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IS THE ETHANOL ADDITIVE MORE ENVIRONMENTALLY FRIENDLY FOR A SPARK IGNITION (SI) ENGINE OR FOR A COMPRESSION IGNITION (CI) ENGINE?

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Abstract

Clearly, the purpose of this paper is to find an answer to the following question "*Is the ethanol additive more environmentally friendly for an SI engine or for a CI engine?*". The tests, therefore, were conducted on both an SI and a CI engine for the same parameters under both same conditions and laboratory. Ethanol was blended into neat diesel (D100) and neat gasoline (G100) at the same proportion (10 vol. %) and two blends were prepared in the study, namely D90E10 and G90E10, respectively. Then the tests were conducted on different engine speeds varying from 2250 to 3250 rpm with an interval of 250 rpm. In the experimental results achieved in the study, the most reductions among exhaust emissions, as compared to reference-D100 and reference-G100 fuel type, were achieved in HC and CO emissions with the presence of ethanol. With the addition of ethanol, HC and CO emissions in the SI engine reduced by 47.9% and 47.0%, respectively; on the other hand, these emissions also reduced by 28.5% and 25.1%, respectively in CI engine. An interesting result from this paper is that NOx emission was slightly reduced by 2.3% for SI engine with the addition of ethanol, whilst it is observed an increase of approximately 40% for the CI engine. This study showed that the addition of ethanol can be used in both SI and CI engines without any modification and can result in a significant reduction in exhaust emissions. In conclusion, this paper is distinctly reporting that the presence of ethanol into diesel fuel has presented better results than those of gasoline fuel in terms of exhaust emissions.

Key words: diesel, ethanol, emission, gasoline, pollutant, greenhouse gas

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1. Introduction

The beginning of the studies regarding internal combustion engines (ICEs) dates to the late the 1800s. The famous scientists such as Rudolph Diesel, Nicolaus Otto, Henry Ford, Gottlieb Daimler and Karl Benz made great successes on the automobile world particularly from the late 1800s to 1930s and they invented to the spark ignition engine (SI) and compression ignition engine (CI) (Ağbulut et al., 2018). Even today, the vehicles on the road have been powered by ICEs at the rate of more than 99%. The fact that ICEs have such a high utilization rate has caused the rapid depletion of fossil-based fuels. On the other hand, fossil fuels are not a clean energy source for ICEs and played a vital role in the increase of air pollution. For example, burning 1-liter diesel fuel is responsible for 2.9 kilograms of greenhouse gas emissions (GHG) whilst this amount is equal to 2.7 kilograms of GHG as a consequence of burning 1-liter gasoline fuel (Ağbulut and Bakır, 2019).

In the literature, a number of studies have conducted on minimizing the exhaust gas emissions arising from burning fossil fuels in ICEs. In general, these studies focused on the modification of the fuels (Ağbulut et al., 2018; Ağbulut et al., 2020; Atmanli, 2016; Efe et al., 2018; Emiroğlu and Şen, 2018; Karagöz et al., 2020a; Rajak and Verma, 2018; Rajak et al., 2019; Sarıdemir and Ağbulut, 2019; Saridemir et al., 2020; Şen, 2019; Uyaroğlu et al., 2018; Yilmaz

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and Atmanli, 2017) or design modification (i.e. material design or modifying on the vehicular system) (Aydın et al., 2019; Aydın and Karaağaç, 2019; Mayda et al, 2017; Şen et al., 2016; Şen and Baykal, 2019; Özçelik and Gültekin, 2019; Sen, 2020). The fuel modification is accomplished by adding different additives into conventional diesel and gasoline fuels at certain proportions. However, it is of great important issues to use the modified fuel without requiring any modifications to ICEs. In this regard, the most acceptable and promising additives in the literature are probably biofuels that can be produced from animal fats and vegetable oils such as sunflower, corn, soybean, canola, cottonseed, waste cooking oils, etc. (Efe et al., 2018; Saridemir and Albayrak, 2015; Şen, 2019; Uyaroğlu et al., 2018). Actually, alcohols can be also considered an additive that has the big alternative fuel additive potential to conventional diesel and gasoline fuels. They have produced by fermentation of biomass feedstocks such as; cereals, sugars, corns and microbial. Similar to biofuels, alcohols are, therefore, renewable and sustainable energy sources. In literature, alcohol types such as ethanol, methanol, butanol, propanol and pentanol have been frequently tested by the fuel researchers (Ağbulut et al., 2018; Atmanli, 2016; Emiroğlu and Şen, 2018; Yilmaz and Atmanli, 2017).

