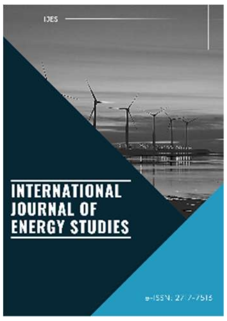
INTERNATIONAL JOURNAL OF ENERGY STUDIES

e-ISSN: 2717-7513 (ONLINE); homepage: https://dergipark.org.tr/en/pub/ijes



Research Article

Int J Energy Studies 2023; 8(4): 859-878

DOI: 10.58559/ijes.1376612

Received	:	16 Oct 2023
Revised	:	20 Nov 2023
Accepted	:	21 Nov 2023

The effects of E80D20 ethanol-diesel blend on combustion and exhaust emissions in SI engine

Esenay Arslan[a]*, Mehmet İlhan İlhak[b]

[a] *Kayseri University, Department of Electricity and Energy, Kayseri, Turkiye, ORCID: 0000-0001-5429-5471*

[b] *Niğde Ömer Halisdemir University, Department of Mechanical Engineering, Niğde, Turkiye, ORCID: 0000-0003-1591-6161*

*(*Corresponding Author:esenayarslan@kayseri.edu.tr)*

Highlights

- E80D20 fuel mixture was investigated in SI engine.
- The mixture of E80D20 provided 15% reduction in BSFC compared to the use of pure ethanol at 25Nm.
- The use of E80D20 in lean mixture decreased HC and NO_x emissions.

You can cite this article as: Arslan E, Ilhak MI. The effects of E80D20 ethanol-diesel blend on combustion and exhaust emissions in SI engine. Int J Energy Studies 2023; 8(4): 859-878.

ABSTRACT

One of the renewable fuels is ethanol, which is widely used in internal combustion engines. Ethanol is produced from renewable sources such as sugar cane, corn, potato, and biomass. It has high octane number, however, lower calorific value than that of gasoline and diesel. Since ethanol is a corrosive fuel, it cannot be used completely pure, so it is used as a mixture in internal combustion engines. Therefore, ethanol was mixed with diesel fuel to both eliminate its corrosive effect and increase its calorific value and used to examine engine performance and exhaust emissions in an SI engine at partial loads in this study. Four-stroke and four-cylinder test engine was used, and the experiments were carried out at constant speed of 2000 rpm, at 25 Nm and 50 Nm load, and with different excess air ratios (λ). The fuel mixture used in experimental studies was set as 80%Ethanol+20%Diesel (E80D20). To see the effect of the E80D20 mixture more clearly, the same experiments were also repeated with pure gasoline and pure ethanol, and these three fuel conditions were presented comparatively. At 25Nm and λ= 0.9, the use of E80D20 resulted in a 15% reduction in BSFC compared to the use of pure ethanol. When emissions are considered, the use of E80D20 in lean mixture (λ=1.1) showed a decreasing trend in HC and NO_x emissions.

