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RESEARCH ARTICLE-MECHANICAL ENGINEERING

Application of Higher-Order Alcohols (1-Hexanol-C6 and 1-Heptanol-C7) in a Spark-Ignition Engine: Analysis and Assessment

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Abstract

Studies on the usage of gasoline–alcohol blends as an alternative fuel in spark-ignition engines have recently gained momentum. In the present research, energy, exergy, environmental, enviroeconomic, exergoenvironmental, and exergoenviroeconomic analyses were conducted with the performance and emission values acquired by utilizing gasoline, gasoline–heptanol, and gasoline–hexanol fuels (G100, HEX5-20, and HP5-20) as a fuel under different powers at a constant speed of 1600 rpm in a single-cylinder four-stroke spark-ignition engine. As the ratio of alcohol in fuel blends increases, fuel consumption also increases. NO_X emission is higher, and CO and HC emissions are lower in alcohol-based fuel blends than G100 fuel. The highest thermal efficiency is 41.09% in G100 fuel at a power of 5 kW. As the ratio of alcohol in fuel blends increases, thermal efficiency decreases. The highest exergy destruction and entropy generation were determined to be 6.25 kW and 0.02134 kW/K, respectively, in HP20 fuel at a power of 5 kW. Entropy generation increases with an increase in the ratio of alcohol in alcohol-based fuels. HEX20 and HP20 fuels produce 25% and 30% more entropy, respectively, compared to G100 fuel. The mass and financial costs of the damage caused by the CO_2 emission of fuels to the environment were determined by conducting four different analyses using energy and exergy analysis data. According to the exergoenvironmental and exergoenviroeconomic analyses, HP20 fuel reached the highest environmental pollution values of 4538.19 kg CO_2/month and 65.804 $/month, respectively. The environmental cost of the CO_2 emission released from the exhaust to the atmosphere is higher in alcohol-based fuels than G100 fuel. As a result of all analyses, it was concluded that hexanol and heptanol could be alternative fuels in spark-ignition engines under particular conditions.

Keywords Spark-ignition engine · Hexanol · Heptanol · Exergy · Environment

List of symbols

C_{co_2}	CO_2 emission (kg CO_2/time)
Cp	Specific heat capacity (kJ/kgK)
E_{co_2}	Envireconomic parameter ($/time)
Ex_{co_2}	Exergoenvireconomic parameter ($/time)
E_{fuel}	Energy of fuel (kW)
$\dot{E}x$	Exergy rate (kW)
Hu	Heat value of fuel (kJ/kg)
h	Enthalpy (kJ)
$\dot{m}$	Mass flow rate (kg/s)
n	Engine speed (rpm)
N	Total usage time (year)
N_{co_2}	CO_2 emission value (kg CO_2/kWh)
P	Pressure (kPa)
P_0	Pressure of the environment (kPa)
P_{co_2}	CO_2 emission value ($/kW$CO_2$)
$\dot{Q}$	Heat transfer rate (kW)
R	Gas constant (kJ/kgK)
$\overline{R}$	Universal gas constant (8.314 J mol/K)
T	Temperature (K)
T_0	Temperature of the environment (K)
T	Torque (Nm)
t_w	Cumulative working hours (h)
rpm	Revolutions per minute
s	Entropy (kJ/kg.K)

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S_{gen}	Entropy produced (kW/K)
y^e	Component mole fraction (%)
$\dot{W}$	Work (kW

Greek symbols

η	Thermal efficiency
μ	Gas viscosity
ψ	Fuel exergy factor
ε	Flow exergy

Subscripts

a	Air
chem	Chemical
cw	Cooling water
CI	Capital investment
cw	Coolant water
dest	Destruction
g	Exhaust gas
heat	Heat transfer
in	Inlet
k	Kinetic
out	Outlet
p	Potential
phy	Physical
ref	Reference
s	Source
w	Work
0	Environmental conditions

Abbreviations

C_8H_{18}	Gasoline
$C_7H_{16}O$	Heptanol
$C_6H_{14}O$	Hexanol
CO	Carbon monoxide
CO_2	Carbon dioxide
EXENEC	Exergoenviroeconomic
EXEN	Exergoenvironment
G100	100% Gasoline
HC	Hydrocarbon
HEX5	5% Hexanol and 95% gasoline blend
HEX10	10% Hexanol and 90% gasoline blend
HEX15	15% Hexanol and 85% gasoline blend
HEX20	20% Hexanol and 80% gasoline blend
HP5	5% Heptanol and 95% gasoline blend
HP10	10% Heptanol and 90% gasoline blend
HP15	15% Heptanol and 85% gasoline blend
HP20	20% Heptanol and 80% gasoline blend
N	Nitrogen
NO_x	Nitrogen oxide
O_2	Oxygen
PFI	Port fuel injection
TEP	Total environmental pollution

1 Introduction

Internal combustion engines have been used in various fields such as the main power source of planes, ships, agricultural machinery, heavy equipment, generators, and automobile vehicles since the day of their invention. Nowadays, most of the internal combustion engines used in automotive vehicles utilize non-renewable fossil gasoline and diesel fuels [1, 2]. According to the study performed by the United States Energy Information Administration, petrol consumption, which is 86.1 million barrels per day (b/d), was reported to reach approximately 110.6 million barrels per day (b/d) with a 28.45% increase by 2035. Adverse effects such as the decrease in fossil fuel reserves, e.g., petrol, coal, and natural gas, increasing fuel prices, increasing excess demand, environmental threats caused by engine exhaust emissions, effects on global warming, and the availability of reserves in limited regions have caused studies on developing renewable energy sources that can replace fossil fuels used in internal combustion [3–5].

Comprehensive research and development studies are conducted on the use of alcohols, which are among renewable energy sources, in internal combustion engines instead of non-renewable fossil fuels. Alcohols are renewable liquid substances that can be easily acquired from herbal products. Although alcohols have low cetane numbers and low thermal values, their density is higher than gasoline. Alcohol fuels are alternative renewable fuels that can be used by blending pure gasoline and pure diesel fuels in different proportions to reduce air pollution or the use of fossil fuels. Alcohols create less harmful exhaust emissions during combustion due to the oxygen in their molecular structures. Therefore, they are used as additives for fossil fuels in internal combustion engines. Methanol, ethanol, propanol, butanol, pentanol, hexanol, heptanol, octanol, and decanol are known to be alcohols that can be used in internal combustion engines instead of fossil fuels nowadays. Unlike other fuels, these alcohol fuels can be utilized directly in internal combustion engines without any modification, or they can be used by blending them at certain proportions with fossil fuels [6–9].

Hexanol and heptanol are called long-chain alcohols due to carbon numbers in their molecular structures. Hexanol and heptanol contain 6 and 7 carbons in their molecular structure, respectively. The chemical formula of hexanol is $C_6H_{13}OH$, and the chemical formula of heptanol is used in the abbreviated form as $C_7H_{15}OH$ [10–12]. Hexanol is a colorless, poorly water-soluble alcohol with a characteristic odor, high boiling point, and low volatility. Hexanol can be produced industrially by the oligomerization of ethyl alcohol, which can be produced easily and at low cost from waste from agricultural products such as corn, crops, and vegetables. Hexanol is used in various fields such as adhesives, lubricants, odorants, dyes, plasticizers, solid release agents, solvents, hair care products, agricultural products, and textile products [13, 14]. When the research in the literature is reviewed, it is observed that there are few research and development studies on the use of hexanol and heptanol as a fuel in spark-ignition engines. Many studies have been performed to examine the effects of hexanol and heptanol fuels on engine performance and exhaust emissions by blending them with pure diesel fuel in different proportions by volume or mass.

The chemical energy of the fuel is converted into heat energy in internal combustion engines. In these engines, heat energy is converted into mechanical energy, with the sudden temperature and pressure increases of the gases burning in the cylinder, the expansion movement of the piston [15]. Since some of the motion energy obtained with the combustion of fuel in internal combustion engines decreases due to various losses, all input energy cannot be converted to work [16].

In terms of thermodynamics, the performance of internal combustion engines is verified by the first and second law of thermodynamics (availability-exergy analysis) [17]. Energy analysis in engines is based on the first law of thermodynamics [18]. Engine energy distribution, losses, and thermal efficiency are calculated in this analysis. Energy distribution and thermal efficiency are important decision parameters in the design of engines, selection of systems, and determining the system's operating conditions [19, 20].

Based on the second law of thermodynamics, exergy is the usable part of energy. Determining the source of thermodynamic inefficiencies in an engine by exergy analysis is important to obtain more efficient designs [21]. Flow, heat transfer, exhaust and friction losses in engines cause a decrease in the quality of energy and the possibility to do work [22]. Entropy generation is the measure of a decrease in the quality of energy [23]. Entropy generation is calculated based on exergy destruction.

Global warming and increased emission rates of greenhouse gases in the atmosphere cause environmental problems [24]. CO_2 gas is the most important one among greenhouse gases [25]. To reduce the use of gases causing the greenhouse effect, the Kyoto Protocol was adopted on December 11, 1997. Engines that use fossil fuels cause an increase in CO_2 emission in the atmosphere [26]. The way to reduce CO_2 emission in these engines is to consume less fuel and use more environmental-friendly fuels [27]. CO_2 emission provides information about the combustion quality in engines [28]. In the environmental analysis conducted following the laws of thermodynamics, studies that determine the amount and cost of emissions come to the forefront [29, 30].

Aloko et al. [31] experimentally examined different features of hexanol fuel, such as flash point, aniline point, diesel index, distillation range, viscosity, and density, by blending it with pure diesel fuel in different proportions (5%, 10%, 15%, 20%, 25%, 30%, 35%, 40%, and 45%). Furthermore, experimental studies, reported that 5% hexanol–95% diesel fuel blend had similar properties to pure diesel fuel.

In 2017, De Poures et al. [32] performed experimental studies at three different injection times (21°, 23°, and 25° CA bTDC), three different EGR rates (10, 20, and 30%), and at a constant engine speed of 1500 rpm using 10%, 20%, and 30% hexanol–diesel fuel blends in a single-cylinder, four-stroke compression ignition engine and examined performance and exhaust emission values. The researchers stated that with the increase in hexanol ratio in blend fuels, hydrocarbon (HC) and nitrogen oxide (NO_X) emissions increased, and carbon monoxide (CO) emission and smoke density decreased. According to the results, it was reported that NOx emissions increased by approximately 3% and smoke density decreased by 35.9%.

Babu et al. [33] performed experimental studies at a constant engine speed of 1500 rpm using biodiesel/diesel/*n*-pentanol and biodiesel/diesel/*n*-hexanol fuel blends containing 5% and 10% alcohol in a single-cylinder, four-stroke direct injection diesel engine and investigated their effects on characteristics such as brake-specific fuel consumption, brake thermal efficiency, cylinder pressure, ignition delay, combustion duration, and harmful exhaust emission.

Doğan et al. [34] conducted experimental studies under full load and at different engine speeds (2000, 2500, 3000, 3500, 4000, and 4500 rpm) using ethanol and gasoline (E0, E10, E20, E30, and G100) fuel blends in a four-cylinder, water-cooled, four-stroke spark-ignition engine. Using the experimental results, the irreversibilities that occurred in the engine were investigated by energy and exergy analyses. According to the results obtained from the thermodynamic analysis, it was reported that the highest exergy and energy efficiency were obtained at an average engine speed of 2500 rpm. The results obtained from the study revealed that the value of the exhaust exergy lost as waste was high.

Suhaimi et al. [35] investigated the impacts of HE5, HE10, and HE20 hexanol–diesel fuel blends on combustion characteristics, engine performance, and exhaust emission values in a single-cylinder, four-stroke compression ignition engine by performing experimental studies at a constant engine speed of 1800 rpm for different engine loads (25%, 50%, 75%, and 100%). In the experimental studies carried out under the same conditions, it was stated that the brake-specific fuel consumption value of HE5, hexanol–diesel fuel blend at a 100% engine load value had a lower and higher brake thermal efficiency value than other fuel blends.

Pandian et al. [36] performed experimental studies for different engine loads (25%, 50%, 75%, and 100%) at a constant engine speed of 1300 rpm with 10% and 20% hexanol fuel blends into biodiesel produced from cashew nut shell oil in a single-cylinder, four-stroke, air-cooled Kirloskar compression ignition engine and investigated their effects on values such as brake-specific fuel consumption, brake thermal efficiency, and harmful exhaust emission. According to the results obtained, it was observed that hexanol–biodiesel fuel blends had a lower effect on reducing CO, HC, NO_X, and smoke emissions in comparison with biodiesel fuel.

Ashok et al. [37] created triple blend fuels in different proportions using conventional diesel, pure biodiesel, decanol, and hexanol fuels. The effects of the resulting triple fuel blends on characteristics such as brake thermal efficiency, brake-specific fuel consumption, brake-specific energy consumption, cylinder pressure, and harmful exhaust emissions were investigated in a single-cylinder, four-stroke, water-cooled, Kirloskar TV1 compression ignition engine.

Kumar et al. [38] conducted combustion analysis and examined harmful exhaust emission values by using fuel blends obtained by mixing four different alcohols, isobutanol, *n*-pentanol, *n*-hexanol, and *n*-octanol with ultra-low sulfur diesel fuel at 30% by volume in a single-cylinder, four-stroke, air-cooled, Kirloskar compression ignition engine. In another study performed by Kumar et al. [39], it was stated that methanol and ethanol, short-chain alcohols, were the most investigated renewable fuels in internal combustion engines. Furthermore, it was reported that a significant number of research and development studies recently accelerated in high-chain alcohols due to the development of modern fermentation processes increasing efficiency. The researchers emphasized that studies on heavy alcohols such as hexanol and heptanol were insufficient and more investigations were required.

