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TECHNICAL PAPER

Experimental investigation of combustion, performance and emission characteristics of a diesel engine fuelled with diesel–biodiesel–alcohol blends

Ümit Ağbulut[1] · Suat Sarıdemir[1] · Serdar Albayrak[2]

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Abstract
The purpose of this study is to investigate the impacts of diesel–biodiesel–alcohol blends on the combustion, performance and emissions characteristics of a single-cylinder diesel engine. Tests were conducted at different engine speeds of 1750, 2250, 2750 and 3250 rpm and under full load. In this study, different fuels [called as reference diesel (D100), 20 vol% cottonseed methyl ester (D80C20), 10 vol% ethanol (D90E10) and finally the ternary type of their derivations (D70C20E10)], were used. The experimental results showed that the highest reduction values were observed on CO emission by 42%, 30% and 8% for the D90E10, D70C20E10 and D80C20 fuels, respectively. These reductions for HC emission were achieved as 40%, 31% and 23% for the D90E10, D70C20E10 and D80C20, respectively. On the other hand, the reductions of NO_x and CO_2 emissions were not sharp and varied between 2–7%. Besides the reductions on the exhaust emissions, biodiesel–ethanol blend presented better results in terms of HRR_{max} and CP_{max} than using biodiesel alone. Additionally, ignition delay of the biodiesel blends was longer than that of D100 fuel owing to their low cetane numbers. Combustion duration was shortened with the increment in engine speed because the turbulence increased in the combustion chamber at high engine speed. This case also improved the homogeneity of test fuels and increased the quality of the combustion process. As a consequence, this paper clearly reported that it is possible to achieve fewer emissions, the highest CP_{max} values with the presence of ethanol in biodiesel fuels rather than using biodiesel alone for diesel engines.

Keywords Biodiesel–alcohol blends · Combustion · Diesel engine · Emission · Ternary blends

List of symbols

ACP_{max}	Angle of maximum cylinder pressure (°CA)
AIP_{max}	Angle of maximum pressure increase (°CA)
BSFC	Brake-specific fuel consumption (g/kWh)
BTE	Brake thermal efficiency
C20	20% cottonseed biodiesel + 80% diesel (v/v)
C100	100% cottonseed methyl ester
CD	Combustion duration
CO	Carbon monoxide
CO_2	Carbon dioxide
CP_{max}	Maximum cylinder pressure
D80C20	20% cottonseed biodiesel + 80% diesel (v/v)
D70C20E10	20% cottonseed biodisel + 10% ethanol + 70% diesel (v/v)
D90E10	10% ethanol + 90% diesel (v/v)
D100	Reference fuel—conventional diesel
DF	Diesel fuel
EP_i	Pollutant mass, i, referenced to P_{eff} (g/kWh)
$EV_{i,d}$	Exhaust emission value of components on dry basis, i, as volume share (ppm)
$EV_{i,w}$	Exhaust emission value of components on wet basis, i, as volume share (ppm)
HC	Hydrocarbon
HRR_{max}	Maximum heat release rate (J/°CA)
ICEs	Internal combustion engines
k	Polytrophic constant

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✉ Ümit Ağbulut
umitagbulut@duzce.edu.tr

[1] Department of Mechanical and Manufacturing Engineering, Faculty of Technology, Düzce University, 81620 Düzce, Turkey

[2] Department of Aircraft Technology, Vocational School, Istanbul Istinye University, 34010 Istanbul, Turkey

M_i	Molecular mass of the components, *i* (g/mol)
$M_{\mathrm{Exh,d}}$	Molecular mass of the exhaust gases on dry basis (g/mol)
$M_{\mathrm{Exh,w}}$	Molecular mass of the exhaust gases on wet basis (g/mol)
NaOH	Sodium hydroxide
NO_x	Nitrogen oxide
P_e	Brake power (kW)
PM	Particulate matter
RMS	Root-mean-square
SOC	Start of combustion (°CA)
SOI	Start of injection (°CA)
TDC	Top dead centre
T_{ex}	Exhaust temperature (°C)
UHC	Unburnt hydrocarbon
wt%	Weight per cent
v/v	Volumetric ratio
vol%	Volumetric per cent
Ø or °CA	Crank angle
∂P	Change in-cylinder pressure
$m_{\mathrm{Exh,d}}$	Exhaust mass flow (kg/h)
p_{eff}	Power output (kW)
∂V	Change in-cylinder volume
λ	Air–fuel equivalence ratio

1 Introduction

Today, the most important problem faced by the world is probably the energy issue. Global energy demand has increased every passing day compared with 1 day before [1]. Currently, 85% of the world energy consumption has been provided by fossil-based fuels [2]. The fact that the reserves of the existing fossil fuels are limited forced human beings to seek alternative sources.

Clearly, one of the dominant usage fields for fossil fuels has been in the transportation sector because nearly all vehicles on the roads have been powered by internal combustion engines burning fossil fuels. In this regard, scientists and researchers started a tendency to various applications to reduce fossil fuel consumption in transportation. For this purpose, some of them tried to modify the vehicle design [3–5] such as light-material usage and geometry changes, while others tried to modify the fuels [6].

Diesel engines are widely used for high combustion efficiency, adaptability, reliability and cost efficiency, and so they are one of the major contributors to harmful exhaust emissions [7]. Particularly with increasing environmental awareness and concerns, the studies focusing on reducing the harmful exhaust emissions have been intriguing. The adopted view in the literature is that clean combustion can be only provided by fulfilling if diesel fuel reformulation is coupled with engine developments [8]. In this context, biodiesel fuels can be considered as the most promising fuel type because they are a renewable energy source produced from vegetable or animal origin oils. In general, biodiesel fuels have a lower cetane number than conventional diesel fuels. Moreover, they have fewer sulphur contents, no aromatics and also have nearly 10–11% oxygen content by weight [9, 10]. Biodiesel fuels have close combustion characteristics as compared with conventional diesel fuel. However, these close characteristics by using some methods can be made much closer to conventional diesel fuels. Transesterification method is undoubtedly the most acceptable one among these methods.

