fuel conversion efficiency and durability; high amount of the nitrogen oxide (NO_x) and particulate matter (PM) emitted from their exhausts appear to be the problems still to be solved and improved. Due to the harmful effects of these emissions on environment and human health, a prohibition has come to the agenda that the diesel vehicles in some major cities of both European and other countries will no longer be used in 2025 and the following years, and in some countries it has been banned

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to be used for vehicles over 20 years old [1-3]. On the other hand, due to the decrease in oil reserve despite increase in energy need and environmental problems, renewable energy sources have become an important research subject as well as insufficient use of domestic energy resources increases the dependence on foreign energy [4]. The fact that the dependency on petroleum in the whole world is increasing leads to significant problems for not only the countries of the world including EU members but also for Turkey. Because of all these reasons, biofuels with their renewable characteristics and producibility in the country itself come forefront among alternative fuel researches against petroleum [5], and their use in internal combustion engines as an alternative to petroleum origin fuels has been widespread. For example, EU increased the share of renewable energy sources (great majority is expected to be biofuels) to 10% in transportation in 2020 [6]. Biodiesel, which is the one of most used and researched biofuel in diesel engines, can be produced from oilseed plants such as sun-flower, canola, soya, aspire, cotton and from waste oils and fats [7] and biodiesel exhibits similar properties to petroleum-based diesel fuels in terms of physical and chemical features [8]. The cetane number of biodiesel is generally higher than conventional diesel; and biodiesel contains molecular oxygen, which enhances the combustion producing lesser emissions than petroleum diesel [9]. Regarding bioalcohols, they are the other most important biofuels used in internal combustion engines, mostly in gasoline engines due to their high octane number [10]. However, their low viscosity and cetane number along with poor lubricity hinders their application to diesel engines as a fuel in addition to phase separation problem in diesel-alcohol mixtures, which is stated to be an unwanted case [11]. Among the alcohol fuels, bioethanol, as a cleaner and more renewable fuel, is more preferred as it is produced from renewable sources by means of fermentation of biomass and its field of use is larger than that of methanol [12]. Their use in diesel engines as fuel leads to considerable decrease in exhaust emissions such as CO, HC and soot emissions since the oxygen that is present in alcohol in high amounts improves the combustion in some fuel-rich regions inside the cylinder during combustion [13]. A comparison of these biofuels in terms of emission reduction shows that the bioethanol blends provides higher oxygen than biodiesel blends, which leads to the reduction of particulate emissions [14]. On the other hand, recently, bio-butanol, which is a more advantageous alcohol compared to ethanol and methanol, is recommended [15] and it is better than ethanol in terms of fuel characteristics and miscibility with diesel fuel [16]. Butanol with economic and sustainable potential to be able to make a blend with diesel fuel instead of ethanol has the characteristics of forming a blend in high rates with diesel fuel thanks to high carbon number, thermal value and phase stability [17]. It was reported that adding n-butanol into B20 fuel blend at the rates of 10% and 20% led to a great deal of decrease in soot emissions [18]. Kattela et al. used fuel mixtures obtained by adding 10%, 20% and 30% butanol in volumetric ratio to diesel fuel in a single-cylinder diesel engine at constant speed and nominal load. They stated that as the butanol ratio in the mixtures increased, CO, NO_x and smoke opacity reduced by 23%-46%, 9%-23% and 24%-79%, respectively, while HC emissions increased by 30%-75% [19]. When the literature is examined, few studies where propanol diesel blend is used in diesel engines can be seen; and in these studies, low CO, NO_x and smoke emissions are reported [20]. Furthermore, propanol is also reported to remain stable up to 45% in mixture with diesel [12]. Bio-n-butanol is simply obtained via fermentation of acetone-nbutanol-ethanol (ABE) or isopropanol-nbutanol-ethanol (IBE) [21]. As obtaining pure butanol by fermentation using the distillation method is an expensive method due to its high boiling point, using the ABE or IBE directly as a biofuel provides a wider commercial application of this fuel by eliminating the additional cost of distillation [22]. Therefore, as alternative fuels, ABE or IBE, the intermediate fermentation products, have been suggested to be used directly in diesel engines so as to attain clean combustion [23]. Despite the fact that studies conducted before have successfully displayed the application of ABE/diesel mixtures to diesel engines, the mixture of IBE seems to be more conspicuous than ABE, while acetone is potentially abrasive to some parts of engine especially made of rubber or plastic [24]. Therefore, it has been shown in studies that IBE products show superior properties than ABE products in terms of biofuel properties and that isopropanol is a more advantageous product than acetone [25]. For example, as stated in a study by Hu et al., the application of IBE directly to IC engines as fuel has attracted attention, and research has begun on the combustion and emissions of IBE-fueled engines [26]. On the other hand, a comparison of blends of IBE or n-butanol with diesel fuel showed that blended fuels containing IBE rather than n-butanol had lower soot emissions while higher NO_x emissions at the same blend ratio [27]. However, due to the insufficient number of studies on the use of IBE in diesel engines, the differences in emission results and since it has been tested at low rates in general, it is of great importance to further investigate the IBE as a fuel. In this context, in a previous work [28], the effect of biodiesel addition to a conventional diesel-IBE blend on diesel engine combustion was studied after investigating and analyzing the results obtained in present study. On the other hand, fuel injection timing, which has a great effect on engine performance, is known as an important factor for controlling exhaust emissions in diesel engines. Nevertheless, a high efficiency with low fuel consumption by maintaining both NO_x and PM emissions at low levels was reported with the optimizing the injection process when IBE-diesel blends were used as fuel in a single-cylinder common rail direct engine [29].

The significance of present study lies on a single cylinder CRDI engine using IBE-diesel blends (up to 30% v/v) following the varying load conditions at a constant engine speed of 2400 rpm and investigating the effect of fuel injection timing on combustion of an IBE-diesel fuel mixture. Therefore, the present study aimed to investigate the combustion and emissions of the engine that were fueled with IBE-diesel fuel mixtures at different injection timings based on the investigations below: in-cylinder pressure, heat release, knock density and the emissions of CO, UHC and NO_x, and also smoke opacity.

2. MATERIAL and METHOD

2.1. Test Fuels

In this work, iso-propanol (99.5%), nbutanol (99.5%) and ethyl alcohol (99.8%) obtained from a commercial company were used for the preparation of isopropanol-butanol-ethanol (IBE). The mixtures of IBE with diesel fuel were prepared by utilizing alcohols mentioned above with volumetric rates of 3:6:1 in order to simulate the mixtures obtained from the fermentation process of butanol. After IBE was obtained, the blends of commercial diesel fuel obtained from a local fuel station (also used as reference fuel) with IBE were prepared in volumetric proportions. These are IBE10 (10% IBE + 90% diesel), IBE20 (20% IBE + 80% diesel) and IBE30 (30% IBE + 70% diesel) fuel mixtures. The physical and chemical properties of the fuels are given in Table 1 [30]. Higher IBE ratios than 30% by volume, which is also studied in literature, were not considered as test fuel owing to low cetane and the high latent heat of IBE (as in fact the most of alcohol fuels) that might led to cold start and combustion inefficiency problems. The blended fuels were prepared just before their use as fuel in test engine. Also, fuel tank on test engine was continuously stirred by the effect of fuel recycling.