Nour et al. [40] performed experimental studies for different engine loads (25%, 50%, 75%, and 100%) at constant engine speeds of 900 and 1500 rpm in a single-cylinder, direct injection, naturally aspirated, air-cooled, Deutz F1L511 compression ignition engine. In this study, fuel blends were obtained by blending butanol, heptanol, and octanol fuels with pure diesel fuel at 10% and 20% by volume. According to the results obtained, they reported that the brake-specific fuel consumption and brake thermal efficiency values increased for all test fuels in comparison with pure diesel fuel. It was indicated that while NO_X and smoke emissions decreased, CO and HC emissions increased for all test fuels compared to pure diesel fuel.

Devarajan, Y. [41] conducted experimental studies for D100, M100, M90H10, and M80H20 fuel blends obtained by blending mustard oil biodiesel fuel and *n*-heptanol fuel at 10% and 20% by volume in a four-stroke, single-cylinder, air-cooled, compression ignition engine. They examined performance and emission values in the experimental studies carried out at an engine speed of 1800 rpm and different engine loads (25%, 50%, 75%, and 100%) for all fuels. It was indicated that adding *n*-heptanol to mustard oil biodiesel fuel reduced brake-specific fuel consumption and increased brake thermal efficiency. Moreover, it was reported that carbon monoxide (CO), hydrocarbon (HC), and nitrogen oxide (NO_X) harmful exhaust emissions significantly decreased as the ratio of heptanol in fuel blends increased.

Yesilyurt [42] carried out experimental studies for D100, B100, B20D80, H20D80, and B20H20D60 fuel blends by blending peanut oil biodiesel fuel, *n*-heptanol fuel, and pure diesel fuels in different proportions in a single-cylinder, four-stroke, water-cooled, Kirloskar TV1 compression ignition engine. In the experimental studies carried out at an engine speed of 1500 rpm and various engine loads (25%, 50%, 75%, and 100%) for all fuels, their effects on engine performance characteristics including brake thermal efficiency, brake-specific fuel consumption, brake-specific energy consumption, exhaust gas temperature, and harmful exhaust emissions were investigated.

Uslu and Çelik [43] performed experimental studies for different compression ratios (8.0:1, 8.5:1, and 9.0:1) and different engine speeds (2600, 2800, 3000, and 3200 rpm) using isoamyl alcohol and gasoline (A10, A20, A30, and G100) fuels in a single-cylinder, air-cooled, four-stroke, spark-ignition engine. In accordance with the findings obtained, it was reported that as the alcohol ratio in fuel blends increased, there was a significant increase in engine torque, engine power, brake thermal efficiency, brake-specific fuel consumption, and brake-specific energy consumption values. Furthermore, it was indicated that the harmful exhaust emissions of carbon monoxide (CO), hydrocarbon (HC), and nitrogen oxide (NO_X) decreased significantly at all compression ratios. In the experimental studies performed under the same conditions, it was stated that A30 isoamyl alcohol–gasoline fuel blend exhibited the best performance in comparison with other fuel blends.

Doğan et al. [44] conducted exergy, exergoeconomic, and environmental analyses with the performance and emission values acquired by utilizing ethanol–gasoline fuel blends in different proportions under full load in a four-cylinder, four-stroke, spark-ignition engine. The researchers examined the exergy of the fuel entering the engine and the exergy lost through the cooling system, exhaust, and radiation under various conditions. In the study, they determined the annual cost of CO_2 emission, which represents a significant environmental hazard for the world we live in. Moreover, the researchers compared the cost of exergy losses and exergy destruction occurring in the engine with the cost of engine power. Sustainability analysis, which has a significant effect on the efficient and effective usage of energy resources, was conducted, and the most sustainable fuel blends were identified based on the SI index. As a result of the analysis conducted, it was stated that the annual CO_2 consumption cost of E30 fuel at 2500 rpm was 0.31 USD/year CO_2, while it was 0.39 USD/year CO_2 for E10 fuel. It was reported that as the ratio of ethanol in fuel blends increased, the amount of CO_2 released into the atmosphere decreased by approximately 25.8%.

Şöhret et al. [45] experimentally investigated the effects of different compression ratios and different spark-ignition times on engine performance at a constant engine speed of 1600 rpm and 100% throttle opening values using 99.99% pure hydrogen fuel in a single-cylinder, air-cooled, four-stroke, spark-ignition engine. Energy and exergy analyses were conducted using these engine performance results obtained. It was stated that the increase in the compression ratio caused a positive increase in the performance parameters of the engine, and there was a positive decrease in exergy destruction. However, it was reported that deviating from the optimum value of spark-ignition timing resulted in a decrease in engine performance parameters and an increase in exergy destruction.

Bhatti et al. [46] conducted experimental studies using gasoline as a fuel at various compression ratios (6.0:1, 7.0:1 8.0:1, 9.0:1, and 10.0:1) and different engine speeds (1200, 1300, 1400, 1500, 1600, 1700, and 1800 rpm) in a single-cylinder, four-stroke, water-cooled, Kirloskar TV1 spark-ignition engine. These engine performance results indicated that the maximum energy and exergy efficiencies were obtained to be 28.55% and 27.35%, respectively, at 1200 rpm and a compression ratio of 9.0:1.

Rufino et al. [47] performed experimental studies under different engine characteristics using E27 and E95 ethanol–gasoline fuels, which are commercially available in Brazil, in a three-cylinder, four-stroke, water-cooled, spark-ignition engine. They conducted energy and exergy analyses using the two-zone combustion model they developed.

Doğan et al. [48] conducted experimental studies to reveal the performance characteristics of the engine for various engine load values (25%, 50%, 75%, and 100%) at a constant engine speed of 1500 rpm with fuel blends obtained by adding 1-heptanol at 5%, 10%, and 20% by volume to diesel fuel in a single-cylinder, water-cooled, four-stroke direct injection compression ignition engine. They conducted the energy, exergy, exergoeconomic, enviroeconomic, and sustainability analyses of the engine by utilizing the acquired engine performance and exhaust emission data. The exergoenvironmental analysis in this study indicated that the highest CO_2 emission cost was 66.94 USD/year at a 100% load in Hp10 fuel. Moreover, the exergoenviroeconomic analysis was carried out in the study, reporting that the highest engine output power cost was obtained to be 1.6 USD/MJ in Hp20 fuel at a 25% load. Since the physical and chemical features of pure heptanol are very close to those of conventional diesel fuel, it was stated that the heptanol–diesel blends utilized in the research had similar performance values to pure diesel fuel concerning energy and exergy efficiency.

When studies in the literature are reviewed, it is observed that research has been conducted with gasoline/heptanol/hexanol fuel blends within the scope of studies on alternative fuel in spark-ignition engines. Alcohols in the range of 5–30% by volume were added to gasoline in these fuel blends. A decrease occurred in engine power and engine torque due to adding alcohol to gasoline. The reason for this is the low heat values and densities of alcohols compared to gasoline. Studies in the literature revealed an improvement in emissions due to the oxygen content in the structures of alcohol-based fuels. The self-ignition temperature of alcohols, their high octane number, and high evaporation temperature provided an advantage. However, for the use of alcohols as an alternative fuel, the fuel equipment of engines must be adapted to alcohol fuels, and the cost of alcohol fuels must be reduced.

Nowadays, thermodynamic analysis studies that use engine performance and emission measurement data are widely used in alternative fuel research and engine technologies. This analysis gives the distribution of fuel energy spent by various engine components. Furthermore, it enables the comparison of engine performance with maximum performance with exergy analysis. The environmental cost of the CO_2 emission of alcohol-based fuels is higher than that of gasoline.

As can be observed from the studies presented above, it is observed that the research on the use of hexanol and heptanol in spark-ignition engines is limited, and studies are generally conducted on the exhaust emissions and engine performances in compression ignition engines. In this study, thermodynamic evaluations were made in terms of the energy, exergy, environmental, enviroeconomic, exergoenvironmental, and exergoenviroeconomic analysis of hexanol–gasoline and heptanol–gasoline fuel blends in a spark-ignition engine. In the energy analysis, energy flow, heat losses, and thermal efficiency were calculated. In the exergy analysis, exhaust exergy, exergy destructed, and exergy efficiency were found. Considering the amount of CO_2 released into the environment from the engine exhaust, environmental assessments based on energy and exergy analysis were made.

When studies in the literature are reviewed, there are generally energy and exergy analyses in gasoline/ethanol/butanol blends and diesel/ethanol/butanol/heptanol blends. This study conducted energy and exergy analyses in fuel blends using heptanol and hexanol in a spark-ignition engine. Furthermore, a new evaluation criterion was determined for alcohol-based fuels by finding the total environmental pollution costs of fuel blends in the environmental analysis.

This study's purpose is to determine the ratio that gives the most suitable engine performance and emission values in fuel blends of gasoline/hexanol/heptanol. In line with this purpose, the energetic and exergetic performances of different mixing ratios were evaluated. Environmental analysis is important for a fuel to become an alternative to gasoline. This study determined the environmental costs of fuel blends.

2 Material and Method

Environmental impact is a parameter that is important as much as performance in studies on the use of alternative fuels in internal combustion engines. In the study, the performance and emissions of fuels used in a spark-ignition engine were tested. After these tests, evaluations in terms of losses and efficiency were made with thermodynamic analysis. Environmental damage was calculated with the thermodynamic analysis results.

2.1 Experimental Setup

In the analyses, the recorded data from the engine trials were used to evaluate different hexanol/gasoline and heptanol/gasoline blends from the energy, exergy, environment, and economic perspectives under various engine loading conditions. The engine tests were performed by using an SI engine with single-cylinder and four-stroke. The tested engine was loaded with the help of an eddy-current dynamometer ranging from 1 to 5 kW by steps of 1 kW. The tested SI engine had a water-cooling system for descending the temperature of the engine parts in order not to damage them. Furthermore, the compression ratio was kept constant at 10:1 during the engine experiments. The cylinder volume of the engine was 661.5 cc, and it had two valves on the cylinder. A port-fuel injection (PFI) system was applied to inject the fuel specimens into the engine cylinder, and it was facilitated by the electronic ignition unit. The technical specifications of the experimental research engine are tabulated in Table 1. In addition, the schematic presentation of the experimental test rig used in the present work is shown in Fig. 1.

The operating conditions for the tested gasoline engine were balanced before beginning the observation of the measured data for each type of fuel (1-hexanol/gasoline blends, 1-heptanol/gasoline blends, and neat gasoline). With a focus on avoiding faults, the trials were actualized three times at each working status for all tested fuel samples, and the average findings were used in the thermodynamic analyses. The impacts of the test fuels on the harmful pollutants and engine performance were addressed, and the recorded data were used in analyzing the SI engine from the energy, exergy, environmental, enviroeconomic, exergoenvironmental, and exergoenviroeconomic perspectives. For the purposes mentioned above, the assorted sensors and pieces of equipment such as encoder, pressure sensor, temperature indicators, fuel measurement system, air consumption measurement system, etc., were attached to the researched engine carry out the measurements for the engine behaviors. Furthermore, many connectors were put into action to feed the signals supplied from the dissimilar sensors to the amplifier following the data acquisition system connected to the work station. The work station was accessed to write down the data and plot them on the screen in various figures. To sum up, the following flowchart was applied for the experiment point by point: (i) the tested fuels were placed into the fuel tank shown in Fig. 1, (ii) pieces of equipment, as well as sensors, were authorized, (iii) the steady-state conditions were ensured throughout the test engine operations. After commencing the SI engine, it was consent out of loading for a short time on account of warming the engine and then the records were addressed.

Table 1 Technical specifications of the experimental engine

Item	Specification
Brand	Kirloskar VCR engine
Model	TV 1
Number of the cylinder	1
Number of the stroke	4
Maximum engine power	7.5 kW
Maximum engine speed	1850 rpm
Bore x Stroke	87.5 mm × 110 mm
Length of the connecting rod	230 mm
Cylinder volume	661.5 cc
Variable compression ratio	6–10
Type of the cooling system	Water-cooled
Idling speed	750 rpm
Inlet valve opening	4.5° before TDC
Inlet valve closing	35.5° after BDC
Exhaust valve opening	35.5° before BDC
Exhaust valve closing	4.5° after TDC
Loading capacity	0 to 50 kg

To measure the harmful pollutant gases throughout the trials from the SI engine operating with the tested fuel samples, an exhaust gas analyzer device (Bosch brand BEA 350 model) was used. This apparatus has implemented a non-dispersive infrared method and an electrochemical transmitter technique ascribable to analyze the pollutants. During the measurements, five diverse gases such as carbon monoxide, carbon dioxide, nitrogen monoxide, oxygen, and unburnt hydrocarbon were taken into account as exhaust emissions. The gas analyzer device was calibrated with the aid of the reference gases to ensure the truthfulness of the recorded outcomes. Coupled with it, numerous thermocouples (K-type) that can measure the temperature ranging from 0 to 1200 °C were placed in suitable locations such as inlets and outlets of the