The previous studies in the literature have generally reported that the presence of biodiesel into diesel fuels leads to higher fuel consumption and higher nitrogen oxides (NO_x) emission, lower unburnt hydrocarbon (UHC), carbon monoxide (CO) and particulate matter (PM) emissions, while there is a small change on the brake thermal efficiency (BTE) [11, 12]. On the other hand, the low volatility, high viscosity and poor cold-flow properties affect its combustion yield and quality particularly at a low temperature of biodiesel [13, 14]. Also, low volatility and high viscosity of biodiesel fuels caused the adverse effects on the atomizability of the fuel blends from the injector. On the other hand, poor cold-flow properties of biodiesels have caused the formation of crystals and solidification of fatty acid compounds particularly at the low temperature [15]. Actually, these poor properties of the biodiesel fuels can be improved by blending with alcohol types because alcohols have pretty high oxygen content compared to biodiesels [16]. Since ethanol has good cold-flow and low viscosity properties, the biodiesels can be blended with ethanol in order to improve their cold-flow properties, their volatility and viscosity properties [13, 15]. As compared with conventional diesel and biodiesel fuels, ethanol has more oxygen content [16], and ethanol has been considered as one of the most promising mixing additives for diesel fuels in the literature [17].

However, there are many technical obstacles to directly use ethanol into diesel fuels owing to some properties of ethanol such as poor solubility in diesel fuel particularly in cold weather, lower flash point and low cetane number. In reality, diesel engines are not able to run normally on ethanol–diesel blend without any special additives [18, 19]. That is why, many additives have improved to ensure solubility of ethanol–diesel blends. Effects of some of the additives into ethanol–diesel blend were reviewed by Hansen et al. [20]. Moreover, it is possible to find many studies in the literature which have investigated the performance of diesel–ethanol blends with some special additives (emulsifiers). However, general conclusions from these studies have encouraged small quantity usage of ethanol in diesel fuel.

Some countries (e.g. Turkey since 1 January 2018) made it compulsory to blend ethanol into diesel and gasoline fuels [6].

The additives help for solubility of the blend, while they cannot improve the properties of the blend [21–23]. Additionally, the low flashpoint of the diesel–ethanol blends is an obstacle to the application of this fuel blend because studies in the literature clearly indicated that the presence of emulsifiers has no effect on flashpoint property [19]. On the other hand, many studies in the literature have reported that biodiesel fuels derived from different sources can act as an emulsifier for ethanol [24–26]. Therefore, blending biodiesel with diesel and ethanol has improved the solubility of ethanol into diesel fuel. The diesel–ethanol–biodiesel (namely bioethanol or ternary blends in some studies) fuel blends are well-stable below zero temperature and have close or superior fuel properties as compared with conventional diesel fuel [27]. The studies in the literature on diesel–ethanol–biodiesel blends have obviously indicated that adding biodiesel into ethanol–diesel fuel blend could improve the properties of blends and showed a promising case to formulate a new form of biodiesel, ethanol and diesel fuel blends. In this respect, this study focused on the experimental investigation of combustion, performance and emission characteristics of a diesel engine fuelled with diesel–biodiesel–alcohol blends.

Table 1 Technical specification of the test engine

Model	Lombardini 15 LD 350
Engine type	Naturally aspirated, air-cooled, DI diesel engine
Cylinder number	1
Maximum power	7.5 HP/3600 rpm
Maximum torque	16.6 N m/2400 rpm
Displacement	349 cm^3
Compression ratio	20.3/1
Bore × stroke	82 mm × 66 mm
Injection pump type	QLC type
Injection nozzle	0.22 × 4 holes × 160°
Nozzle opening pressure	207 bar
Fuel delivery advance (°CA)	20 BTDC
Intake valve open/close (°CA)	10 BTDC/42 ABDC

2 Experimental equipment and procedure

2.1 Equipment

The engine tests were carried on direct injection, air-cooled, naturally aspirated and single-cylinder diesel engine. Table 1 shows the main characteristics of the used diesel engine in this study.

A Kemsan brand DC dynamometer was used to load the test engine in the experiments that it is capable of generating 15 kW at 3000 rpm. All experiments were performed at full load. The engine was loaded by this dynamometer, and the engine speed was then reduced. Additionally, Kistler rotor-type 4550A torque measuring system was used in the study. This system is able to measure the engine torque values from the shaft via a magnetic system.

To sum up, the schematic view of the experimental setup used in this study is shown in Fig. 1.

2.2 Experimental procedure

In this paper, ethanol, cottonseed methyl ester and conventional diesel fuels and their derivation were used as test

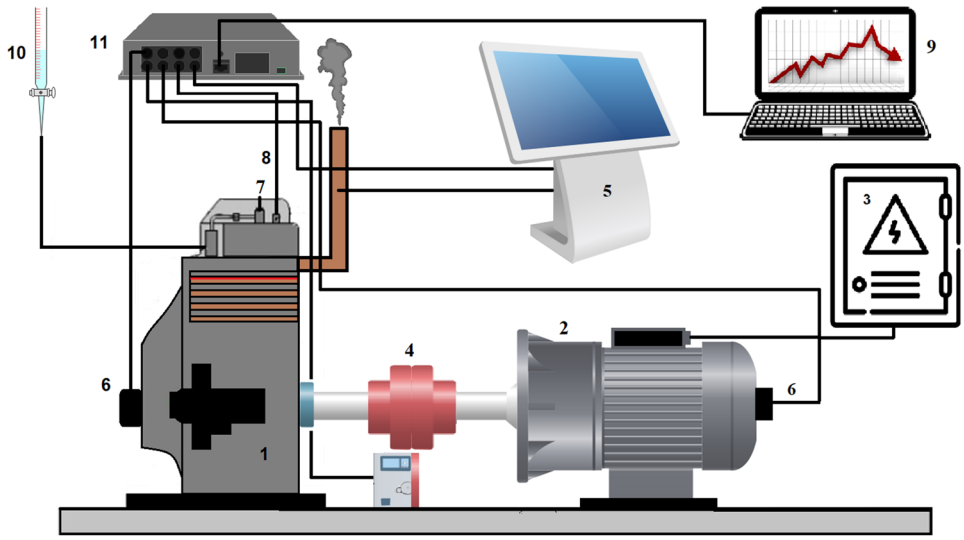


Fig. 1 Schematic view of the experimental setup. 1. Test engine; 2. Dynamometer; 3. Control panel; 4. Torque meter; 5. Exhaust gas emissions analyser; 6. Encoder; 7. Injector; 8. In-cylinder pressure sensor; 9. Computer; 10. Fuel consumption level; 11. Mainboard

fuels. The fuel types, ratios and abbreviation used in the study are shown in Table 2.