Owing to the main characteristics of alcohols, they are able to improve the combustion quality, to enhance the thermal efficiency (BTE) and also to reduce the exhaust emissions of ICEs (Zaharin et al., 2017). Additionally, low viscosity values of alcohol enhance the fuel atomization capability during the fuel injection. Another important characteristic of alcohols is that they have a remarkable amount of oxygen molecules. This case promotes more complete combustion and causes lower exhaust emissions. In literature, it is possible to find many studies regarding changes in the exhaust emissions of blending alcohol within neat diesel or gasoline fuels nearly at every proportion. For example, Singh et al. (2016) experimentally investigated the effects of blending ethanol at various proportions of 5%, 10% and 20% into gasoline fuel in an SI engine. The tests were conducted on 12 various engine speeds (from 1000 rpm to 6500 rpm at a regular interval of 500 rpm). The obtained results of the related study showed that ethanol, due to its higher octane number and oxygen content, reduced the specific energy consumption improved the combustion process and so marginally enhanced the engine performance. Additionally, ethanol reduced the exhaust emissions of CO and HC while increasing the emissions of NO_x and CO_2 (Singh et al., 2016). In another study, Sayin (2010) investigated the exhaust emission changes by blending ethanol and methanol alcohols at different rates (5% and %10 into conventional diesel fuel by volume). The experiments performed by various the engine speeds between 1000 and 1800 rpm at a constant engine load of 30 Nm. The results indicated that brake specific fuel consumption (BSFC) and NO_x emission increased whilst smoke opacity, BTE, CO and HC emissions decreased with methanol–diesel and ethanol–diesel fuel blends (Sayin, 2010). Ceviz and Yüksel (2005) determined the optimum ethanol-gasoline rate in their study. For this purpose, four different ethanol-gasoline blends (E5, E10, E15 and E20) were preferred and the engine was operated at 2000 rpm. The study showed that the lowest CO and HC emissions were achieved with E10 blend while the highest CO_2 emissions were achieved with E10 blend (Ceviz and Yüksel, 2005). Bata and Roan (1989) experimentally studied the effects of blending ethanol into conventional gasoline fuel on CO, CO_2 and HC exhaust emissions. The study indicated that CO and HC emissions were reduced while the concentration of CO_2 increased with increasing the alcohol content in the fuel blend. Also, the best alcohol blend into the gasoline ratio was determined as 10% (Bata and Roan, 1989). Huang et al. (2009) investigated the changes in the exhaust emissions for ethanol blends in CI engine at 1500 and 2000 rpm. The results showed that CO, NO_x emissions generally decreased while CO_2 emission increased. Additionally, depending on the increasing engine speed, BSFC and BTE, smoke emission also increased (Huang et al., 2009). Rakopoulos et al. (2007) conducted on the exhaust emissions of ethanol content-diesel fuels. The obtained results from the related study showed that BSFC, BTE, HC emission slightly increased and NO_x and CO slightly decreased while smoke density significantly reduced with the use of ethanol in a CI engine (Rakopoulos et al., 2007).

Considering the literature studies, it is clearly seen that ethanol has blended to both conventional diesel and gasoline fuels in different ratios. In literature studies, some variables such as base fuel and ethanol ratio, engine speed and load have generally preferred to observe the changes in a CI or SI engines. The previous studies are reporting that more complete combustion has been reached due to the high oxygen content of ethanol and therefore the presence of it in the blend generally reduces CO and HC emissions and increases CO_2 emissions. This means that fossil-based conventional diesel and gasoline fuel types are transformed into a cleaner energy source. However, there is no study investigating the addition of ethanol for both diesel and gasoline fuel because the studies, heretofore in the literature, have focused on the investigation the impacts of ethanol blends for only either an SI engine or the CI engine. To the best of the Author knowledge, this study will be the first study discussing the same blend ratios of ethanol-diesel and ethanol-gasoline fuel types in the same parameters, under the same experimental conditions and laboratory and there is also no study reporting that addition of ethanol is more appropriate to convert conventional diesel or gasoline fuels into cleaner energy sources.

In line with this, the same rates (10%) of ethanol were separately blended into both conventional diesel and gasoline fuels at the different engine speeds (2250, 2500, 2750, 3000 and 3250 rpm) in this study. The same tests were also performed under the same conditions for reference neat diesel

(D100) and reference neat gasoline (G100) fuel types in order to make a better evaluation of the results. Briefly, it is aimed to determine whether the addition of ethanol is a more environmental-friendly fuel additive for the SI engine or for the CI engine with this study.

2. Materials and method

In this study, the tests were firstly conducted on a direct injection, air-cooled, naturally-aspirated and single-cylinder diesel engine (Lombardini 15 LD 350). The technical specification of this diesel engine (CI) engine is given in Table 1. Then, this CI was replaced with a spark ignition (SI) engine (Honda GX390). This engine is 4-stroke, air-cooled and single cylinder. The experiments were secondly conducted on this engine at the same parameters with the first experiments in CI engine. These experiments under the same condition and the same laboratory were already performed one of the previous studies (Ağbulut et al., 2018). That is why the data of the related previous study was used in this paper for the CI engine. The technical specification of this SI engine is given in Table 2. In the experiments, a Kemsan brand DC dynamometer was attached to both SI and CI test engines that this dynamometer is also capable of 15 kW.

