A Study Toward Analyzing the Energy, Exergy and Sustainability Index Based on Performance and Exhaust Emission Characteristics of a Spark-Ignition Engine Fuelled with the Binary Blends of Gasoline and Methanol or Ethanol

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Abstract
The anxieties regarding global warming upon increasing greenhouse gas emission grades worldwide and the presence of petroleum-based fuels have directed the researchers to focus on the development of biofuels as well as the utilization of reformulated gasoline fuels by adding oxygenated additives resulting in an extensive application to improve fuel properties. In this study, engine performance and exhaust emission tests were performed using pure gasoline and volumetrically 10% ethanol-C2 or methanol-C1/gasoline blends (G100, E10, and M10). The engine experiments for all test fuels were carried out in a single-cylinder, four-stroke, water-cooled, spark-ignition (SI) engine under fixed engine speed (1500 rpm) and various loading conditions (25%, 50%, 75%, and 100%). In the tested engine, the brake specific fuel consumption (BSFC) values of G100, M10, and E10 fuels under full load condition were found to be as 0.279 kg/kWh, 0.296 kg/kWh and 0.307 kg/kWh, respectively. When the exhaust emissions were examined, E10 and M10 fuels were observed to have lesser CO, CO_2, NO_X, and HC emissions in comparison with pure gasoline. The lowest CO emission was determined as 3.15% for E10 fuel at a 75% load. NO_X emissions descended with the increase of engine load in all fuel blends meanwhile the best performance is measured as 908.86 ppm in E10 fuel at 100% load. The minimum HC emission for E10 fuel was measured as 116.36 ppm at a 75% load. Compared with G100 fuel, E10 and M10 blends emitted 39% and 35% fewer HC emissions, respectively at 75% load. Besides, E10 and M10 fuels generated 8% and 5% less CO_2 emissions at all engine loads, respectively, when compared to G100 fuel. As a result of thermodynamic analyses; The highest exergy efficiency values were found to be at 21.0% for G100, 17.92% for E10, and 16.85% for M10, respectively. Besides, the energy efficiencies were obtained to be as 30.01% for G100, 28.33% for E10, and 29.90% for M10, respectively. According to the sustainability analysis, E10 fuel performed better results than M10 fuel in order to be an alternative to G100 fuel.

Key Words
"Spark-ignition engine, gasoline, ethanol, methanol, exergy analysis, sustainability index."

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Nomenclature

C_{pc}	*Specific heat of the coolant (kJ/kg)*
C_{peq}	*Specific heat of the exhaust gases (kJ/kg)*
$\dot{E}_{fuel}$	*Energy rate of the fuel (kW)*
$\dot{E}x$	*Exergy rate (kJ/s)*
$\dot{E}x_a$	*Ambient exergy rate (kJ/s)*
$\dot{E}x_{air}$	*Air exergy rate (kJ/s)*
$\dot{E}x_c$	*Cooling water exergy rate (kJ/s)*
$\dot{E}x_{dest}$	*Exergy destruction rate (kJ/s)*
$\dot{E}x_{eq}$	*Exhaust exergy rate (kJ/s)*
$\dot{E}x_{eq,ch}$	*Chemical exergy of exhaust gases (kJ/s)*
$\dot{E}x_{eq,t}$	*Thermomechanical exergy of exhaust gases (kJ/s)*
$\dot{E}x_{fuel}$	*Fuel exergy rate (kJ/s)*
$\dot{E}x_w$	*Exergetic work (kJ/s)*
Hu	*The lower calorific value of fuel (kJ/kg)*
$\dot{m}_{air}$	*Air mass flow rate (kg/s)*
$\dot{m}_{fuel}$	*Fuel mass flow rate (kg/s)*
$\dot{m}_c$	*Cooling water mass flow rate (kg/s)*
P	*Pressure (kPa)*
P_0	*The pressure of the environment (kPa)*
P_3	*Exhaust inlet pressure (kPa)*
T	*Temperature (K)*
T_a	*Average engine surface temperature (K)*
T_{cout}	*Cooling water exist temperature (K)*
T_{cin}	*Cooling water entrance temperature (K)*
T_0	*The temperature of the environment (K)*
T_3	*Exhaust inlet temperature (K)*
$\dot{Q}_a$	*Ambient energy flow (kW)*
$\dot{Q}_c$	*Cooling water energy flow (kW)*
$\dot{Q}_{eq}$	*Exhaust energy flow (kW)*
$\bar{R}$	*Universal gas constant (8.314 J mol/K)*
R	*Gas constant (kJ/kgK)*
$\dot{W}$	*Work (kW)*
$\dot{S}_{gen}$	*Entropy generation (kW/K)*
y	*Exhaust gas mole fraction*
y^e	*Component mole fraction*
n	*Engine speed (rpm)*
η	*Energy efficiency (%)*
Ψ	*Exergy efficiency (%)*
φ	*Chemical exergy factor*

Abbreviations

ABDC	*After the bottom dead centre*
BBDC	*Before the bottom dead centre*
ATDC	*After the top dead centre*
BTDC	*Before the top dead centre*
BTE	*Brake thermal efficiency*
B10	*10% butanol+90% gasoline*
B20	*20% butanol+80% gasoline*
B60	*60% butanol+40% gasoline*
BSFC	*Brake specific fuel consumption*
C_8H_{18}	*Gasoline*
CH_3OH	*Methanol*
C_2H_5OH	*Ethanol*
CO_2	*Carbon dioxide*
CO	*Carbon monoxide*
CR	*Compression ratio*
E0, G100	*100% gasoline*
E5	*5% ethanol+95% gasoline*

E10	*10% ethanol+90% gasoline*
E15	*15% ethanol+85% gasoline*
E20	*20% ethanol+80% gasoline*
E25	*25% ethanol+75% gasoline*
E30	*30% ethanol+70% gasoline*
E40	*40% ethanol+60% gasoline*
E50	*50% ethanol+50% gasoline*
E60	*60% ethanol+40% gasoline*
E85	*85% ethanol+15% gasoline*
E100	*100% ethanol*
EGT	*Exhaust gas temperature*
HC	*Unburned hydrocarbon*
ICE	*Internal combustion engine*
M5	*5% methanol+95% gasoline*
M10	*10% methanol+90% gasoline*
M15	*15% methanol+85% gasoline*
M20	*20% methanol+80% gasoline*
M25	*25% methanol+75% gasoline*
M30	*30% methanol+70% gasoline*
M60	*60% methanol+40% gasoline*
M85	*85% methanol+15% gasoline*
M100	*100% methanol*
N_2	*Nitrogen*
NO_X	*Nitrogen oxides*
O_2	*Oxygen*
S10	*10% solketal+90% gasoline*
SI	*Spark-ignition*
SIN	*Sustainability index*

1. Introduction

Energy is needed for areas such as domestic utilization and transportation in daily life. A significant part of the energy demands has been met from fossil-based fuels. As a result of using fossil-based fuels in the internal combustion engines (ICEs), many harmful pollutants, like carbon monoxide (CO), unburned hydrocarbon (HC), and nitrogen oxides (NO_X), have been released into the atmosphere (Awad et al., 2018; Barreto, 2018; Mwangi, 2015). Moreover, it is predicted that the reserves of fossil-based fuels will be depleted in the next 50 years all over the world. Such adverse factors have directed the developed countries to alternative, sustainable, and renewable energy resources that can be produced from domestic sources within the country (Bussar et al., 2016; Connolly et al., 2014; Connolly et al., 2016; Hansen et al., 2019; Krakowski et al., 2016; Lund et al., 2009).

Nowadays, alcohol-based choices like ethanol and methanol have been widely preferred instead of fossil-based fuels in SI engines. Ethanol and methanol have attracted more attention due to their advantages, such as less emission in combustion products, high octane numbers, and being able to run without knocking at high compression ratios (CRs) (Chen et al., 2018; Masum et al., 2013; Sayah et al., 2011; Sezer et al., 2013; Zhen et al., 2015).

Methanol and ethanol have been evaluated with blending both pure diesel and gasoline fuels in various proportions without making too many arrangements on the fuel system of engines. The most common ethanol and methanol blend applications have been known as E10 - E85 and M10 - M85, respectively, and they contain 10% and 85% of ethanol or methanol, respectively (Elfasakhany, 2017; Thangavelu et al., 2016).

Yücesu et al. (2006) examined the performance and emission values in a single-cylinder, four-stroke, SI engine with ethanol-gasoline fuel blends (E10, E20, E40, and E60) at dissimilar CRs and engine speeds. It could be reported that the infusion of ethanol into gasoline improved brake torque and BSFC along with descended HC and CO harmful exhaust emissions.