Cottonseed methyl ester-biodiesel was used in the current study, and it was prepared by the transesterification method. In this regard, the biodiesel production steps and rates were preferred in light of the literature studies which previously performed and achieved high performance [28, 29]. Biodiesel production steps are given in Fig. 2.

As shown in Fig. 2, NaOH was preferred as catalyst, and methanol was preferred as reactor alcohol in this study. Additionally, some important properties of diesel, cottonseed oil methyl ester and ethanol are given in Table 3.

In this study, ethanol and diesel fuels were blended together in volumetric ratios. Just before the tests, the fuels were mixed for 10 min at 500 rpm by using a thermomagnetic (at room temperature) mixer, and so it was provided to obtain a homogeneous blend. All tests were performed when the oil temperature of the test engine reached 40 °C. The tests were conducted for four different engine speeds (1750, 2250, 2750 and 3250 rpm) and four fuel blends are derived from conventional diesel (D), cottonseed oil methyl ester (C) and ethanol (E) (D100, D90E10, D80C20, D70C20E10). Prior to starting the experiments, the test engine was operated for approximately 5 min to reach a steady-state test condition. The obtained values from the tests were repeated three times under the same conditions. The given values in the result section of this study were obtained by calculating the arithmetic mean of these three measurements.

In order to analyse the combustion characteristics, Febris software was preferred in this paper. The cylinder volume, position of crank angles and cylinder gas pressure values were measured by the sensors that are attached to this software. With an accuracy of 0.1°, a kübler sendix brand encoder was connected onto the crankshaft. The top dead centre (TDC), crank angle and engine speed were measured by using this encoder. Combustion data were processed by an Optrand Auto PSI-TC pressure sensor with a precision of 1% in the combustion chamber during

Table 2 A brief summary of test fuel type and blends

Fuel number	Test fuel						Abbreviation
	Diesel, vol%				Cottonseed methyl ester, vol%20, C20	Ethanol, vol%10, E10	
	%100, D100	90%, D90	80%, D80	70%, D70			
1	✓						D100
2			✓		✓		D80C20
3		✓				✓	D90E10
4				✓	✓	✓	D70C20E10

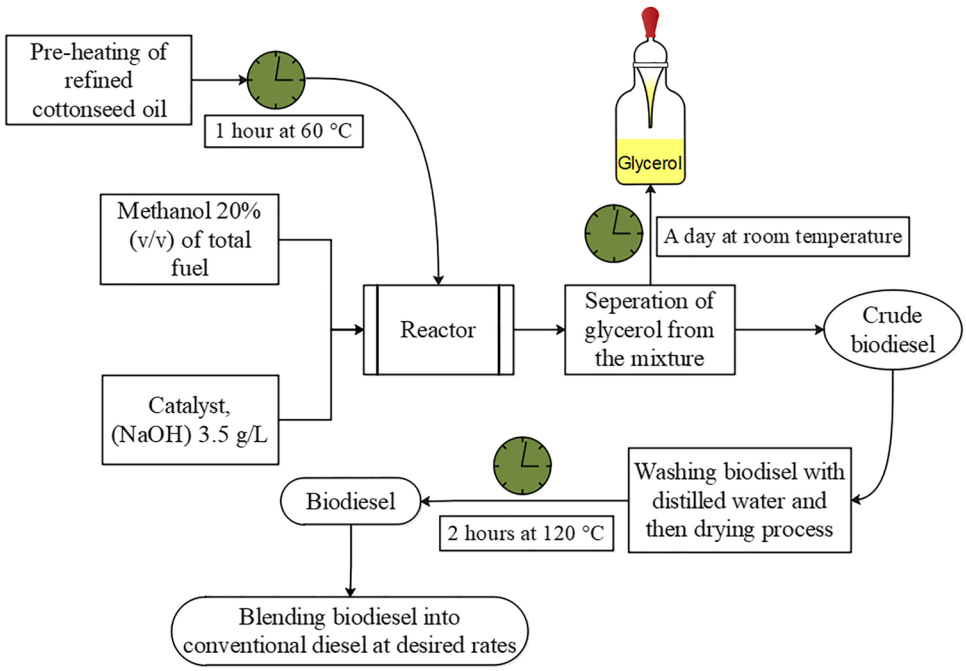


Fig. 2 Biodiesel production steps (transesterification method)

Table 3 Some important properties of conventional diesel, cottonseed oil methyl ester and ethanol [17, 30, 31]

Property	Diesel	Cottonseed methyl ester	Ethanol	Method
Purity (%, v/v)	–	–	99.5	–
Density (mm^2/s; 15 °C)	842	887	789	EN ISO 3675
Cetane number	54.92	46.84	53.44	–
Kinematic viscosity (mm^2/s; 40 °C)	2.93	4.42	1.13	EN ISO 3104
Cloud point (°C)	−4.3	7.0	–	EN 23015
Lower heating value (kJ/kg)	42.93	36.87	26.84	UNE 51123
Flash point (°C)	92.5	150	–	TS EN ISO 2719
Carbon (wt%)	86.74	70.96	52.14	–
Hydrogen (wt%)	13.26	11.43	13.13	–
Oxygen (wt%)	0	9.46	37.73	–
Water content (mg/kg)	74.25	456.3	2024	EN ISO 12937
Pour point (°C)	−21.0	6.0	–	ASTM D97

all experiments. Depending on the changes of crank angle (°CA), the heat release rate was obtained by considering the start and finish of the combustion. All data regarding the combustion process were calculated by taking an average of at least 1000 cycles. This software calculated the heat release rate by using the following equation in accordance with the first law of thermodynamics.

$$\frac{\partial Q_\eta}{\partial \emptyset} = \frac{k}{k-1} P \frac{\partial V}{\partial \emptyset} + \frac{1}{1-k} V \frac{\partial P}{\partial \emptyset} \tag{1}$$

where k shows the polytrophic constant and taken as 1.37 in this paper, and n shows the polytropic index. ∂P shows the change in-cylinder pressure and ∂V changes in the volume. $\emptyset$ is the crank angle. In the calculations, it was assumed that the mass into the cylinder cannot change because of the gas leakage from the valves and segments, and the thermodynamic properties of all fuel blends in the combustion chamber are regular during the tests. Also, it was assumed that the gas in the cylinder acts as an ideal gas.