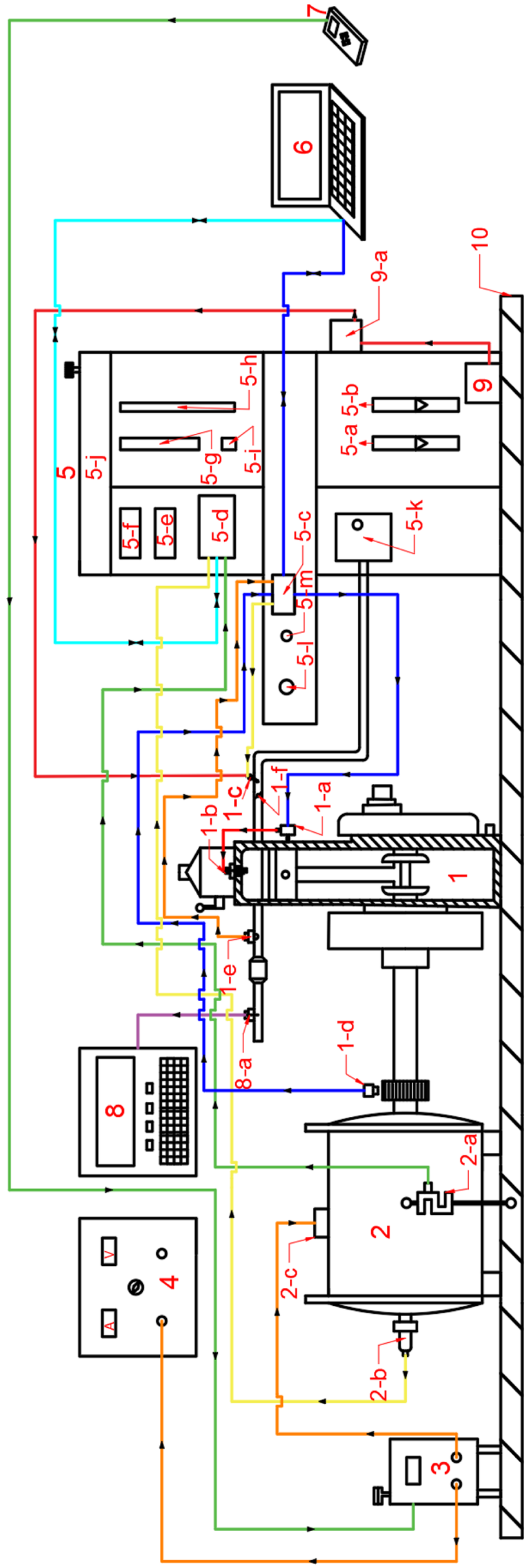

Fig. 1 Schematic layout of the experimental apparatus

1 variable compression ratio engine (VCR), 1-a ignition coil, 1-b spark plug, 1-c gasoline injector, 1-d hall effect sensor, 1-e O_2 sensor,
1-f throttle valve
2 AC dynamometer, 2-a load-cell, 2-b encoder, 2-c dynamometer and speed control connection
3 dynamometer speed control unit 11 kW
4 AC dynamometer 380 V board
5 measuring component stand, 5-a engine cooling rotameter, 5-b exhaust calorimeter rotameter, 5-c gasoline engine ECU, 5-d data acquisition device, 5-e load indicator, 5-f rpm indicator, 5-g manometer, 5-h fuel measuring burette, 5-i fuel cock, 5-j fuel tank, 5-k engine intake air measurement system, 5-l throttle position control system, 5-m system switch
6 computer interface
7 load and speed control interface device
8 gas analyser device
9 gasoline injection pump, 9-a gasoline filter

exhaust pipe and exhaust calorimeter apparatus to observe the temperature. The technical specifications of the exhaust gas analyzer device used in this study are presented in Table 2.

In the current study, a square-root technique that was recommended by Holman was implemented to the findings from the engine trials to calculate the uncertainties [49]. A

Table 2 Uncertainties of the measured and calculated parameters

Parameter	Measurement range	Accuracy	Uncertainty (%)
Engine power	0–7.5 kW	±0.33 kW	±0.683
Torque	0–112 Nm	±0.1 Nm	±0.500
Fuel flow rate	-	-	±0.563
Air flow rate	-	-	±0.606
Temperature	-	±0.1 ℃	±1.0
Digital stop watch	-	±0.2 s	±0.3
Engine load	0–20 kg	±0.1 kg	±0.714
Engine speed	0–9999 rpm	±1 rpm	±0.063
Burette system	0–100 cc	±0.1 cc	±1.0
BSFC	-	-	±0.856
CO	0–10% vol	±0.001%	±0.771
HC	0–9999 ppm	±1 ppm	±0.399
CO_2	0–18% vol	±0.01%	±0.653
NO	0–5000 ppm	±1 ppm	±0.598
O_2	0–22% vol	±0.01%	±0.593
λ	0.500–9.999	0.001%	±0.530
Energy efficiency	-	-	±0.871
Exergy efficiency	-	-	±0.929

Table 3 Technical properties of test fuels used in the experimental study [34, 36, 40, 50, 51]

Specifications	Unit	Gasoline	1-hexanol	1-heptanol
Molecular formula	-	C_8H_{18}	$C_6H_{13}OH$	$C_7H_{15}OH$
Molecular weight	kg/kmol	114	102.18	116.19
Carbon content	wt. %	84.21	70.52	72.16
Hydrogen content	wt. %	15.79	13.70	13.71
Oxygen content	wt. %	0	15.70	14.13
Octane number	-	95	69.3	-
Density at 20 °C	g/mL	0.715	0.820	0.822
Self-ignition temperature	°C		285	275
Flash point	°C		60	100
Lower heating value	MJ/kg	44.25	39.10	34.65
Boiling point	°C	210	157	175
Latent heat of vaporization	kJ/kg	380–400	486	574.95
Kinematic viscosity at 40 °C	cSt	0.494	3.64	3.32
Vapor pressure	mmHg		1	0.5
Assay	%	-	≥98	≥99

general form of the above-mentioned formula is shown in Eq. (1);

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \ldots + \left(\frac{\partial R}{\partial x_n} w_n \right)^2 \right]^{1/2} \tag{1}$$

where w_1, w_2, …, w_n indicate the uncertainties of the independent variables, w_R infers the uncertainty value of the results, $x_1, x_2, x_3, \ldots, x_n$ denote the independent variables, and R defines the dependent factor. Moreover, R is a function of independent variables.

Table 2 summarizes the uncertainty values of the measured and calculated parameters during the engine tests.

2.2 Test Fuels

In the present study, 1-hexanol ($C_6H_{14}O$) and 1-heptanol ($C_7H_{16}O$) long-chain alcohols were procured from Merck (Darmstadt, Germany) and Sigma-Aldrich Chemical Company (St. Louis, Missouri, USA), while unleaded gasoline was bought from a local petroleum station located in Kirikkale, Turkey. To prepare the alternating alcohol/gasoline blends, the splash blending technique was applied to the test fuels. The fuel blends were effectuated on a volume basis. In the test fuels, 1-hexanol and 1-heptanol were used as oxygenated alternating additives for gasoline fuel. Test fuels and their blending ratios were presented as follows: 100% gasoline (G100), 5% 1-hexanol/95% gasoline (HEX5), 10% 1-hexanol/90% gasoline (HEX10), 15% 1-hexanol/85% gasoline (HEX15), 20% 1-hexanol/80% gasoline (HEX20), 5% 1-heptanol/95% gasoline (HP5), 10% 1-heptanol/90% gasoline (HP10), 15% 1-heptanol/85% gasoline (HP15), and 20% 1-heptanol/80% gasoline (HP20). As a result of the investigations and observations made, it was found that there was no phase separation in the fuel blends. Some of the significant physical and chemical properties were adopted from recent literature and are presented in Table 3. The flowchart of the analyses in the study is shown in Fig. 2.

3 Analyses

Using inadequate technologies and low-efficiency fuels in internal combustion engines increases costs and affects the environment adversely. Energy and exergy analyses attract the attention of researchers in the evaluation of the fuels used in these engines. Exergy is an interdisciplinary concept related to economics and the environment. The flowchart of the analyses in the study is shown in Fig. 3. Therefore, exergy analysis is necessary to evaluate the possible effects on the environment during the use of fuels. In this study, the following assumptions were made while performing the exergy analysis of gasoline, hexanol–gasoline, and heptanol–gasoline fuel blends. The control volume used in the current study for thermodynamic analysis is displayed in Fig. 4.

- Air and exhaust gases are ideal gases.

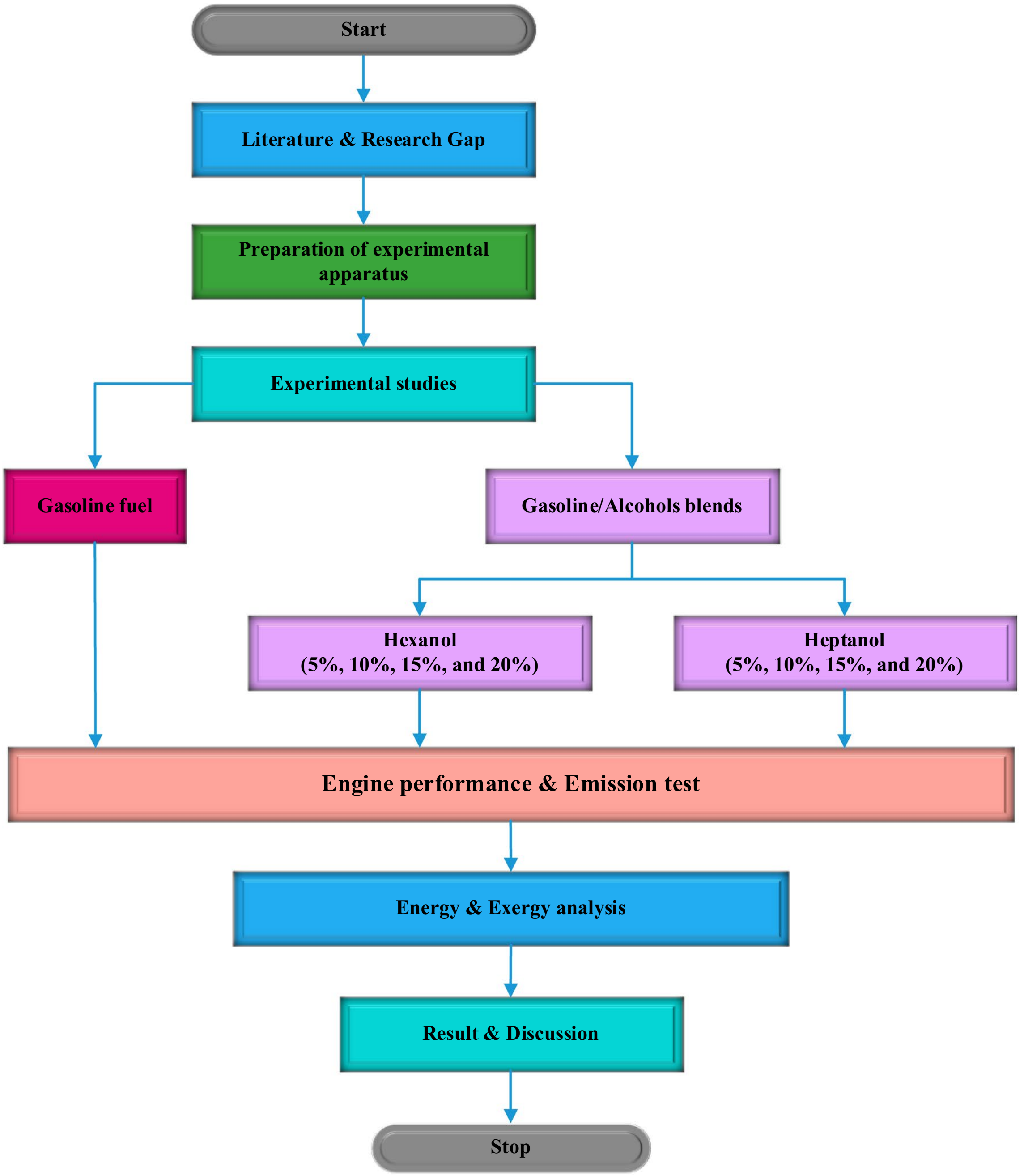

Fig. 2 Flowchart of the experimental studies

- For specific heat capacity values, the constant temperature approach was applied by averaging inlet and outlet temperatures.
- Changes in kinetic and potential exergy were neglected.
- The combustion reaction was completed.
- The engine runs in a steady state.
- Combustion products are in chemical balance.
- In the tests, the ambient temperature was taken as 293 K, and the pressure was taken as 1 atm.

3.1 Energy Analysis

The power generated in the engine depends on the angular velocity (ω) and the engine torque (T), as given in Eq. (2). In an internal combustion engine, if the system takes as much heat as Q and does work as much as W, the energy balance of the system is expressed by Eq. (4) [52–56].