Keywords: Renewable fuels, Ethanol, Gasoline, Diesel, SI engine

1. INTRODUCTION

Ethanol is the most suitable alternative fuel to replace high octane fuels (gasoline) due to high octane number. However, it has a positive effect on exhaust emissions because it has a smaller molecular structure than diesel, contains oxygen, and does not contain sulphur, carcinogenic substances, and heavy metals as in diesel fuel. Fuels such as ethanol and methanol have higher octane numbers than gasoline. Therefore, alcohol-gasoline mixtures can be used in engines with higher compression ratios. As a result, higher thermal efficiency can be obtained in an alcohol mixture fuelled engine. Alcohol fuels have higher octane number and latent heat of evaporation than gasoline, resulting in achieving higher compression ratios as well as a denser air/fuel mixture [1]. With the addition of alcohols to diesel fuel, there are some chemical and physical changes in the properties of diesel fuel. In particular, the cetane number, viscosity, and lower calorific value decrease. There are some difficulties in the use of alcohols in diesel engines due to changing properties. To overcome these difficulties, different techniques are developed to ensure the compatibility of alcohol-containing diesel fuels with diesel engine technology. The use of ethanol in diesel engines has come to the fore in recent years since ethanol is a renewable fuel and has better mixing properties with diesel fuel. Ethanol-diesel fuel mixtures can be used up to 20% without the need for radical changes on the engine. An ultrasonic atomizer was used in a study to develop an SI engine running on diesel fuel [2]. They concluded that the applicability of this system used in two-stroke engines in commercial engines is possible. In studies in the literature on gasoline/diesel mixtures, the conversion of diesel engine to spark ignition engine has been discussed, and the use of fuels that can be used in gasoline engines in dual fuel mode with diesel has been the focus of attention. Zapata-Mina et al. [3] used the different fuel condition in diesel engine converted to spark ignition engine to investigate exergy analysis. In addition to diesel and ethanol fuels, a modified engine was tested by using a gasoline/ethanol mixture at different compression ratios and engine speeds. The 50% gasoline/ethanol mixture showed positive results in terms of exergy efficiency. Du et al. [4] tested the mixture of gasoline/diesel in CRDI engine. The effects of this mixture on combustion and emissions were investigated under different test conditions. The increase in the amount of gasoline in the mixture prolonged the combustion period and contributed to the reduction of nitrogen oxide and soot emissions. Liu et al. [5] studied on cold-start control strategy by using two stroke engine diesel fuelled with spark-ignition. Significant engine parameters were considered in these tests. It was concluded that the correct amount of fuel injection as well as the adjustment of the ignition timing can improve the start-up and warm-up processes. Ning et al. [6] studied the combustion characteristics of a methanol fuelled direct injection spark ignition (DISI) engine using lean mixture (at high EAR values). Tests at constant rpm and wide injection timing range showed that ITE increased and all emissions, including NO_x, decreased as EAR increased. Lee et al. [7] tested the effects of using different ratios of ethanol in a diesel engine on performance and emissions. While the increase in the amount of ethanol reduced NO_x emission, it caused a decrease in ignition energy at min load and sudden pressure increases at max loads. Jamrozik et al. [8] studied the effects on engine performance and emissions and fuel consumption by adding gasoline and ethanol to diesel fuel separately. The use of ethanol had an increasing effect on thermal efficiency and IMEP. Although the increase in the amount of ethanol and gasoline did not cause a significant change in HC emissions, it was effective in reducing CO emissions. In their study, Chen et al. [9] numerically analysed the effects of diesel use in a direct injection spark ignition engine. Using diesel in SI mode caused higher knocking than gasoline but better results were obtained in terms of performance with diesel fuel at low EAR. Wang et al. [10] aimed to characterize the change in PN size by examining the combustion of diesel/gasoline mixture in premixed and compression ignition mode. As the gasoline ratio in the mixture increased, the total particle number increased, but the particle size decreased. It was also seen that EGR, and injection pressure had a slight effect on particle size. Kaleemuddin and Rao [11] converted the diesel engine to a gasoline engine and aimed to reach emission standards by using LPG and CNG fuels. Ignition timing has also been considered to obtain the best performance from gaseous fuels. LPG was effective in reducing CO emissions, and CNG in reducing HC, CO_2, and NO_x emissions. Liu et al. [12] investigated the effects on performance, fuel consumption and emissions by using diesel in a two-stroke spark ignition engine. The low load condition is considered according to parameters such as ignition advance and EAR. It was concluded that the ignition advance reduces HC emissions, but the best ignition time for power and fuel consumption is 60 CA BTDC. The use of mixtures at high EAR ratios also resulted in a reduction in CO and HC emissions. Zhang et al. [13] experimentally investigated basic performance parameters such as emissions and efficiency in a gasoline engine using both CTL and diesel. The use of CTL has improved ITE compared to diesel use, and at the same time, its use with gasoline has been effective in reducing particulate emissions. Liu and Dumitrescu [14] converted diesel engine into spark ignition mode fuelled with natural gas and analysed flame properties in detail. Combustion images taken from the optical engine gave information about the flame distribution and turbulence in the combustion zone. It was also concluded that with this type of engine, higher cylinder pressure and less knocking can occur. There are also studies in the literature considering different ethanol ratios and engine parameters. The power, efficiency and emission parameters of the diesel engine were examined in experiments carried out by adding ethanol to diesel at different rates. According to the test results performed at different loads and two different engine speeds, as the percentage of ethanol contribution to diesel increased, BTE increased and BSFC decreased. While ethanol addition improved CO emissions, it caused an increase in NOx emissions [15]. Iodice et al. [16] conducted a review study on the effects of ethanol/gasoline mixtures on NOx emissions of the SI engine. In tests conducted between 2000-4500 rpm, the NOx emissions of the mixture containing 30% ethanol remained at lower levels than diesel and mixtures containing lower amounts of ethanol. It has been shown that ethanol-gasoline mixtures containing 50% and 85% ethanol by volume provide significant advantages over gasoline in NOx emissions. In the study where the diesel-ethanol mixture in an SI engine was tested under lean conditions, features such as cylinder pressure, HRR, emissions and thermal efficiency were examined. It has been stated that in lean conditions, that is, when the excess air coefficient is above 1.0, the gasoline-ethanol mixture increases cylinder pressure and thermal efficiency and reduces NOx and HC emissions [17]. In the study conducted to compare the environmental aspects of the use of ethanol in SI and CI engines, 10% ethanol was added to diesel and gasoline separately and tested under the same conditions. In tests conducted in the 2250-3250 rpm range, the addition of ethanol to conventional fuels reduced HC and CO emissions in both SI and CI engines, while it increased CO2 and NOx emissions in SI engines [18]. Tutak et al. [19] added methanol and ethanol (E85) to diesel fuel and examined the effects of different additive rates and fuel energy percentages on basic performance parameters. As a result of the experiments carried out at full and partial loads, the changes in pressure, HRR and emissions are given comparatively. Huang et al. [20] examined the effects of ethanol and n-butanol additives in a diesel engine at different engine loads from low to high and at speeds of 1500-2000 rpm. The ethanol contribution is in the range of 10-30%, and 5% n-butanol was used in the mixture to prevent alcohol-diesel separation. While 30% ethanol additive caused the highest BSFC among the test fuels, it also increased HC emissions. In an experimental study conducted by adding 5% and 10% ethanol to diesel fuel in a heavy vehicle diesel engine, performance and exhaust emissions were examined. As a result of the tests performed at two different engine speeds and partial loads, the NOx emissions of ethanol-added diesel and pure diesel were almost similar, but the ethanol addition provided a decrease in CO emissions. While a slight increase in BSFC was observed with the addition of ethanol, an increase in BTE was also achieved [21].

As seen in the literature review, studies on the use of ethanol/diesel mixture in gasoline engines are limited. In this study, the effect of diesel-ethanol mixture on engine performance and emissions in a spark ignition engine was investigated. The main reason for choosing this mixture is to test the interaction of ethanol, which is an alcohol derivative with high octane number, with diesel fuel, which has a disadvantage in terms of viscosity and octane number in spark ignition engines. One of the reasons for adding diesel to ethanol is to evaluate the high energy density advantage of diesel and it has also a lubricating property. On the other hand, the purpose of the ethanol and diesel mixture is to reduce the corrosive effect of ethanol and to increase the calorific value of the mixture with diesel fuel. At the same time, it has been observed in which EAR range the created mixture can work in the engine. In the studies in the literature, diesel-ethanol mixtures have found more application areas in diesel engines. In this study, it is aimed to evaluate the working range and performance of the mixture in spark ignition engine.

2. MATERIAL AND METHOD

2.1. The Experimental Setup

The view of the experimental setup used the studies is displayed in Figure 1. In addition, in the experimental setup, there is an electronic card that can change the ignition advance independently from the engine electronic control unit, and an electronic card system that can change the opening time of the fuel injectors between 0-20 milliseconds for the flow rate adjustment of the liquid fuel. Commands are given to these electronic cards via computer interface programs.

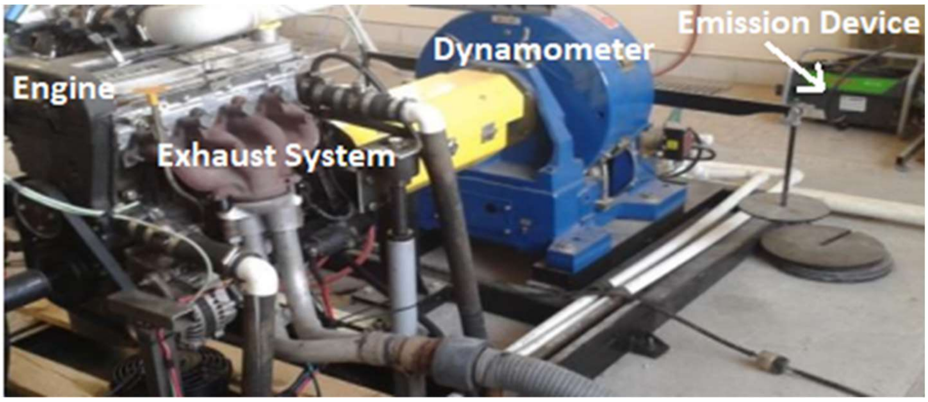


Figure 1. View of the experimental setup

In this study, FORD MVH 418 spark-ignition gasoline engine connected to the experimental setup was used. The technical data of the 4-stroke, water-cooled, electronically controlled and common rail gasoline engine are given in Table 1.