Additionally, exhaust emissions for each variable in the study were measured by using K Test brand exhaust emission analyser.

The exhaust emissions from the internal combustion engines are typically measured by the aid of a gas analyser and generally given in parts volume per cent (vol%) and per million (ppm) in the previous studies. In order to do better comparison, it is actually important to use these emissions according to the standards of European vehicle emissions which are usually referred in g/km for light-duty and passenger vehicles and g/kWh for heavy-duty vehicles. Previous studies have reported an interesting relationship between vehicle emission concentration and SFC [32, 33]. The mentioned relations are defined by Eq. (2).

$$\mathrm{EP}_i = \mathrm{EV}_{i,\mathrm{d}} \times \left(\frac{M_i}{M_{\mathrm{exh,d}}} \times \frac{m_{\mathrm{Exh,d}}}{P_{\mathrm{eff}}} \right) = \mathrm{EV}_{i,v} \times \left(\frac{M_i}{M_{\mathrm{exh,w}}} \times \frac{m_{\mathrm{Exh}}}{P_{\mathrm{eff}}} \right) \tag{2}$$

Empirical constants were calculated how the literature studies [32, 33] calculate them. In calculations, the following Eqs. (3) and (4) were used:

$$k_{\mathrm{d}} = \frac{m_{\mathrm{Exh,d}}}{P_{\mathrm{eff}}} = 3.873\,\mathrm{g/kWh} \tag{3}$$

$$k_{\mathrm{w}} = \frac{m_{\mathrm{Exh,w}}}{P_{\mathrm{eff}}} = 4.160\,\mathrm{g/kWh} \tag{4}$$

Taking into account the empirical constants in Eq. (2), the values of $\mathrm{EP}_{i,\mathrm{d}}$ and $\mathrm{EP}_{i,\mathrm{w}}$ in Eqs. (5) and (6) were derived.

$$\mathrm{EP}_{i,\mathrm{d}}\left(\frac{\mathrm{g}}{\mathrm{kWh}}\right) = \frac{\mathrm{EV}_{i,\mathrm{d}}\,(\mathrm{ppm})}{1 \times 10^6} \times \left(\frac{M_i}{\frac{30.21\,\mathrm{g}}{\mathrm{mol}}} \times 3873\,\frac{\mathrm{g}}{\mathrm{kWh}} \right) \tag{5}$$

$$\mathrm{EP}_{i,\mathrm{w}}\left(\frac{\mathrm{g}}{\mathrm{kWh}}\right) = \frac{\mathrm{EV}_{i,\mathrm{w}}\,(\mathrm{ppm})}{1 \times 10^6} \times \left(\frac{M_i}{\frac{24.84\,\mathrm{g}}{\mathrm{mol}}} \times 4160\,\frac{\mathrm{g}}{\mathrm{kWh}} \right) \tag{6}$$

Under this assumption, Eqs. (7–10) may be used to estimate the corresponding SFC of exhaust emission values in this study. The general conversion from exhaust gas emission (ppm and vol%) to SFC (g/kWh) for heavy-duty vehicles is summarized as follows:

$$\mathrm{CO}\left(\frac{\mathrm{g}}{\mathrm{kWh}}\right) = 3.591 \times 10^{-3} \times \mathrm{CO}\,(\mathrm{ppm}) \tag{7}$$

$$\mathrm{NO}_x\left(\frac{\mathrm{g}}{\mathrm{kWh}}\right) = 6.636 \times 10^{-3}\ \mathrm{NO}_x\,(\mathrm{ppm}) \tag{8}$$

$$\mathrm{HC}\left(\frac{\mathrm{g}}{\mathrm{kWh}}\right) = 2.002 \times 10^{-3} \times \mathrm{HC}\,(\mathrm{ppm}) \tag{9}$$

$$\mathrm{O}_2\left(\frac{\mathrm{g}}{\mathrm{kWh}}\right) = 41.024 \times \mathrm{O}_2\,(\mathrm{Vol,\%}) \tag{10}$$

Table 4 briefly gives the measuring range and accuracy values of all instruments in this paper.

3 Results and discussion

In this paper, an experimental study was conducted to determine the combustion, performance and emission characteristics of a diesel engine fuelled with diesel–biodiesel–alcohol blends. Diesel–biodiesel and alcohol blends (ethanol as alcohol and cottonseed methyl ester as biodiesel) were used as the test fuels, namely D100, D90E10, D70C20E10 and D80C20. Tests were performed at various engine speeds (1750, 2250, 2750 and 3250 rpm) and under 207 bar injection pressure. As a consequence, the following results from the present study were achieved.

3.1 Combustion

Combustion characteristics such as in-cylinder pressure (CP) and heat release rate (HRR) are the most important parameters that strongly affect engine performance as well as exhaust emissions. The chemical and physical properties of the test fuels largely influence the combustion characteristics and performance of diesel engines. As well-known, the time interval between the start of injection and the start of combustion has been stated as ID in the diesel engines. The viscosity, cetane number and oxygen content of the test fuels strongly influence the ignition delay. On the other hand, ID also strongly influences the CP, SOC and HRR. Table 5 gives the numerical data obtained for the injection and combustion characteristics of all fuel types depending on the various engine speeds.

As shown in Table 5, the SOI of the D80C20 fuel was earlier in nearly all experiments. This is due to the fact that the fuel line pressure earlier reached to 207 bar (the injector opening pressure) owing to the high density and viscosity of the D80C20 fuel. Moreover, SOI and SOC were closer to TDC for all fuel types depending on engine speed. The low cetane number of the D80C20 fuel caused the ID to last longer than other fuel types. For all fuel types, the CD values decreased with the increment in the engine speed. It is thought to be caused by higher turbulence in the combustion chamber as a result of increasing the engine speed. In addition, with high turbulence, the homogeneity of the fuel blends improved, and then combustion process accelerated.