Shenghua et al. (2007) experimentally researched the engine performance and emission values in a three-cylinder, four-stroke, SI engine with methanol-gasoline fuel blends with different ratios (M10, M15, M20, M25, and M30) at various engine speeds and full load. The addition of methanol to gasoline led to reduce engine power and torque insignificantly but to decline HC and CO harmful exhaust emissions significantly.

Yanju et al. (2008) performed power, thermal brake efficiency (BTE), and emission features using methanol-gasoline fuel blends at different concentrations coded as M10, M20, and M85, respectively. The researchers stated that the addition of methanol to gasoline descended CO and NO_X emissions by 25% and 80%, respectively.

Bilgin et al. (2008) carried out experimental studies in a single-cylinder, four-stroke, SI engine with methanol-gasoline fuel blends (M5, M10, M15, and M20) at several CRs of 7.5:1, 8:1, and 8.5:1, spark timings of 7.5°, 10°, and 12.5°, and engine speeds (900-1600 rpm). The researchers reported that the maximum brake means effective pressure values occurred when the tested engine was run on the M5 mixture. Besides, the highest BTE was indicated to be observed with the M20 fuel blend. Furthermore, the best performance for all spark timings was also noted to be provided with the M20 fuel blend.

Koç et al. (2009) investigated the performance and emission characteristics by conducting experimental studies in a single-cylinder, four-stroke, SI engine with ethanol-gasoline fuel blends (E50, and E85) at different CRs and engine speeds. The researchers noticed that the addition of ethanol to gasoline caused to turn in up the fuel consumption, power, and engine torque as well as reduction in CO, NO_X, and HC emissions.

Sezer et al. (2009) in their thermodynamic modelling study, worked the utilization of ethanol and methanol as alternating and clean fuels for an SI engine with exergy analysis. The outcomes coming from the analysis showed that the oxygenated fuels were stated to be exergetically appropriate alternatives because of declining heat losses and entropy generation. In contrast to pure gasoline, ethanol and methanol were reported to decrease the irreversibility by 7.44% and 4.29%, respectively. However, the oxygenated fuels were observed to increase the BSFC and descend the effectual power output.

Eyidogan et al. (2010) examined the performance and combustion characteristic values using ethanol-gasoline blends (E5 and E10) and methanol-gasoline blends (M5 and M10) in a vehicle having a four-cylinder, water-cooled, multi-point injection, SI engine. During the experimental studies, alcohol blends brought about to increase the BSFC and cylinder gas pressures compared to pure gasoline fuel.

Li et al. (2011) converted a single-cylinder, four-stroke, direct-injection, diesel engine into an SI engine by making modifications. They investigated the effects of injection and ignition timings with optimum injection and ignition timings on engine performance and emissions experimentally by using pure methanol as an alternative fuel in the engine SI mentioned above. With this developed test setup, experimental studies were carried out using different CRs and a multi SI system and a single-SI system at a constant engine speed of 1500 rpm. According to the outcomes coming from the experimentations, the ignition system, CR, and methanol injector modifications were found to have positive influences on the performance of the tested engine by 27% in the BTE.

Turner et al. (2011) carried out experimental studies in a single-cylinder, four-stroke, SI engine using E0, E10, E20, E30, E50, E85, and E100 fuels at different spark ignition timings along with a constant engine speed of 1500 rpm. They examined the performance and exhaust emission behaviours. According to the obtained results, it was to be noted that with the increase in the ratio of ethanol in the fuel blends, the combustion rate and in-cylinder gas pressure increased, and CO, NO_X, and HC emissions decreased.

Schifter et al. (2011) investigated the effects of ethanol-gasoline blends (E10, E15, and E20) on the brake power, BSFC, and exhaust emission characteristics in a single-cylinder, electronically-controlled, SI engine by performing experimental studies at a constant engine speed of 2000 rpm for different lambda values. Since the energy content of the E20 fuel blend was 8% less than that of pure gasoline, the BSFC was determined to be higher, approximately 6%. The addition of ethanol to gasoline was observed to decline CO and HC emissions while increasing NO_X emission.

Ozsezen et al. (2011) studied the performance and exhaust emissions of the vehicle running on ethanol-gasoline blends (E5 and E10) and methanol-gasoline blends (M5 and M10) at various speeds (from 40 km/h to 100 km/h) in a vehicle with a four-cylinder, SI engine. During the experimental studies, alcohol blends were determined to increase BSFC and wheel power compared to pure gasoline fuel. Moreover, in all of the tests conducted with alcohol-gasoline blends, it was noticed that CO emission increased while HC emissions decreased in comparison with pure gasoline.

In the experimental study conducted by Farkade et al. (2012) the performance and emission values by blending butanol, ethanol, and methanol alcohol fuels with gasoline at different ratios (10%, 20%, and 30% by volume) were investigated. In general, the researchers detected that lower CO and HC emissions along with higher carbon monoxide (CO_2) emission were generated as a result of more stable combustion of the fuels inside the cylinder. This situation was because of the increase in the number of oxygen molecules due to the aforementioned three different tested fuel mixtures when compared with the neat gasoline fuel. Besides, they stated that the M30 fuel blend exhibited the best engine power performance compared to other fuel blends and that the lowest fuel consumption, as well as the best thermal efficiency, were provided with the M10 fuel blend.

Gravalos et al. (2013) researched the exhaust emission properties of low and high molecular weight alcohol-gasoline fuel blends in a single-cylinder, two-stroke, SI engine. The results showed that among the harmful exhaust emissions, CO and HC emissions decreased. In contrast, NO_X and CO_2 emissions increased in comparison with pure gasoline regarding the increase in alcohol content in the blend fuels.

Canakci et al. (2013) investigated the BSFC, exhaust gas temperature (EGT), and harmful exhaust emissions using ethanol-gasoline blends (E5 and E10) and methanol-gasoline blends (M5 and M10) in a vehicle operated with a four-cylinder, water-cooled, multi-point injection system, gasoline engine at speeds of 80 km/h and 100 km/h. The findings exhibited that CO, CO_2, HC, and NO_X emissions decreased at a speed of 80 km/h while CO emission increased at a speed of 100 km/h. The researchers stated that the highest emission values among alcohol-gasoline fuel blends were found with the M10 fuel blend.

Altun et al. (2013) investigated the impacts of using 5% and 10% ethanol or methanol in gasoline on the engine performance and exhaust emissions by conducting experimental studies at different engine speeds and 75% throttle opening. According to the experimental results, it was determined that for all alcohol-gasoline tested fuels, CO and HC emissions decreased compared to pure gasoline fuel while the BSFC, BTE, and CO_2 emission increased. In contrast to pure gasoline, the use of E10 and M10 fuel blends in the tested engine led to a decrease in HC emissions by 15% and 13%, respectively, and CO emissions by 9.8% and 10.6%, respectively. The best results were observed to be in the E10 and M10 fuel blends in terms of emissions.

Agarwal et al. (2014) conducted experimental studies in a four-cylinder, four-stroke, water-cooled, Maruti Suzuki Zen 2001 model SI engine with methanol-gasoline blends (M10 and M20) at various engine speeds and loads. They scrutinised some parameters, such as BSFC, BTE, EGT, cylinder pressure, and harmful emissions. Based on the results, the researchers reported that methanol-gasoline fuel blends had higher BTE compared to pure gasoline. In addition, they observed that CO and NO_X emissions decreased. The researchers stated that when the combustion characteristics of methanol-gasoline fuel blends were compared to those of pure gasoline, almost similar values were obtained.

Balki and Sayin (2014) studied the performance, combustion characteristic and harmful exhaust emission values of a single-cylinder, air-cooled, four-stroke, SI engine fuelled with pure methanol (M100), ethanol (E100), and gasoline (G100) fuels under a constant engine speed of 2400 rpm with different CRs. The researchers found that BTE and volumetric efficiency in pure ethanol and methanol alternating fuels were occurred to be higher than those of pure gasoline for all CRs. Besides that, the BSFC increased as expected due to the lower calorific values of ethanol and methanol compared to pure gasoline. Moreover, the usage of pure ethanol and methanol for all CRs caused drop the CO, HC, and NO_X emissions.