Table 1. The properties of the fuels

Property/Fuel	Diesel Fuel	Iso-propanol	Butanol	Ethanol	IBE
Chemical formula	$C_{10}H_{22}$	C_3H_7OH	C_4H_9OH	C_2H_5OH	-
CN	52.65	12	15.92	8	13.952
ON	-	112	87	100	95.8
Oxygen (%)	-	26.6	21.6	34.8	24.4
Density (kg/m^3)	820-860	786	813	795	803.1
LHV (MJ/kg)	42.7	30.4	33.1	26.8	31.7
Boiling point (°C)	282-338	84	118	78	-
Heat of evaporation (kJ/kg)	260	758	582	904	667
Stoichiometric ratio	14.3	10.4	11.2	9.0	10.7

2.2. Engine Test Set

Engine test set was available in Engine Test Laboratory of Fırat University which was established by the financial support of TUBITAK within project numbered 118M650 [31]. The test set included a test engine, exhaust gas emission and smoke measurement devices, digital thermometers, stopwatch, anemometer, Gensan GSA 271 S/4 dynamometer, combustion analysis system, scaled fuel tank, fuel control unit and load control unit. The experiments were performed on a four-stroke, single-cylinder KIPOR KDP40E model diesel engine. The properties of the test engine are shown in Table 2. The common rail injection (CRDI) system was also integrated to the engine for better control of fuel injection. In Figure 1, the test set is shown.

Table 2. Technical features of test engine

Number of Cylinder and Stroke	1 and 4 Stroke
Cylinder Diameter x Sroke	86 mm x 70 mm
Volume of Cylinder	406 cm^3
Compression Rate	18.1
Maximum Engine Power	7.4 kW@3000 rpm
Maximum Torque	27.5 Nm@2400 rpm
Fuel Injection System	Common-rail direct injection (CRDI)
Injection Pressure	300 bar

In the experimental studies, Gensan GSA 271 S/4 model 10 kW electric dynamometer and Zemic L6W brand load cell (± 0.02) were used to load the engine. The dynamometer load indicator was used to load the test engine at different levels by means of the dynamometer, to adjust and control the dynamometer voltage, and see how much load was applied to the engine with the Mervesan MS-305 DII Regulated Power Supply Tester. In order to adjust and control the test engine at the desired engine speed, injection pressure and injection advance, a fuel control unit was used. In the experiments, Bosch BEA 350 brand emission measuring device was used for the measurement of exhaust gas emission and smoke opacity. After the probe of the measuring device was placed in the exhaust pipe and the engine reached operating temperature, the measurements were

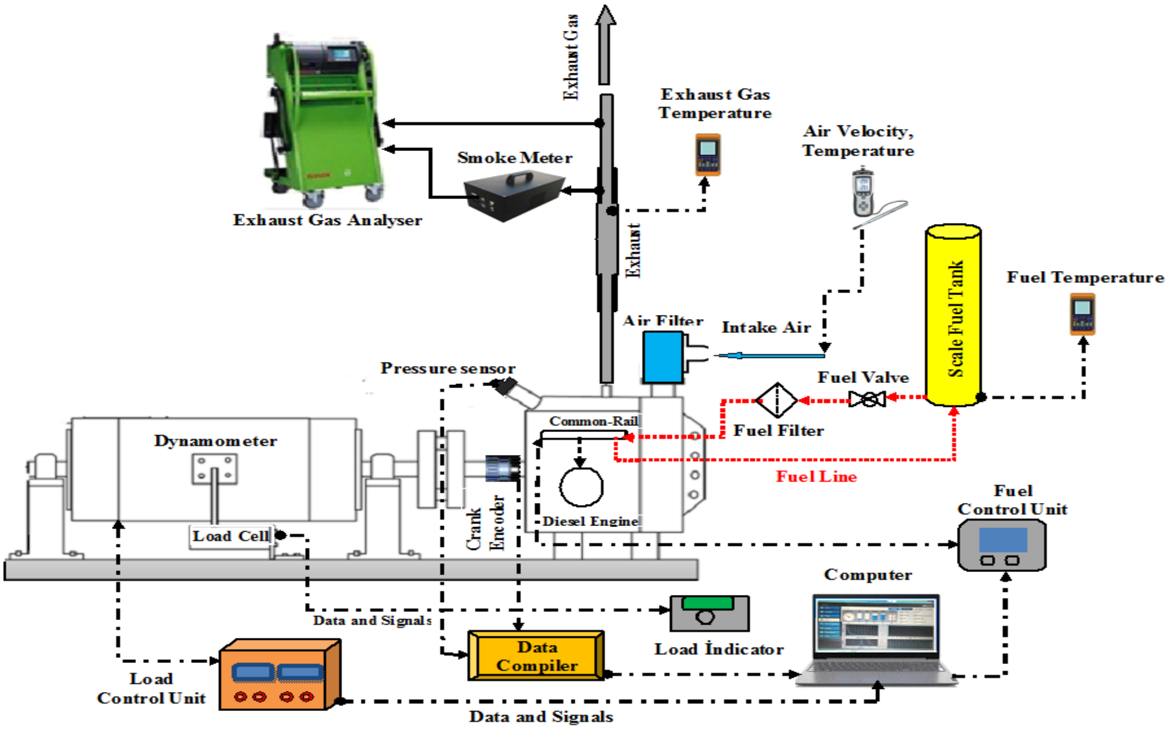

Figure 1. Schematic view of engine test set.

performed. Then it was possible to be printed out with a printer.

Thermocouple and digital indicators CEM DT-612 digital thermometer were used to measure the exhaust gas temperature of the test engine, fuel temperature and air temperature entering the intake manifold. The thermocouple measuring tip of the thermometer is placed in the exhaust outlet of the engine, and the thermometer body is fixed on the motor test setup in a certain position. 1000 ml scaled cylindrical fuel measuring tube used as fuel tank in experimental studies. In order to determine the fuel consumption time of the engine used in the experiment, the fuel in the scaled fuel tank was measured for a certain time with the help of a digital stopwatch Kenko KK-613D. The stopwatch can measure digitally with high split-second precision. For the measurements of the speed, temperature and flow rate of the intake air of the test engine, the CEM-DT 8880 Digital anemometer with a telescopic, elongating probe and capable of very sensitive measurement was used.