In Figs. 3, 4, 5 and 6, the impacts of test fuels on CP and HRR are seen depending on the various engine speeds. As seen from these figures, the highest maximum in-cylinder pressure (CP_{max}) values for each fuel type were achieved at 1750 rpm, and CP_{max} values were generally decreased depending on increasing the engine speed. The starting point of the combustion process gets closer to the top dead centre (TDC) because of the increment in engine speed. However, this case shortened the time of the uncontrolled combustion phase (or premixed combustion phase) at high speeds and so decreased the CP_{max} values by increasing the diffusion-controlled combustion time. The reduction in the CP_{max} depending on the engine speed also similarly reduced the maximum HRR values (HRR_{max}) of each fuel type depending on the engine speed. On the other hand, the ignition delay (ID) was another parameter which can significantly affect the combustion process. The oxygen content of the used fuels, the atomization rate and also cetane number markedly affect the ID. The lower cetane numbers of both biodiesel and ethanol prolonged the duration of ID and caused more fuel accumulation in the combustion chamber prior to starting of the combustion process. In addition, the high evaporation latent heat of ethanol caused longer ID. That is why the highest CP_{max} and HRR_{max} values were achieved by the fuels of D90E10 and D90C20E10, respectively. Owing to the more O_2 content of the ethanol and biodiesel-content fuel types, they provided to reach to higher CP_{max} and HRR_{max} values than D100 fuel.

Table 4 Specifications of the measurement equipment

No.	Measuring instrument	Range	Units	Accuracy
1	Pressure (Optrand Auto PSI-TC)	0–20.68	MPa	±1%
2	Crank angle encoder (kübler sendix)	0–360	°	±1%
3	Thermocouple (K-Type)	0–1200	°C	±0.1 °C
4	Fuel flow burette	0–100	ml	±1%
5	Torque measuring system (Kistler Rotor-type 4550A)	0 ± 100–0 ± 5000	Nm	0.05%
6	K Test exhaust gas emission analyser			
6a	O_2	0–25	%	±0.1%
6b	CO	0–15	%	±0.01%
6c	NO_x	0–5000	ppm	±0.1%
6d	HC	0–20.000	ppm	±1%
6e	CO_2	0–20	%	±0.1%

Table 5 Combustion and injection characteristics of test fuel types at different engine speeds in this study

Engine speed (rpm)	Fuel types	SOI [°CA]	ID [°CA]	SOC [°CA]	CD [°CA]	CP_{max} [bar]	ACP_{max} [°CA]	AIP_{max} [°CA]	HRR_{max} [J/°CA]
1750	D100	−12	5	−7	45	93.52	6	−4	40.24
	D80C20	−13	6	−7	47	95.86	5	−4	45.55
	D90E10	−12	5	−7	45	96.80	7	−3	54.58
	D70C20E10	−12	5	−7	46	95.01	5	−4	45.12
2250	D100	−10	5	−5	43	86.71	7	−1	47.69
	D80C20	−11	6	−5	46	90.06	6	−1	49.69
	D90E10	−11	6	−5	44	91.74	7	0	57.29
	D70C20E10	−12	7	−5	45	90.65	8	0	40.89
2750	D100	−9	6	−3	41	78.80	6	3	38.13
	D80C20	−10	7	−3	43	80.11	8	2	46.10
	D90E10	−9	6	−3	43	83.15	9	3	47.94
	D70C20E10	−9	6	−3	42	82.59	9	2	46.82
3250	D100	−8	6	−2	40	76.84	9	3	36.36
	D80C20	−10	7	−3	38	75.00	7	3	32.78
	D90E10	−8	6	−2	40	79.94	9	5	45.99
	D70C20E10	−8	6	−2	40	78.00	7	4	41.02

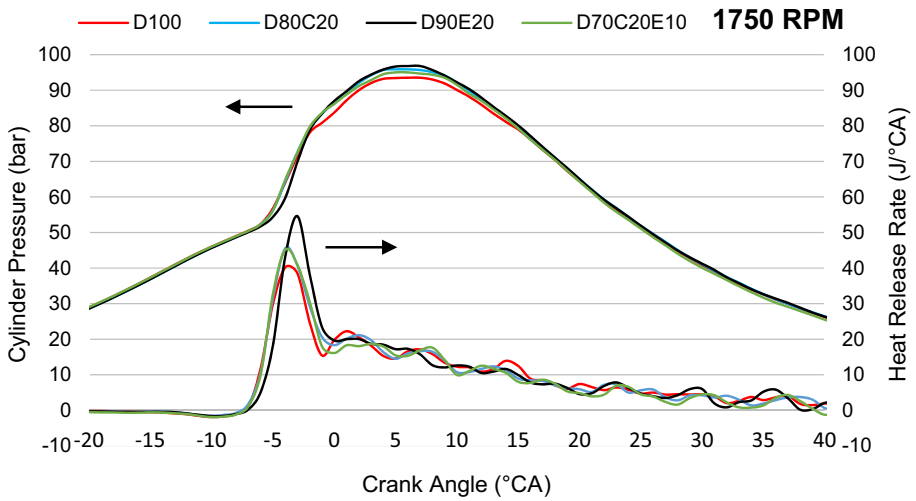


Fig. 3 Change of the CP and HRR according to the crank angle at 1750 rpm

The higher density of the used biodiesel drew the start of the fuel injection earlier. The earlier injected fuel flowed to an ambient in which the pressure and the temperature in the combustion chamber were partially lower. This was why the ID increased for the fuel blends containing biodiesel. Moreover, the low cetane number and high viscosity of this biodiesel caused a higher CP_{max} and HRR_{max} compared with D100 at all engine speeds except for 3250 rpm. On the other hand, the lower heating value of D100 was higher than ethanol and cottonseed methyl ester, and it has a lower density than cottonseed methyl ester. These properties of D100 caused to the highest CP_{max} and HRR_{max} at 3250 rpm compared with D80C20.