2.2. Test Method

In this study, the test engine was operated with E80D20, and the engine performance and emission data obtained as a result of the experiments were analysed comparatively. The experiments were carried out at partial loads of 25 Nm and 50 Nm, and since the engine did not operate stably with the E80D20 mixture at loads above 50 Nm, the engine load was limited to this value. Before starting the experiments, the test engine was run until it reached 90 ºC regime temperature.

Table 1. Technical data of the Ford MVH 418 engine [22]

Brand	Ford
Model	MVH 418
Number of cylinders	4
Stroke volume	1796 cm^3
Bore	80.6 mm
Stroke	88 mm
Compression ratio	10:1
Maximum BMEP	10.7 bar
Maximum power	75 kW / 5500 rpm
Maximum torque	150 Nm / 4000 rpm
Idle speed	800 rpm
Maximum speed	6000 rpm
Minimum fuel consumption at full load	268 g/kWh @3000 rpm
Dimensions (H x W x L)	673 x 559 x 628 mm
Weight	123 kg

In the experiments, the flow rate of the mixture fed to the engine as fuel was adjusted by changing the liquid fuel line pressure with the throttling valve and setting the injectors opening time in milliseconds. In the experiments, speed, torque, fuel flow, cooling water inlet and outlet temperatures, air temperature, air pressure, in-cylinder pressure change, exhaust gas temperature and emissions were measured and used to determine engine performance and emission values. In-cylinder pressure changes were taken as the average of 100 cycles. E80D20 was prepared with 80% ethanol (96% purity) and 20% diesel fuel using volumetric measuring cups. Ethanol in diesel fuel causes phase separation as seen in Figure 2.

a) After mixed

b) After waiting

Figure 2. Ethanol-Diesel blend

Therefore, the mixture was stirred continuously with the fuel pump return to prevent phase separation during the experiment. The properties of each fuel that make up the mixture are given in Table 2 [23, 24].

Table 2. Thermophysical properties of fuels used in the mixture

Properties	**Gasoline**	**Ethanol**	**Diesel**
Chemical formula	$C_4 - C_{12}$	C_2H_5OH	$C_8 - C_{20}$
Auto ignition temperature (°C)	257	363	254
Density (kg/m^3)	720-780	790	820-860
Octane number	95	108	45-50
Boiling point(°C)	30-225	79	210-235
Min. ignition energy (MJ)	0.23	0.23	-
Calorific value MJ/kg	43	26.7	42.5

By using the data obtained from the experiment, the mole numbers were found by the combustion equation in the calculation of the emission values (CO, CO_2, NO_X, and HC) as g/kWh. Emission values obtained using the Equations (1), (2), (3), (4) [25] are presented in the graphics. In the equation, SFC is specific fuel consumption, M_f is molar mass of fuel and n_x is number of moles of gases.

$$NO(\frac{g}{kWh}) = \frac{n_{NO} \times M_{NO}}{n_f \times M_f} \times SFC \quad (1)$$

$$CO(\frac{g}{kWh}) = \frac{n_{CO} \times M_{CO}}{n_f \times M_f} \times SFC \quad (2)$$

$$CO2(\frac{g}{kWh}) = \frac{n_{CO2} \times M_{CO2}}{n_f \times M_f} \times SFC \quad (3)$$

$$HC(\frac{g}{kWh}) = \frac{n_{HC} \times M_{HC}}{n_f \times M_f} \times SFC \quad (4)$$

2.3. Uncertainty Analysis

Experimental set and measurement tools may cause errors in experimental studies. It is possible to calculate the error analysis of experimental results with uncertainty analysis. To calculate it, a dimension called R is measured in the system and n independent variables (x_1, x_2, x_3, ……, x_n) affect R. Hence, the Equation (5) can be written as follows;

$$R = R(x_1, x_2, x_3, \dots\dots\dots, x_n) \quad (5)$$

The w_1, w_2, w_3, …, w_n are error rates for each independent variable are and the error rate for R is w_R. In this case, the Equation (6) can be written as follows [26];

$$w_R = \left[\left(\frac{\delta R}{\delta x_1} w_1\right)^2 + \left(\frac{\delta R}{\delta x_2} w_2\right)^2 + \cdots + \left(\frac{\delta R}{\delta x_n} w_n\right)^2\right]^{1/2} \quad (6)$$

Using the experimental data, the uncertainties of the measured parameters were calculated with Equation 6, and by taking the uncertainties of each parameter into account, the total uncertainty in the experiment was obtained as 2.2%. The characteristics and accuracy of the test instruments used in the measurement of cylinder pressure, fuel consumption, emissions, etc. are given in Table 3.

Table 3. Accuracies of instruments used in the experiment

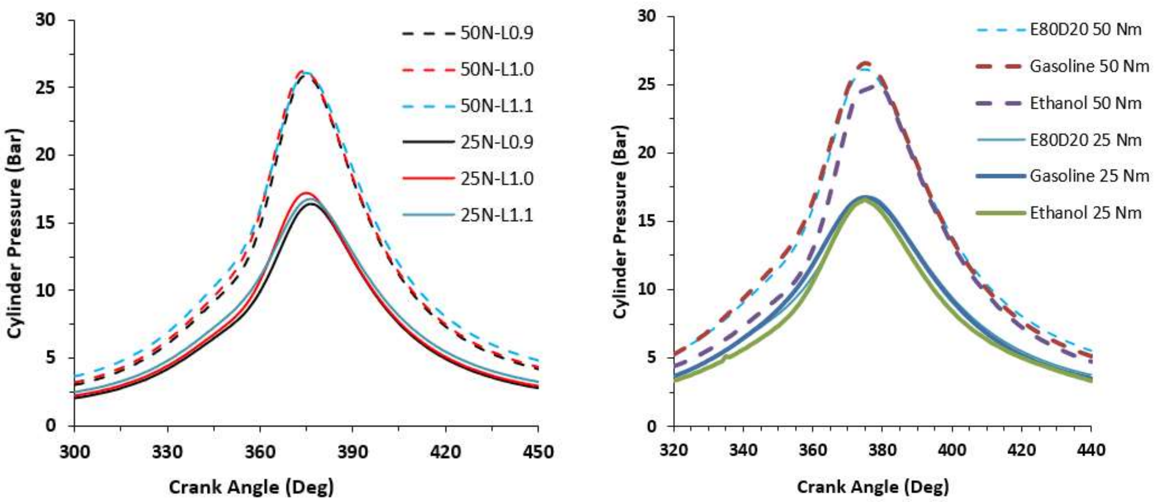
Instruments	Brand Name	Range	Accuracy
Pressure transducer	PCB 113B22	0-5000 psi	± 1% of actual reading
Liquid mass flow meter	Krohne	1.2-130 kg/h	± (0.1% of actual measured flow rate + zero stability)