Balki et al. (2014) tested methanol (M100), ethanol (E100), and gasoline (G100) fuels in a single-cylinder, air-cooled, four-stroke, Datsu LT 200 model SI engine at several engine speeds so as to investigate the influences of alcohol-based fuels on the BSFC, engine torque, BTE, EGT, cylinder gas pressure, and harmful exhaust emissions. The outcomes demonstrated that the usage of alcohol fuels led to increasing engine torque, BSFC, BTE, and combustion efficiency values. Furthermore, they observed that the utilization of pure ethanol and methanol reduced CO, HC, and NO_X emissions and that it increased CO_2 emission due to combustion recovery.

Ghazikhani et al. (2014) calculated the exergy terms after experimentally determining the emission amounts in a two-stroke engine operating with alcoholic fuel additives (E5, E10, and E15). The experiments were carried out between 2500 rpm and 4500 rpm engine speed intervals. It has been determined that the irreversibility increased in alcoholic mixtures and therefore, the second law efficiency turned down. The formation of pollutants like HC, CO_2, CO, and NO_X was substantially dropped when the tested engine fueled with the addition of ethanol to gasoline in all test conditions.

Elfasakhany (2014) examined volumetric efficiency, brake power, engine torque, BSFC, EGT, cylinder pressure, and emission characteristics of a single-cylinder, air-cooled, four-stroke, SI engine operating with ethanol-gasoline blends (E3, E7, and E10) at different engine speeds. The experimental findings demonstrated that volumetric efficiency, brake power, engine torque, EGT, and cylinder pressure values increased for all test fuels, and BSFC decreased in comparison with pure gasoline fuel. Furthermore, CO and HC emissions were decreased. As a result of this study, the fuel blend exhibiting the best performance values considering all experimental parameters was stated to be the E10 blend.

Elfasakhany (2015) compared the performance and exhaust emission behaviours of an SI engine powered with ethanol-gasoline and methanol-gasoline blends, including 3-10 vol.% alcohol concentration. It was to be noted that alcohol-treated fuel blends reduced CO and HC emissions. Also, the lowest emission values and highest power values were obtained with methanol-gasoline blends among alcohol-gasoline fuel blends.

Kapusuz et al. (2015) performed the effects of the use of ethanol-gasoline blends (E5, E10, and E15) and methanol-gasoline blends (M5, M10, and M15) on the engine performance of a single-cylinder, SI engine at different engine speeds (1000-2500 rpm) by applying an artificial neural network model that they developed.

Wu et al. (2016) investigated the impacts of methanol and gasoline (M100 and G100) fuels on the combustion characteristics and exhaust emissions of a four-cylinder, continuous variable valve timing, SI engine by conducting experimental studies at idle engine speed (800 rpm) for different lambda values ($\lambda = 1.0$, 1.2, and 1.4). According to the results, they reported that the use of pure methanol reduced CO, HC, and NO_X emissions. Methanol fuel was stated to provide higher BTE than that of pure gasoline fuel.

Li et al. (2017) observed the effects of using ethanol-gasoline blends (E10, E30, and E60), methanol-gasoline blends (M10, M30, and M60) and butanol-gasoline blends (B10, B30, and B60) on the performance, and harmful pollutants under a constant engine speed of 1200 rpm with full throttle opening. The researchers indicated that the lowest NO_X emission values were obtained with methanol-gasoline fuel blends, while the lowest HC emissions were obtained with ethanol-gasoline fuel blends. Furthermore, it was reported that butanol-gasoline fuel blends provided lower BSFC compared to pure gasoline fuel.

Alexandru et al. (2017) conducted experimental studies in a single-cylinder, four-stroke, Honda CN 250 model gasoline engine using methanol-gasoline blends (M5, M10, M15, M20, and M25) at various engine speeds and loads to investigate the engine performance and harmful exhaust emission characteristics. They explained that methanol-gasoline blends reduced engine power and torque by about 10% while also reducing CO and HC emissions significantly. Moreover, methanol-gasoline blends were stated to ensure 13% higher CO_2 emission compared to pure gasoline fuel.

Doğan et al. (2017) performed exergy and energy analyses of a four-cylinder, four-stroke, SI engine operating with gasoline-ethanol mixtures (E10, E20, and E30). Hence, the exergy amounts lost by the exhaust, cooling water and radiation were calculated in each fuel. The most considerable exergy destruction occurred in neat gasoline while the maximum exergy efficiency was found to be at 53% in the E0 fuel at 3000 rpm along with minimum exergy efficiency was observed to be like 45% in the E30 fuel at 4500 rpm.

Mithaiwal et al. (2017) performed an exergy analysis based on the engine performance results using both 100% ethanol and ethanol-gasoline blends (E25 and E40). The results exhibited that the mechanical efficiency values decreased by 10-16% for E25, 9-5% for E40, and 5-3% for E100 at different loads as compared to E0 fuel. The exergy efficiencies reduced by 3-5% at medium load for the entire ethanol-gasoline mixture. On the other hand, the availabilities of E25, E40, and E100 fuels increased by 9-13%, 13-19%, and 3-5%, respectively, when compared to pure gasoline under variable load conditions.

Hasan et al. (2018) carried out experimental researches in a single-cylinder, air-cooled, four-stroke, SI engine using alcohol-gasoline blends which contained 10% and 20% ethanol at different CRs and a constant engine speed of 2500 rpm with full load conditions. The researchers investigated the effects of ethanol-gasoline blends on the BTE, BSFC, and harmful exhaust emission values. Based on the results, the CR had essential influences on harmful pollutants. It was indicated that HC and CO emissions were affected very little by a change in the CR and that the NO_X emission values increased a lot in the emission amount depending on the increase in the CR. The lowest HC and NO_X emissions were obtained with the E20 blend at 4:1 of the CR while the lowest CO emission was observed with the E10 blend.

Özcan and Çakmak (2018) investigated the influences of oxygenated fuel additive-gasoline blends involving 10% (by volume) ethanol (E10), methanol (M10), and solketal (S10) on the exergy parameters in an SI engine. With the use of oxygenated fuel additives, the maximum cylinder pressures increased, but exergy efficiencies decreased in comparison with pure gasoline. The researchers reported that the maximum decrement in the exergy efficiency occurred as 8.42% with S10 fuel. It was explained that maximum irreversibilities occurred with pure gasoline fuel, and minimum irreversibility occurred with E10 fuel. However, oxygenated fuel additives were observed to reduce the first law efficiency.

Tian et al. (2020) conducted a study in a four-cylinder, water-cooled, turbocharged, four-stroke, SI engine implementing the GT-Power simulation platform. In the trials, ethanol-gasoline blends (E10 and E20), methanol-gasoline blends (M10 and M20), and butanol-gasoline blends (B10 and B20) were used depending on different engine speed, load, flame kernel radius, and ignition time values. In this simulation study, they investigated the effects of different alcohol-gasoline blends on the BSFC, BTE, EGT, and harmful exhaust emission values.

In recent years, the studies on engines have been based on reducing exhaust emissions, increasing efficiency, and minimizing power losses. When the researches in the literature have been meticulously reviewed, it has been reported that many works and developments have been carried out on the use of ethanol and methanol alcohols in SI engines. These studies have been observed to aim at determining the performance and emission characteristics mainly. Exergy analysis studies involving the use of ethanol-gasoline and methanol-gasoline blends in SI engines have been limited (Doğan et al., 2017; Ghazikhani et al., 2014; Mithaiwal et al., 2017; Sezer et al., 2009). For this reason, in the present study, the performance and exhaust emissions obtained by using pure gasoline, ethanol-gasoline, and methanol-gasoline blends at different loads in an SI engine were experimentally determined. By using experimental data, energy, exergy, and sustainability analyses were conducted. Exergy performance in the case of using two different alcohol-based fuel blends as an alternative fuel instead of gasoline was investigated. In the study, it was determined which alcohol-based additive would be suitable as alternative fuel instead of gasoline by using sustainability analysis. In the fuels used in the test engine; Exergy losses from the exhaust, coolant, and engine body were analyzed. In addition, entropy production resulting from irreversibilities was detected in all engine loads.

2. Material and Methods

2.1 Data preparation

In experimental studies, three different fuels, namely pure gasoline (G100), 90% gasoline - 10% ethanol (E10), and 90% gasoline - 10% methanol (M10), were used. Pure gasoline with octane number 95 used in this work was procured from a local petroleum station in Samsun, which is one of the leading fuel companies in Turkey. The fuel mixtures were prepared by splash blending technique with adding 99% purity ethanol and methanol to gasoline to obtain the fuel mentioned above mixtures. The physical and chemical properties of pure gasoline, ethanol, and methanol are presented in Table 1. The highest carbon content in fuel mixtures used in the test engine is in G100 fuel. The amount of oxygen contained in the fuel is high in methanol. While the densities of methanol and ethanol are very close to each other, both fuels have a much higher density than gasoline. Gasoline's lower heating value is 37.3% and 53.4% higher than ethanol and methanol, respectively.