2.3. Combustion Analysis

In the experiments, FebriS was used for the analysis of combustion data. Crankshaft position and pressure sensors were placed on the test engine and measurements were made simultaneously with the engine operation. The information received from the encoder (crank angle sensor) that determines the position of the crank angle and the pressure sensor mounted in the cylinder are transferred to the FebriS software with a data recorder. Kübler brand sensor (encoder) with 360 signal frequency is mounted at the engine crankshaft output by a belt-pulley system. The sensor for in-cylinder pressure collects each 1o crank angle as an average of 100 cycles. The pressure sensor is mounted on the engine's glow plug. The data obtained from the crank and pressure sensors are collected on the data compiler card, analyzed with FebrisS combustion analysis software, transferred and recorded to the computer instantly and these studies are monitored instantly on the computer screen. Heat release rate and knock density were calculated by using following equations given in FebriS User Guide.

$$\frac{dQ}{d\theta}=\frac{\gamma}{\gamma\text{-}1}p\frac{dV}{d\theta}+\frac{1}{\gamma\text{-}1}V\frac{dp}{d\theta} \quad (1)$$

where $\frac{dQ}{d\theta}$ is net heat release rate, γ specific heat ratio, θ is the crank angle, P is the cylinder pressure, and V is the cylinder volume. Knock density was calculated with follow equation;

$$dp(\theta)=\frac{[86(p_{i\text{-}4}\text{-}p_{i+4})+142(p_{i+3}\text{-}p_{i\text{-}3})+193(p_{i+2}\text{-}p_{i\text{-}2})+126(p_{i+1}\text{-}p_{i\text{-}1})]}{1118d\theta} \quad (2)$$

where dp(θ) is the derivative crank angle, p (θ),p(θ-1) and p(θ-1) is the original signal, preceding value and succeeding value, respectively.

2.4. Method

Before starting the engine experiments, after checking the general physical conditions and electrical connections of the test set and connected devices, some engine maintenance such as changing the engine oil and cleaning the air filter were performed. The auxiliary apparatus required for the tests are placed in their places. Before starting the experiments, the engine was run with diesel fuel for 15 minutes in order to ensure that the engine becomes stable in ideal operating conditions. The test engine was started with diesel fuel and the first measurements was recorded, and then the experiments were started with IBE-diesel blended fuels. Tests were carried out for each test fuels at 2400 rpm and under 5 Nm, 10 Nm and 15 Nm load values, which correspond to 20%, 40% and 60% of the maximum torque. The engine speed of 2400 rpm is the max torque region specified by manufacturer at full-load; that is why it was preferred for the experiments. Engine loads were also selected to simulate the operating points such as low, mid and mid-high loads, which are typical operating points of the single-cylinder diesel research engines. During the tests, the fuel injection pressure was determined as 300 bar, the injection advance of 19 ºCA before TDC. When the measurements for a fuel type were completed, after each test, the engine was stopped for a certain period of time to cool down and rest before starting the tests with another fuel. In order for the fuel tested in the previous experiment to be completely finished, the engine continued to run until it stopped, and after each test the fuel tank was emptied, the fuel in the fuel pipes was emptied with the pump in the engine, and the fuel tank was filled with the new fuel to be tested and new experiments were started.

In the tests, exhaust gas components such as CO, CO_2, HC, NO_x, O_2 and smoke opacity and parameters such as intake air temperature, exhaust gas temperature, fuel temperature, intake air speed, engine torque, cylinder pressure values and crankshaft position for each fuel were measured. Using these values, the values such as air/fuel ratio, effective power, specific fuel consumption, heat release rate and knock density were calculated. Firstly, the test fuels were experimented under different loads corresponding to 20%, 40% and 60% of the maximum torque value at 2400 rpm. In the second stage, experiments were repeated for IBE30 with different injection timings such as 15 ºCA, 17 ºCA, 19 ºCA, 21 ºCA and 23 ºCA BTDC in order to study in detail the effect of IBE30 under 60% load without any structural changes.

Overall uncertainty was calculated as follow:

Overall uncertainty = (uncertainty of CO)^2+ (uncertainty of HC)^2+ (uncertainty of NO_x)^2 +(uncertainty of smoke)^2 +(uncertainty of BP)^2+(uncertainty of FC)^2}

Square root of {$(2.1)^2 + (4.7)^2 + (1.25)^2 + (3)^2 + + (0.678)^2 + (0.0695)^2$

=6.125% (3)

In addition to applications stated above in order to reduce the experimental errors, each tests are repeated three times and their arithmetic mean are used. Moreover, the error caused in various parameters has been evaluated as per the standard procedure. The experimental uncertainties for engine parameters and accuracy for devices are provided and given in Table 3.

Table 3. Accuracy of measuring instruments and experimental uncertainties.

Parameters	Unit	Measurement range	Accuracy	Uncertainty (%)
Speed	rpm	0–9500	±3	0.125
Torque	Nm	0–50	±0.1	0.67
Pressure	psi	0–3000	±1(%)	-
Inlet air	°C	0–800	±0.5	2
Exhaust gas	°C	0–800	±0.5	0.19
Fuel	mL/s	0–1000	±0.01	0.0695
CO	%	0–10.0	±0.001	2.1
NO_x	ppm	0–5000	±1	1.25
Smoke	%	0–99.99	±0.1	3

3. RESULTS AND DISCUSSIONS

In this section, the effect of the addition of IBE into diesel fuel on combustion of a CRDI engine fueled by diesel under varying operating conditions was explored. At each subsection, firstly, the results obtained in the CRDI engine operating on different IBE- diesel fuel blends (IBE10, IBE20 and IBE30) at 2400 rpm and loads of 20%, 40% and 60% of max torque in comparison to diesel fuel. After presenting and discussing the change of combustion characteristics with IBE blends, the effect of Injection Strategies of IBE30 on combustion was discussed under constant conditions (2400 rpm and 60% load).

3.1. Combustion Diagnostics

The change of cylinder gas pressure (CGP) with crank angle degree (CAD) and maximum pressure values for test fuels under varying loadings is given in Figure 2. In the figure, it is seen that with the use of IBE-diesel fuel mixtures, the start of combustion was delayed compared to diesel fuel use, but this delay shortened with the increase of the load, which was more obvious for IBE30 under all load conditions. Maximum CGP was measured by IBE30 under high load conditions after TDC. At 20% load, maximum CGP was measured as 59.01 bar at 3 ºCA after TDC with IBE20, while it was 57.40 bar at 1 ºCA after TDC with diesel fuel. Therefore, the increase in CGP was realized for IBE20 by 2.80%. Maximum CGP at 40% load is measured as 61.39 bar at 4 ºC after TDC by IBE30; it was 58.06 bar at 4 ºCA after TDC for diesel fuel. For IBE30, there was an increase of 5.74% in CGP compared to diesel fuel. At 60% load, maximum CGP