Dynamometer	(SAJ SE 150)	0-500 Nm	± 0.25% of FS ± 1 rpm
Exhaust gas analyser			
CO	Bosch BEA 060	0-10% Vol.	±0.06% vol.
CO_2		0-18% Vol.	±0.5% vol.
O_2		0-22% Vol.	±0.1% vol.
HC		0-9999 ppm	±0.5%
NO		0-5000 ppm	±0.12%

3. RESULTS AND DISCUSSIONS

The findings obtained from the experiments carried out with a diesel-ethanol mixture at different EAR (λ=0.9, λ=0.95, λ=1.0, λ=1.05, λ=1.1) in a spark ignition test engine, are discussed and presented graphically. During the experiments carried out at 2000 rpm, two different torques were applied to the motor shaft by means of an electric dynamometer, and the optimum ignition advance was adjusted manually with an electronic card and computer software added to the experimental setup to give the maximum torque. In conditions where the load and engine speed are kept constant, the EAR is adjusted by interfering with the fuel flow and the position of the throttle. Experiments were started with a rich mixture with an EAR (λ=0.9) and completed with a lean mixture at λ=1.1.

Figure 3a shows the variation of the in-cylinder pressures with respect to the crankshaft angle, obtained as a result of experiment with ethanol-diesel mixture at 2000 rpm and 25 Nm and 50 Nm constant torque for λ=0.9, λ=1.0 and λ=1.1. Maximum peak pressure values were obtained under stoichiometric mixture conditions (λ=1.0) for both torque values, and peak pressures occurred around ATDC 15 CA. Figure 3b shows the comparison of the ethanol-diesel mixture for λ=1.1 with ethanol and gasoline only, in terms of pressure. Although the pressure is high at high torque, it is seen that the pressure value obtained with the E80D20 fuel mixture is very close to the pressure value obtained by using gasoline. With the E80D20 fuel mixture, approximately 6% higher cylinder pressure was obtained than using ethanol alone. The reason why the pressure of ethanol is lower than gasoline and E80D20 is due to the high latent heat of ethanol fuel. This reduces the temperature of the air taken into the cylinder and causes the pressure released as a result of combustion to decrease. On the other hand, as seen in Table 2, the high calorific value of diesel fuel causes the calorific value of the E80D20 mixture to increase, causing the cylinder pressure value to be close to that of gasoline. According to a study on an SI engine, the cylinder pressure value of ethanol fuel is higher than that of gasoline, and an average cylinder pressure of around 25-30 bar was obtained under stoichiometric conditions [24].



a) Pressure variation for E80D20 b) Pressure variation for test fuels (λ=1.1)

Figure 3. In-cylinder pressure variation versus to CA

The heat release rate (HRR) variation for three different fuel conditions and λ=1.1 is given in Figure 4. While there was a high heat release with ethanol by virtue of its contain oxygen component, the HRR values of using E80D20 blend and gasoline was close to each other. On the other hand, the oxygen concentration in ethanol caused more significant fluctuations in the HRR curve compared to other experimental fuels. Mixing ethanol with 20% diesel fuel reduced the HRR of ethanol, bringing it to approximately the same HRR as gasoline. When these three fuels are compared in terms of burning times, it is seen that there are no obvious differences. While the ethanol flame speed is 0.61 m/s, this value is around 0.4 m/s in gasoline [24]. Since the flame speed of ethanol is higher than that of gasoline, the HRR value is also higher than the HRR value of gasoline and the mixture. It has been stated [28] that the ethanol additive in the diesel engine shows a similar trend at partial loads.

In a typical spark ignition engine, the temperature of the exhaust gases is between 400-600 °C and may vary according to the engine speed, load condition, ignition advance and EAR [25,27]. The variation of the exhaust gas temperatures according to the EAR is shown in Figure 5. In all operating conditions, the temperature reaches its maximum value around λ=1.0 and λ=1.05. At 50 Nm engine load, the temperatures generally increased, and the maximum temperature was observed for the E80D20 mixture.

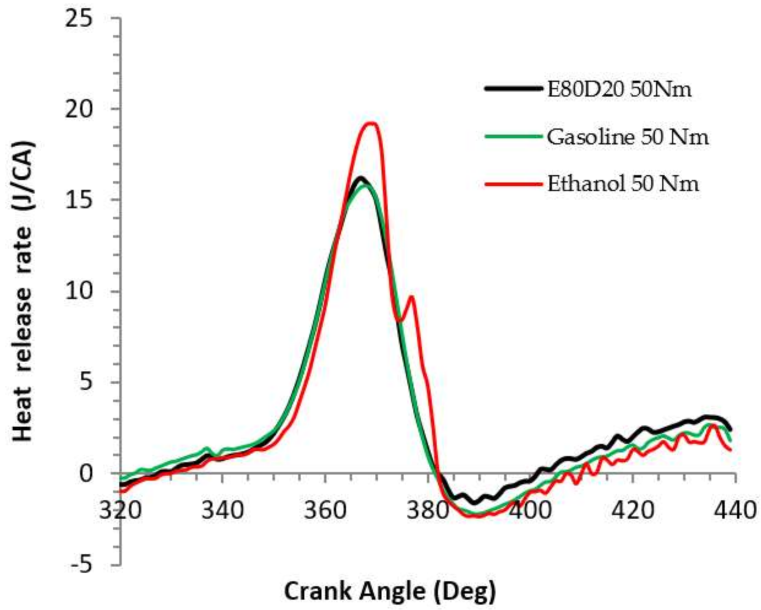


Figure 4. Heat release rate variation at 50 Nm torque

When the situation where the E80D20 mixture is used is compared in terms of engine load, there is a temperature difference of at least 17% and at most 27%. Since the evaporation pressure of diesel is lower than gasoline and ethanol, it is difficult to evaporate and will start to burn later in an SI engine. For this reason, the combustion of diesel will cause higher exhaust temperature as it shifts to exhaust timing. Similar trend in exhaust temperature change depending on engine load and ethanol ratio was also reported by Han et al [28].