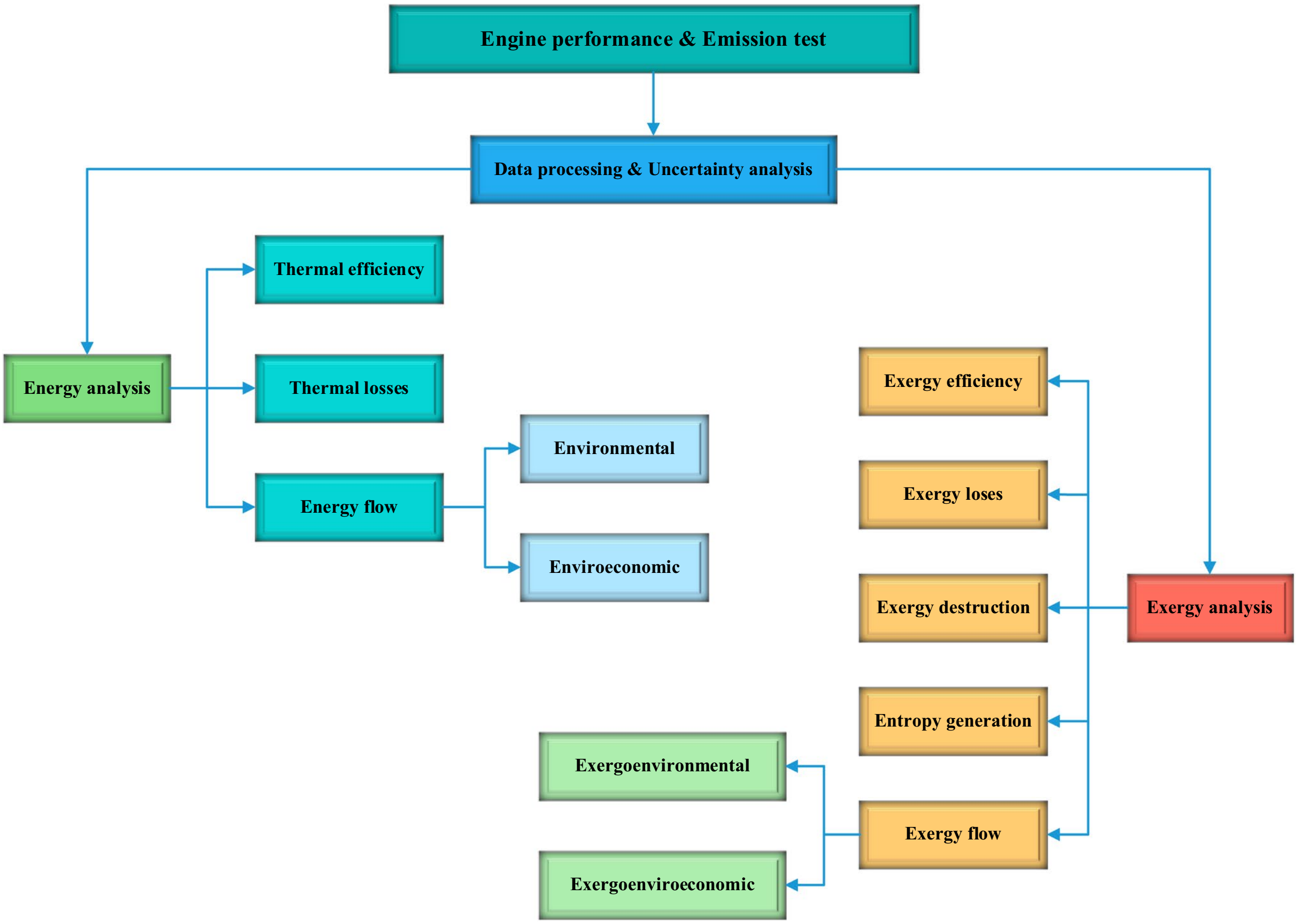

Fig. 3 Flowchart of the energy, exergy, and environmental analysis

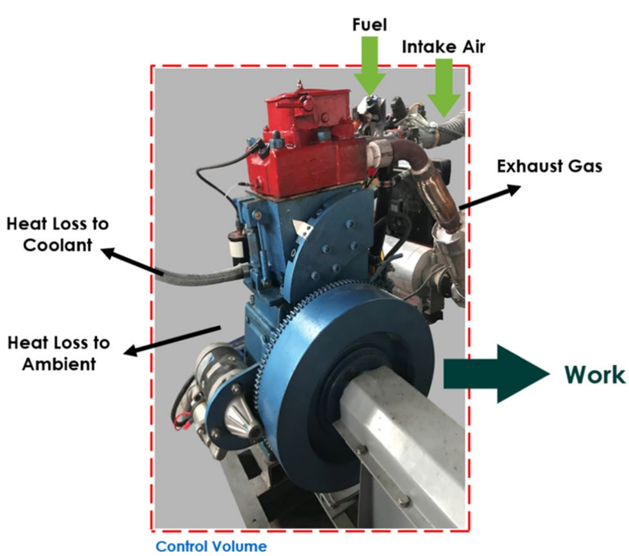

Fig. 4 Control volume for thermodynamic analysis

$$\dot{W} = \omega \cdot T \tag{2}$$

$$\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}} \tag{3}$$

$$\dot{Q} - \dot{W} = \sum \dot{m}_{\text{out}} h_{\text{out}} - \sum \dot{m}_{\text{in}} h_{\text{in}} \tag{4}$$

In the energy analysis, the energy entering the control volume can be calculated from Eq. (5) using the lower heating values (H_u) and mass flow rates ($\dot{m}_{\text{fuel}}$) of different gasoline–heptanol–hexanol fuel blends, and the total heat losses in the engine ($\dot{Q}_{\text{lost}}$) can be calculated using Eq. (6). The thermal efficiency, which represents the ratio of the work acquired from the engine to the energy given, can be found using Eqs. (7) [57, 58].

$$\dot{E}_{\text{fuel}} = \dot{m}_{\text{fuel}} H_u \tag{5}$$

$$\dot{Q}_{lost} = \dot{E}_{fuel} - \dot{W} \tag{6}$$

$$\eta = \frac{\dot{W}}{\dot{E}_{\text{fuel}}} \tag{7}$$

3.2 Exergy Analysis

In the exergy analysis in internal combustion engines, the exergy entering the control volume $\left(\dot{E}x_{\text{in}}\right)$ is equal to the exergy flow exiting the system $\left(\dot{E}x_{\text{out}}\right)$. For unit mass, total exergy (ε) is the sum of kinetic (ε_k), physical (ε_{phy}), potential (ε_p), and chemical ($\varepsilon_{\text{chem}}$) exergy [34, 59].

$$\dot{E}x_{\text{in}} = \dot{E}x_{\text{out}} \tag{8}$$

$$\varepsilon = \varepsilon_k + \varepsilon_{\text{phy}} + \varepsilon_p + \varepsilon_{\text{chem}} \tag{9}$$

In the present study neglected changes in kinetic and potential energy. Chemical and physical exergies are calculated using Eqs. (10) and (11). Here, the terms h, s, $\overline{R}$,y^e and T_0 are enthalpy, entropy, gas constant, mole fraction, and ambient temperature, respectively. The index "0" indicates the dead state. While calculating the chemical exergy, (y^e) is the mole fraction of the component given in Table 4 in the reference environment [50, 61].

$$\varepsilon_{\text{phy}} = \left[\left(h - T_0 s\right) - \left(h_0 - T_0 s_0\right)\right] \tag{10}$$

$$\varepsilon_{\text{chem}} = \overline{R} T_0 \ln \frac{1}{y^e} \tag{11}$$

Exergy is the portion of the energy given by the fuel that transforms into another form of energy (mechanical energy). As given in Eq. (12), the usable energy of the fuel, in other words, its useful part, is equal to the work exergy [63].

$$\dot{E}x_w = \dot{W} \tag{12}$$

In accordance with the second law of thermodynamics, the exergy balance can be written using Eq. (13). Here, $\left(\dot{E}x_{\text{heat}}\right)$ refers to the exergy flow generated by heat transfer, $\left(\dot{E}x_W\right)$ denotes exergetic power, and ($\dot{E}x_{\text{dest}}$) denotes irreversibility (exergy destruction). Exergy destruction originates from the inadequate use of fuel exergy to produce useful mechanical work in an internal combustion engine [64].

$$\sum \dot{m}_{\text{in}} \varepsilon_{\text{in}} = \sum \dot{m}_{\text{out}} \varepsilon_{\text{out}} + \dot{E}x_{\text{heat}} + \dot{E}x_W + \dot{E}x_{\text{dest}} \tag{13}$$

The exergy flow due to the heat transferred to the environment in the test engine is calculated using Eq. (14), and the heat discharged from the cooling water at temperature T_{cw} to the environment (Q_{cw}) is calculated using Eq. (15) [65].

$$\dot{E}x_{\text{heat}} = \sum \left(1 - \frac{T_0}{T_{\text{cw}}}\right) \dot{Q}_{\text{cw}} \tag{14}$$

$$\dot{Q}_{\text{cw}} = \dot{m}_{\text{fuel}} H_u - \left(\dot{E}x_w + \dot{m}_{\text{out}} \Delta h_{\text{out}}\right) \tag{15}$$

The exergy of the fuel supplied to the engine is calculated using Eq. (17). Here, the chemical exergy factor (φ) of the fuel is found using Eq. (18) with the hydrogen (h), carbon (c), sulfur (α), and oxygen (o) ratios. Since the air entering the engine cylinders was under atmospheric conditions, its exergy was neglected [44].

$$\dot{E}x_{in} = \dot{m}_{fuel} \varepsilon_{fuel} \tag{16}$$

$$\varepsilon_{fuel} = H_u \varphi \tag{17}$$

$$\varphi = 1.0401 + 0.1728\frac{h}{c} + 0.0432\frac{o}{c} + 0.2169\frac{\alpha}{c}\left(1 - 2.0628\frac{h}{c}\right) \tag{18}$$

After completing all exergy calculations, exergy destruction can be computed using the exergy balance in Eq. (13). Exergy efficiency is calculated from the ratio of the exergy of the fuel entering the engine and exergetic power, as given in Eq. (19). Exergy is also destroyed with the production of entropy $\left(\dot{s}_{\text{gen}}\right)$ in internal combustion engines. The entropy generated can be calculated using Eq. (20) [66–69].

$$\Psi = \frac{\dot{E}x_W}{\dot{E}x_{\text{in}}} \tag{19}$$

$$\dot{s}_{\text{gen}} = \frac{\dot{E}x_{\text{dest}}}{T_0} \tag{20}$$

Table 4 Mole fraction of the reference environment [62]

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

3.3 Life Cycle-Based Environmental and Enviroeconomic Analyses

Environmental awareness, which has recently been developed with the Kyoto Protocol and Paris Agreements, has accelerated studies on reducing harmful emissions released into the environment from internal combustion engines. Alternative fuels have gained importance due to the depletion of fossil fuels used in these engines and their harmful impacts on the environment. In the selection of an alternative fuel, its environmental impact and being economical, as well as performance, are the reasons for its preference.

In spark-ignition engines, a certain amount of carbon dioxide is formed between the products of combustion as a result of the combustion of the fuel. CO_2 is a natural component in the atmospheric air. Therefore, CO_2 formed after combustion in the engine is not classified as a pollutant. However, the continuous increase in carbon dioxide in the atmosphere leads to climate change. In the simplest way, the amount of CO_2 released from the exhaust can be reduced by decreasing the amount of the fuel used. This can be ensured by developing engines with low fuel consumption. Another way is to use fuels with low CO_2 emissions in the engine.

3.3.1 Environmental Analysis

Emissions from spark-ignition engines affect the environment adversely. To calculate this adverse effect, the power obtained from the engine and the amount of CO_2 released into the atmosphere should be determined while conducting the environmental analysis. At the next step, the running time of the engine can be predicted. In this study, CO_2 emission in the range of 15–16% was revealed in all fuel blends in the test engine. Using these data, the CO_2 emission released into the atmosphere during a certain working period (10 h/day) is calculated by mass in Eq. (21). Here, N_{CO_2} is the CO_2 emission (kg CO_2/time) released during the operation time, E_{in} is the energy flow of the fuel that enters the engine (kW), and t_w (h/time) is the operation time at a certain period [70, 71].

$$C_{CO_2} = N_{CO_2}\dot{E}_{in}t_w \tag{21}$$

3.3.2 Enviroeconomic Analysis

Enviroeconomic analysis is a methodology developed to economically determine the environmental impacts of CO_2 emissions causing global warming. The economic value of mass CO_2 emission obtained from the environmental analysis can be calculated using Eq. (22). Here, (EC_{CO_2}) is the enviroeconomic parameter ($/time) and ($P_{CO_2}$) is the price of carbon dioxide emission ($/kg CO_2). In this study, it was taken as P_{CO_2} = 0.0145 $/kg CO_2 [72, 73].

$$EC_{CO_2} = C_{CO_2}\, P_{CO_2} \tag{22}$$

3.3.3 Exorgoenvironmental (EXEN) Analysis

EXEN analysis is an exergy-based method employed in the environmental evaluation of internal combustion engines. In this analysis, the exergy analysis of fuel blends should be performed first. Furthermore, it is necessary to determine the CO_2 emission released from the engine exhaust and predict the operating hours. With EXEN analysis, the CO_2 emission released to the atmosphere at a certain time (10 h/day) is determined using the exergy flow $\dot{E}x_{in}$ of the fuel blends entering the engine C_{xCO_2} by Eq. (23) [74, 75].

$$C_{xCO_2} = N_{CO_2}\dot{E}x_{in}t_w \tag{23}$$

3.4 Exorgoenviroeconomic (EXENEC) Analysis

The economic calculation of the exergetic value of CO_2 emission determined by mass with EXEN analysis is performed with EXENEC analysis using Eq. (24). Here, the exergetic price of the CO_2 emission released from the engine is calculated with.

In the study, the total environmental pollution cost (TEP) of the fuels used in the test engine was found. These costs were calculated with different parameters according to the energy analysis (E_{TEP}) and exergy analysis (Ex_{TEP}) results. In these calculations, considering the CO_2 released into the environment with the combustion of fuels that provide energy throughout the engine's life, it is used in Eqs. (25) and (26). Here, N is the total useful life of the engine (20 years), and t_w was taken as the operating hours for a day (10 h/day) [76, 77].

$$ExC_{CO_2} = C_{xCO_2}\, P_{CO_2} \tag{24}$$

$$E_{TEP} = C_{CO_2}\left(N_{CO_2}\dot{E}_{in}t_wN\right) \tag{25}$$

$$Ex_{TEP} = C_{CO_2}\left(N_{CO_2}\dot{E}x_{in}t_wN\right) \tag{26}$$

4 Results and Discussion

In the present research, first, the performance and emission measurements of G100, HEX5-20, and HP5-20 fuels were made in a spark-ignition engine. Using the data obtained from these measurements, efficiency, losses, and entropy generation were calculated with energy and exergy analysis. The environmental impact of CO_2 emission was determined by mass and on a cost basis with life cycle-based environmental, exergy, environmental, and economic analyses using exhaust emissions.

The fuel consumption of the fuel blends in the measurements taken at different powers in the test engine is presented in Fig. 5. Fuel consumption increases with an increase in engine power. The lowest fuel consumption at all engine powers is obtained to be 0.197 kg/kWh in G100 fuel at a power value of 5 kW, and the highest fuel consumption is obtained to be 0.625 kg/kWh in HP20 fuel at a power value of 1 kW. Since the engine operating condition is realized at full load and full throttle opening, in other words, the throttle opening is not changed under all engine operating conditions, the lowest and highest amount of the fuel taken into the cylinder was observed to be at least in G100 fuel and at most in HP20 fuel.