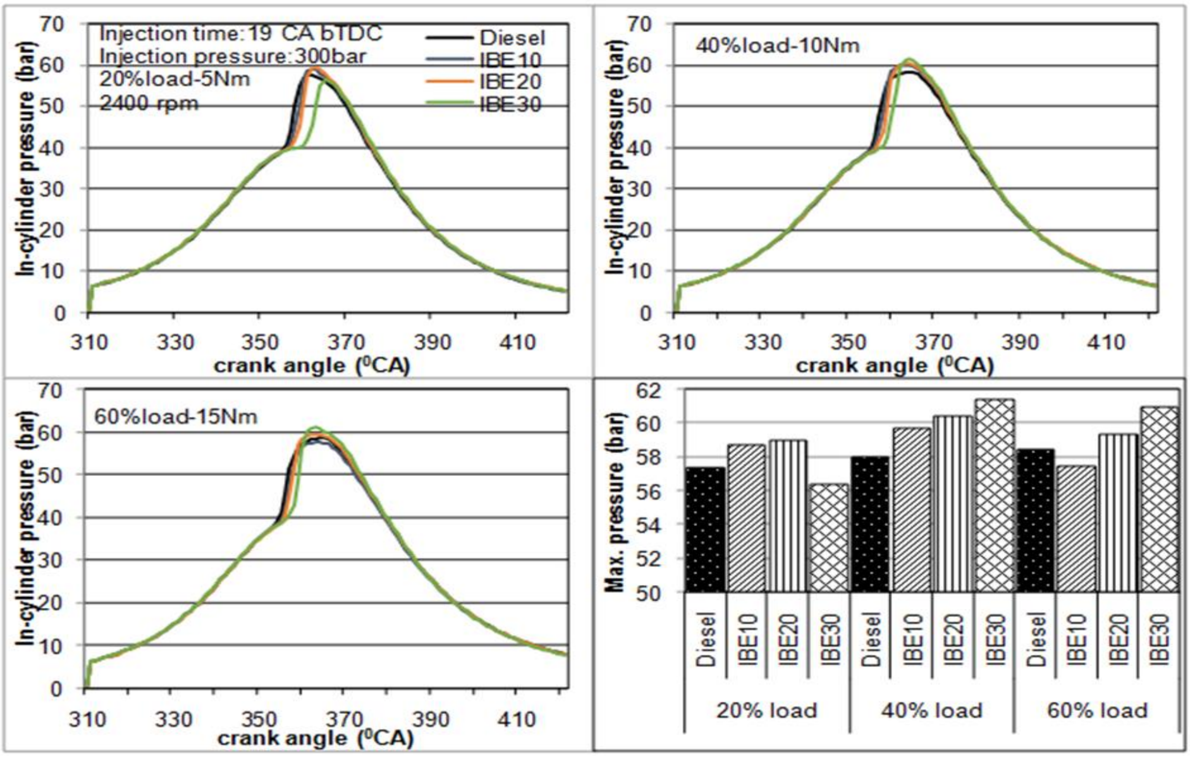

Figure 2. The change in cylinder gas pressure (CGP) with CA for fuels tested under different load

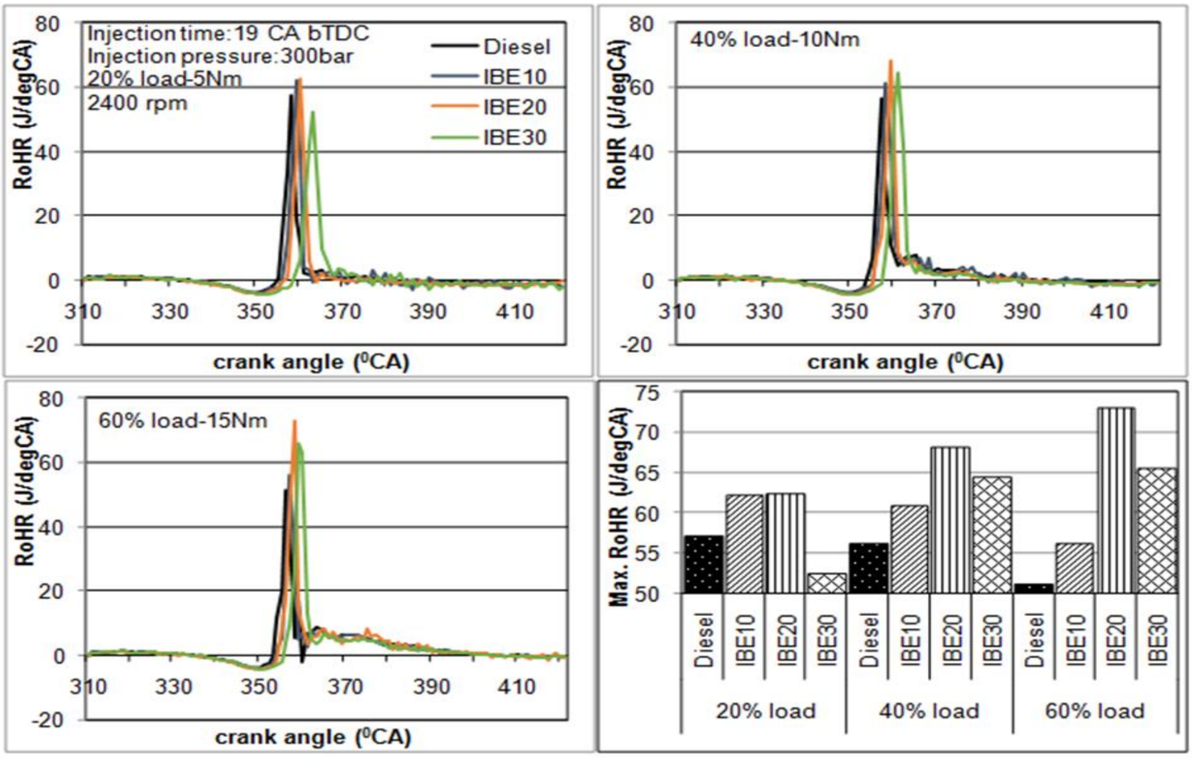

Figure 3. The rate of heat release (RoHR) and maximum RoHR for fuels tested under different loads**.**

occurred with IBE30 as 60.98 bar at 3 ºCA after TDC while it was recorded as 58.50 bar at 4 ºCA after TDC with diesel fuel. Increased CGP by using IBE-diesel fuel blends may be due to low CN and high heat of vaporization for fuel blends, as seen in Fig. 2, compared to diesel fuel, which led to longer ID period and therefore enhanced the premixed combustion [23]. This was also reported in the study of Nour et al. who studied the effect of adding alcohol to diesel fuel on the combustion [32].

The change in the rate of heat release (RoHR) with crank angle degree (CAD) for test fuels under varying loadings is given in Figure 3. Looking at the RoHR graphs in Fig. 3, an increasing trend was realized in all tested fuels with increasing load. The reason is that the amount of fuel consumed increased with increasing load. In this case, the amount of heat released increased depending on the amount of fuel consumed and combusted. It is seen that RoHR is higher for IBE20 than for other fuels at all load stages. Maximum heat release was 8.96% greater for IBE20 than diesel fuel at 20% load. This ratio was up to 21.20% and 42.44%, respectively, under 40% and 60% load conditions. On the other hand, it was observed that RoHR was lower and moved away from the top dead center in the use of IBE30 mainly due to its low cetane number compared to other test fuels. As it can be understood from these values, increasing the injection advance by using IBE30 causes an increase in CGP and RoHR values thanks to increased premixed combustion duration with help of the fuel properties of IBE such as low CN and high heat of evaporation, which causes the delayed combustion process [33]. In Table 4, the increase in ignition delay period is clearly seen with the increase of IBE content in the fuel. At 60% load, ignition delay period was calculated as 13.3 ^{o}CA with diesel fuel, while it was 16.6 ^{o}CA with IBE30. Conversely, combustion duration was found to be decreased with the addition of IBE into diesel fuel, and it was calculated as 28.3 ^{o}CA with IBE30 at 60% load. Under this condition, combustion duration was 34.97 ^{o}CA for diesel fuel. The change of ignition delay and combustion duration with injection timing for IBE30 is also given in Table 5.