Ethanol was used as the fuel-additive in the experiments. It was blended at a rate of 10% (v/v) within conventional diesel and gasoline fuels. There are several reasons to blend the ethanol into D100 and G100 fuels at 10% proportion. The first of these, 10% blending ethanol is one of the most preferred blending ratios according to the Republic of Turkey Ministry of Energy and Natural Resources (MENR, 2019). Therefore, it may be compulsory to blend ethanol at a rate of up to 10% into neat gasoline or diesel fuels in the future because 10% of ethanol is currently blending into gasoline in India (Singh et al., 2016). Additionally, previous studies showed that 10% ethanol blend in neat fuel gave the best results (Ceviz and Yüksel, 2005; Bata and Roan, 1989).

Firstly, both engines were operated with conventional fuels i.e. CI engine was firstly run with 100% diesel fuel and called D100 fuel in the study. Secondly, it was run with the blend contains 90% diesel fuel and 10% ethanol, called D90E10 fuel. Experiments were performed with five different engine speeds (2250, 2500, 2750, 3000, and 3250 rpm). After that, the SI engine was changed with the CI engine. The same procedure was applied to the SI engine. Similar to the first experiments, two different test fuel was also used in this engine. The first one is containing 100% gasoline fuel and called G100. The second one was a fuel blend that contains 90% gasoline fuel and 10% ethanol, called G90E10. Just as the experiments in the CI engine, the experiments were performed with four same engine speed in SI engine. In this study, ethanol into neat diesel and neat gasoline fuels were blended in volumetric ratios. Just before the experiments, the fuels were mixed for 10 minutes at 500 rpm by using a thermal-magnetic (at room temperature) mixer and so it ensured to obtain a homogeneous blend. Some important properties of diesel, gasoline and ethanol were given in Table 3.

Table 1. Technical specification of the CI 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 Nm/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

Table 2. Technical specification of the SI engine

Model	***Honda GX 390 H VTE5***
Engine type	4-stroke single-cylinder OHV petrol engine
Ignition system	Digital CDI with variable ignition timing
Cylinder number	1
Fuel type	Unleaded gasoline
Bore x Stroke	86 x 64 mm
Displacement	389 cm^3
Compression ratio	8.2:1
Specific fuel consumption	230 gr/BG-h
Max. net torque	26.5 Nm / 2.7 kgfm / 2500 rpm
Nominal speed	3600 rpm
Max. engine power	9.6 kW

Table 3. Some important properties of diesel, gasoline and ethanol

Property	*Diesel*	*Gasoline*	*Ethanol*	*Test Method*
Purity (%, v/v)	-	-	99.5	-
Density (mm^2/s; 15 °C)	831	785	789	EN ISO 3675
Cetfane/Octane number	53.8	95	53.5	-
Kinematic viscosity, (mm^2/s; @40 °C)	2.86	0.5	1.1	EN ISO 3104
Lower heating value (MJ/kg)	43.1	43.5	26.8	UNE 51123
Latent heat of evaporation, kJ/kg	250	305	840	-
Carbon (wt%)	86.7	87.5	52.1	-
Hydrogen (wt%)	13.3	12.5	13.1	-
Oxygen (wt%)	0	0	37.7	-

It is aimed to investigate the effects of neat diesel and neat gasoline fuels and also their blends with the same proportion ethanol content on the exhaust emissions under the same test conditions for the same engine speeds. Exhaust emission values were measured by using K-Test brand exhaust emission measurement device. The exhaust emission equipment measures the emissions as volume percent (vol %) and per million (ppm). However, it is crucial to converting the emission units into g/kWh because this undoubtedly provides a better comparison.

Actually, it is crucial to give the emission units 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 (Ağbulut et al., 2019; Khanlari et al., 2020a; Khanlari et al., 2020b; Pilusa et al., 2012). The mentioned relations are defined by Eq. (1).

$$EP_i = EV_{i,d} \times \left(\frac{M_i}{M_{exh,d}} \times \frac{m_{exh,d}}{P_{eff}}\right) = EV_{i,v} \times \left(\frac{M_i}{M_{exh,w}} \times \frac{m_{exh,w}}{P_{eff}}\right) \quad (1)$$

Empirical constants were taken from the literature studies (Pilusa et al., 2012; Ağbulut et al., 2019). In calculations, Eqs. (2) and (3) were used:

$$k_d = \frac{m_{exh,d}}{P_{eff}} = 3.873\ g/kWh \quad (2)$$

$$k_{dw} = \frac{m_{exh,w}}{P_{eff}} = 4.160\ g/kWh \quad (3)$$

Considering the empirical constants in Eq. (2), the values of $EP_{i,d}$ and $EP_{i,w}$ in Eqs. (4) and (5) were derived.

$$EP_{i,d}\left(\frac{g}{kWh}\right) = \frac{EV_{i,d}(ppm)}{1\times 10^6} \times \left(\frac{M_i}{\frac{30.21g}{mol}} \times 3873\frac{g}{kWh}\right) \quad (4)$$

$$EP_{i,w}\left(\frac{g}{kWh}\right) = \frac{EV_{i,w}(ppm)}{1\times 10^6} \times \left(\frac{M_i}{\frac{24.84g}{mol}} \times 4160\frac{g}{kWh}\right) \quad (5)$$