Table 1. Technical properties of fuels (Shenghua et al., 2007; Sezer et al., 2013; Elfasakhany, 2015)

Property	Unit	Gasoline	Methanol	Ethanol
Typical formula	-	C_8H_{18}	CH_3OH	C_2H_5OH
Carbon content	wt. %	84.21	37.50	52.17
Hydrogen content	wt. %	15.79	12.50	13.04
Oxygen content	wt. %	0	50.00	34.79
Carbon/Hydrogen ratio	-	5.333	3.000	4.000
Molecular weight	g/mol	114.0	32.0	46.0
Density at 20ºC	g/cm^3	0.715	0.791	0.793
Motor octane number	-	87.2	91	92
Research octane number	-	95	112	111
Lower heating value	MJ/kg	43.00	20.05	26.95
Copper strip corrosion (3 h at 50ºC)	Degree of corrosion	1a	-	1a
Auto ignition temperature	ºC	257	480	425
Latent heat of evaporation	kJ/L	223	920	725
Kinematic viscosity at 40ºC	cSt	0.494	-	1.221
Water content	ppm	775	-	<0.1%
Lead	g/L	0.004	-	0
Stoichiometric air/fuel ratio	-	10.5-14.1	15.5-13.7	13.2-14.1
Stoichiometric laminar flame speed	m/s	0.34	0.43	0.41

Ethanol and methanol are alternative fuels that can be used in ICEs with different methods. In the present research, gasoline, ethanol-gasoline blend (E10), and methanol-gasoline blend (M10) were used in a single-cylinder, four-stroke, water-cooled, SI engine at different loads (from 25% to 100%) without changing the CR to achieve the performance and emission values. The technical specifications of the tested engine were tabulated in Table 2.

Table 2. Technical specifications of the experimental engine

Parameters	Specification
Engine supplier	Apex Innovations Pvt. Ltd.
Brand-Model	Kirloskar- TV1
Cylinder number	1
Engine cycle	4
Maximum engine power	4.5 kW at 1800 rpm
Engine speed range	1200-1800 rpm
Powertrain	Camshaft in the block with pushrod
Valve system	2 valves per cylinder
Type of fuel injection	Carburator
Ignition	Spark-ignition
Cooling system	Water-cooled
Swept volume	661.45 cm^3
Bore x Stroke	87.50 mm x 110.00 mm
Compression ratio	Variable: 6-10
Exhaust valve opening advance	35.5º BBDC
Exhaust valve closing delay	4.5º ATDC
Spark timing	10º BTDC
Intake valve opening advance	4.5º BTDC
Intake valve closing delay	35.5º ABDC

Engine maintenance was carried out prior to the experiments commenced. Before each test, the engine was run for 30 minutes with the fuel used in the relevant test in order for it to be stabilized. After the engine was stabilized, the tests were performed in the experimental setup at a fixed speed of 1500 rpm and four different engine loading conditions. The pictorial view of the experimental setup was presented in Figure 1. Fuel consumption was measured volumetrically, and mass fuel consumption was determined by multiplying the measured volumetric flow rate by the fuel density. The tests were repeated four times to verify the experimental data obtained for each fuel type, and the average values were used in the analyses.

Figure 1. Pictorial view of the experimental setup

K-test brand gas analyszer was used for the measurement of exhaust gas emissions. The measurement range and sensitivities of the devices used during the experimental studies can be observed in Table 3. Before conducting tests with each type of fuel, the exhaust emission device was subjected to zeroing, and the sampling probe was cleaned with compressed air. The emission measurements were made at 60 seconds intervals to avoid exhaust emission measurement errors. The cooling water flow rate of the engine was measured by means of a liquid flowmeter. Pt 100 type temperature measurement probes were used to determine the engine cooling water inlet and outlet temperatures. EGTs were measured using a K-type thermocouple. During the experimental studies, ambient temperature and humidity were continuously controlled using a digital thermometer and a humidity measurement device. Table 3 also exhibited the uncertainties of the calculated and measured parameters.

Table 3. The uncertainties of the calculated and measured parameters

Parameter	Measurement Range	Accuracy	Uncertainty (%)
Brake torque	0-90 Nm	±0.1 Nm	±0.88
Engine speed	0-9999 rpm	±1 rpm	±0.83
Engine load	0-12 kg	±0.1 kg	±0.07
Air flow rate	-	-	±0.80
Digital stopwatch	-	±0.2 s	±0.20
Burette system	0-100 cc	±0.1 cc	±1.00
Fuel flow rate	-	-	±0.90
Liquid flowmeter	40-400 L/h	±5 L/h	±1.25
Temperature	-	±0.1 °C	±1.00
BTE	-	-	±1.55
CO	0-10 % vol.	±0.001%	±0.98
HC	0-4000 ppm	±1 ppm	±1.25
CO_2	0-20 % vol.	±0.01%	±0.85
NO_X	0-4000 ppm	±1 ppm	±0.80
O_2	0-25 % vol.	±0.01%	±0.80

2.2 Theoretical consideration

In the analyses conducted in this current study, it is assumed that the engine runs steadily, and the inlet air and exhaust gases formed as a result of combustion are ideal gases. Moreover, potential and kinetic energies of fuel, combustion air, and exhaust gases were neglected in the energy and exergy analyses. In the analysis, the reference state was identified as $T_0 = 25$ °C and $P_0 = 1$ atm.

SI engines convert the chemical energy of fuel into mechanical energy after the combustion process. The test engine was accepted as an open thermodynamic system with continuous flow, and its energy balance is given below in Eq. (1). In this statement, $\dot{E}_{fuel}$ includes the fuel energy flow, $\dot{W}$ identifies the engine power, $\dot{Q}_{eq}$ is the exhaust energy flow, $\dot{Q}_c$ indicates the energy flow going to cooling water, and $\dot{Q}_a$ represents the heat flow going from the engine surface to the environment and other losses.

$$\dot{E}_{fuel} - \dot{W} = \dot{Q}_{eq} + \dot{Q}_c + \dot{Q}_a \tag{1}$$

The exergy balance for the control volume shown in Figure 2 is given in Eq. (2). Here, $\dot{E}x_{fuel}$ indicates the exergy of the tested fuel, $\dot{E}x_{air}$ is the inlet air exergy, $\dot{E}x_{eq}$ is the exhaust exergy, $\dot{E}x_c$ is the cooling water, and $\dot{E}x_a$ indicates the engine surface-related exergy flows. Furthermore, $\dot{E}x_W$ expresses the exergetic power, and $\dot{E}x_{dest}$ identifies the irreversibility (exergy destruction) (Gümüş et al., 2013). In the calculations in the present study, it was accepted that $\dot{E}x_{air} = 0$. Combustion air enters the test engine under environmental conditions. Accordingly, the exergy of the air is considered as zero by assuming that the air is found dead. Thus, it can be said that the input exergy consists of the exergy of the fuel.

$$\dot{E}x_{fuel} + \dot{E}x_{air} = \dot{E}x_{eq} + \dot{E}x_W + \dot{E}x_c + \dot{E}x_a + \dot{E}x_{dest} \tag{2}$$
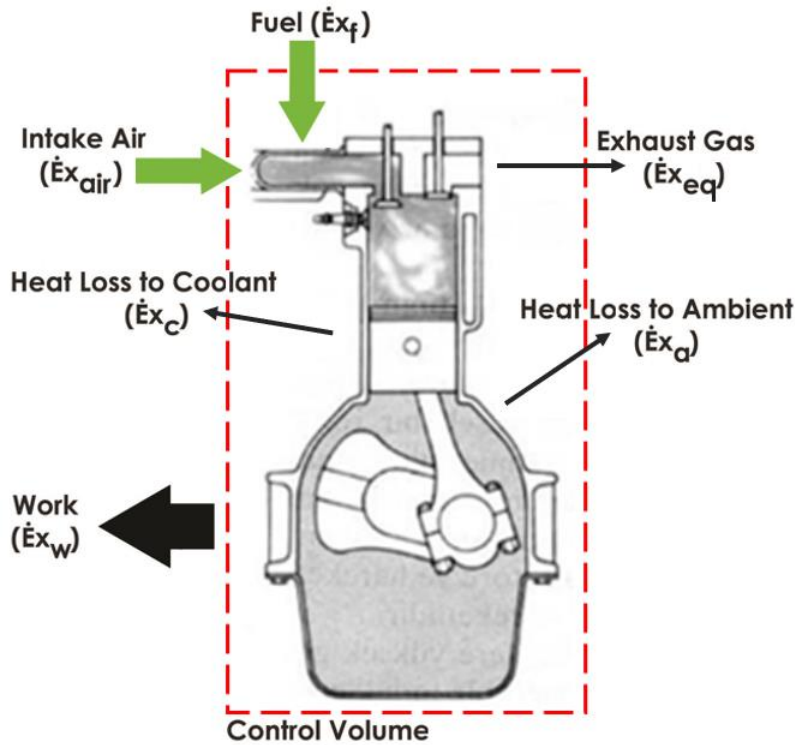
Figure 2. Schematic view of the control volume

