Journal of Chemical and Petroleum Engineering (JChPE)
Print ISSN: 2423-673X Online ISSN: 2423-6721

Thermodynamic Performance and Exergy Analysis of a Four-Cylinder Gasoline Engine Fueled with Gasoline-Alcohol Blends: The Role of Pentanol, Butanol, Ethanol, and Propanol

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ARTICLE INFO

Article History:
Received: 07 March 2025
Revised: 03 July 2025
Accepted: 07 July 2025
Published: 07 July 2025

Article type: Research

Keywords:
Bio-alcohols,
Engine,
Exhaust,
Heat,
Irreversibility,
Work

ABSTRACT

This study examines the combustion behavior and exergy performance of a water-cooled, four-cylinder gasoline engine operating on pure gasoline and various gasoline–bio-alcohol blends, including those containing ethanol, butanol, propanol, and pentanol. Experiments were conducted at engine speeds of 1,000, 1,500, and 2,000 rpm. Exergy analysis was employed to assess the input fuel exergy, exergy losses, and exergy efficiency. Increased engine speed was correlated with higher input exergy rates across all fuels, resulting from increased fuel consumption. Among the fuel blends, higher pentanol content led to lower input exergy but enhanced exergy transfer through heat due to improved combustion characteristics. Exhaust exergy rates were higher for alcohol-containing fuels, especially at elevated speeds, due to higher combustion temperatures. The exergy work rate increased with engine speed in all blends, with the G60Pe10E10Bu10Pr10 blend achieving the highest work output (46.48 kW at 2000 rpm). However, exergy destruction also rose with speed and alcohol concentration, particularly in blends with 20% pentanol. Overall, blends with moderate alcohol content (particularly pentanol) showed favorable exergy behavior compared to pure gasoline. This study confirms the potential of optimized alcohol–gasoline blends to enhance engine thermodynamic performance while reducing dependence on fossil fuels. Among the alcohols, pentanol blends demonstrated the most favorable exergy performance at higher engine speeds.

Introduction

Since internal combustion engines are increasingly common, the need for alternative energy sources has grown in response to the growing environmental issues and depletion of fossil fuel reserves. Alternative energy sources for internal combustion engines have recently included alcohol-based fuels, such as pentanol, propanol, butanol, and ethanol [1, 2]. For engines with high compression ratios and octane numbers, alcohol-based fuels are a top choice. Moreover, this might decrease the transportation sector's need for petroleum-based energy [3-5]. Energy analysis can determine the work capacity lost in the energy of internal combustion engines. Because of this, analyzing exergy analysis is crucial for determining the optimal operating

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Journal of Chemical and Petroleum Engineering, 2025, 59(2): 387-403.
Publisher: University of Tehran Press *DOI: 10.22059/jchpe.2025.391312.1601*

parameters and enhancing the efficiency of internal combustion engines. In internal combustion engines, energy analysis plays a crucial role in helping researchers and designers make informed decisions. Adding alcohol to gasoline helps provide complete combustion conditions, increases combustion efficiency, and reduces environmental pollution due to the extra oxygen in its structure [6]. According to exergy analysis, an energy-efficient system could not be exergy-efficient, which assesses a system's proximity to an ideal state. Exergy is the maximum practical work that the system achieves during a reversible process from a thermodynamic state to a real dead state (equilibrium of the system with the environment) [6]. Increasing efficiency and lower fuel consumption is another way to deal with countries' emission laws. Consequently, a deeper examination of a gasoline engine's properties is warranted rather than just measuring its performance, emissions, and combustion characteristics. Thermodynamic models can analyze a gasoline engine's performance and identify the various losses that occur during engine operation [7, 8]. While the first law of thermodynamics is typically used to assess internal combustion engines, fuel stability and renewability are often overlooked. The second law of thermodynamics, which deals with the quality of energy utilized, must be assessed to achieve this [9]. The essential idea behind the second law of thermodynamics is the concept of energy. The most theoretically practical work that a system may produce is known as its energy. Exergy degradation, also known as irreversibility, is the primary cause of the decline in diesel engine efficiency [10]. Examples of study work related to the exergy analysis of gasoline engines with bio-alcohols are mentioned below. In a study conducted by Khoshkname et al. in 2023 [1], on a water-cooled 4-cylinder gasoline engine, it was demonstrated that increasing the percentage of alcohol in the fuel at various engine speeds enhanced the engine's performance while employing fuel blends of ethanol, butanol, propanol, and pentanol with gasoline in varying volume percentages. Feng et al (2021) [11] conducted an exergy analysis on a gasoline, ethanol, and methanol-fueled turbocharged SI engine in another study. The analysis revealed that increasing the amount of alcohol increased the total irreversibility in exergy and improved exergy and thermal efficiencies to a limited extent. Özkan and Shakmak (2018) [12] explored how blended gasoline with a 10% volume percentage of oxygenated fuel additives (solketal, methanol, and ethanol) affected the exergy parameters measured by the crank angle in an SI engine. In line with the study's findings, adding alcohol to gasoline increases the cylinder's peak pressure, with methanol compounds exhibiting the most significant increase in pressure. According to reports, the energy efficiency of gasoline decreases when ethanol, methanol, and solketal are added. Sayin et al. [13] (2007) examined the effects of different research octane numbers on the exergy and performance of gasoline engines. It has been noted that utilizing higher octane fuel causes the engine to run less efficiently in terms of energy, and vice versa. Higher octane levels also increase exhaust exergy losses, which result from heat dissipation. The study revealed a direct link between engine speed and fuel mixture uniformity in the combustion chamber, resulting in enhanced exergy parameters.