Table 4 Ignition delay and combustion duration data for test fuels

	Ignition Delay (^{o}CA)			Combustion Duration (^{o}CA)		
Load	20%	40%	60%	20%	40%	60%
Diesel	14.1	13.8	13.3	14.86	24.13	34.97
IBE10	15.2	14.7	14.1	11.77	21.77	32.88
IBE20	16.1	15.4	14.8	10.37	20.55	30.1
IBE30	18.7	17.1	16.65	9.27	18.08	28.3

Table 5 Ignition delay and combustion duration data for IBE30 at different injection timings

Injection Timing (^{o}CA bTDC)	Ignition Delay (^{o}CA)	Combustion Duration (^{o}CA)
15	16.25	28.9
17	16.45	28.6
19	16.65	28.3
21	17	25.7
23	18.45	23

In Figure 4**,** the change of CGP and RoHR values obtained by using IBE30 under 60% load and 2400 rpm and different injection timings depending on the CAD is shown. In the figure, it is seen that the CGP and RoHR increase with the increase of the injection advance. However, when compared with the reference advance value of 19 ^{o}CA bTDC, CGP and RoHR decreased in case of 15 ^{o}CA bTDC and 17 ^{o}CA bTDC injection advances. On the other hand, in case that injection advance is adjusted as 21 ^{o}CA bTDC and 23 ^{o}CA bTDC an increase was observed. Therefore, the maximum pressure is measured as 66.65 bar at 23 ^{o}CA bTDC while lowest was measured in case of 15 ^{o}CA bTDC injection advance. In addition, the maximum heat release was measured as 94.61 J/degree at 2 ^{o}CA bTDC when injection advance was adjusted as 21 ^{o}CA bTDC. In case of 23 ^{o}CA bTDC injection timing, this value is calculated as 94.48 J/degree at 3 ^{o}CA bTDC. With diesel fuel, the maximum pressure and heat release at the reference advance of 19 ^{o}CA bTDC are realized as 58.49 bar at 4 ^{o}CA aTDC (after top dead center) and 51.29 J/degree at 5 ^{o}CA bTDC, respectively. In the use of IBE30, these values are 60.97 bar at 2 ^{o}CA aTDC and 65.67 J/degree at 2 ^{o}CA bTDC. In a study where the mixtures of ABE and diesel fuel were tested in the different injection timings, similar results were reported and explained by increased ID period where more premixed charge was formed and burned by advancing the injection timing with using the ABE20 as fuel [34]. The increase in premixed combustion and the fact that most of the heat released is around the TDC increases the combustion rate and causes an increase in amount of heat converted to work.

It is understood from Figure 5 that the maximum knock density values at 20%, 40% and 60% loads occurred at positions between 350 ^{o}CA and 370 ^{o}CA and did not change significantly with the increase in load. For all load stages, the knock density increased with the increase in IBE amount in the fuel, and the knock density started to increase closer to TDC when compared to diesel. For example, at 60% load, the maximum knock density was realized as 0.154 at 7 ^{o}CA before TDC and 0.152 at 6 ^{o}CA after TDC with IBE30. For diesel fuel, it was 0.107 at10 ^{o}CA before TDC and 0.082 at 3 ^{o}CA after TDC. Therefore, for IBE30, there was an increase in knock density by 43.93% before TDC and 85.37% after TDC compared to diesel fuel. This situation was also observed in other load stages. On the other hand, in case of maximum pressure, the high knock density shows that the pressure increase is effective in the formation of the knock density. For IBE30, at 60% load where the maximum pressure is higher than the other types of fuel, the knock density values are also higher for the same fuel. Similarly, the maximum pressure at 20% load conditions is the lowest in the use of IBE30 and the knock density is also low. As a result, it can be said that due to the properties such as low cetane number of IBE added to diesel fuel, the maximum pressure increased and consequently the knock density increased due to effects

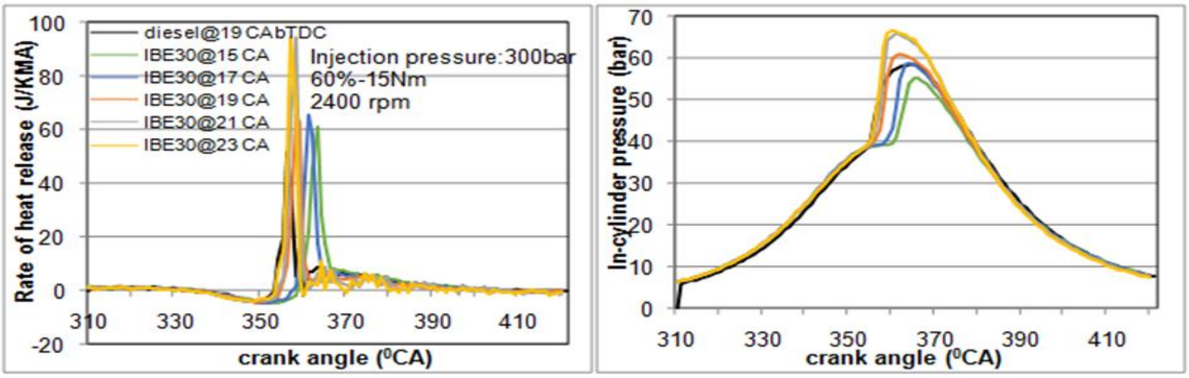

Figure 4. The variation of CGP and RoHR for diesel and IBE30 at different injection timings under 60% load.