When calculating the fuel exergy with using Eq. (3) called input exergy, only chemical exergy was taken into account (Çakmak et al., 2017; Chaudhary et al., 2020). Here, $\dot{m}_{fuel}$ is the mass flow of the fuel, H_u is the lower heating value of the fuel, and φ is the chemical exergy factor. The chemical exergy factor was computed in Eq. (4) according to the mass ratio of hydrogen, oxygen, carbon, and sulfur contents in the composition of the fuel (Çakmak et al., 2017). Exergetic power ($\dot{E}x_w$) is taken equal to the engine power.

$$\dot{E}x_{fuel} = \dot{m}_{fuel} H_u \varphi \tag{3}$$

$$\varphi = 1.0401 + 0.1728 \frac{h}{c} + 0.0432 \frac{o}{c} + 0.2169 \frac{s}{c} (1 - 2.0628 \frac{h}{c}) \tag{4}$$

$$\dot{E}x_w = \dot{W} \tag{5}$$

The exhaust exergy flow is the sum of the chemical and thermomechanical exergies of the exhaust gases, as given in Eq. (6). Here, T_3 is the exhaust inlet temperature, P_3 is the exhaust inlet pressure, and C_{peq} is the specific heat at constant pressure. The chemical exergy of the exhaust gases can be found from Eq. (8). When calculating chemical exergy, $(\bar{R})$ is the gas constant, (y) is the exhaust gas mole

fraction, and (y^e) is the component mole fraction given in Table 4 in the reference ambient conditions (Caliskan et al., 2009; Aghbashlo et al., 2016).

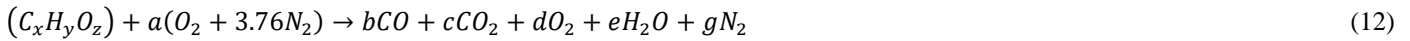$$\dot{E}x_{eq} = \dot{E}x_{eq,ch} + \dot{E}x_{eq,t} \tag{6}$$

$$\dot{E}x_{eq,t} = (\dot{m}_{fuel} + \dot{m}_{air})\left[C_{peq}\left(T_3 - T_0 - T_0\ln\left(\frac{T_3}{T_0}\right)\right) + RT_0\ln\left(\frac{P_3}{P_0}\right)\right] \tag{7}$$

$$\dot{E}x_{eq,ch} = \bar{R}T_0\ln\frac{y}{y^e} \tag{8}$$

Table 4. Mole fraction of the reference environment (Moran et al., 2010).

Reference component	Mol Fractions (%)
N_2	75.6700
O_2	20.3500
H_2O	3.03000
CO_2	0.03450
SO_2	0.00020
CO	0.00070
H_2	0.00005
Others	0.91455

The actual combustion equations of G100, E10, and M10 fuels are needed when calculating the exhaust exergy flow. The theoretical combustion equations of these fuels are given in Eqs. (9-11). The emission data measured in the tests to obtain the actual combustion equation were used in Eq. (12) (da Costa et al., 2019; Taghavifar et al., 2019; Verhelst et al., 2019).

$$C_8H_{18} + 12.5(O_2 + 3.76N_2) \rightarrow 8CO_2 + 9H_2O + 47N_2 \tag{9}$$

$$C_2H_5OH + 3(O_2 + 3.76N_2) \rightarrow 2CO_2 + 3H_2O + 11.28N_2 \tag{10}$$

$$CH_3OH + 1.5(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 5.64N_2 \tag{11}$$

$$\left(C_xH_yO_z\right) + a(O_2 + 3.76N_2) \rightarrow bCO + cCO_2 + dO_2 + eH_2O + gN_2 \tag{12}$$

The exergy loss caused by cooling water is given in Eq. (13). Here, $\dot{m}_c$ is the cooling water mass flow rate, T_{cout} is the cooling exist water temperature, T_{cin} is the cooling water entrance temperature, T_0 is the dead state temperature, and C_{pc} is the specific heat at constant pressure (Aghbashlo et al., 2015).

$$\dot{E}x_c = \dot{m}_c C_{pc}\left(T_{cout} - T_{cin} - T_0\ln\left(\frac{T_{cout}}{T_{cin}}\right)\right) \tag{13}$$

Exergies originated from the engine surface can be calculated from Eq. (14). Here, $\dot{Q}_a$ is the heat loss of the engine surface calculated from the energy balance, and T_a is the average engine surface temperature (Douvartzides et al., 2004).

$$\dot{E}x_a = \dot{Q}_a\left(1 - \frac{T_0}{T_a}\right) \tag{14}$$

After determining all exergy flow values from Eqs. (3, 5, 6, 13, 14), exergy destruction can be calculated using Eq. (2). Exergy efficiency is computed using Eq. (15) (Khanali et al., 2013).

$$\Psi = \frac{\dot{E}x_W}{\dot{E}x_{fuel}} \tag{15}$$

The sustainable index can be calculated with Eq. (16) by using exergy efficiency (Ψ). Entropy production can be found from Eq. (17) by using exergy destruction and dead state temperature (Aghbashlo et al., 2017).

$$SIN = \frac{1}{1-\Psi} \tag{16}$$

$$\dot{S}_{gen} = \left(\frac{\dot{E}x_{dest}}{T_0}\right) \tag{17}$$

3. Results and Discussion

In this study, engine performance and exhaust emission tests were performed firstly by using pure gasoline and then by using the fuel blends obtained by blending volumetrically 10% ethanol and methanol as alcohol with pure gasoline. Fuel consumption in ICEs generally depends on operating parameters such as load and speed. In this study, fuel blends were tested at different engine loads. As is observed in Figure 3, the consumption of E10 and M10 alcohol-gasoline fuel blends is higher than the consumption of G100 pure gasoline fuel. The reason for this is that ethanol and methanol have lower heating values than gasoline. At all engine loads, E10 and M10 blends were found to be consumed approximately by 8.5% and 5.5% more, respectively, than pure gasoline fuel.
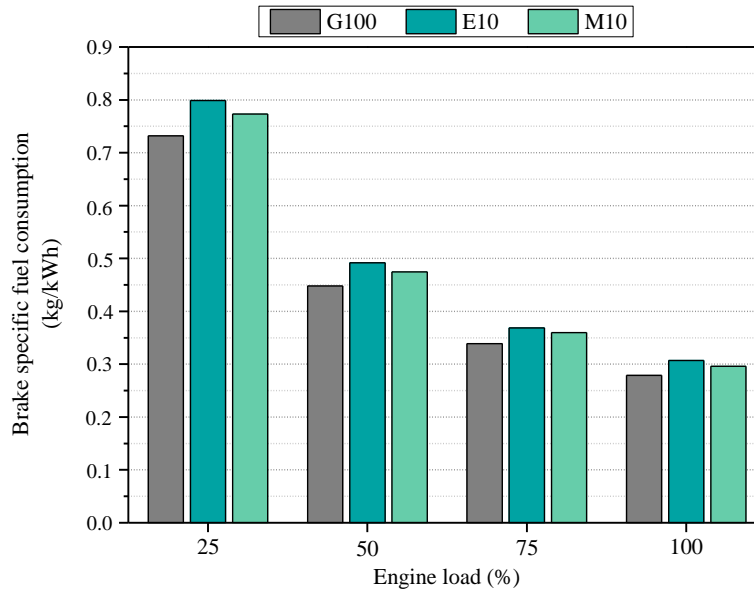
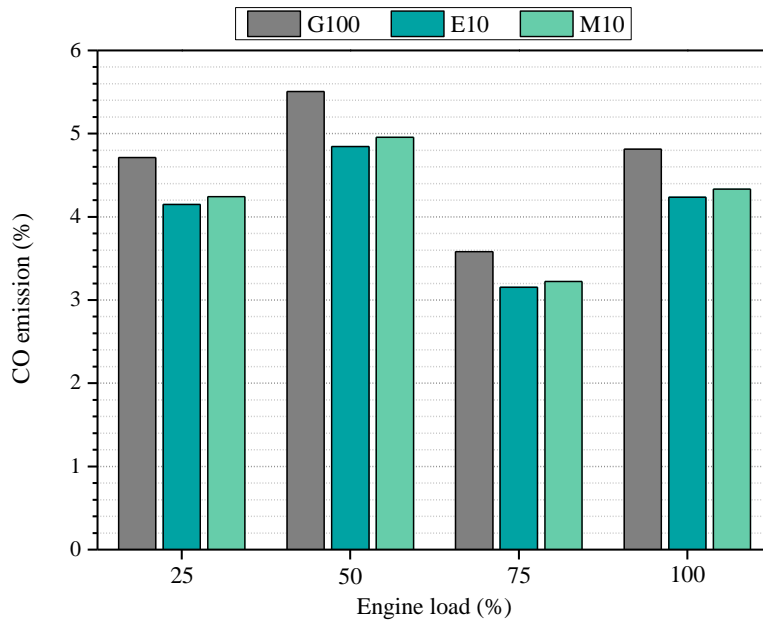**Figure 3.** BSFC values for tested fuels at various engine loads