Concerning the lower heating value in internal combustion engines, more fuel should be used to obtain the same power. In studies in the literature, fuel consumption increases in blends in which alcohol-based fuels are used [30, 34, 39]. When the lower heating values of the fuels in the study are examined, the lower heating value of gasoline is 11% and 21% higher than that of hexanol and heptanol, respectively. Hexanol and heptanol blends consumed approximately 1.94–24.1% and 6.82–35.71% more fuel, respectively, compared to G100 fuel at all powers. The high density of the fuels used in engines means more fuel consumption in the same volume. The density of G100 fuel is higher compared to heptanol and hexanol. In the tests performed in the study, it was observed that the fuel consumption did not change depending only on the lower heating value and density, and parameters such as the fuel–air blend degree and homogeneity, the boiling/evaporation points of the blend and the air/fuel ratio were effective.

CO_2 emission in spark-ignition engines gives important information about the combustion quality. In these engines, the amount of CO_2 in the combustion products is directly related to the fuel properties such as the C/H ratio and O_2 concentration of the fuel. The C/H ratio of hexanol and heptanol is low compared to G100 fuel. As can be observed from Fig. 6, the CO_2 ratio is higher in blend fuels in comparison with G100 fuel. Therefore, CO_2 emission was measured to be 15.292% at least in G100 fuel at a power of 1 kW and 15.599% at most in HEX20 fuel at a power of 5 kW. At all power values, the CO_2 ratio of HEX5-20 and HP5-20 fuel blends is higher than that of G100 fuel. The increase in CO_2 emission continued as a result of the increase in the amount of the fuel blend taken into the cylinder with the increasing engine power. In their study, Balki et al. [78] demonstrated that ethanol, an oxygenated fuel, increased the combustion efficiency and caused low CO_2 and HC emissions. In the study carried out by Yao et al. [79], CO_2 emission decreased with the increase in the ratio of alcohol in gasoline/ethanol blends. Çelik et al. [80] obtained similar results at a high compression ratio in a spark-ignition engine. However, the study obtained high CO_2 emission, differently from the literature.

Incomplete combustion occurs when there is not a sufficient amount of oxygen to burn fuel as a result of the low A/F ratio in the cylinder. CO and HC emissions occur as a result of the combustion of fuels with low oxygen content and rich blends. Within the exhaust emissions, CO is a type of emission harmful to human health, indicating incomplete combustion. In the study, when the results of the emission measurements given in Fig. 7 were examined, significant decreases in CO emissions were obtained compared to G100 fuel in the tests with alcohol fuels. Compared to G100 fuel, HEX5-20 and HP5-20 fuel blends produced 0.68–16.31% and 0.19–13.68% less CO emission to the atmosphere, respectively. The reason for this decrease is the O_2 content in the fuel and the effect of the stoichiometric ratios of alcohol fuels. CO emission decreases with the increase in engine power. The engine tests in this study were conducted with the highest power of 5 kW, and the CO emissions of all fuels at this power value were close to each other. It was observed that the higher the alcohol ratio in the fuel blends

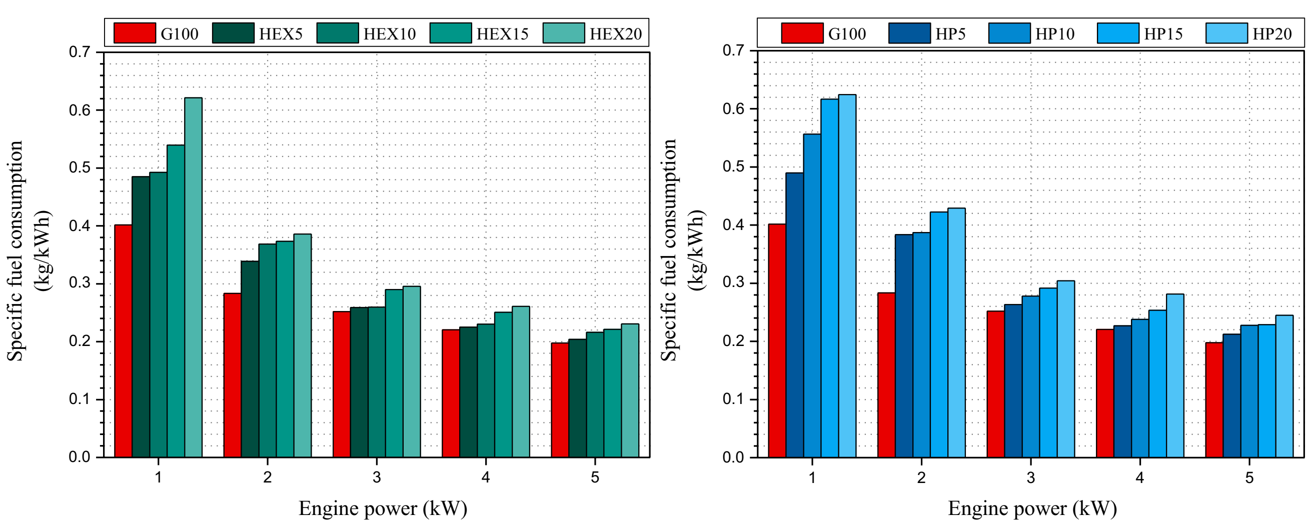

Fig. 5 Change in the fuel consumption of fuel blends at different engine powers

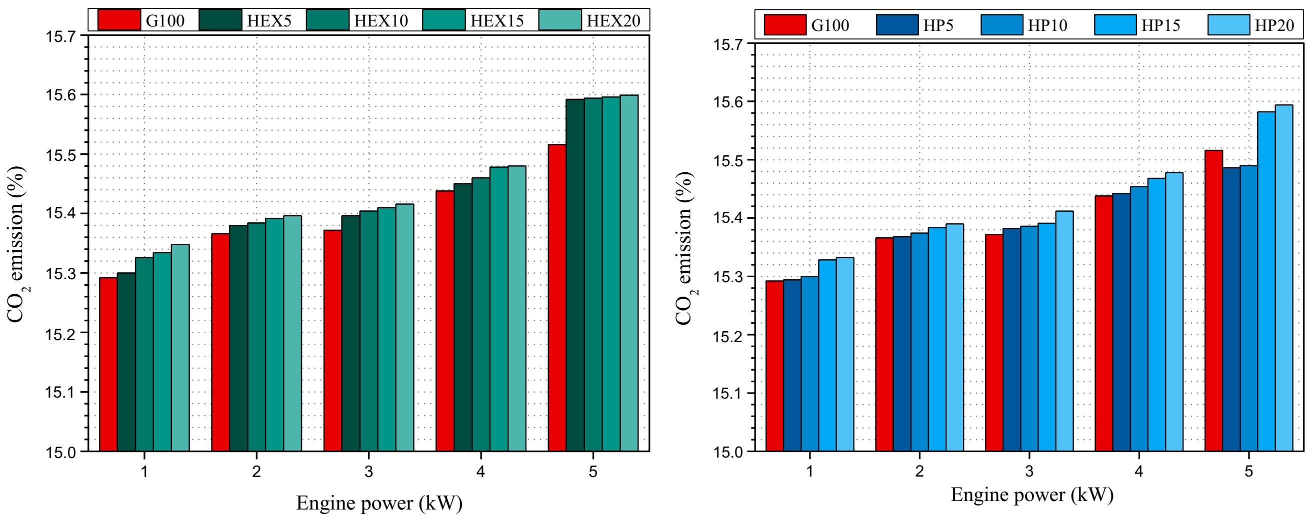

Fig. 6 CO_2 emissions of fuel blends at different engine powers

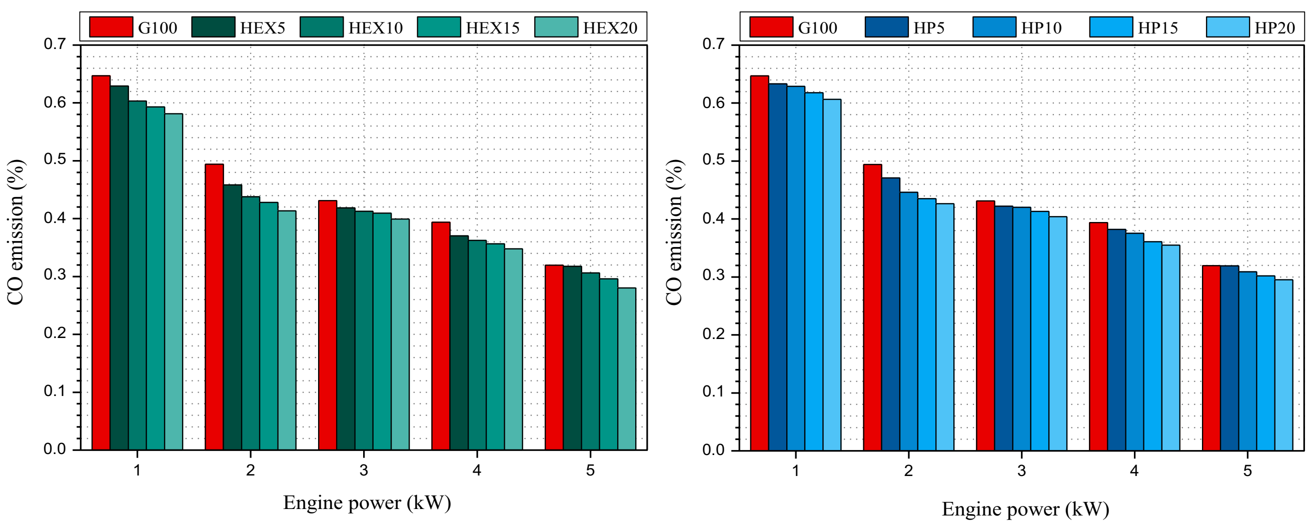

Fig. 7 CO emissions of fuel blends at different engine powers

was, the higher the decrease in CO emissions was. Since the O/C ratio of hexanol in fuel blends is higher than that of heptanol, it yielded environmental-friendly results in terms of CO emissions. Since hexanol and heptanol have higher vaporization latent heat in comparison with gasoline, the CO emission released to the atmosphere was lower in alcohol-containing fuel blends. As a result of examining the emission data in the study, decreases in CO emission and increases in CO_2 emission in all fuel blends at the same power values develop interdependently. In the tests they conducted at a constant speed of 2000 rpm, Gravalos et al. [81] determined that the CO ratio was lower in alcohol-based fuels compared to gasoline. Gravalos et al. [82] explained that they reduced CO emissions from the engine with the addition of more ethanol in fuel blends. Yacoub et al. [83] reported that CO emission decreased significantly in propanol, butanol, and pentanol blends.

HC emission occurs as a result of discharging products that do not burn completely through the exhaust in spark-ignition engines. In these engines, incomplete combustion, flame extinguishing in cold regions close to the cylinder walls, carbon deposits in the combustion chamber, and fuel retention of the oil layer in the cylinder wall reveal HC emissions. As can be seen in Fig. 8, it is observed that HC emission decreases in all three fuel types with the increase in engine power, but it is lower in alcohol fuels compared to G100 fuel. Furthermore, it is observed that as the alcohol ratio in fuel blends increases, HC emission reaches lower values. In alcohol blends, there is a decrease in HC emissions due to the low C content in the fuel and

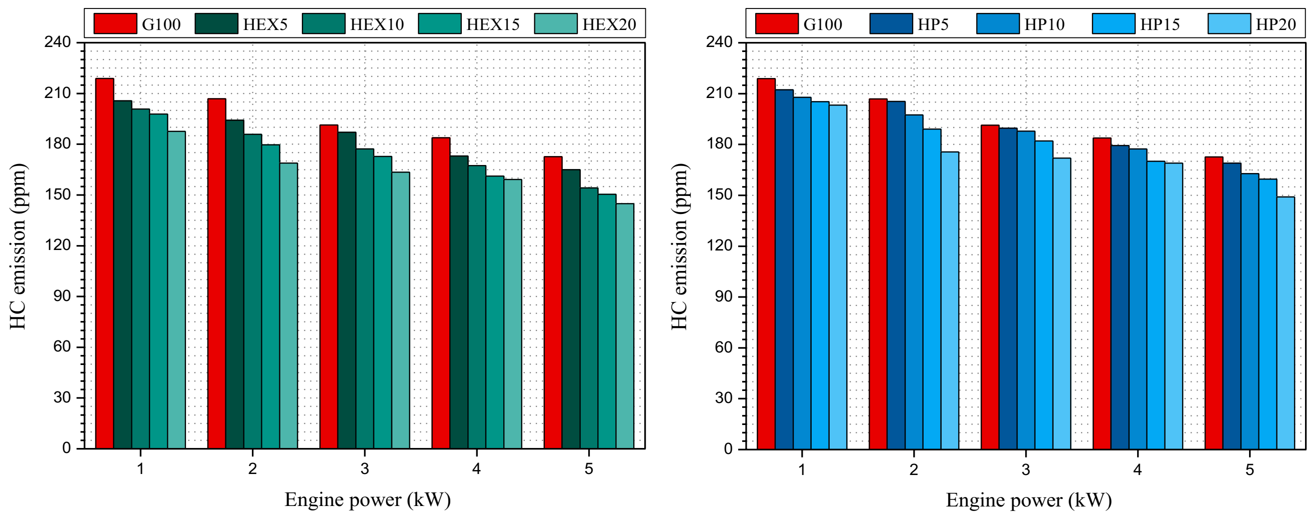

Fig. 8 HC emissions of fuel blends at different engine powers

the presence of O_2 in its structure, which improves the combustion quality. In the tests, the lowest value of HC emission was found to be 144.8 ppm in HEX20 fuel at a power of 5 kW, and the highest value was found to be 218.8 ppm in G100 fuel at a power of 1 kW. Hexanol blends produced approximately 2.3–7.6% lower HC emission at all power values compared to heptanol blends. This is because although hexanol has the same O_2 amount as heptanol, it contains fewer C atoms. Balki et al. [78] showed in their study that the use of ethanol and methanol reduced HC emission by 13.6%. In the study conducted by Chen et al. [84], while ethanol–gasoline blends gave HC emission values close to gasoline in E5 and E10 fuels, a considerable decrease occurred in E20–E40 fuels. In their study, Canakci et al. [85] demonstrated a decrease in the range of 10–17% in the HC emissions of E5, E10, M5, and M10 fuels compared to gasoline.