Under this assumption, Eqs. (6–10) can be used to estimate the corresponding specific fuel consumption of exhaust emission values in this paper. The general conversion from exhaust gas emission (ppm and vol.%) to specific fuel consumption (g/kWh) for heavy-duty vehicles is shortly given as follows:

$$CO\left(\frac{g}{kWh}\right) = 3.591 \times 10^{-3} \times CO(ppm) \quad (6)$$

$$NOx\left(\frac{g}{kWh}\right) = 6.636 \times 10^{-3} \times NOx(ppm) \quad (7)$$

$$HC\left(\frac{g}{kWh}\right) = 2.002 \times 10^{-3} \times HC(ppm) \quad (8)$$

$$O_2\left(\frac{g}{kWh}\right) = 41.024 \times 10^{-3} \times O_2(ppm) \quad (9)$$

$$CO_2\left(\frac{g}{kWh}\right) = 63.470 \times 10^{-3} \times CO_2(ppm) \quad (10)$$

Converting the emission units into g/kWh was done using the Eqs. 6-10 in this study. On the other hand, the measurement range and accuracy values of the device is given in Table 4.

Table 4. Emission measuring range of K-Test exhaust emission measuring system

Emissions	*Range*	*Accuracy*
CO (%)	0-15	0.01 %
HC (ppm)	0-20.000	1 %
CO_2 (%)	0-20	0.1 %
NO_x (ppm)	0-5000	0.1 %

The experimental setup used in this study was schematically given in Fig 1. To sum up, this study was carried out on two different engines (CI and SI engines), four different fuel types (G100, G90E10, D100 and D90E10) and 5 different engine speeds (2250, 2500, 2750, 3000 and 3250). As a consequence, CO, CO_2, HC and NO_x emissions were experimentally obtained. The flow chart and experimental procedure of the study are summarized in Fig. 2. Prior to starting the tests, each engine type was operated for nearly five minutes to achieve a steady-state test condition. Each emission value was taken throughout 2 minutes after transiting the steady-state test conditions. The measured values from the tests in this study were repeated three times under the same conditions. This confirms the repeatability of the results under the same conditions and also to ensure the reality of the achieved results. Then the averages of these three measurements were given in the section of the results and conclusions of this study.

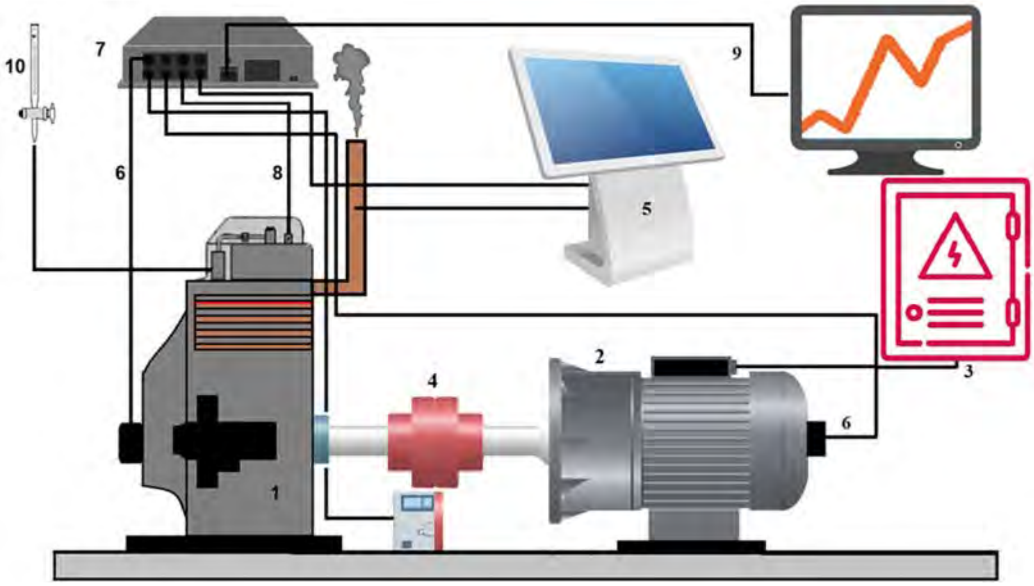


Fig. 1. Schematic view of the experimental setup 1. Test engine (first experiments in CI engine then replaced with SI engine) 2. Dynamometer 3. Control panel 4. Torque meter 5. Exhaust gas emissions analyzer 6. Encoder 7. Mainboard 8. Cylinder pressure sensor 9. Computer 10. Fuel consumption level