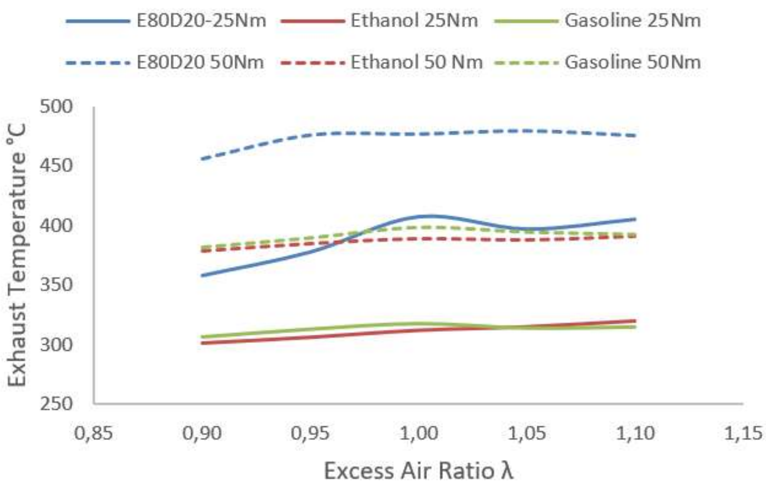
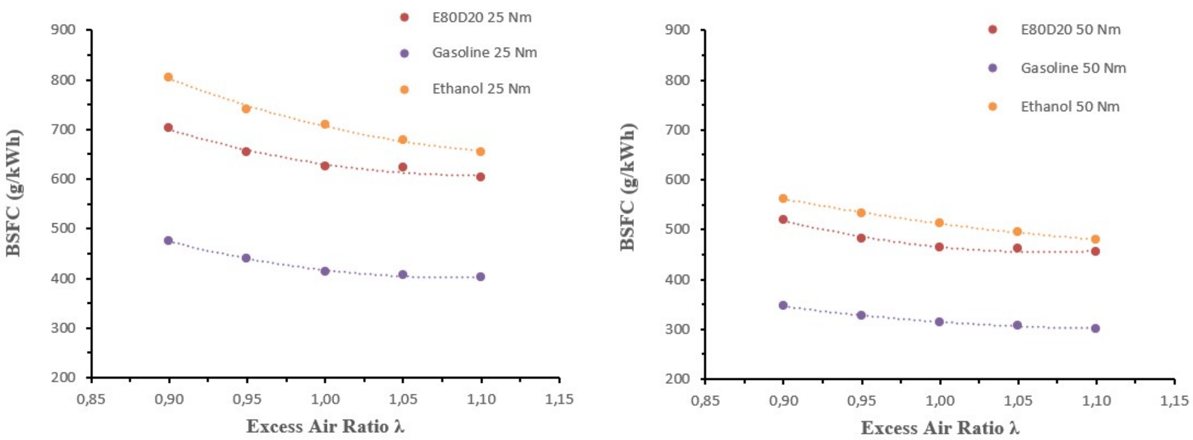


Figure 5. Variation of exhaust temperature versus to EAR

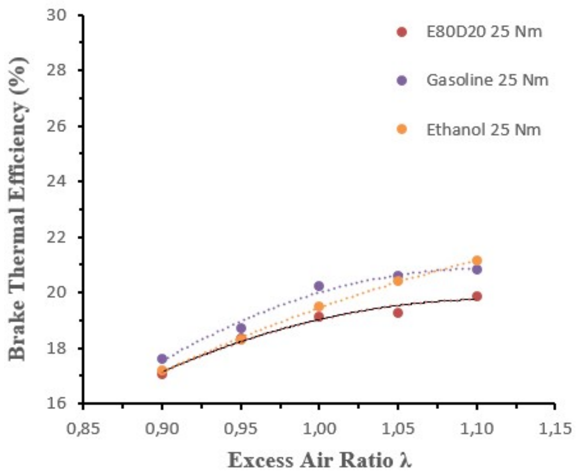
Figure 6 shows the variation of brake specific fuel consumptions according to the EAR. In the experiments, since the λ was increased by opening the throttle, the pumping losses decreased due to the increase in the manifold pressure and the BSFC decreased as seen in the Figure. In addition, the increase in engine load resulted in a decrease in fuel consumption. With the use of E80D20, BSFC increased compared to gasoline, but decreased compared to the use of ethanol alone. There is a maximum difference of about 15% in BSFC between E80D20 and ethanol at 25 Nm load. As can be seen from Table 2, among the test fuels, the fuel with the highest calorific value is gasoline and the one with the lowest is ethanol. Since ethanol has a low calorific value, more fuel will be consumed to deliver the same power, thus resulting in a higher BSFC. For this reason, the BSFC value of the E80D20 mixture has a value between gasoline and ethanol. Roso et al. [17] stated that ethanol additive in a SI engine increases the BSFC due to the difficulty of ignition compared to pure gasoline fuel. It has been stated [21] that in the CI engine, the ethanol additive provides a slight increase in BSFC compared to diesel due to its low calorific value.



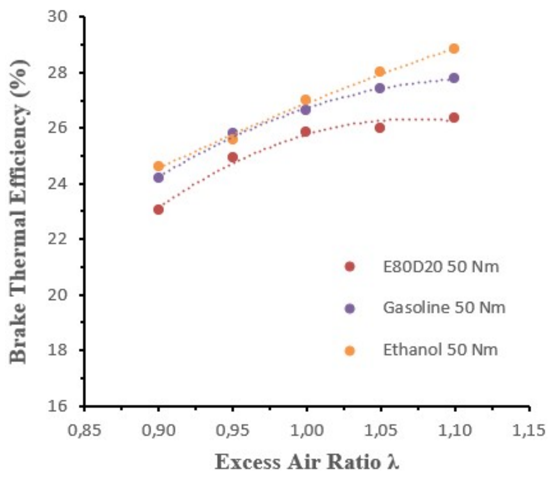
a) BSFC change at 25Nm load b) BSFC change at 50Nm load

Figure 6. Variation of brake specific fuel consumption versus to EAR

Thermal efficiency is defined as the ratio of the energy obtained from the engine to the thermal energy of the fuel. The variation of the brake thermal efficiency with the EAR is given in Figure 7. As can be seen in the figure, due to the increase in the EAR in all operating conditions, the complete combustion of the fuel with the excess air taken into the cylinder and the reduction of pumping losses have increased the brake thermal efficiency. The use of lean mixture at 50 Nm load and λ=1.1 resulted in the highest BTE values. The max. BTE was revealed in the use of ethanol with approximately 29%. Since the calorific value of ethanol is significantly lower compared to other test fuels, it is concluded that the BTE value of ethanol is higher, based on the ratio of engine power to the fuel calorific value in the BTE calculation. It has been stated that [17] the ethanol mixture increases the thermal efficiency of the SI engine due to its lower calorific value in lean mixture conditions compared to the stoichiometric condition. Rakopoulos et al. [21] stated that 5% and 10% ethanol addition to diesel provided a slight increase in BTE.