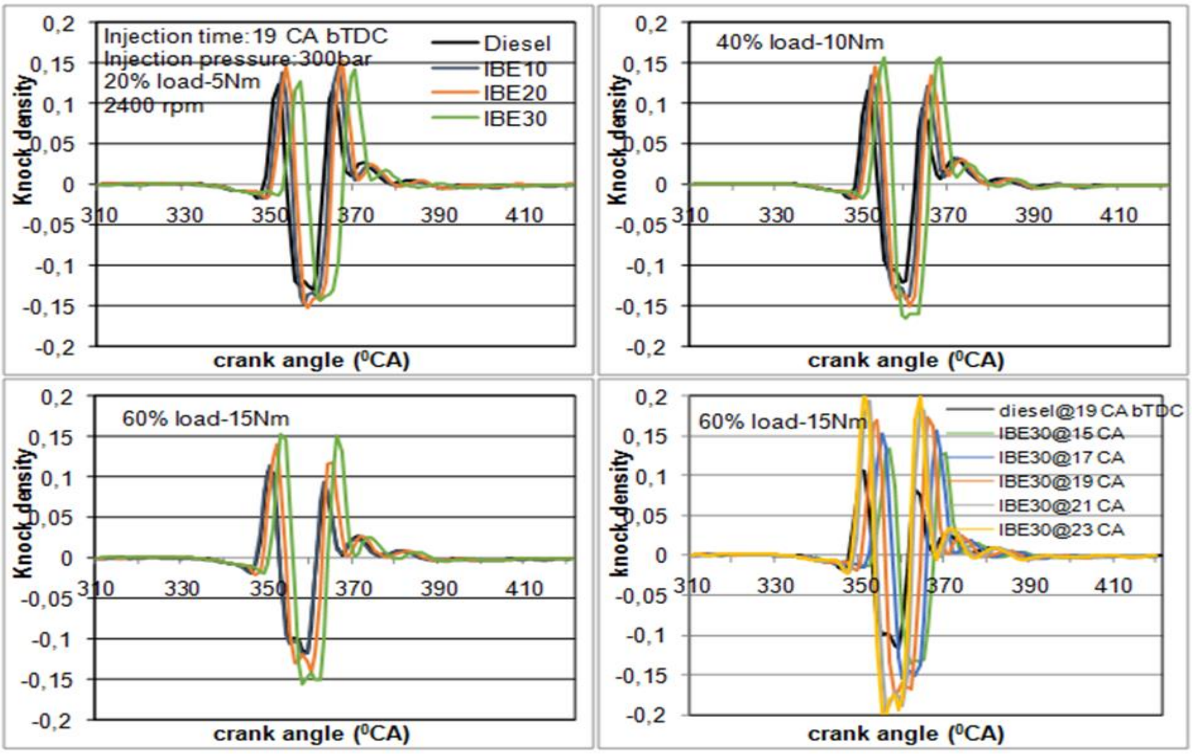

Figure 5. Knock density for fuels tested under different loadings and injection timings

such as prolongation of ignition delay. Similarly, Aydın who studied the knock density in a diesel generator fueled with biodiesel also reported that prolonged ID period might lead to increase of knock as more fuel is quickly burned in premixed phase [35]. In the same Figure, in case the use of IBE30 at 60% loads in different injection advances, the change of knock density with CAD is given. The knock density increases as the advance increases. The maximum knock density was realized with 23 ^{o}CA bTDC injection advance as 0,206 at 8 °CA before TDC and as 0.186 at 5 °CA after TDC. In case of 19 °CA bTDC injection advance, when compared with the values obtained with diesel, there was an increase with IBE30 as 55.14% before and 81.71% after TDC.

3.2. Exhaust Emissions

Figure 6 shows the variation of CO and unburned HC emissions with load and fuels tested. As shown in the figure, CO emissions were significantly lower at 20% and 40% load conditions compared to 60% load conditions. This may be due to the low fuel/air ratio in general under low load conditions. However, it is seen that there is a slight increase in CO emissions at low loads (20%) with IBE mixtures compared to diesel fuel. It is thought that the cooling effect of IBE blends due to their high latent heat of evaporation has an effect on this result at low load conditions where in-cylinder temperature is relatively low. At 60% load, lower CO emissions were obtained with IBE20 and IBE30 except that with IBE10 they were higher than diesel use. Average CO emissions were calculated as 7.41% higher for IBE10 while 51.85% and 55.55% lower for IBE20 and IBE30, respectively, compared to diesel fuel, and this reduction is thanks to

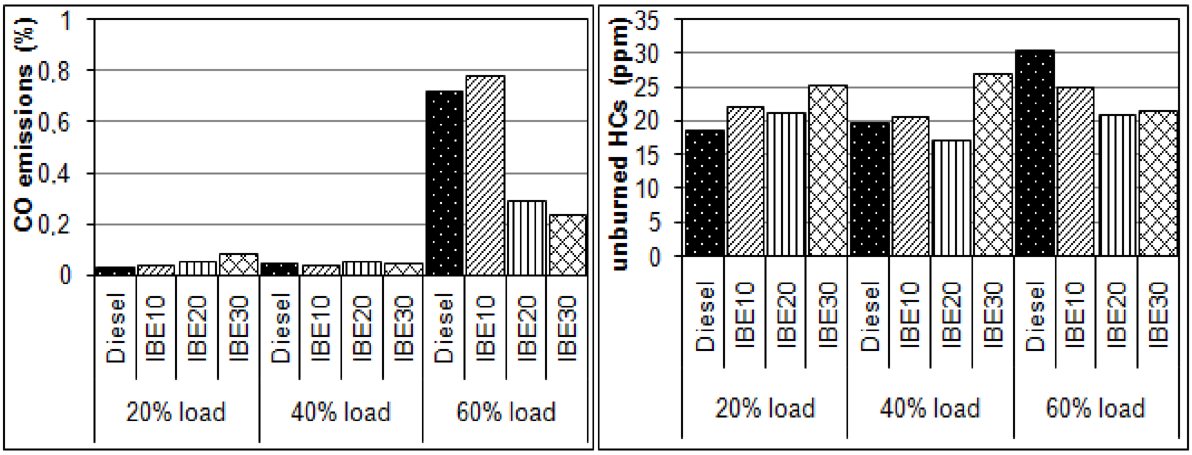

Figure 6. The change in CO and unburned HC emission with load for fuels tested.

the higher in-cylinder oxygen concentration, as reported in studies where oxygenated fuels are used [36]. Unburnt HC emissions also showed a different variation according to the load, as shown in Fig. 6. While slightly high HC formation was observed with IBE blends with diesel fuel under low load conditions, HC emissions were measured to be lower in high load conditions than diesel. For example, at 20% load, the highest HC emission was measured as 25.25 ppm with IBE30 while the lowest value was with diesel as 18.67 ppm. In the use of IBE10, IBE20 and IBE30, the unburned HC values at 20% load were 19.1%, 14.6% and 35.2% higher, respectively, than that of diesel fuel. This result may be due to the quenching effect leading to higher HC emissions under low loads with relatively low temperature as IBE blends have a high latent heat and low cetane number [36]. Thus, while the highest HC was obtained with IBE30 at 40% load when the load was increased to 60%, the highest value was measured with diesel fuel as 30.5 ppm. Although, in present study, a similar tendency for both emissions can be seen in Figures 6 and 8 when they are examined carefully, some minor variations may be evaluated within experimental error. Moreover, both CO and unburned HC are incomplete combustion products, however, in their formation mechanism, there are some differences such as crevices and quenching regions contributes to unburned HC formation while in-cylinder oxidation level is responsible for CO emissions. Also, unburned emissions are greatly affected by longer ignition delay. Considering the high oxygen and latent heat content of IBE, it can be said that it exhibits different combustion behavior for each emission under different operating conditions.