NO_X emission occurs as a result of the oxidation of the nitrogen atom in the air because of the high temperature and pressure formed during the combustion of the fuel in the engine. As the temperature and pressure increase or the amount of the air entering the cylinders increases, the amount of nitrogen oxides (NO_X) in the exhaust gas also increases. As seen in Fig. 9, as the engine power increases, the amount of NO_X also increases. Alcohol fuel blends produced higher NO_X emissions at all power values in comparison with G100 fuel. The reason for this is that when the actual combustion equations of the fuel blends were examined, the highest nitrogen ratio of the combustion products at all power values was observed in G100, HP5-20, and

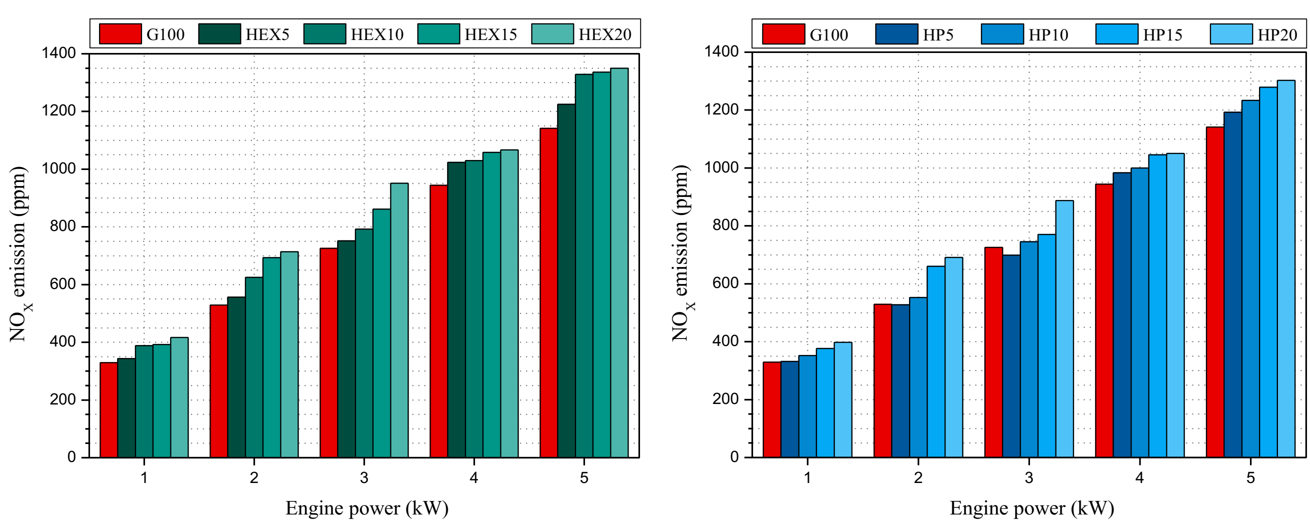

Fig. 9 NO_X emissions of fuel blends at different engine powers

HEX5-20 fuels, respectively. The highest NO_X emission was 1349.8 ppm in HEX20 fuel at a power of 5 kW, 1302.8 ppm in HP fuel, and 1141.4 ppm in G100 fuel. Wang et al. [86] determined that the NO_X emission of gasoline–alcohol blends increased at an engine speed of 2000 rpm depending on the engine speed. The latent heat of vaporization and cooling effects of alcohol-added fuel blends increased NO_X emission. Li et al. [87] demonstrated that NO_X emissions increased in gasoline and alcohol-based fuel blends with the addition of alcohol. In studies in the literature, in general [79, 83], alcohol-based fuels emit more NO_X than gasoline.

The presence of O_2 between combustion products is an important parameter that provides information about the conditions under which combustion occurs. The O_2 emission values of the alcohol fuel blends used in the study were higher compared to G100 fuel, as seen in Fig. 10. This can be explained by the excess amount of O_2 in the emission values of the combustion products as a result of the excess amount of oxygen in the chemical structure of alcohol fuels. When the alcohol fuel blends were evaluated among themselves, it was observed that although the O_2 number in the fuel content was the same, HEX5-20 fuel blends released less O_2 emissions than HP5-20 fuel blends at all power values. The reason for this is the higher air consumption of heptanol fuels compared to hexanol fuels in the engine tests. In studies in the literature [81, 84, 86], as a result of the combustion of alcohol-based fuels, higher O_2 emission occurs in products due to O_2 content in its structure.

In the study, the energy of fuels, total energy loss, and thermal efficiency were calculated in the energy analysis. The maximum energy that G100, HEX5-20, and HP5-20 fuels used in the test engine can give is determined to be the fuel energy. Engine power energy is the useful energy taken from the output shaft of the engine. Total energy losses are the difference between these two values, and they represent unusable energy. The largest part of the total energy losses that occur in the test engine is the energy discharged into the environment due to heat transfer and exhaust losses. As seen in Table 5, alcohol blends in the fuels used in the study have more fuel energy than G100 fuel. In alcohol blends, higher fuel energy was calculated in HP5-20 fuels compared to HEX5-20 fuels. The reason for this is that although the lower heating value of heptanol blends is lower than hexanol blends, the amount of fuel consumed is high. In their study, Doğan et al. [34, 48] showed that the energy flow increased depending on fuel consumption in the fuel blends of alcohol-based fuel with gasoline.

In the energy analysis, the magnitude expressing how much of the thermal energy obtained as a result of the combustion of the fuel in the engine was used for power was calculated as thermal efficiency. In cases when energy losses are minimum, thermal efficiency reaches the maximum level. In the energy analysis performed according to the data obtained from the tests conducted in the study, as shown in Fig. 11, G100 fuel produced a better thermal efficiency than alcohol blends. In all tests, the highest value for thermal efficiency was found to be 41.08% in G100 fuel at a power of 5 kW, and the lowest value was found to be 11.34% in HP20 fuel at a power of 1 kW. In alcohol fuel blends, lower thermal efficiency was obtained in heptanol blends compared to hexanol. To obtain the same power value in all fuels, the thermal efficiency performance is worse since the highest fuel consumption and the lowest lower heating value are obtained in heptanol blends. Doğan et al. [34] demonstrated that the thermal efficiency of alcohol-based fuels (E0, E10, E20, and E30) was lower than gasoline. Studies in the literature determined that thermal efficiency generally tended to increase with an increase in engine load [88, 89].

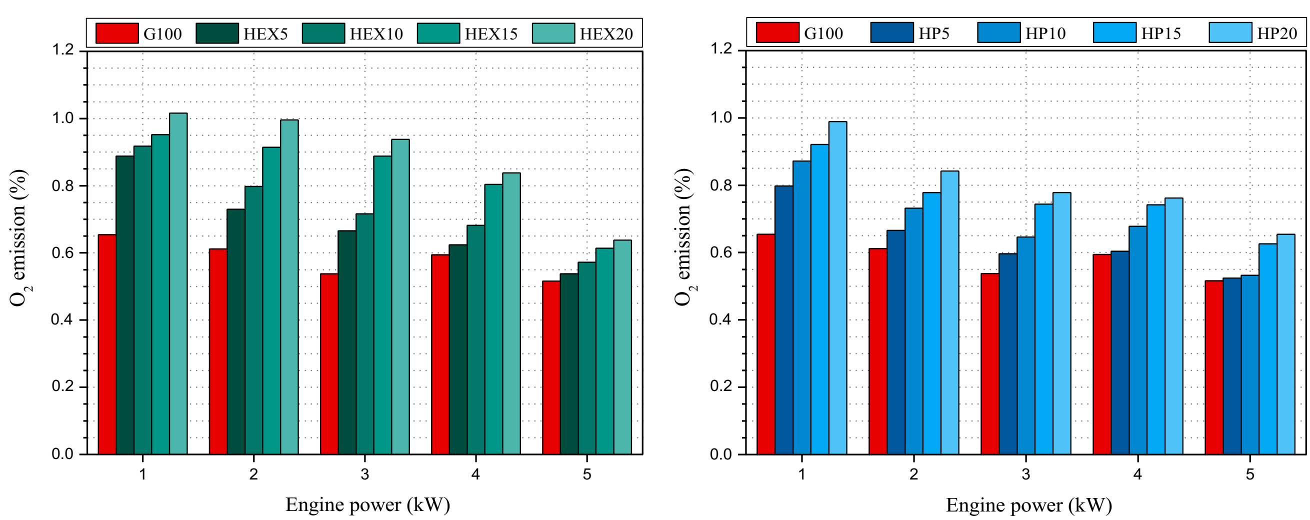

Fig. 10 O_2 emissions of fuel blends at different engine powers

Table 5 Energy flows and heat losses of fuels

Power (kW)	Energy current (kW)					Heat loss (kW)				
	G100	HEX5	HEX10	HEX15	HEX20	G100	HEX5	HEX10	HEX15	HEX20
1	6.638	6.721	7.654	7.729	8.044	5.638	5.721	6.654	6.729	7.044
2	7.744	8.310	8.868	9.299	9.484	5.744	6.310	6.868	7.299	7.484
3	9.342	9.654	9.840	10.386	11.165	6.342	6.654	6.840	7.386	8.165
4	10.448	10.998	11.298	11.956	12.366	6.448	6.998	7.298	7.956	8.366
5	12.169	12.587	13.121	13.526	14.167	7.169	7.587	8.121	8.526	9.167
	G100	HP5	HP10	HP15	HP20	G100	HP5	HP10	HP15	HP20
1	6.638	7.660	7.816	8.205	8.819	5.638	6.660	6.816	7.205	7.819
2	7.744	8.632	9.259	9.989	10.347	5.744	6.632	7.259	7.989	8.347
3	9.342	9.848	9.981	10.703	11.053	6.342	6.848	6.981	7.703	8.053
4	10.448	11.429	11.905	12.367	12.817	6.448	7.429	7.905	8.367	8.817
5	12.169	12.888	13.588	13.794	14.228	7.169	7.888	8.588	8.794	9.228

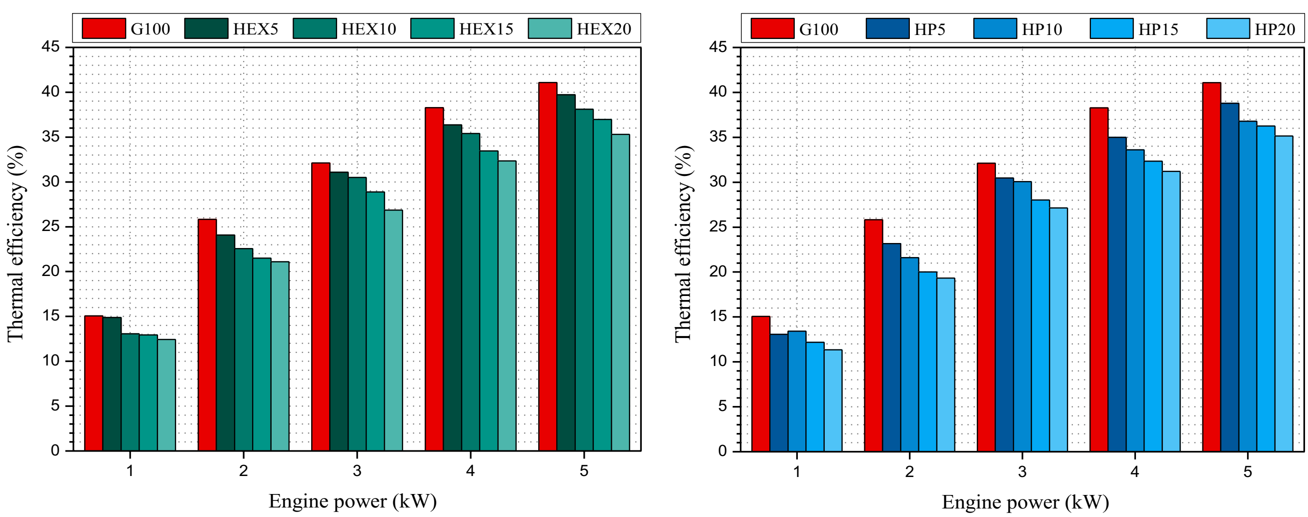

Fig. 11 Thermal efficiency of fuel blends at different engine powers

In the exergy analysis, chemical exergy, exhaust exergy, exergy destruction with exergy accompanying heat transfer, and exergetic power were calculated for each fuel using the engine test data. Moreover, entropy generation and exergy efficiency, which is the ratio of exergetic power to fuel exergy, were found. Since the chemical exergy calculated for all fuels is a function of the lower heating value and fuel consumption, similarities to the results obtained in the energy analysis were observed. The exhaust exergy of the fuels tested in this study at different engine powers is illustrated in Fig. 12. Exhaust exergy, which determines the heat energy discharged with exhaust emissions, increases in direct proportion to the exhaust gas temperature. Furthermore, the exhaust exergy increases with the increase in engine power. G100 fuel had a higher exhaust exergy value at all power values compared to alcohol fuels. The reason for this is thought to originate from the fact that the heating value and the pressure and temperature at the end of the combustion of alcohol-containing fuels are lower in comparison with G100 fuel. The exhaust exergy of heptanol fuels was found to be close to that of hexanol fuels in alcohol-containing fuels. In the engine tests, the exhaust exergy was calculated to be maximum 2.797 kW in G100 fuel at a power of 5 kW and minimum 0.811 kW in HEX10 fuel at a power of 1 kW. Kiani et al. [90] showed that the chemical exergy of the fuel was at the maximum level at the compression stroke. It was stated that the heating value of gasoline was higher than fuel blends with different bioethanol percentages.