a) BTE change at 25Nm load

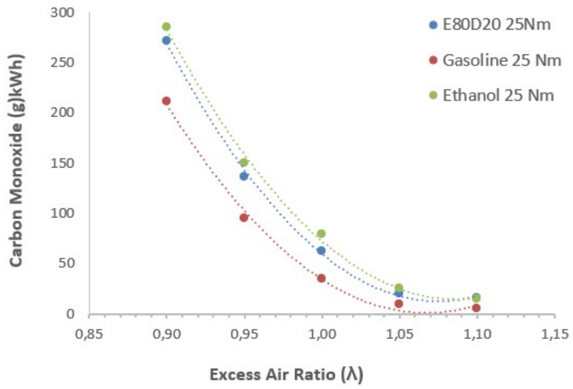


b) BTE change at 50Nm load

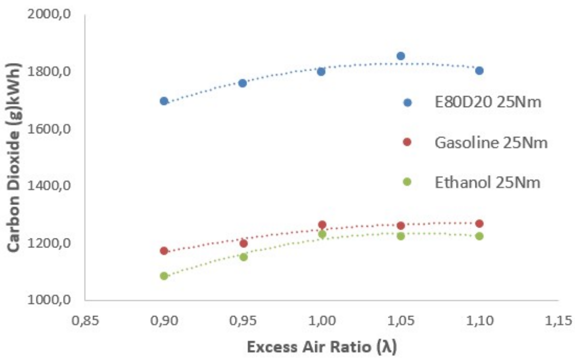
Figure 7. Variation of brake thermal efficiency according to EAR

The variation of carbon monoxide emission at 25 Nm load according to the EAR for three different fuel types is given in Figure 8a. CO emission is a strong function of lambda and decreases rapidly with increasing lambda as seen in the Figure. Since fuel consumption is high at 25 Nm torque, CO emissions are also higher at this torque value. After gasoline, the lowest CO emission was revealed for the E80D20. CO is a product of incomplete combustion and is the emissions released when fuel is not burned completely. The reason why ethanol and E80D20 fuels have higher CO emissions than gasoline can be attributed to the fact that diesel evaporates much slower than gasoline and therefore starts to burn later. This causes some of the fuel to be removed from the exhaust without being burned. In addition, Jie et al. [15] showed that as the amount of ethanol additive increases in a diesel engine, CO emissions increase and the reason for this is the high latent heat of ethanol and therefore the lower temperature.

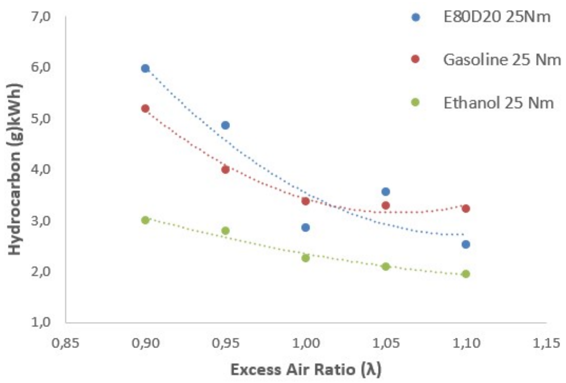
The variation of carbon dioxide emissions is given in Figure 8b. CO_2 is a product of complete combustion and its ratio in exhaust gases varies inversely with the ratio of CO and HC emissions. At the same time, a reduction in specific fuel consumption will normally lead to a reduction in CO_2 emissions. In the experiments carried out with a λ=0.9, CO_2 emissions were low because there was not enough oxygen in the cylinder for the complete combustion of the air-fuel mixture. A significant increase in CO_2 emission was observed for E80D20 compared to other fuel types. This is because the E80D20 mixture contains higher carbon content than gasoline and ethanol, and CO_2 emissions at the end of combustion are higher than others. Similar trend was reported by Agbulut and Sarıdemir [18].