3.2 Performance characteristic

The impacts of test fuels on torque values depending on engine speed are given in Fig. 7. The high volumetric efficiency at lower engine speeds increased the combustion period and decreased friction loss, gas leakage per cycle and also heat loss. This case caused to the increments in the torque values up to 2500 rpm for each test fuel, and then they started decreasing. Because the lower heating values of both biodiesel and ethanol additives were lower than D100, the torque values were obtained lower than D100 at all engine speeds. Even though ethanol has the smallest lower heating value among all test fuels, its O_2 content was approximately

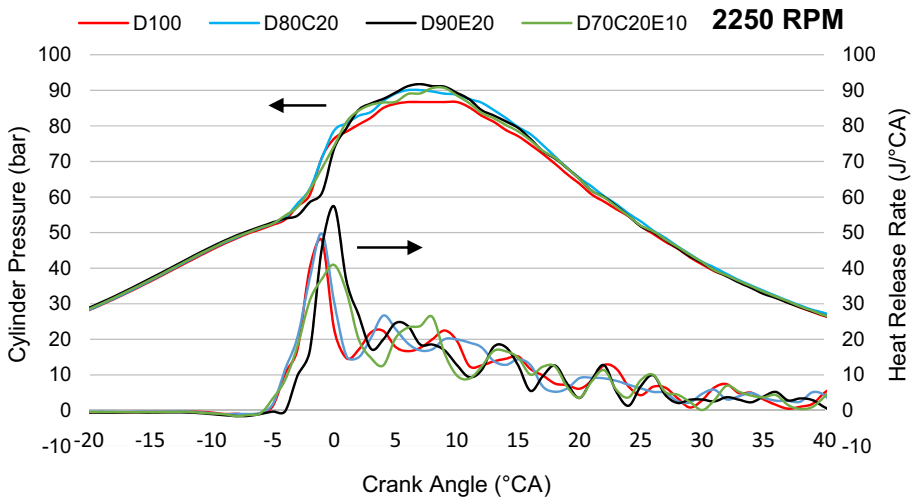


Fig. 4 Change of the CP and HRR according to the crank angle at 2250 rpm

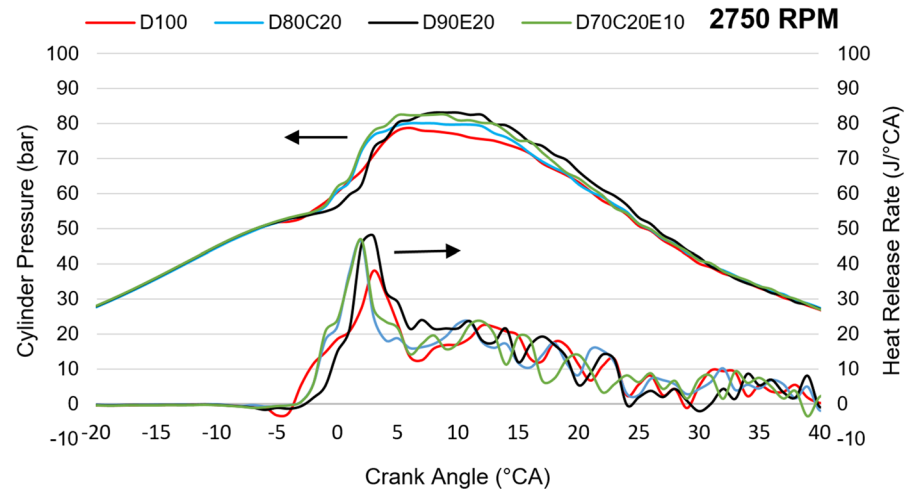


Fig. 5 Change of the CP and HRR according to the crank angle at 2750 rpm

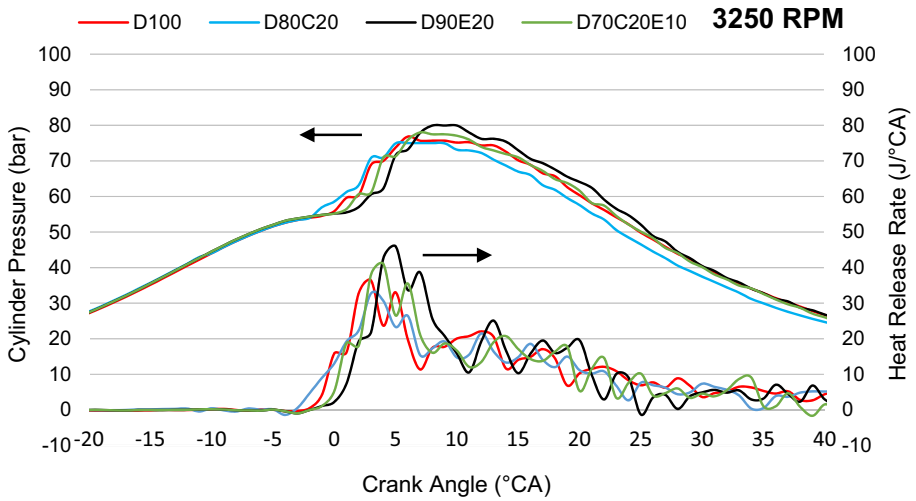


Fig. 6 Change of the CP and HRR according to the crank angle at 3250 rpm

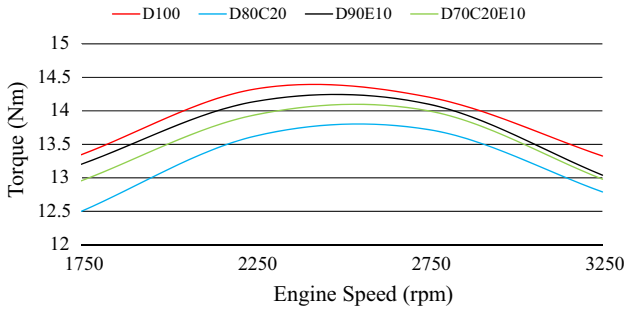


Fig. 7 Changes of torque values according to engine speeds

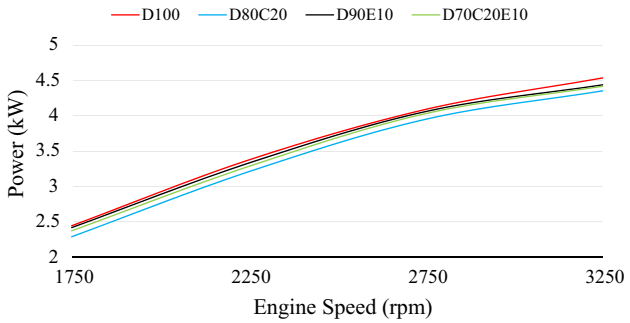


Fig. 8 Changes in power according to engine speeds

4 times higher than that of biodiesel. Moreover, its density was the lowest among the test fuels and its present ratio in the blends was lower. Therefore, the highest torque values were obtained with D100. Then, D90E10, D70C20E10 and D80C20 fuels followed to D100 fuel.