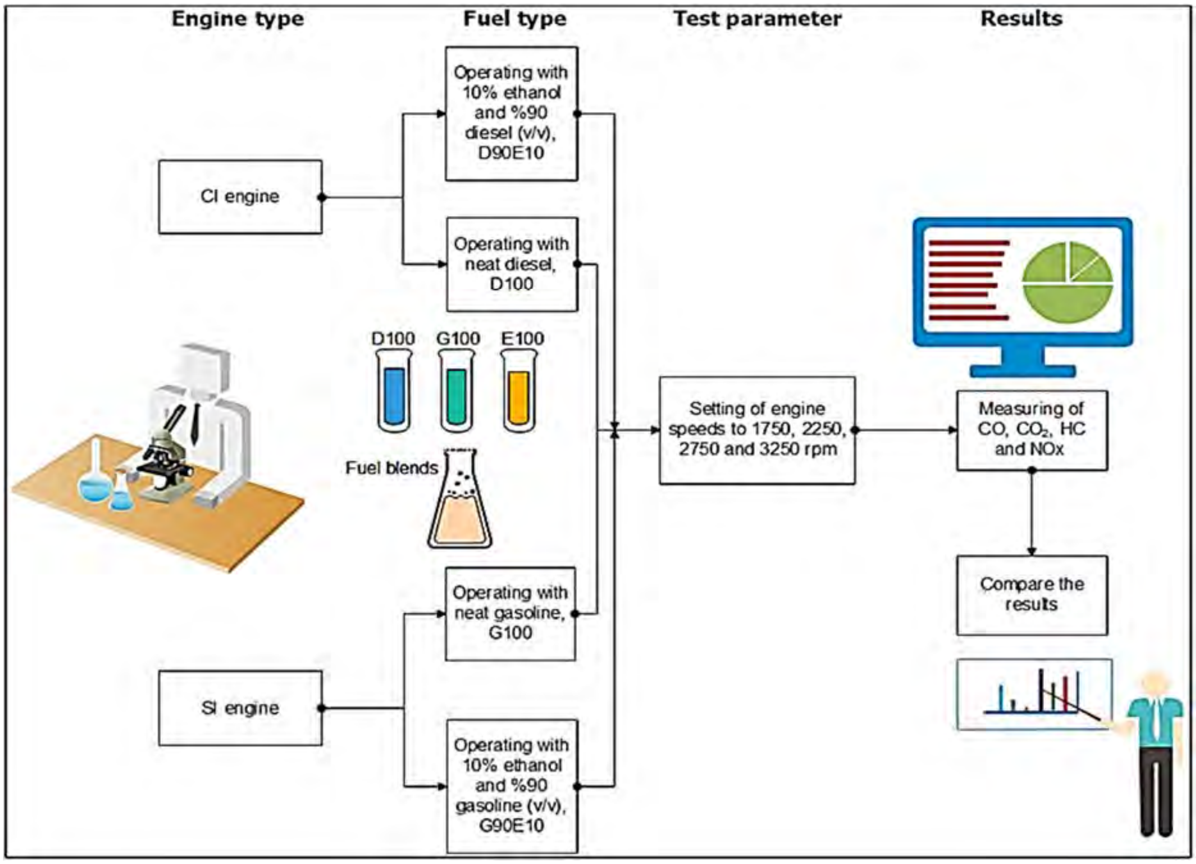


Fig 2. A brief view of the flow chart and experimental procedure steps of this study

3. Uncertainty analysis

The experiments cannot be measured by 100% accuracy owing to many factors. To minimize the impacts of these factors, it is suggested that the test in the experimental studies were conducted at least three times in order to ensure the reality of the obtained results and to confirm the repeatability under the same conditions, and then reported the averages of these three measurements (Ağbulut et al., 2018). Also, the uncertainty analysis is one of the most effective methods to determine and evaluate the obtained test results. Therefore, the accuracy rates of all measurement devices using the experimental section of this study are given in Table 4. Total uncertainty analysis in this paper was calculated using the following Eq. (11) (Ağbulut et al., 2019; Karagöz et al., 2020b; Khanlari et al., 2020a; Uluer et al., 2018).

$$W_R = \left[\left(\frac{\partial R}{\partial x_1} w_1\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 \ldots + \left(\frac{\partial R}{\partial x_n} w_n\right)^2\right]^{0.5} \quad (11)$$

where: W_R is the total uncertainty, %, w is the dimensional shape factor, R is the function uncertainty, and w_1, w_2, w_n represents the uncertainties in the independent variables (Güler et al., 2020; Khanlari et al., 2020b).

4. Result and discussion

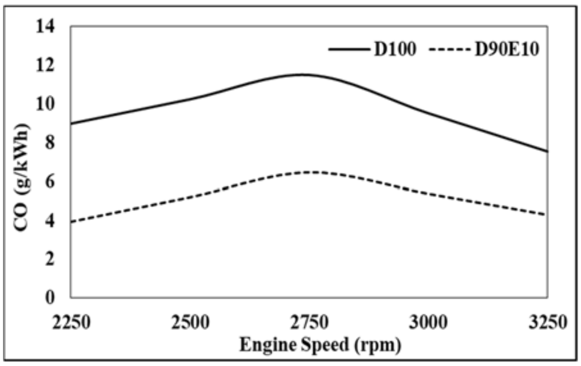
In this paper, the effects of ethanol addition to conventional diesel and gasoline fuels on a CI and SI engines exhaust emissions at variable engine speeds were investigated. Fig. 3 shows the effects of test fuels on CO emissions. CO is an incomplete combustion product and the most important factor affecting the incomplete combustion is the air-fuel ratio of the fuel blend. The incomplete blend of air and fuel, and the lack of time for the formation of O_2 and CO_2 triggers CO emissions. CI engines operate with a larger air-fuel coefficient than SI engines. Naturally, this case for CI engines results in lower CO emissions than SI engines. As shown in Fig. 3, less CO emissions were achieved in both engines at low and high engine speeds.