In Figure 7, the change of NO_x emissions and smoke opacity is shown. Although some decrease in NO_x emissions was observed with the increase of IBE ratio in fuel at low loads, it was measured higher for IBE blends at 40% and 60% load conditions. Figure 7 also shows that at 20% load, the highest NO_x emission value among the test fuels is measured as 500.50 ppm with IBE10 while the lowest is 395.75 ppm with IBE30. In this case, the NO_x with diesel fuel was measured as 458.67 ppm. When the load was increased to 40% and 60%, NO_x emission was measured as 650.25 ppm and 594 ppm with IBE30, respectively. In this case, NO_x values measured with diesel fuel were 552 ppm and 476.67 ppm, respectively. Average NO_x emissions for all loads were 3.07%, 3.69% and 10.26% higher for IBE10, IBE20 and IBE30, respectively, compared to diesel fuel. The fact that NO_x emissions of IBE blended fuels are lower at low loads may be due to the cooling effect that occurs due to the

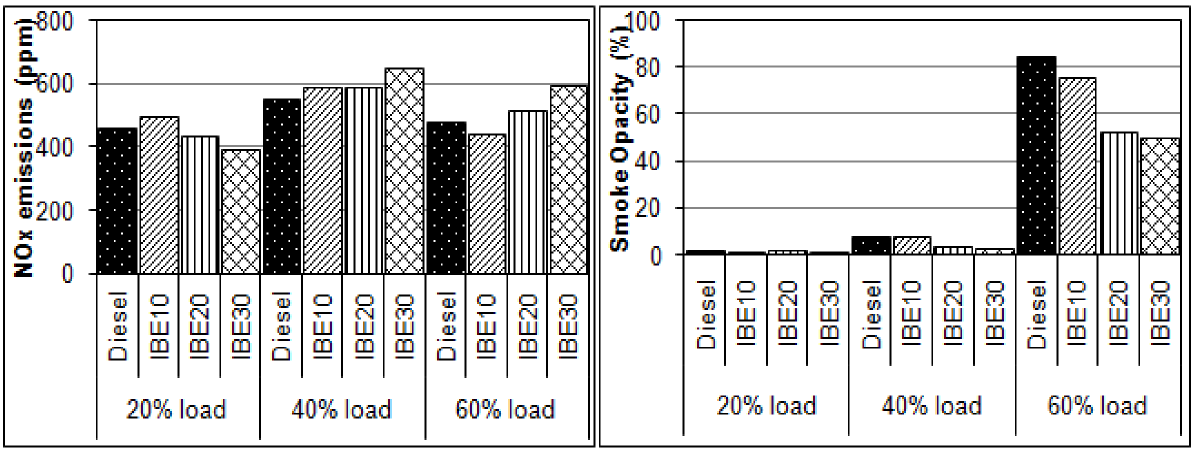

Figure 7. The change in NO_x and smoke opacity with load for fuels tested

high heat of evaporation and less fuel injected and lower energy content. On the other hand, the increase in NO_x emissions under high load conditions can also be attributed to the increase in combustion temperature together with the increase in ignition delay due to low cetane number and oxygen content of IBE, as found by other researchers [37]. It can be said that the oxygen content is more dominant than the latent heat in these high load conditions due to the increase in the combustion temperature with the increase in the amount of fuel injected to chamber at high loads. Similarly, in a study where IBE and diesel fuel blends were used with the addition of water, NO_x emissions were reported to be higher with blended fuels than diesel fuel mainly due to high combustion temperature from the oxygen content of fuels [38].

injection timing for IBE30 at 60% load in comparison to diesel at 19 ºCA before TDC. As seen in the Figure, CO emissions increased with the increase in injection timing and reached its maximum value at 21 ºCA before TDC. However, with further increase of the injection advance, there was a decrease in CO emissions. Similarly, the unburned HC emissions decreased from 21.60 ppm at 19ºCA before TDC to 16.50 ppm at 23ºCA before TDC by increasing the injection advance. When the injection is advanced the ID period prolongs and leads to higher combustion temperature that reduces the incomplete combustion. By increasing the injection advance, there is more time for better mixture of air and fuel which is provocative to reduce CO and HC emissions with the support of the oxygen content of IBE stated also in Ref.[34]. As for the smoke opacity, it gradually increased by increasing the injection advance from 15 ºCA bTDC

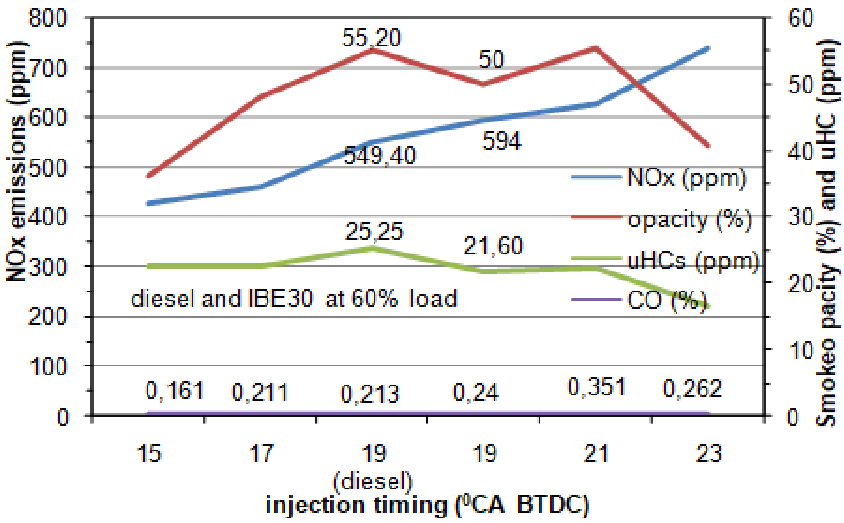

Figure 8. The change in emissions with injection timing.

In Figure 7, it is seen that highest smoke opacity value under 20% load conditions was realized with diesel fuel as 2.35% and the lowest value was measured as 1.50% with IBE30. However, for all test fuels, it is understood from Figure 7 that the smoke opacity increased with the increase of the load. The increase in the amount of fuel sent to the cylinder with the increase of engine load decreased the air/fuel ratio, in other words, it caused a rich mixture that increased the smoke amount due to the high temperature and pressure in the combustion chamber. Another important result is that the lowest smoke opacity at each load stage was obtained with IBE30, which has the highest IBE ratio among the tested fuels. With this fuel, there was a decrease of 64.33% at 40% load and 41.18% at 60% load compared to the use of diesel fuel. In this study, it can be said that by increasing IBE in the fuel mixtures leads to increase the oxygen concentration in the combustion chamber, thus reducing the smoke formation. Similarly, Abubakar et al. reported that oxygen present in blended fuels promoted the lesser opacity formation [38].

Figure 8 shows the change in emissions of CO, unburnt HC and NO_x and smoke opacity depending on the

to 21 ºCA bTDC with intervals of 2 ºCA. However, it started to decrease at 23 ºCA bTDC. When compared with the smoke opacity value obtained with diesel fuel under the same conditions and reference advance, the smoke opacity was lower for IBE30 (up to 34%) in all injection advance cases. It was observed that NO_x emissions increased with increasing the injection advance, but decreased in the case of decreasing the advance of injection. While the highest NO_x emission value was 737.50 ppm at 23 ºCA bTDC, the lowest value was measured as 427.20 ppm at 15 ºCA bTDC. NO_x value of diesel fuel was measured as 549.40 ppm at 19 ºCA bTDC, which is the reference injection advance, and compared with this value, NO_x decreased by 10.38% and 3.50%, respectively, by IBE30 at 15 ºCA bTDC and at 17 ºCA bTDC. At 19 ºCA, 21 ºCA and 23 ºCA bTDC, there was an increase in NO_x emissions by 15.26%, 31.20% and 54.72% when compared with 19 ºCA bTDC for diesel fuel. As seen, there is a gradual increase in NO_x emissions with increase of injection advance mainly due to enhanced premixed combustion. Increased NO_x emissions were also reported by other researchers who used the blends of ABE with diesel fuel [34]. They explained this trend by increase in-cylinder temperature when the injection timing was advanced owing to the premixed combustion. Finally, the increase of injection advance for IBE30 at 60% load decreased CO, unburned HC emissions and smoke opacity while increased NO_x emissions. This is mainly due to the enhanced premixed combustion; however, the diffusion part is weakened and oxygen is increased when IBE is added to diesel fuel, as reported in Ref. [29].