With the increment in engine speed, the flow rate of the injected fuel into the combustion chamber has also increased. This case increased both the released energy amount and the engine power for all fuel types depending on the engine speed (Fig. 8).

3.3 Emission characteristic

The lambda values (λ), O_2, CO, NO_x and HC emissions and exhaust temperature (T_{ex}) values were measured depending on the various engine speeds. As well-known, emissions have negative effects on both the environment and living things. Therefore, it is important to reduce the emission levels in the vehicles. To do a more accurate comparison, the emission units are converted into g/kWh in this paper.

3.3.1 Lambda (λ): air–fuel equivalence ratio

λ represents the air–fuel equivalence ratio. The changes of λ value are given in Fig. 9 for each test condition. For all fuel types, the highest lambda values were obtained at D70C20E10, D90E10, D80C20 and D100 fuels,

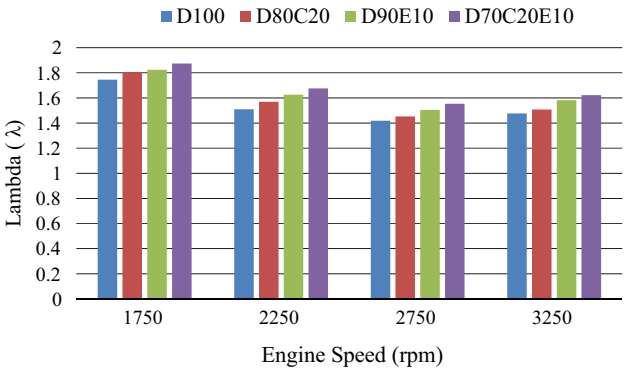


Fig. 9 Changes in lambda (λ) according to engine speeds

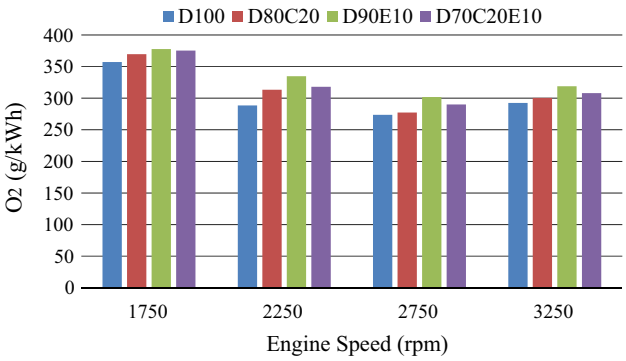


Fig. 10 Changes in O_2 emissions according to engine speeds

respectively. The reason for the lowest lambda value in the D100 fuel was that the D100 fuel has the lowest oxygen content among all test fuels. In addition, the lambda value of the D80E10 was higher than that of the D80C20. This is due to the fact that the oxygen content of ethanol was about 4 times higher (see Table 3) than cottonseed methyl ester.

3.3.2 O_2 emission

The presence of O_2 (%) in exhaust emissions is one from the indicators of combustion quality. O_2 content of ethanol in this study was about 4 times greater than the used biodiesel. On the other hand, the O_2 content of conventional diesel D100 is equal to zero. As shown in Fig. 10, there are, therefore, the highest O_2 emission levels achieved with ethanol-blended fuels among all fuel types at all engine speeds. The high volumetric efficiency at low engine speeds increased the O_2 content of all fuel types. With the increment in the engine speed, turbulence in the combustion chamber provided more ideal mixing conditions, and therefore the flame speed also increased. Hereby, the O_2 consumption increased with the increase in combustion quality, and the amount of O_2 released from the engine to the atmosphere reduced.

The maximum combustion efficiency around the maximum torque cycle area reduced the O_2 ratio. Despite the low volumetric efficiency at the high engine speeds, the inadequate time for combustion process also increased the O_2 rate.

3.3.3 CO emission

The most important reason for the presence of CO emission in combustion products is that the oxygen in the combustion chamber was insufficient, so the fuel could not be oxidized or partially oxidized. Figure 11 shows the CO changes depending on the engine speeds.

The reasons for lower CO emissions at low engine speeds were the high volumetric efficiency and high O_2 ratio, and also sufficient combustion time. CO emissions increased due to the decreasing volumetric efficiency and shortening of the combustion duration (CD), while engine speeds were increasing up to 2750 rpm. However, it was observed that CO emission values were reduced after exceeding to 2750 rpm for all fuel types. It is thought that the main reason for this reduction was due to the better homogeneous mixture with the increasing speed and as a result of that the improvement of combustion quality. The high O_2 content of ethanol additive also encouraged more complete combustion and resulted in lower CO emissions with ethanol-containing fuels at all engine speeds.

3.3.4 Unburned HCs emission

The main reason for the presence of unburned HCs in the combustion products is due to the inability of the fuel to reach the ignition temperature or not to have sufficient oxygen in the combustion chamber. In general, this case has caused the fuel to become non-oxidized or semi-oxidized. Figure 12 shows the HC changes with the engine speeds for all fuel types.

For all fuel types, the changes in HC emissions depending on the engine speed were quite small and changed from 0.03 to 0.09 g/kWh. However, it was observed that the O_2 ratio was higher at low and high engine speeds, and it decreased HC emission values for all fuels. The high O_2 ratios of both ethanol and biodiesel fuels decreased the HC emissions by improving the combustion quality.