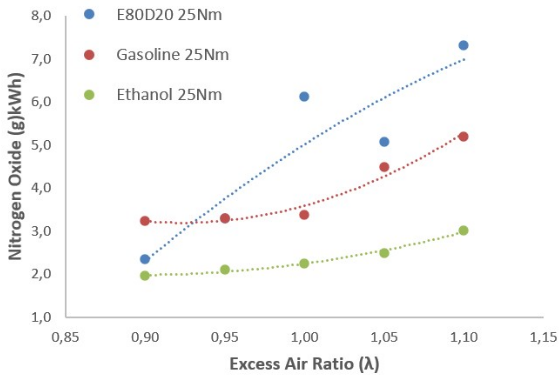
a) CO emission change



b) CO_2 emission change



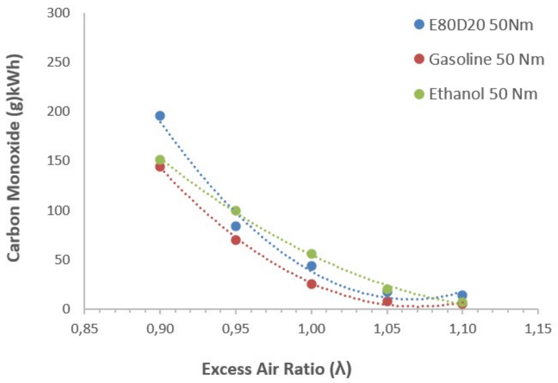
c) HC emission change



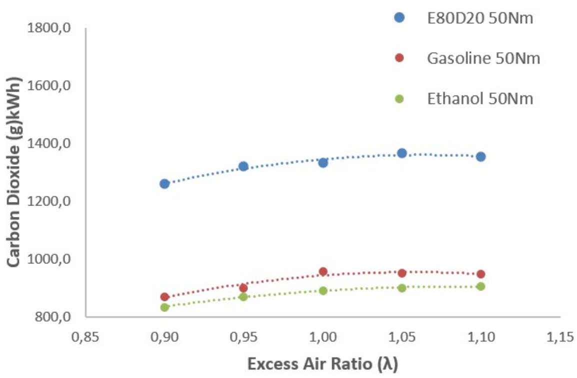
d) NO_x emission change

Figure 8. Variation of exhaust emissions versus to EAR at 25Nm

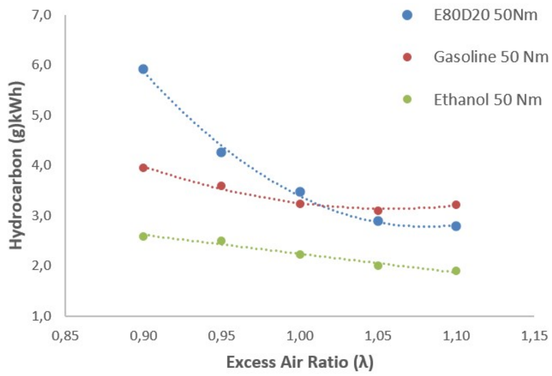
The variation of unburned hydrocarbon (UHC) emissions according to the EAR is given in Figure 8c. Unburned hydrocarbon emissions were high due to high fuel consumption and CO emission values at low load. For λ=0.9 and λ=0.95 values, that is, in rich mixtures, these emission values reach their maximum. HC emission was lower for E80D20 in lean mixture compared to gasoline. While alcohol fuel provides the lowest HC emissions since its combustion is better compared to other fuels due to the presence of oxygen in its structure, diesel fuel in the E80D20 mixture increased HC emissions due to late evaporation. The variation of nitrogen monoxide emissions according to the EAR is given in Figure 8d. Since there is no nitrogen (N_2) in the composition of diesel and ethanol, NO_x is not a fuel-related combustion product, but as a result of the reaction of nitrogen and oxygen molecules, which are the basic components of the intake air, with the effect of the in-cylinder temperature, the maximum NO_x emission is not at 1.0 λ, where the in-cylinder temperature is the highest. It is formed around λ=1.1 where free oxygen molecules are also present in the combustion products with the temperature. Figure 8d shows that the rate of increase in NO_x emissions decreases due to the decrease in in-cylinder temperatures due to excess air and reduced fuel consumption in lambdas higher than λ=1.05. As can be seen from the exhaust temperature graph, the high exhaust temperature of the E80D20 mixture containing diesel caused an increase in NOx formation at this temperature. On the other hand, in a study conducted with gasoline-ethanol mixtures containing 10%-30% ethanol in a gasoline engine, it was stated that NOx emissions decreased compared to gasoline [16].



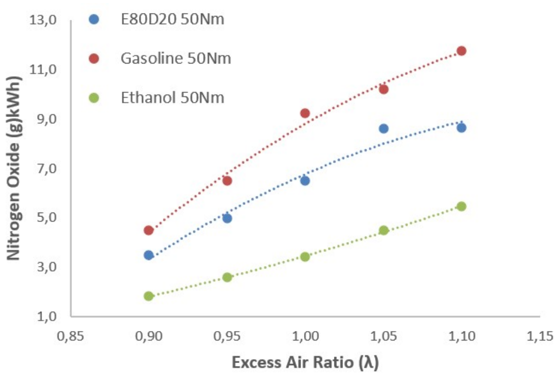
a) CO emission change



b) CO_2 emission change



a) HC emission change



b) NO_x emission change

Figure 9. Variation of exhaust emissions versus to EAR at 50Nm

In a literature study conducted on a diesel engine, it was stated that the oxygen in ethanol would increase NOx formation on the one hand, and on the other hand, it was stated that its high latent heat could reduce NOx emissions due to low temperature [15]. Figure 9 shows the effect of EAR on emissions at 50 Nm. Emission changes at 50 Nm tend to be similar with changes at 25 Nm, however the increase in load resulted in an overall reduction in emissions values. Although there are no significant differences in CO emissions between the three fuel alternatives, the E80D20 mixture has lower CO emissions than ethanol and higher than gasoline under stoichiometric conditions (Figure 9a). When the same EAR values are considered, the CO_2 emission levels at 50 Nm have decreased and maximum values have been observed in the E80D20 mixture depending on the combustion state and the amount of carbon in the fuel composition (Figure 9b). In HC emissions, E80D20 mixture showed better results than gasoline in case of lean mixture (λ=1.1) combustion (Figure 9c). The NO_x levels increased for all three fuel conditions at 50 Nm load, but unlike the 25Nm load, the maximum NO_x values occurred in the use of gasoline. For the λ=1.1, there was a 36% difference in NO_x emission between gasoline and E80D20 (Figure 9d). Agbulut and Sarıdemir [18] stated that ethanol additive reduces CO and HC emissions and increases NOx and CO_2 emissions in the SI engine. The lower carbon amount of ethanol in low CO emissions and the decrease in HC emissions are attributed to the oxygen level in ethanol.

4. CONCLUSION

This study investigated emissions and performance of E80D20 at 2000 rpm engine speed and partial loads at an SI engine. With the E80D20 fuel mixture, approximately 6% higher cylinder pressure and slightly lower BSFC were obtained than using pure ethanol. E80D20 mixture caused an increase in exhaust emissions compared to ethanol in almost all EAR. As a result of the experiments carried out, it was concluded that the use of ethanol-diesel mixtures in SI engines is not practical, considering the increase in exhaust emissions and the phase separation problem.

The results obtained in the study can be generally summarized as follows:

- The cylinder pressure of the E80D20 mixture was higher than that of pure ethanol, and this was caused by the high latent heat of ethanol. This situation has been partially prevented by adding diesel to ethanol.
- Since the evaporation pressure of diesel is lower than gasoline and ethanol and therefore it starts to burn later in the SI engine compared to other test fuels, the E80D20 mixture increased exhaust temperatures.

- Among the test fuels, the fuel with the highest calorific value is gasoline, the lowest is ethanol, and since the calorific value of the fuel increases thanks to the diesel component in the E80D20 mixture, the BSFC value is lower than pure ethanol.
- While approximately 26% BTE was obtained with E80D20 at 50 Nm load and lean mixture condition, a maximum of 29% BTE was achieved with ethanol due to its low calorific value compared to diesel and the mixture.
- Since the diesel in the E80D20 mixture evaporates much slower than gasoline and therefore starts to burn later, it caused higher CO emissions in the mixture at both engine loads.
- When all three test fuels were compared, the higher carbon content of the E80D20 mixture compared to gasoline and ethanol resulted in increased CO_2 emissions at the end of combustion at both loads and in the lean mixture condition.
- As stated in CO emissions, due to the evaporation problem of the diesel additive in the E80D20 mixture, the HC emissions of the mixture were higher than other test fuels.
- While ethanol had the lowest value in NOx emissions due to its low calorific value and therefore low exhaust temperature, an increase was observed in the NOx emissions of gasoline and the E80D20 mixture containing diesel at both engine loads.

NOMENCLATURE

ATDC	After Top Dead Centre	EGR	Exhaust Gas Recirculation
BMEP	Brake Mean Effective Pressure	HC	Hydrocarbons
BSFC	Brake Specific Fuel Consumption	HRR	Heat Release Rate
BTDC	Before Top Dead Centre	IMEP	Indicated Mean Effective Pressure
BTE	Brake Thermal Efficiency	ITE	Indicated Thermal Efficiency
CA	Crank angle	LPG	Liquefied Petroleum Gas
CNG	Compressed Natural Gas	N_2	Nitrogen
CO	Carbon Monoxide	NO_x	Nitrogen Oxide
CO_2	Carbon Dioxide	PN	Particle Number
CRDI	Common Rail Direct Injection	SFC	Specific Fuel Consumption
CTL	Coal-to-Liquid	SI	Spark Ignition
EAR	Excess Air Ratio	UHC	Unburned Hydrocarbons

ACKNOWLEDGMENT

The authors would like to thank Erciyes University Mechanical Engineering, Engines Laboratory, where the experiments were carried out, and Prof. Dr. Nafiz Kahraman, Prof. Dr. S. Orhan Akansu and Prof. Dr. Bilge A. Çeper for their support.