One of the greenhouse gases that cause global warming is CO_2 (Kim et al., 2016). The CO_2 emission figures for all tested fuel samples under various engine loads were portrayed in Figure 4. With the increase in engine load, CO_2 emissions decrease in all fuel blends. Since ethanol and methanol have lower C/H ratios than gasoline, as a result of combustion, they emitted a smaller amount of CO_2 compared to gasoline, as is seen in Fig. 4. The average amount of emission decreased with the utilization of E10 and M10 blends compared to pure gasoline fuel was obtained to be 13% and 8%, respectively.
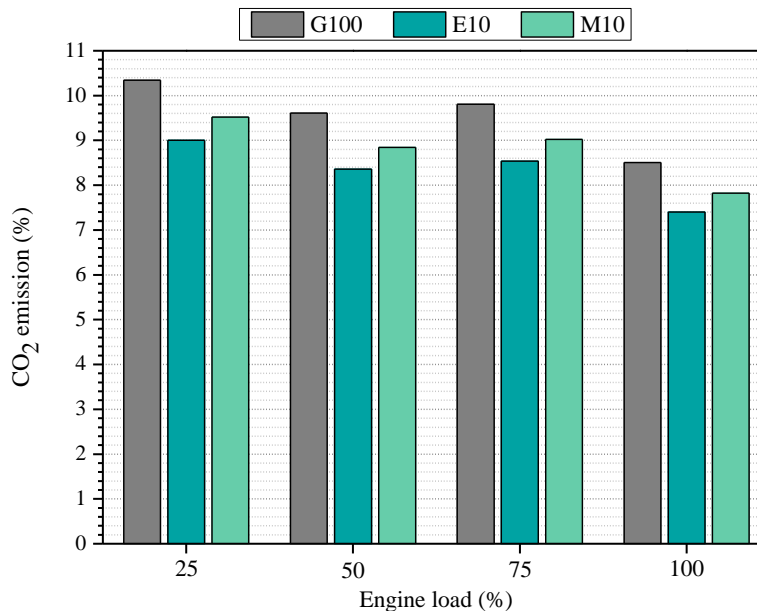
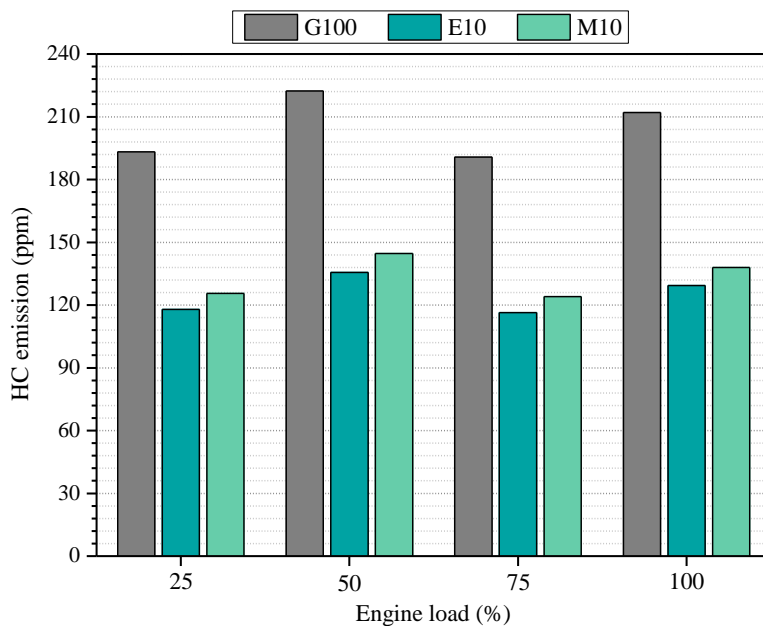**Figure 4.** The change of CO_2 emissions for tested fuels at various engine loads

The effect of fuel blends on CO emission is observed in Figure 5. The molecular structure of the fuel, air-fuel ratio, cylinder gas temperature, and turbulence in the combustion chamber effect on the CO emission (Pulkrabek, 2004; Elsemary et al., 2016). In terms

of CO emission, the average amount of decrease in the case of using E10 and M10 blends compared to gasoline is 12% and 10%, respectively. Ethanol and methanol contain oxygen in their chemical structures. With the increase in the ratio of ethanol and methanol in the fuel blends used in the study, the air-fuel ratio needed for complete combustion also decreases. Accordingly, the oxygen content of the fuel is observed to be quite useful in combustion. However, even though the oxygen content of methanol is higher than that of ethanol, a further decrease in CO emissions with ethanol was achieved. This situation can be explained by the latent heat of evaporation characteristics of the tested fuels. The fact that the latent heat of evaporation of methanol is higher than that of ethanol and thereby it may have slowed the oxidation rate of the fuel by reducing the cylinder gas temperature and increased CO emission slightly (Awad et al., 2018).
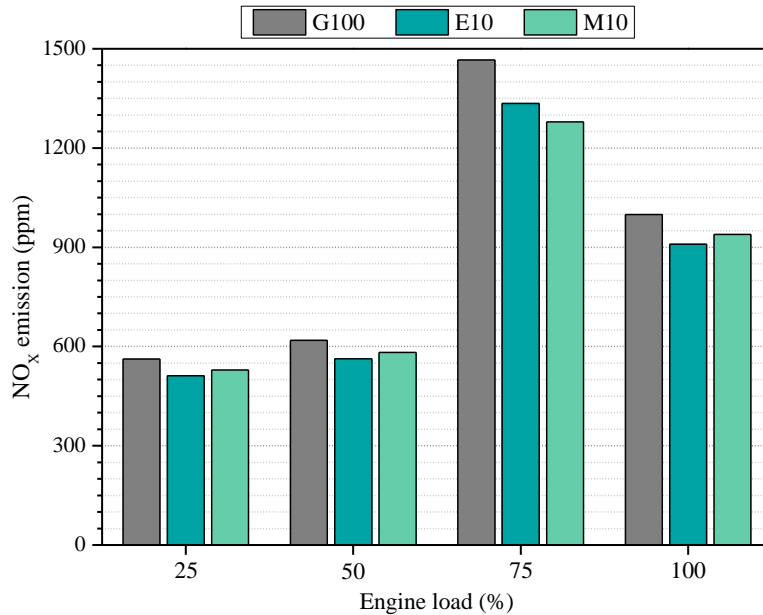