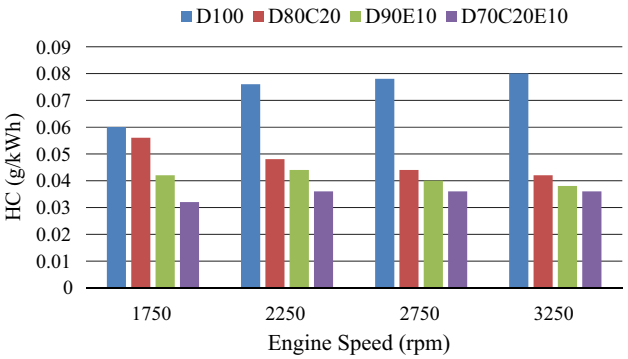


Fig. 12 Changes in HC emissions according to engine speeds

3.3.5 NO_x emission

There are actually three important factors in the formation of NO_x emissions. These are oxygen content, temperature and duration, respectively. In the cases that in-cylinder temperature exceeds 1600 K, oxygen will react with nitrogen to form NO_x if there are sufficient oxygen and time. In Fig. 13, NO_x changes with engine speeds are given. The highest NO_x values were achieved with D100 fuel at all engine speeds. The lower heating value of both biodiesel and ethanol decreased NO_x values by decreasing the exhaust temperature. In terms of NO_x emissions, no significant change was observed.

3.3.6 Exhaust gas temperature

The change in the exhaust temperature (T_{ex}) is shown in Fig. 14. The increase in the number of burns per unit time

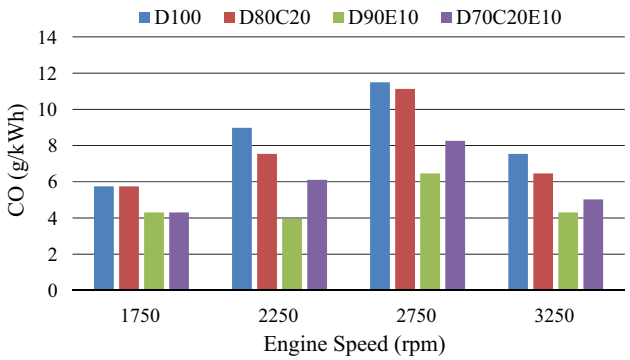


Fig. 11 Changes in CO emissions according to engine speeds

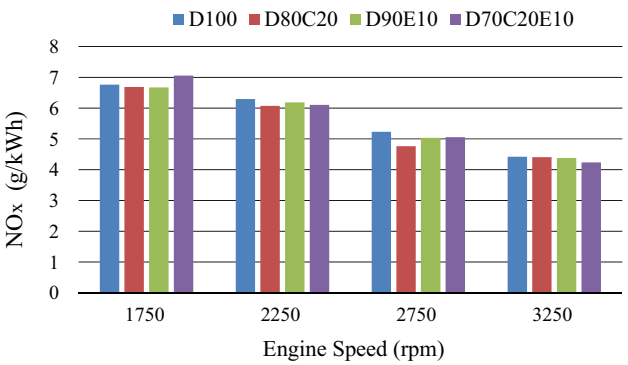


Fig. 13 Changes in NO_x emissions according to engine speeds

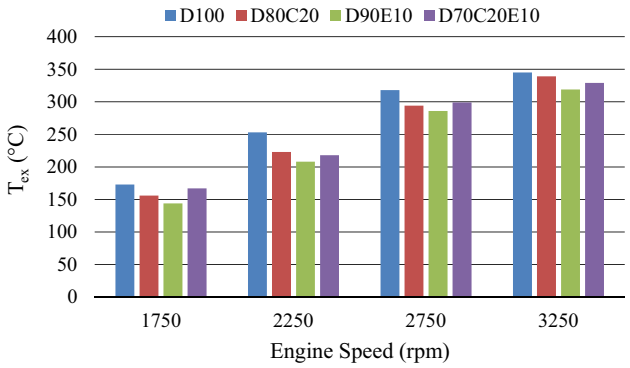


Fig. 14 Changes in T_{ex} according to engine speeds

and the amount of injected fuel from the injector, depending on the engine speed, also increased the exhaust temperature with the engine speed for all fuels. In addition, prolongation of diffusion combustion duration depending on engine speed contributed to the increase in exhaust temperature with the increasing engine speed. The lower thermal values of ethanol and biodiesel-containing fuels compared with D100 fuel caused a lower exhaust temperature at all engine speed.

4 Conclusion

The objective of this paper is to experimentally investigate the combustion, performance and emission characteristics of a single-cylinder diesel engine fuelled with diesel–biodiesel–alcohol blends. In order to compare the obtained results, the experiments were also performed for reference conventional diesel fuel—D100. Based on the achieved experimental results, the following conclusions can be summarized below:

- Comparing with the D100 reference fuel, the highest reduction percentages in CO emissions were achieved by 42%, 30% and 8% of the D90E10, D70C20E10 and D80C20 fuels, respectively. The sharp reduction in D90E10 fuel is due to the high oxygen content of ethanol.
- Similarly, the highest reduction percentages in HC emissions were obtained 52%, 43% and 33% for the D70C20E10, D90E10 and D80C20, respectively.
- The high oxygen content of biodiesel and alcohol additives caused an increase in O_2 emissions. This increase was 10%, 6.7% and 4% for D90E10, D70C20E10 and D80C20, respectively.
- Compared with D100 fuel as a result of the addition of biodiesel and ethanol, the reduction of NO_x emissions is very low (varied from 1.5 to 3.5%).
- The experimental results clearly show that a cleaner source of energy can be obtained with the addition of alcohol and/or biodiesel into conventional diesel fuels.
- The increase in the presence of ethanol in the fuel blends resulted in the highest torque values. Therefore, the highest torque values for the blends were achieved by D90E10, D70C20E10 and D80C20, respectively.
- With increasing engine speed, the power obtained in all fuel types increased approximately linearly.
- Due to the higher latent heat of vaporization, ethanol content fuels resulted in long ignition delay time. Therefore, the CP_{max} and HRR_{max} values were obtained D90E10 and D90C20E10, respectively.
- Ethanol acted as an emulsifier for biodiesel fuel. The blending of ethanol with diesel–biodiesel blend improved the solubility of biodiesel into diesel fuel.
- The presence of ethanol in biodiesel fuels both caused to fewer emissions and provided to reach to the highest CP_{max} values. Therefore, this paper strongly suggests using alcohol into the diesel–biodiesel blends (ternary blends) rather than using diesel–biodiesel alone for diesel engines.
- The ID of D80C20 fuel was longer than other fuel types because of the low cetane number of this fuel type.
- With the increment in engine speed, turbulence in the combustion chamber increased. High turbulence improved the homogeneity of the fuel blends and accelerated the combustion process. On the other hand, CD decreased with the increment in engine speed.

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