3.3. Brake-specific fuel consumption

As seen in Figure 9 (left side), brake-specific fuel consumption (bsfc) for all test fuels increased when the engine load increased. It is seen that there is a slight increase in bsfc in comparison to diesel fuel, especially for IBE mixtures at low loads. This increase was approximately 3.5% and 5.4% with IBE30 at 20% and 40% loads, respectively. At 60% load, lower bsfc was obtained with all blended fuels compared to diesel. In the use of alternative fuels with low energy content, the amount of fuel used should be increased to maintain the power output as diesel fuel. In low load conditions where combustion conditions are relatively poor, a slight increase in bsfc is expected due to the properties of IBE blends such as lower heating value, but it is thought that the properties such as oxygen content of the blended fuels become more dominant with the increase of the amount of fuel injected under high load conditions. On the other hand, improved bsfc was widely reported in the literature review [39] when alcohol based fuels are used mainly due to the high oxygen content and latent heat of the alcohols, which enhance the combustion rate and decrease in-cylinder temperature, which results in heat losses to be reduced.

The influence of the timing on bsfc for IBE30 at %60 load is also shown in the right side of Figure 9. The figure shows that the bsfc significantly reduced if the injection advance is increased or decreased. With IBE30 at the standard advance value of 19 ^{o}CA bTDC, bsfc is calculated as 310 g/kWh, this value for diesel fuel is 320.88 g/kWh. When the injection advance was increased to 21 ^{o}CA bTDC, the bsfc slightly increased. The influence of the timing on bsfc for IBE30 is supported by researchers who investigated the use of ethanol blends under varying injection timing conditions [40]. At 15 ^{o}CA and 23 ^{o}CA bTDC, the lowest bsfc was obtained that is more retarding or advancing the injection timing led to bsfc to be reduced when IBE30 was used. However, under retarding conditions, although NO_x emissions reduced, they measured as highest at 23 ^{o}CA bTDC in which the lowest bsfc was obtained. In fact, the highest NO_x is an expected result as the injection timing was advanced especially with the fuels having oxygen content, and this was also widely reported in the literature. However, the bsfc is not expected to be reduced as the timing advanced. The possible reasons are: the IBE containing high oxygen content with help of more available time to form air-fuel mix due to advanced injection had the opportunity to oxidize more fuel in the chamber. In these conditions, obtaining the lowest unburned HC emissions can also be shown as an indicator of better oxidation/combustion. Secondly, the fuel measurement at this condition may be evaluated within experimental error.

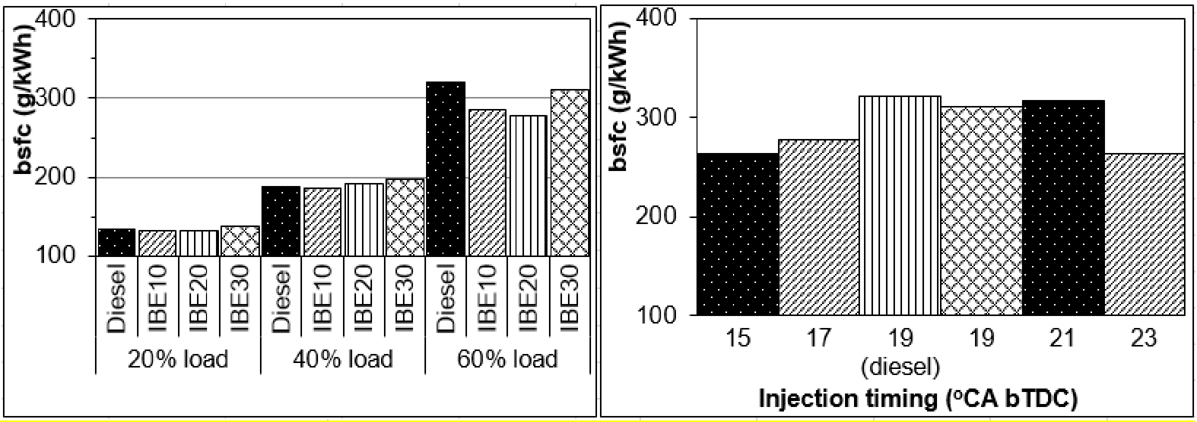

Figure 9. The variation of bsfc with engine load and injection timing.

4. CONCLUSION

A series of experiments were performed to assess the effects of blending ratio and injection timing on combustion and emissions of a single-cylinder CRDI engine using IBE and diesel blends. It is possible to summarize the main results as follows:

• For test fuels, a significant increase is observed in peak pressure and heat release, and as the ignition timing is advanced, these peak values are slowly advanced. With increase in injection advance, heat release increased to 84% by using IBE30 compared with diesel fuel at original advance of 19 ^{o}CA before TDC.

• At low and moderate loads, bsfc for IBE30 increased up to 5.4% in comparison to diesel fuel, although lowering bsfc is observed under high load conditions with IBE.

• CO emissions is found to be a slightly higher for IBE blends at low loads while they reduced with increase IBE amount in the fuel mixture at high load conditions up to 55% by IBE30. The increase in injection advance increased CO emissions and reached its maximum value at 21 ^{o}CA before TDC.

• Unburned HC emissions decreased by %1.35 and %13.40 for IBE10 and IBE20 while an increase about 7% is observed with IBE30 when compared to diesel fuel. Advancing or retarding the injection timing according to 19 ºCA bTDC decreased unburned HC emissions.

• Smoke opacity was found to be greatly reduced as IBE amount increase in fuel mixture and on average, 10.49%, 39.02% and 43.04% decrease was calculated for IBE10, IBE20 and IBE30, respectively, in comparison to diesel fuel. Smoke opacity is further reduced when injection timing is advanced or retarded for IBE30 compared to diesel fuel use in 19 ºCA bTDC.

• NO_x emissions increased with increase in both load and IBE amount in fuel mixture. On average, NO_x emissions increased up to 10% using IBE blends compared with diesel fuel. Increasing the injection advance led to a substantial increase in NO_x emissions.

Experimental results have shown that the main advantage of using IBE in blends with diesel fuel is the significant reduction in smoke opacity, CO and unburned HC emissions, which are beneficial for diesel engines in many respects such as meeting the stringent emission standards and avoiding after treatment systems. Therefore, as the use of IBE by adding it to diesel fuel has yielded important results in terms of exhaust emissions, it is of great importance to advance the work in order to determine the wider blended ratios and the injection advance for these ratios.

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DECLARATION OF ETHICAL STANDARDS

The authors of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

AUTHORS' CONTRIBUTIONS

Kudbettin İlçin: Performed the experiments and analyse the results.

Müjdat Fırat: Performed the experiments and analyse the results..

Şehmus Altun: Performed the experiments and analyse the results, wrote manuscript.

Mutlu Okcu: Performed the experiments and analyse the results.

CONFLICT-OF-INTEREST

No conflict of interest in present study.

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