Figure 5. The change of CO emissions for tested fuels at various engine loads

HC emission is an indicator that reflects the incomplete combustion products inside the combustion chamber and occurs due to rich mixture, low combustion temperature, crevice volume, extinguishment of the flame front on the combustion chamber surfaces, and engine lubricating oil (Tangestani et al., 2020; Uslu et al., 2020). HC emissions that occurred in the study, depending on the engine load, are presented in Figure 6. In the use of E10 and M10, the average amount of HC emissions decreased compared to gasoline was determined to be 39% and 35%, respectively. The main reason for lower HC emissions in ethanol-gasoline and methanol-gasoline blends is that ethanol and methanol contain an excessive amount of oxygen molecules in their structure.
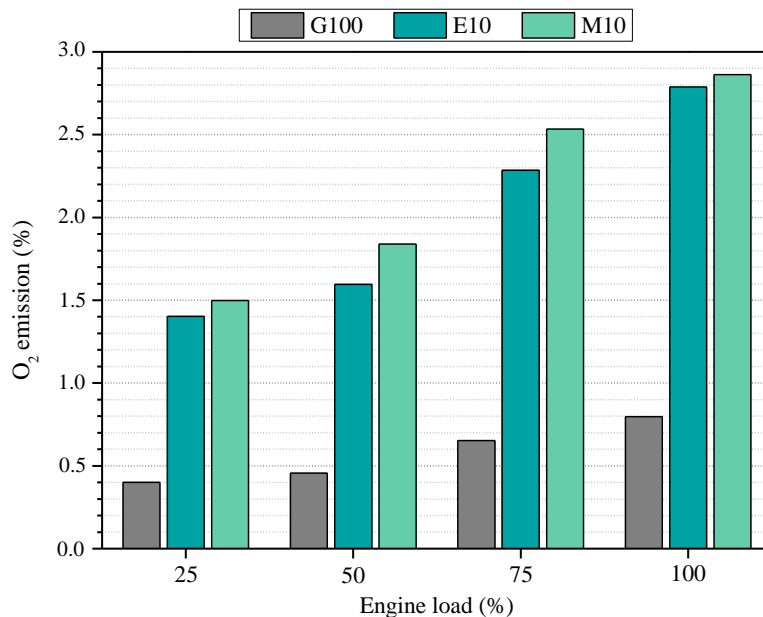

Figure 6. The change of HC emissions for tested fuels at various engine loads

NO_X emissions occur as a result of the nitrogen in the air reacting with oxygen at high temperatures during the combustion process inside the engine cylinder (Fletcher et al., 1971; Şimşek et al., 2020). The alterations of the NO_X emissions for all the tested fuel samples according to the engine load was presented in Figure 7. When Figure 7 is examined, NO_X emission is observed to increase in all fuels with the increase in engine load up to 75%, and it is observed to decrease at 100% engine load. As seen, fewer NO_X emissions were released into the atmosphere when the utilization of E10 and M10 fuel blends in the tested engine compared to G100 fuel. The average amount of decrease in NO_X emissions in the use of E10 and M10 fuels compared to gasoline is 9% and 6%, respectively. The main reason for this is that the latent heat of evaporation values of ethanol and methanol is higher than that of gasoline, and their calorific values are lower in comparison with gasoline. The fact that the high latent heat of evaporation caused to decrease the temperature at the end of the combustion reaction due to the cooling effect resulting in withdrawing the heat from the regions inside the cylinder.

Figure 7. The change of NO_X emissions for tested fuels at various engine loads

The amount of oxygen contained in the exhaust gas of the test engine used in the study is given in Figure 8. The oxygen contents of E10 and M10 blends are high compared to G100 fuel. Therefore, more oxygen emissions were obtained in the combustion of ethanol and methanol blended fuels. This situation can be explained by the high rates of oxygen molecules contained in the structure of alcohols. Since pure gasoline fuel does not contain oxygen in its molecular structure, it caused the lowest O_2 emissions.

Figure 8. The change of oxygen emissions for tested fuels at various engine loads

Energy analysis applied to SI engines gives the distribution of fuel energy flow by various engine components. Exergy analysis applied after this analysis enables the comparison of engine performance with maximum performance. The energy flow, heat loss, net power, and energy efficiency of the fuels were calculated from the energy analysis, and the results were presented in Table 5. Power and efficiency increase in all fuel blends with an increase in engine load. Compared to other fuel samples, G100 fuel performed best in energy efficiency at all loads due to the higher energy content. The highest heat losses in all engine loads occurred in E10 fuel.

Table 5. Energy analysis results

Engine Load (%)	Input Energy (kW)			Total Heat Loss (kW)			Energy Efficiency (%)		
	G100	E10	M10	G100	E10	M10	G100	E10	M10
25	7.50	7.90	7.50	6.64	7.04	6.64	11.44	10.89	11.44
50	9.22	9.73	9.25	7.50	8.01	7.53	18.69	17.68	18.63
75	10.34	10.81	10.38	7.78	8.26	7.82	24.70	23.60	24.60
100	11.34	11.99	11.38	7.93	8.60	7.97	30.01	28.33	29.90

In the exergy analysis conducted in the present work, the quantities, which are given in Table 6 and which belong to different fuel blends, were calculated. While calculating the exhaust exergy flow, the actual combustion equations of the fuels given in Table 7 were found according to the values obtained in the engine tests, and the total exergies of the exhaust gases formed as a result of combustion were determined.

Fuel exergy flow increases with increasing engine load in all fuel blends. Methanol has a lower heating value than ethanol. However, exergy flow is higher since the chemical exergy factor is more significant compared to ethanol by 12%. In the test fuels, EGTs increased with an increase in engine load. When the exhaust emission exergy flow was examined, it was determined that E10 and M10 fuels caused fewer pollutant emissions than G100 fuel. The burn of more fuel due to the increase in the load increases cooling water exergy flows. Accordingly, since the amount of heat transferred from the cylinder wall increases with the increase in fuel consumption, the temperatures of the cooling fluid and engine surface increase. This situation increases the exergy rate transferred to the engine cooling system. The exergy transfer rate from the engine surface to the ambient air increased due to the increase in fuel consumption in SI engines and thereby, the increase in the cylinder temperatures.

Exergy destruction occurs due to engine load increase (the increase in irreversibility due to the blend formation), heat transfer originating from the engine surface and cooling fluid (entropy production due to heat transfer realized from a finite temperature difference), combustion reaction, and friction (Boles et al., 2014). The increase in engine load increases exergy destruction and accordingly, entropy generation.

Table 6. Exergy analysis results

Engine Load (%)	Input Exergy (kW)			Exergy Loss to Ambient (kW)			Exhaust Exergy (kW)			Cooling Water Exergy (kW)			Exergy Destruction (kW)		
	G100	E10	M10	G100	E10	M10	G100	E10	M10	G100	E10	M10	G100	E10	M10
25	10.72	12.48	13.31	1.82	1.76	1.77	2.61	2.42	2.54	0.23	0.27	0.29	5.20	7.17	7.85
50	13.18	15.38	16.42	2.02	1.97	2.01	2.94	2.68	2.78	0.28	0.31	0.35	6.21	8.69	9.56
75	14.76	17.07	18.41	2.19	2.17	2.19	3.41	2.99	3.12	0.30	0.33	0.37	6.31	9.04	10.18
100	16.19	18.95	20.19	2.41	2.39	2.37	3.68	3.23	3.40	0.33	0.36	0.41	6.37	9.57	10.60

Table 7. Actual combustion equation coefficients

Fuel	Load (%)	Real Combustion Equation Coefficients (dimensionless)					
		b	d	c	g	a	e
G100	25	2.504	0.213	0.213	44.917	11.946	9.9700
	50	2.914	0.241	0.241	44.686	11.885	10.200
	75	2.139	0.390	0.390	51.350	13.657	12.674
	100	2.891	0.479	0.479	51.598	13.723	13.380
E10	25	2.334	0.789	0.789	48.097	12.792	11.639
	50	2.715	0.895	0.895	47.744	12.698	11.622
	75	1.996	1.447	1.447	54.476	14.488	13.378
	100	2.694	1.774	1.774	54.437	14.478	13.402
M10	25	2.250	0.795	0.795	44.962	11.958	10.076
	50	2.622	0.974	0.974	44.641	11.873	9.9200
	75	1.921	1.510	1.510	50.790	13.508	11.417
	100	2.601	1.719	1.719	51.040	13.574	11.812