Low CO emissions at low engine speed depend on high volumetric efficiency and sufficient combustion time. As the increase at engine speed, both the volumetric efficiency and combustion time decreased. This phenomenon results in an increase in CO emission values. Furthermore, a better homogeneous mixture with the increment at engine speed provided and consequently improved the quality of the combustion process. This is also another reason why CO emissions reduced. Similar trends were reported by (Yesilyurt et al., 2020a;b).

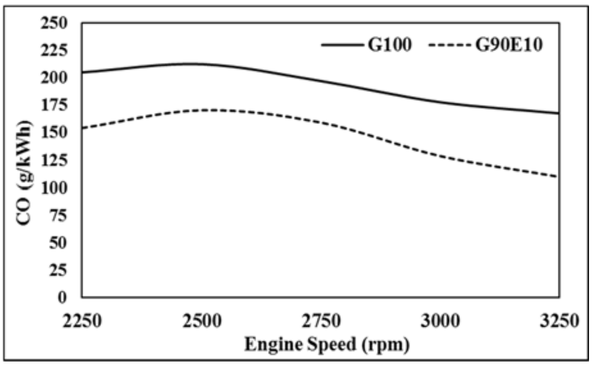
As compared with neat diesel and gasoline fuels, ethanol contains less carbon (C) and high O_2 content. The addition of ethanol into neat fuels, therefore, reduced CO emissions in both engines thanks to more complete combustion as compared to that of neat diesel and gasoline fuels. Carbon is directly converted to CO emission during combustion. However, the high latent heat of evaporation of ethanol and the high O_2 contained in it increase the volumetric efficiency by allowing the air required for combustion to cool during intake. Increased volumetric efficiency allows more O_2 to be taken into the cylinder for combustion. Due to these reasons, adding ethanol into neat diesel and gasoline fuels improved the combustion quality in the experiments and reduced CO emissions.

Generally, CO_2 emission is considered as a product of complete combustion owing to a sufficient amount of air in the air-fuel mixture and an adequate amount of time in the cycle for completion of the combustion process. Fig. 4 shows the changes in CO_2 emissions depending on the test fuels and various engine speeds. During the combustion process, hydrogen components convert into the water with the decomposition of the carbon and hydrogen components in the fuel. The carbon component, on the other hand, converts to CO_2 emissions if it finds sufficient oxygen during the combustion. Otherwise, it converts to CO emissions or smoke.

As shown in Fig. 4, the addition of ethanol into neat fuels reduces CO_2 emissions in the CI engine (Yesilyurt 2019), but increases it in the SI engine. Actually, CO_2 emission can be considered as a product of complete combustion and forms if a sufficient amount of air in the air-fuel mixture and a sufficient amount of time in the cycle for completion of the combustion process. Hereby, with sufficient time and sufficient oxygen for the chemical reactions of the combustion process inside the combustion chamber, the amount of CO_2 emission will be increased. Then, CO emissions will reach its maximum values. On the other hand, the low carbon-hydrogen and high oxygen content of the D90E10 compared to the D100 may be the reason why CO_2 emission reduced in the CI engine.

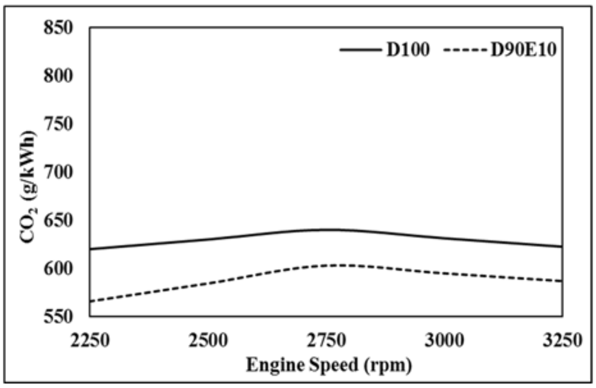


(a)

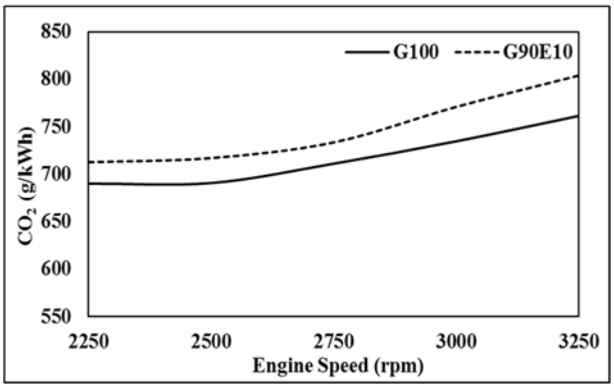


(b)