Exergy destruction in internal combustion engines refers to many irreversible processes, in other words, the lost work potential. As a result of the analysis conducted in the study, exergy destruction increased in all fuels with the increase

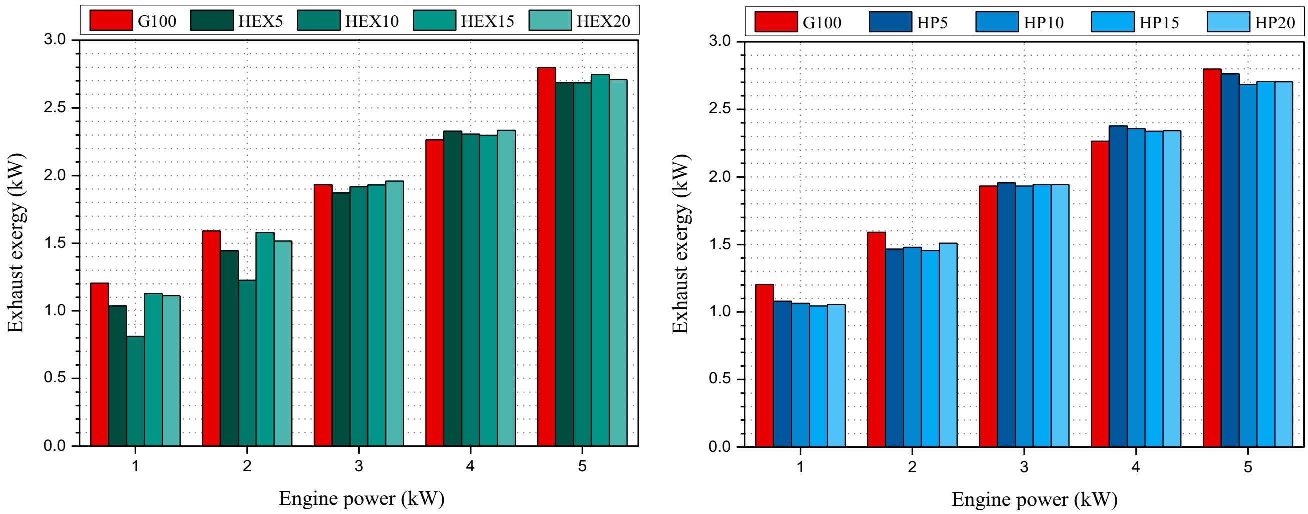

Fig. 12 Exhaust exergy of fuel blends at different engine powers

in engine power. The exergy destruction of G100 fuel was lower in comparison with alcohol fuels at all powers. In alcohol fuel blends, higher exergy destruction was calculated in heptanol blends compared to hexanol blends. It is thought that this increase in exergy destruction observed in alcohol fuels is caused by the amount of the fuel entering the engine and the irregularities occurring during combustion. Upon examining studies in the literature, one of the important exhaust emission problems when alcohol blend fuels are used in internal combustion engines is increased aldehyde emissions. In the present study, since aldehyde emission could not be measured during the tests, its effect on exergy destruction was not addressed. However, it is thought that a large part of the increase in exergy destruction in alcohol-based fuels originates from aldehyde emissions. As can be seen from Fig. 13, the highest exergy destruction was determined to be 6.253 kW in HP20 fuel at a power of 5 kW, and the lowest exergy destruction was found to be 3.939 kW in G100 fuel at a power of 1 kW. In their study, Doğan et al. [28] analyzed that more exergy destruction occurred in alcohol-based fuels than gasoline.

Using the data acquired from the tests conducted in the test engine, the exergy accompanying the heat transfer was calculated. This exergy refers to the heat transfer from the engine body, cooling water, and exergy transitions. In Fig. 14, the heat transfer exergy of all fuels is observed. There is a decrease in heat transfer exergy with the increase in engine power. This was similar to changes in the lost exergy due to engine load in studies in the literature [7, 34, 46].

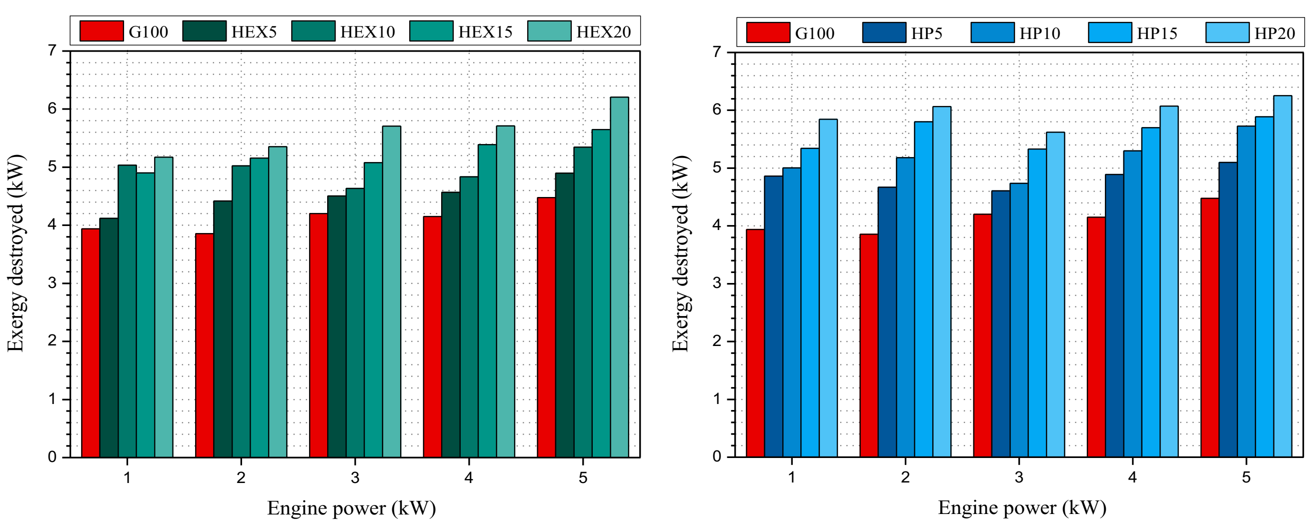

Fig. 13 Exergy destroyed of fuel blends at different engine powers

Irreversibilities occurring in the engine cause entropy generation while passing through a thermodynamic, mechanical, or chemical process. In the study, as can be seen in Fig. 15, entropy generation increases with the increase in power in all fuels. When the entropy generation in all fuels was examined, the highest entropy generation was determined to be 0.02134 kW/K in HP20 fuel at a power of 5 kW, and the lowest entropy generation was found to be 0.0134 kW/K in G100 fuel at a power of 1 kW. Entropy generation originates from exhaust losses, heat discharged into the environment, friction, and other losses. Since alcohol-containing fuel blends had higher exergy destruction, more entropy generation occurred compared to G100 fuel. The quality of energy decreases during the processes that occur in the engine in studies on alcohol-based fuels. As calculated in exergy analysis, while the beneficial part of energy decreases, entropy increases [7, 34, 46, 49].

Exergy efficiency is expressed as the ratio of the actual thermal efficiency to the highest possible thermal efficiency under the same conditions in spark-ignition engines. As a result of the analyses in the study, the thermal efficiencies given in Fig. 11 were lower than the exergy efficiencies given in Fig. 16 at the same engine powers in all fuels. A higher exergy efficiency was calculated for G100 fuel in comparison with alcohol fuels. In alcohol fuels, hexanol blends have higher exergy efficiency than heptanol blends. It was observed that when the exergy efficiency was low, exergy destruction and related entropy generation were higher. The highest exergy efficiency was determined to be 38.31% in G100 fuel at a power of 5 kW. In the study of Kul et al. [88],

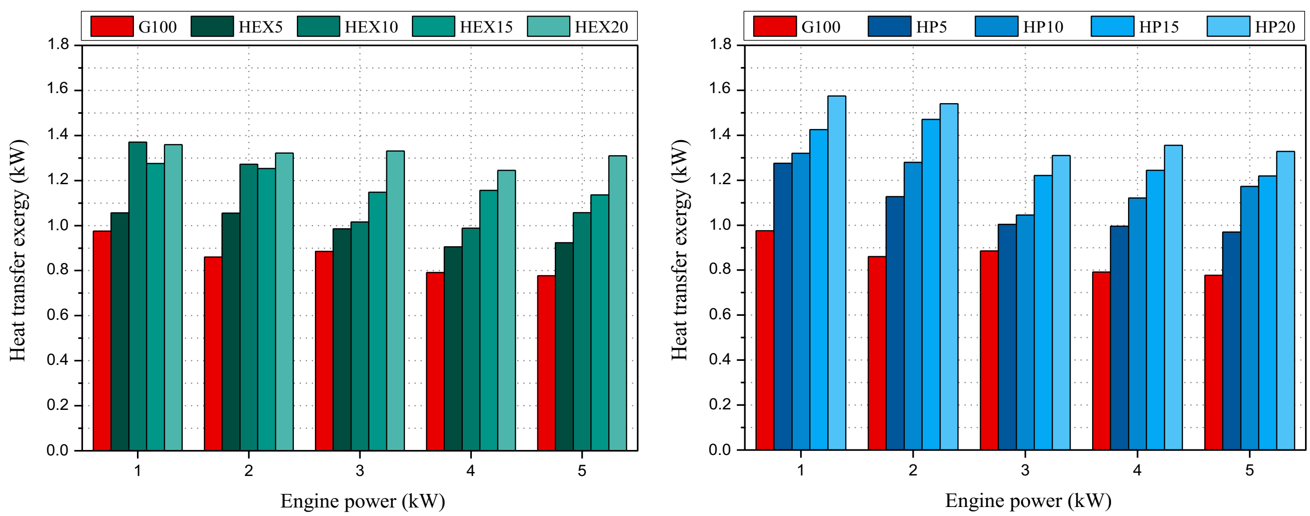

Fig. 14 Heat transfer exergy of fuel blends at different engine powers

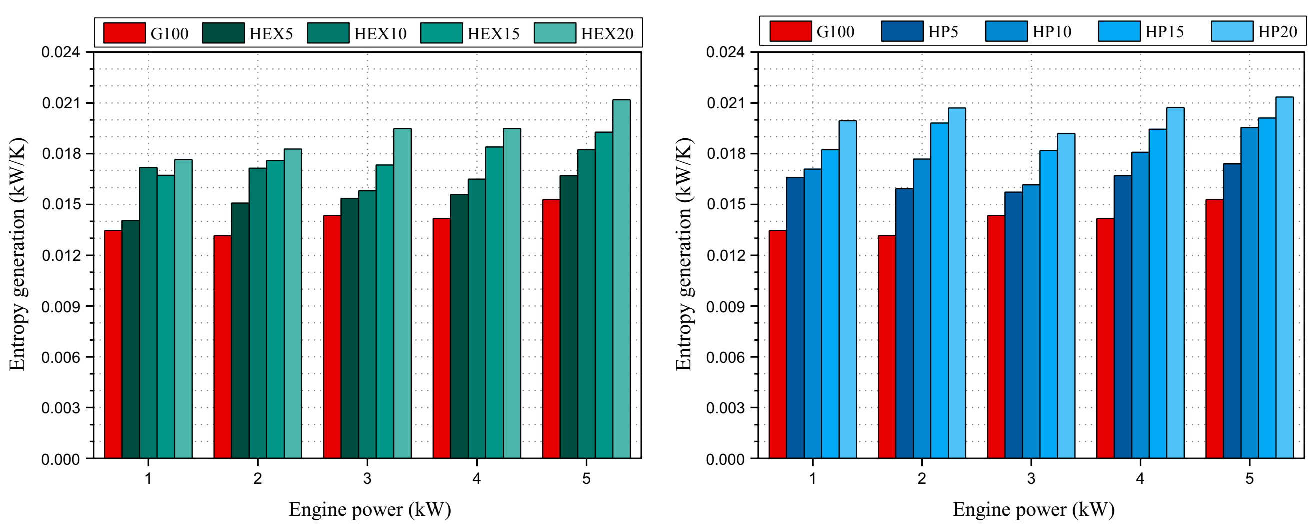

Fig. 15 Entropy generation of fuel blends at different engine powers

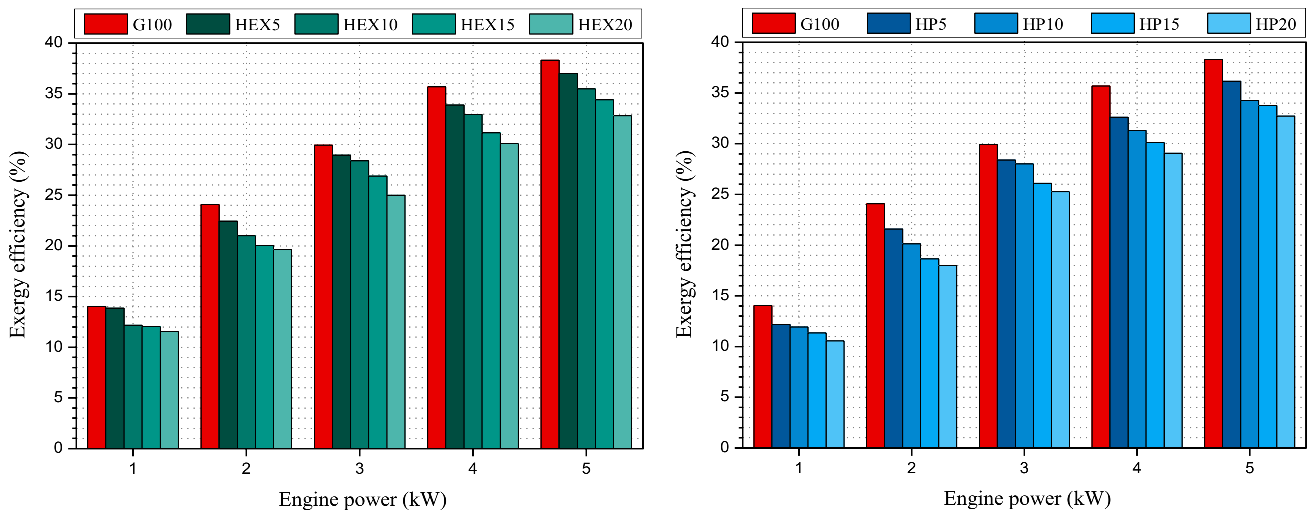

Fig. 16 Exergy efficiency of fuel blends at different engine powers

exergetic efficiency increased in case alcohol-based fuels were added to gasoline. It ensured that the maximum value was achieved for E100 fuel. Tippawan et al. [91] showed the dependence of exergy efficiency on the chemical exergy of fuels and engine output power. In the study performed by Nazzal et al. [92], the exergy and thermal efficiency of alcohol-based fuels acted in a parallel way.