DECLARATION OF ETHICAL STANDARDS

The authors of the paper submitted declare that nothing which is necessary for achieving the paper requires ethical committee and legal-special permissions.

CONTRIBUTION OF THE AUTHORS

Esenay Arslan: Performed the experiments, analyse the results, and wrote the manuscript.

Mehmet İlhan İlhak: Performed the experiments, analyse the results, and wrote the manuscript.

CONFLICT OF INTEREST

There is no conflict of interest in this study.

REFERENCES

[1] Turkcan A, Ozsezen AN, Canakci M. Experimental investigation of the effects of different injection parameters on a direct injection HCCI engine fueled with alcohol gasoline fuel blends. Fuel processing technology 2014; 126: 487-496.

[2] Tsurutani K, Takezi H, Hosogai D. Development of a Diesel Fuel SI Engine Using an Ultrasonic Atomizer. SAE Technical Paper 1991; No. 910667.

[3] Zapata-Mina J, Restrepo A, Romero C, Quintero H. Exergy analysis of a diesel engine converted to spark ignition operating with diesel, ethanol, and gasoline/ethanol blends. Sustainable Energy Technologies and Assessments 2020; 42: 100803.

[4] Du J, Sun W, Guo L, Xiao S, Tan M, Li G, Fan L. Experimental study on fuel economies and emissions of direct-injection premixed combustion engine fueled with gasoline/diesel blends. Energy Conversion and Management 2015; 100: 300-309.

[5] Liu R, Wei M, Yang H. Cold start control strategy for a two-stroke spark ignition diesel-fuelled engine with air-assisted direct injection. Applied Thermal Engineering 2016; 108: 414-426.

[6] Ning L, Duan Q, Liu B, Zeng K. Experimental assessment of lean-burn characteristics for a modified diesel engine operated in methanol direct injection spark ignition (DISI) mode at full throttle condition. Fuel 2020; 279: 118455.

[7] Lee J, Lee S, Lee S. Experimental investigation on the performance and emissions characteristics of ethanol/diesel dual-fuel combustion. Fuel 2018; 220: 72-79.
[8] Jamrozik A, Tutak W, Gruca M, Pyrc M. Performance, emission and combustion characteristics of CI dual fuel engine powered by diesel/ethanol and diesel/gasoline fuels. Journal of Mechanical Science and Technology 2018; 32: 2947-2957.
[9] Chen Z, Qin T, He TP, Zhu LJ. Effect of equivalence ratio on diesel direct injection spark ignition combustion. Journal of Central South University 2020; 27(8): 2338-2352.
[10] Wang Q, Sun W, Guo L, Fan L, Cheng P, Li G, Du J. Particle number size distribution from direct-injection premixed combustion engine fueled with gasoline/diesel blends. Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering 2020; 234(1): 258-269.
[11] Kaleemuddin S, Rao PAG. Conversion of diesel engine into spark ignition engine to work with CNG and LPG fuels for meeting new emission norms. Thermal Science 2010; 14(4): 913-922.
[12] Liu R, Ji H, Wei M. Study of low load performance on a two-stroke direct injection spark ignition aero-piston engine fuelled with diesel. Aircraft Engineering and Aerospace Technology 2020; 93(3): 473-482.
[13] Zhang H, Sun W, Guo L, Yan Y, Li J, Lin S, Sun Y. An experimental study of using coal to liquid (CTL) and diesel as pilot fuels for gasoline dual-fuel combustion. Fuel 2021; 289: 119962.
[14] Liu J, Dumitrescu CE. Flame development analysis in a diesel optical engine converted to spark ignition natural gas operation. Applied Energy 2018; 230: 1205-1217.
[15] Liu J, Liu S, Wei Y, Li Y, Li G, He H. Diesehol CI engine performances, regulated and nonregulated emissions characteristics. Energy & fuels 2010; 24(2): 828-833.
[16] Iodice P, Amoresano A, Langella G. A review on the effects of ethanol/gasoline fuel blends on NOx emissions in spark-ignition engines. Biofuel Research Journal 2021; 8(4): 1465-1480.
[17] Roso VR, Santos NDSA, Alvarez CEC, Rodrigues Filho FA, Pujatti FJP, Valle RM. Effects of mixture enleanment in combustion and emission parameters using a flex-fuel engine with ethanol and gasoline. Applied Thermal Engineering 2019; 153: 463-472.
[18] Ağbulut Ü, Sarıdemir S. Is the ethanol additive more environmentally friendly for a spark ignition (si) engine or for a compression ignition (ci) engine?. Environmental Engineering & Management Journal (EEMJ) 2020; 19(4).
[19] Tutak W, Lukács K, Szwaja S, Bereczky Á. Alcohol–diesel fuel combustion in the compression ignition engine. Fuel 2015; 154: 196-206.

[20] Huang J, Wang Y, Li S, Roskilly AP, Yu H, Li H. Experimental investigation on the performance and emissions of a diesel engine fuelled with ethanol–diesel blends. Applied Thermal Engineering 2009; 29(11-12): 2484-2490.
[21] Rakopoulos DC, Rakopoulos CD, Kakaras EC, Giakoumis EG. Effects of ethanol–diesel fuel blends on the performance and exhaust emissions of heavy duty DI diesel engine. Energy conversion and management 2008; 49(11): 3155-3162.
[22] Preliminary Engine Specification MVH Range MVH 418. Ford Power Products.
[23] Geng P, Cao E, Tan Q, Wei L. Effects of alternative fuels on the combustion characteristics and emission products from diesel engines: A review. Renewable and Sustainable Energy Reviews 2017; 71: 523-534.
[24] İlhak Mİ, Doğan R, Akansu SO, Kahraman N. Experimental study on an SI engine fueled by gasoline, ethanol and acetylene at partial loads. Fuel 2020; 261: 116148.
[25] Stone R. Introduction to Internal Combustion Engines, Third Edition. Macmillan Press Ltd., London, 1999.
[26] Kline SJ. Describing Uncertainties in Single-Sample Experiments. Mechanical Engineering 1963; 75: 3-8.
[27] Pulkrabek WW. Engineering Fundamentals of the Internal Combustion Engines, Prentice Hall, New Jersey, 1997.
[28] Han J, Somers LMT, Cracknell R, Joedicke A, Wardle R, Mohan VRR. Experimental investigation of ethanol/diesel dual-fuel combustion in a heavy-duty diesel engine. Fuel 2020; 275: 117867.