Fig. 3. CO emissions depending on different engine speed (a) CI engine (b) SI engine



(a)



(b)

Fig. 4. CO_2 emissions depending on different engine speed (a) CI engine (b) SI engine

Fig. 5 shows the effects of test fuels on HC emissions depending on the various engine speeds. The reason for the formation of HC emissions is that the combustion due to insufficient air required for combustion in the cylinder is not fully occurred. Since diesel engines operate with more air, HC emissions are very close to each other at all speeds. The increase in engine speed and turbulence in the combustion chamber and the increase in combustion temperature increased the combustion quality of the SI engine and reduced HC emissions. As the temperature inside the cylinder accelerates the final reactions that are important for the formation of HC emissions, and HC emission was reduced. The high oxygen content of ethanol has reduced HC emissions at all engine speeds in both engines.

Fig. 6 shows the impacts of test fuels on NO_x emissions depending on the different engine speeds. The three important factors that trigger NO_x emission are temperature, oxygen rate and time, respectively. This phenomenon increases NO_x emissions because quality combustion will increase post-combustion pressure and temperature. However, due to the lack of sufficient oxygen during combustion in rich mixtures and the slow combustion in poor mixtures, the reduction in post-combustion temperature and pressure reduces NO_x emissions. As shown in Fig. 6, ethanol reduced NO_x emissions for the CI engine and increased for the SI engine. The addition of ethanol into the conventional diesel fuel also reduced the lower thermal value of the fuel mixture and lowered the post-combustion temperature. Therefore, lower NO_x values were obtained in all cycles compared to D90E10 and D100. Similarly, the addition of ethanol into conventional gasoline fuel helped to reduce the lower thermal value of the fuel mixture. However, the ethanol additive into gasoline fuel increased the oxygen-to-fuel ratio in the fuel-rich regions. The most important parameter affecting NO_x emission is clearly the relative air-fuel ratio. The actual air-fuel ratio approaches to stoichiometric as G90E10 increases, and consequently, combustion becomes complete. This complete combustion helps to increase the in-cylinder temperature as well as NO_x emission whilst the HC emission decreases. Therefore, more NO_x emissions were achieved with G90E10 fuel than G100 fuel type at all engine speeds.

To sum up, the change percentages of the emissions depending on the reference fuels (D100 and G100) is separately given in Table 5. As clearly seen in Table 5, the addition of ethanol caused a decrease in all emission values. On the other hand, the addition of ethanol to gasoline increased some emission values.

Fig. 7 depicts the average changes of all emissions for two fuel types in Table 5. The highest reduction was achieved by HC emission for D90E10 fuel type. Then, CO, CO_2, NO_x emissions followed HC emission in terms of the highest decreased emissions in D90E10 fuel type.

Similar to D90E10, for G90E10 fuel type, the highest reduction in emissions was achieved by HC emission. Then, CO emission is the other and last reduction emission type for G90E10 fuel type. Except for HC and CO emissions, the values of other emissions (CO_2 and NO_x) increased.

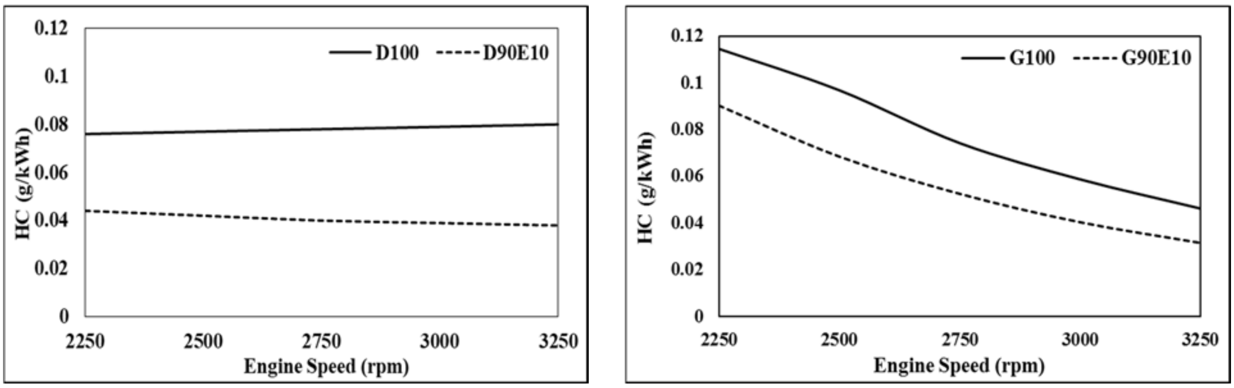
5. Conclusions

This study mainly aims to highlight particularly determining the more appropriate fuel type by which the addition of ethanol has converted into more environmentally friendly. The following conclusions in this paper are drawn based on the experimental results:

- Blending ethanol into neat diesel and gasoline fuels is a fairly simple process and has the big potential to highly change exhaust emission values in the engines.
- The increasing and decreasing rates for CI and SI engines highly differ from each other. The reason of that is the physical and chemical properties of the fuels.
- CI engines run with excess air. This phenomenon caused that CI engine emits less CO emissions than that of SI engine.
- The highest reductions are seen in HC and CO emissions for D90E10 fuel type and these reductions are equal to 47.9 % and 47.0 %,

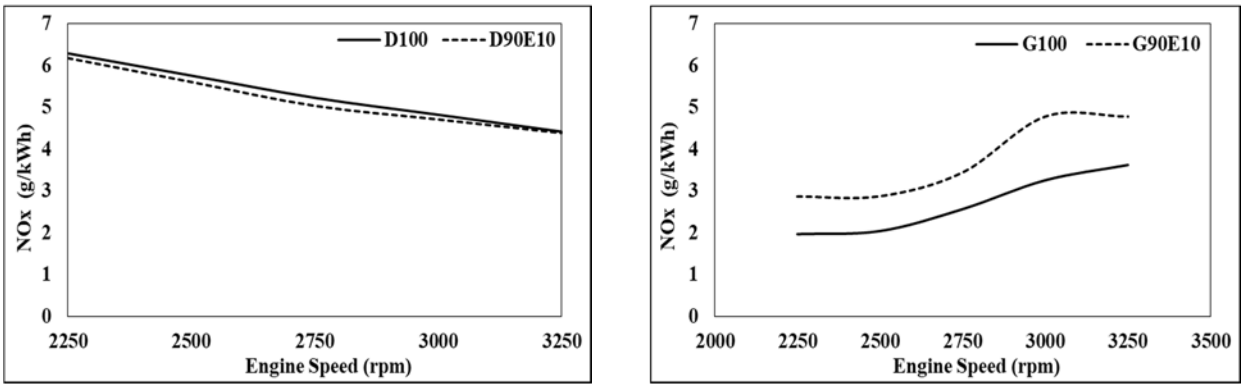
respectively. On the other hand, the highest two reductions for G90E10 similarly are seen in HC and CO emissions and these reductions were equal to 28.5 % and 25.1 %, respectively.

- The addition of ethanol resulted in a reduction of 2.3% in NO_x with respect to diesel fuel and an increase of about 40% in NO_x with respect to gasoline fuel.
- Ethanol additive into neat gasoline fuel caused an increase in CO_2 and NO_x emissions. On the other hand, ethanol additive into neat diesel fuel caused only an increase in CO_2 emission.
- This paper is reporting that the addition of ethanol has presented very positive results in CI engine in terms of exhaust emissions in comparison to those of SI engine.



(a) (b)

Fig. 5. HC emissions depending on different engine speed (a) CI engine (b) SI engine



(a) (b)

Fig. 6. NOx emissions depending on different engine speed (a) CI engine (b) SI engine

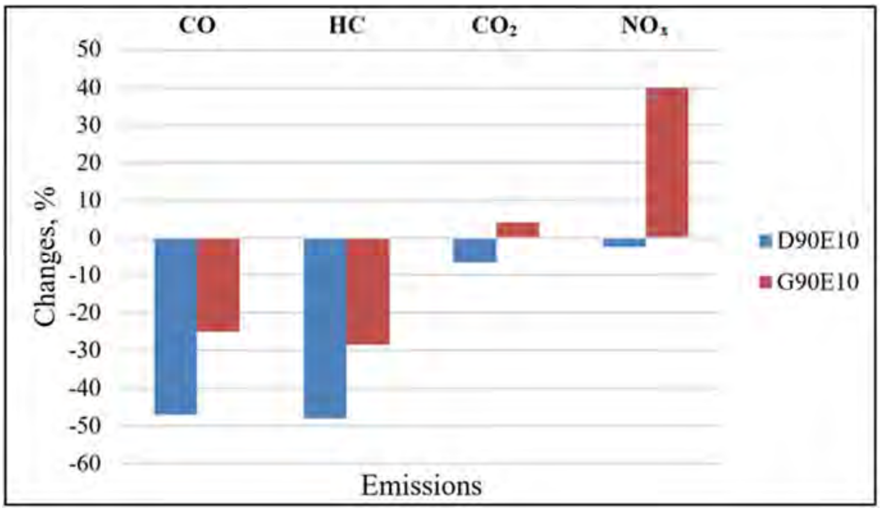


Fig. 7. Percentage **c**hanges on emission values for D90E10 and G90E10 compared to D100 and G100 fuel types

Table 5. Average variation percentages of the emissions depending on the reference fuels (neat diesel and gasoline fuels)

Fuel type	***Engine type***	***Average variation by emission***				***Overall Uncertainty, %***
		CO, %	***HC, %***	***CO_2, %***	***NO_x, %***	
D90E10	CI Engine	-47.0↓	-47.9↓	-6.6↓	-2.3↓	1.01
G90E10	SI Engine	-25.1↓	-28.5↓	4.1↑	39.8↑	1.01

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