It is necessary to reduce carbon dioxide emissions in internal combustion engines to prevent climate change and global warming, which are among the most important environmental problems nowadays. Determining the mass and financial damage of these engines to the environment and collecting the damage from the people causing damage will increase environmental awareness. In this study, the CO_2 emissions released from the engine to the atmosphere were determined experimentally, and the cost of the damage to the environment was computed. The results obtained from the environmental and enviroeconomic analyses conducted using the CO_2 emissions released into the environment from the test engine are presented in Table 6. G100 fuel is regarded as a more environmental-friendly fuel as a result of the analysis based on CO_2 emissions in comparison with alcohol blends. Although heptanol fuels in alcohol-containing blends release less CO_2 emission than hexanol fuels in engine tests, they cause more damage to the environment due to the higher energy flow. The highest environmental and enviroeconomic results were found to be 4224.53 kg CO_2/month and 61.256 \$/month, respectively, in HP20 fuel.

In the tests, the exergoenvironmental and exergoenviroeconomic analyses were conducted using the exergy

Table 6 Results obtained from the environmental and enviroeconomic analysis of fuels at different engine powers

Power (kW)	Environmental analysis (kg CO_2/month)					Enviroeconomic analysis (\$/month)				
	G100	HP5	HP10	HP15	HP20	G100	HP5	HP10	HP15	HP20
1	1932.674	2230.619	2277.087	2394.788	2574.516	28.024	32.344	33.018	34.724	37.330
2	2265.697	2526.036	2710.519	2926.045	3032.193	32.853	36.628	39.303	42.428	43.967
3	2734.290	2884.441	2924.009	3136.475	3243.563	39.647	41.824	42.398	45.479	47.032
4	3071.217	3360.433	3503.087	3642.504	3777.260	44.533	48.726	50.795	52.816	54.770
5	3595.137	3800.222	4007.788	4092.735	4224.530	52.129	55.103	58.113	59.345	61,256
	G100	HEX5	HEX10	HEX15	HEX20	G100	HEX5	HEX10	HEX15	HEX20
1	1932.674	1958.033	2233.499	2256.770	2350.708	28.024	28.391	32.386	32.723	34.085
2	2265.697	2433.499	2597.817	2725.446	2780.398	32.853	35.286	37.668	39.519	40.316
3	2734.290	2830.094	2886.256	3047.565	3277.379	39.647	41.036	41.851	44.190	47.522
4	3071.217	3235.466	3325.897	3523.724	3644.854	44.533	46.914	48.226	51.094	52.850
5	3595.137	3736.844	3895.809	4016.826	4207.758	52.129	54.184	56.489	58.244	61.012

analysis data of CO_2 emissions released from the engine to the atmosphere, and the results are presented in Table 7. The highest exergoenvironmental and exergoenviroeconomic results were found to be 4538.196 kg CO_2/month and 65.804 \$/month, respectively, in HP20 fuel. If the values obtained in these analyses are higher than those presented in Table 6, the exergy flows are larger than the energy flows.

The total environmental pollution costs of the fuels used in the study are presented in Table 8. The highest total environmental pollution (TEP) obtained according to energy analysis data as a result of using the predicted values taken as the operating life of the engine was found to be 14,701.36 \$ in HP20 fuel at a power of 5 kW, and the lowest total environmental pollution (TEP) was found to be 6725.7 \$ in G100 fuel at a power of 1 kW. The same situation is observed in the total environmental pollution calculated using the exergy analysis data. When the results are assessed in this context, the total environmental pollution cost is directly proportional to the engine power in all fuels. In the total environmental pollution costs, G100 fuel has lower values compared to alcohol blends. The total environmental pollution cost of heptanol fuels in alcohol blends is higher than that of hexanol blends. Due to the high oxygen content of alcohol blends, the low C/H ratio, the oxidation of more oxygen as a result of the higher air/fuel ratio at all powers, and the more efficient combustion, a significant increase occurred in environmental costs occurred due to CO_2 emission compared to G100 fuel.

In the study conducted by Yildiz et al. [61], the change of total environmental pollution cost increased with the

Table 7 Results obtained from the exergoenvironmental and exergoenviroeconomic analysis of fuels at different engine powers

Power (kW)	Exergoenvironmental analysis (kg CO_2/month)					Exergoenviroeconomic analysis (\$/month)				
	G100	HP5	HP10	HP15	HP20	G100	HP5	HP10	HP15	HP20
1	2072.793	2393.314	2444.167	2571.551	2765.670	30.055	34.703	35.440	37.287	40.102
2	2429.960	2710.278	2909.401	3142.021	3257.329	35.234	39.299	42.186	45.559	47.231
3	2932.525	3094.824	3138.556	3367.983	3484.393	42.522	44.875	45.509	48.836	50.524
4	3293.880	3605.533	3760.124	3911.362	4057.716	47.761	52.280	54.522	56.715	58.837
5	3855.785	4077.399	4301.856	4394.826	4538.196	55.909	59.122	62.377	63.725	65.804
	G100	HEX5	HEX10	HEX15	HEX20	G100	HEX5	HEX10	HEX15	HEX20
1	2072.793	2101.048	2397.840	2424.042	2526.211	30.055	30.465	34.769	35.149	36.630
2	2429.960	2611.242	2788.964	2927.456	2987.983	35.234	37.863	40.440	42.448	43.326
3	2932.525	3036.804	3098.627	3273.450	3522.068	42.522	44.034	44.930	47.465	51.070
4	3293.880	3471.785	3570.617	3784.902	3916.979	47.761	50.341	51.774	54.881	56.796
5	3855.785	4009.783	4182.463	4314.554	4521.909	55.909	58.142	60.646	62.561	65.568

Table 8 Results obtained from TEP (energy and exergy) of fuels at different engine powers

Power (kW)	Total environmental pollution from energy analysis (\$)								
	G100	HP5	HP10	HP15	HP20	HEX5	HEX10	HEX15	HEX20
1	6725.705	7762.555	7924.263	8333.862	8959.316	6813.955	7772.576	7853.559	8180.462
2	7884.627	8790.605	9432.606	10,182.637	10,552.031	8468.576	9040.402	9484.552	9675.786
3	9515.327	10,037.854	10,175.550	10,914.933	11,287.600	9848.728	10,044.172	10,605.524	11,405.279
4	10,687.835	11,694.306	12,190.743	12,675.912	13,144.864	11,259.423	11,574.122	12,262.559	12,684.094
5	12,511.078	13,224.771	13,947.101	14,242.719	14,701.365	13,004.216	13,557.416	13,978.556	14,642.999
Power (kW)	Total environmental pollution from exergy analysis (\$)								
	G100	HP5	HP10	HP15	HP20	HEX5	HEX10	HEX15	HEX20
1	7213.319	8328.733	8505.700	8948.996	9624.532	7308.779	8330.326	8432.364	8782.051
2	8456.262	9431.766	10,124.716	10,934.232	11,335.504	9080.031	9699.286	10,182.252	10,394.128
3	10,205.189	10,769.986	10,922.174	11,720.580	12,125.688	10,558.469	10,770.622	11,377.560	12,253.617
4	11,462.703	12,547.255	13,085.230	13,611.540	14,120.852	12,075.555	12,420.924	13,162.950	13,629.327
5	13,418.131	14,189.348	14,970.459	15,293.995	15,792.921	13,859.180	14,457.900	15,001.168	15,731.201

increase in engine load. Çaliskan and Mori [72] proposed a filter system to reduce the CO_2 prices released in one day as a result of environmental analysis in the diesel engine. In their study performed with gasoline–ethanol blends, Doğan et al. [44] calculated 25%–30% higher CO_2 emission cost in E30 fuel than E0 fuel at the same speed.

5 Conclusions

In the current research, performance and emission data were obtained using HP5-20 and HEX5-20 fuel blends acquired from pure gasoline, hexanol, and heptanol fuels in a single-cylinder, four-stroke, water-cooled, variable compression spark-ignition engine. Using these data, fuel energy, heat losses, and thermal efficiency values were calculated in the energy analysis.

- The use of gasoline–alcohol blend fuels in the engine increased fuel consumption. The lowest fuel consumption was obtained to be 0.197 kg/kWh in G100 fuel at a power of 5 kW. Heptanol blends consumed more fuel than hexanol blends.
- In CO and HC emissions, gasoline–alcohol blends produced lower emissions than G100 fuel. In these emissions, hexanol blends yielded more environmentally friendly results than heptanol blends. The lowest CO and HC emissions were 0.28% and 144.8 ppm, respectively, in HEX20 fuel at a power of 5 kW.
- Gasoline–alcohol blends had higher CO_2 emissions released from the engine exhaust to the environment than G100 fuel. In the tests, the highest CO_2 emission was obtained with the values of HP20: 15.594%, HEX20: 15.559%, and G100: 15.516% at a power of 5 kW.
- In NO_X emissions, HEX5-20 fuel blends reached the highest values of 1349.8 ppm at a power of 5 kW power. G100 fuel produced lower NO_X emission at all powers than alcohol blends.
- According to the energy analysis results, since the lower heating value of gasoline–alcohol blends is lower, their energy flows are lower than G100 fuel. According to the energy flow calculations, it was found to be 12.88–14.22 kW in HP5-20 fuel blends at an engine power of 5 kW, 12.57–14.16 kW in HEX5-20 blends, and 12.17 kW in G100 fuel. The highest value in heat transfer losses was 9.23 kW in HP20 fuel at an engine power of 5 kW, HEX20:9.17 kW, and G100: 7.17 kW.
- The highest thermal efficiency of the fuels used in the tests was found in the range of G100: 41.08%, HEX5-20: 35.29–39.72%, and HP5-20: 35.14–38.79% at a power of 5 kW.
- G100 fuel performed the best in exergy destruction calculations due to irreversibilities. The highest exergy destruction was HP20: 6.25 kW, HEX20: 6.20 kW, and G100: 4.47 kW at a power of 5 kW. Entropy generation is based on exergy destruction calculations. Therefore, entropy generation values give results parallel to exergy destruction values. The highest entropy generation was revealed to be 0.02134 kW/K in HP20 fuel at a power of 5 kW.
- Exergy efficiency was determined to be lower than thermal efficiency in all fuels. The highest exergy efficiencies were found in the range of G100: 38.31%, HEX5-20: 32.84–37.02%, and HP5-20: 32.71–33.75% at an engine power of 5 kW.
- In all analyses, alcohol-containing blends cause more mass and economic damage to the environment than G100 fuel.
- In the environmental analysis, it was found to be HP20: 4224.53 kg CO_2/month, HEX20: 4207.758 kg CO_2/month, and G100: 3595.137 kg CO_2/month at a power of 5 kW. The economic cost of the mass emission values in this analysis was determined by the enviroeconomic analysis, and the values of HP20: 61.256 $/month, HEX20: 461.012 $/month, and G100: 52.129 $/month were obtained at a power of 5 kW power.
- In the exergoenvironmental analysis, the highest environmental pollution was calculated to be HP20: 4538.196 kg CO_2/month and economically HP20: 64.804 $/month in the exergoenviroeconomic analysis at a power of 5 kW.
- HP5-20 fuel blends reached the highest value at all engine powers in the total environmental pollution throughout the engine's life. In all tests, the highest TEP with energy source was found to be HP20: 14,701.356 $ and the TEP with exergy source was found to be HP20: 15,792.92 $ at a power of 5 kW.

As a result of conducting the energy, exergy, and environmental analyses of gasoline–hexanol–heptanol fuel blends according to the data obtained in a spark-ignition engine, the use of alcohols as alternative fuels was found to be appropriate under certain conditions. If alcohols are utilized as an alternative fuel in spark-ignition engines in the upcoming periods, their economic evaluation can be performed.

Authors' contributions Hayri Yaman has contributed in research objectives, data analysis plan, validation of results, revision/proofreading, and final approval. Battal Doğan to the definition of research objectives, hypotheses, data analysis plan, validation of results, article writing, revision/proofreading, and final approval. Murat Kadir Yeşilyurt has contributed to the definition of research objectives, hypotheses, results interpretation, validation of results, article writing, revision /proofreading, and final approval. Derviş Erol has contributed in research objectives, data analysis plan, designing of the graphs, validation of results, article writing, revision/proofreading, and final approval.

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Data availability The data used and/or analyzed throughout the present study are available from the authors on reasonable request.

Declarations

Conflict of interest The authors pointed out that there is no potential conflict of interest. The authors declared that there is no competing financial interest in this research.

Ethical approval The authors declared that no animal and human studies are presented in this manuscript and no potentially identifiable human images or data are given in this research.

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