Entropy generation is proportional to exergy destruction. Exergy always disappears due to irreversibilities depending on the second law of thermodynamics. The destructed exergy causes the system to have less efficiency than theoretical efficiency. It is observed in Figure 9 that when the engine load increases in all fuel blends, the in-cylinder temperature, the exergy loss caused by heat transfer, and the total exergy will increase. Since entropy generation increases depending on the engine load, the total exergy destruction also increases. Entropy generation was calculated to be the most in M10 fuel and the least in G100 fuel depending on engine load.
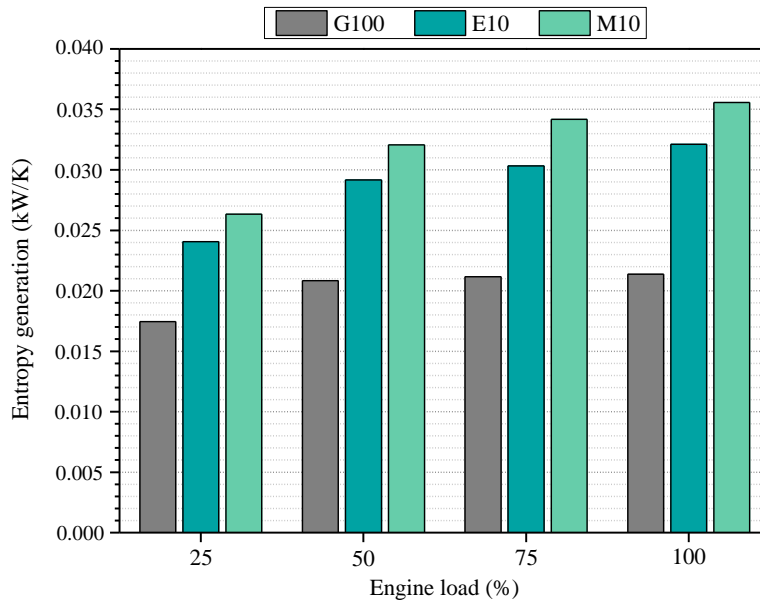

Figure 9. Entropy generation values of tested fuels at various engine loads

How much of the exergy flow of the fuel entering the engine from the intake manifold is converted to power is calculated by exergy efficiency. Exergy efficiency at different loads in the test engine is presented in Figure 10. As alcohol-based fuels are added to gasoline, exergy efficiency decreases depending on the increase in the consumption of fuel. The highest exergy efficiency in all fuels was achieved at 100% load and the exergy efficiency values of G100, E10 and M10 fuels were calculated at this load, approximately 21%, 17.92% and 16.85%, respectively.
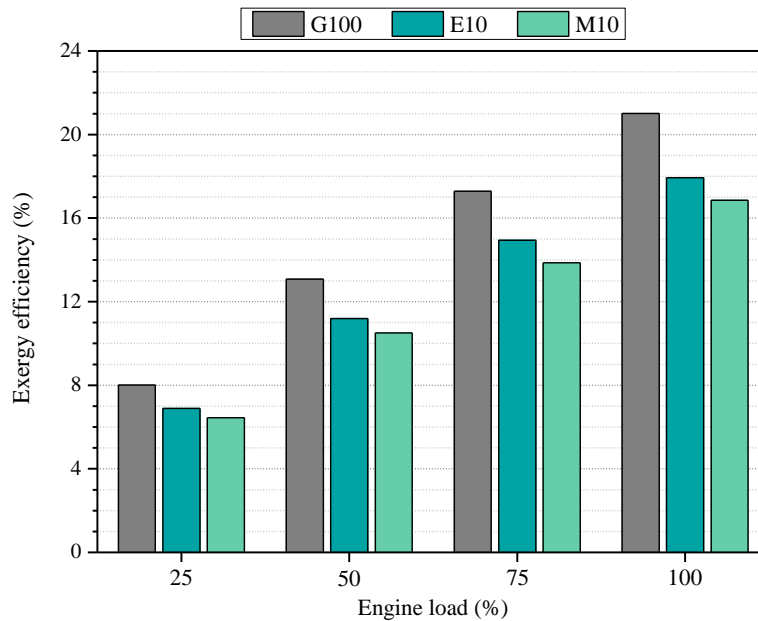
Figure 10. Exergy efficiencies of tested fuels at various engine loads

The sustainability index based on the exergy efficiency of the test engine is given in Figure 11. According to the results, E10 and M10 fuels can be alternatives to G100 fuel. By reducing the thermodynamic irreversibilities that occur in the combustion process, the sustainability index value can be increased. It has been determined that since increasing the engine load increases both energy and exergy efficiencies, the sustainability index also increases. In this study, the highest sustainability index was calculated as 1.26 in G100 fuel at 100% load.
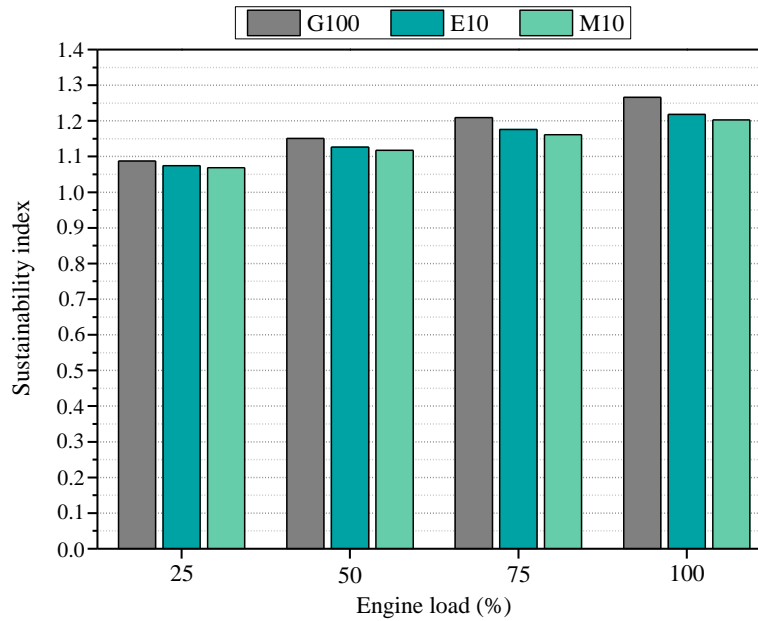
Figure 11. Effect of alcohol addition to gasoline on the sustainability index at various loads

4. Conclusions

In the present research, the performance and exhaust emissions of a single-cylinder, four-stroke, SI engine fueled with G100 (100% gasoline), E10 (10% ethanol + 90% gasoline), and M10 (10% methanol + 90% gasoline) fuel samples at a constant engine speed of 1500 rpm for four different load values (25% - 100%) were experimentally determined. Afterwards, thermodynamic analyses were conducted with the obtained data.

The consumption of E10 and M10 fuels created by adding alcohol-based additives to pure gasoline was approximately 8.8-9.1% and 2.4-3.6% higher in comparison with gasoline, respectively. The lowest BSFC was observed to be as 0.279 kg/kWh in G100 fuel at 100% load.

Due to the inherent oxygen concentrations of ethanol and methanol, CO and HC emissions are lesser in comparison with pure gasoline. In experimental studies, the oxygen emissions measured in the exhaust gas of E10 and M10 fuel blends are higher than that of G100 fuel. In the test engine, the lowest CO emissions in all fuel mixtures occurred at a 75% load. In this load, E10, M10, and G100 fuels emitted 3.15%, 3.22%, and 3.58% CO emissions to the environment, respectively. CO_2 emission is more environmentally friendly since the C/H ratios of alcohols used in the study are lower compared to gasoline. CO_2 emission descended with the rise in load. The lowest CO_2 emission value was determined to be 7.398% in E10 fuel. Since the latent heat of evaporation values of ethanol and methanol are higher than that of gasoline, and their heating values are lower compared to gasoline, NO_X emissions decreased.

Energy and exergy efficiencies increased with the increasing load in fuel blends. At 100% engine load, the highest energy efficiencies of G100, M10, and E10 fuels were calculated to be as 30.01%, 29.90%, and 28.33%, respectively. When the exergy efficiency figures of the tested fuels were evaluated, G100 fuel has performed better than M10 fuel. At the highest engine load operating condition, the exergy efficiency values were calculated to be as 17.92% for E10 fuel, 16.85% for M10 fuel, and 21% for G100 fuel. In comparison with G100 fuel, exergy destruction is higher in E10 and M10 blends, in which the energy content is low, and fuel consumption is high at all engine loads. Entropy generation increases in direct proportion to exergy destruction. The highest entropy generation was occurred to be as 0.0355 kW/K in M10 fuel blend at 100% engine load. According to the calculations of the sustainability index, the highest index coefficient was calculated 1.26 in G100 fuel at 100% load. It was found as 1.21 in E10 fuel and 1.20 in M10 fuel with the same engine load.

As a result, since E10 and M10 fuel blends exhibit similar performance with G100 fuel, they can be assessed as alternative fuels instead of gasoline out of any modification in the engine. However, energy and exergy destruction should be reduced in order to improve the first and second law efficiencies of SI engines. Therefore, in alternative fuel studies, exergo-economic and exergo-enviremental analyses can be performed, and evaluations can be made from an environmental and economic perspective.

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