Recent studies have explored biodiesel–diesel blends to improve engine sustainability. One investigation evaluated engine performance and emissions using blends up to 20%, applying multi-objective optimization to identify optimal conditions. Results showed that engine load and blend ratio have a significant impact on performance and emission outputs. Optimal outcomes were achieved at a 9% biodiesel blend and a 70% load, enhancing efficiency while reducing pollutants [31].

Based on experimental testing findings, Bhatti et al (2019) [15] performed an exergy study of a single-cylinder gasoline engine operating at various engine speeds and compression ratios. They claimed that a more accurate picture of spark ignition engine energy consumption can be obtained through energy analysis. They opened the door for more advancements by identifying and minimizing exergy losses at different sites through exergy analysis. They also recommended using alternative gaseous fuels, lean and low-temperature combustion, and exhaust heat recovery systems to reduce energy losses and enhance engine efficiency. In another study, Fu et al. (2013) [5] demonstrated the application of exergy analysis and stability through the performance and emission outcomes of ethanol and methanol blended with gasoline in SI engines. Exergy and stability analysis results showed that pure gasoline had the highest exergy efficiency, followed by ethanol and methanol.

A study evaluates a new biodiesel blend in a diesel engine at different concentrations (5 to 20 percent). Multi-objective optimization identified optimal conditions at 70% load and 9% biodiesel, which improved performance (BTE: 20%, BSFC: 307 g/kW·h) and reduced emissions (HC: 12 ppm, CO_2: 9%, NO_x: 209 ppm). Results confirm significant effects of load and blend ratio on engine efficiency and emissions, with a desirability of 73% [30].

Furthermore, ethanol fuel was said to have a greater potential than methanol to replace pure gasoline. The impact of various fuel blends, including petrol, natural gas, and methanol, on exergy analysis in a spark-ignition engine operating under varying engine load conditions and constant engine speed was examined by Akbiyik et al. (2023) [6].

The performance, combustion, and emissions of a direct-injection compression ignition engine using B20 biodiesel with MWCNTs were examined in a research study. The addition of 100 ppm MWCNT enhanced engine performance and thermal efficiency while significantly reducing hydrocarbon (HC), carbon monoxide (CO), and particulate matter emissions. However, the improved combustion process resulted in elevated in-cylinder temperatures, leading to an increase in nitrogen oxides (NO_x) emissions [32].

According to the study's findings, the gasoline-natural gas blend produced the lowest pollution level and the best exergy analysis performance out of all the fuel blends tested. Researchers conducted a CFD-based exergy analysis of an SI engine using various fuels, including ethanol, methanol, and gasoline. The study results indicate that increasing the speed, exhaust losses, and heat transfer increases, which leads to more energy losses. Additionally, data suggest higher energy losses with ethanol fuel compared to gasoline and methanol [15].

The continuous use of fossil fuels has depleted their resources worldwide; therefore, many researchers worldwide are seeking solutions to mitigate their use. On the other hand, the low efficiency of biofuels compared to fossil fuels has led many researchers to conduct extensive research to address these issues and improve the conditions for using biofuels with high efficiency. Therefore, no research has been done on engine exergy using ethanol, butanol, propanol, and pentanol compounds in different volume percentages. In this study, the exergy analysis of a four-cylinder gasoline engine coupled to a dynamometer using pure gasoline and ethanol-butanol-propanol and pentanol fuel combinations ($G_{60}Pe_{10}E_{10}Bu_{10}Pr_{10}$, $G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$, $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$, and G_{100}) at speeds of 1000, 1500, and 2000 rpm was investigated. This study aims to conduct a comparative exergy analysis of gasoline-alcohol blends at varying engine speeds.

Materials and Methods

Preparation of Fuel Blends

This test used gasoline from a gas station in Hamedan, Iran. The alcohols used, including ethanol, butanol, propanol, and pentanol, were obtained from the Merck brand in Germany, and their purity was 99.6%. The fuel compounds were combined according to the specified volume percentages after preparing gasoline fuel and ethanol, butanol, propanol, and pentanol alcohols. In Table 1, the composition ratios of the prepared fuels are presented. The letter E represents ethanol, B stands for butanol, Pr stands for propanol, Pe stands for pentanol, and G stands for gasoline. Subscript numbers indicate the volume percentage of each fuel. For example, the fuel blend $G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$ means 55% by volume of gas, 15% by volume of pentanol, 10% by volume of ethanol, and 10% by volume of butanol, as well as 10% by volume of propanol.

Table 1. Volume percentage of alcohols added to gasoline

Fuel Blends	Fuel Name	Gasoline (%)	Pentanol (%)	Ethanol (%)	Butanol (%)	Propanol (%)
G100	Base	100	0	0	0	0
$G_{60}Pe_{10}E_{10}Bu_{10}Pr_{10}$	1	60	10	10	10	10
$G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$	2	55	15	10	10	10
$G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$	3	50	20	10	10	10

Engine Specifications

This research utilized a four-cylinder gasoline engine with direct injection, water cooling, and a dynamometer, available at Bu-Ali Sina University, to conduct experiments (Fig. 1). The test was conducted at a temperature of 25°C and an air pressure of 101,900 Pa.

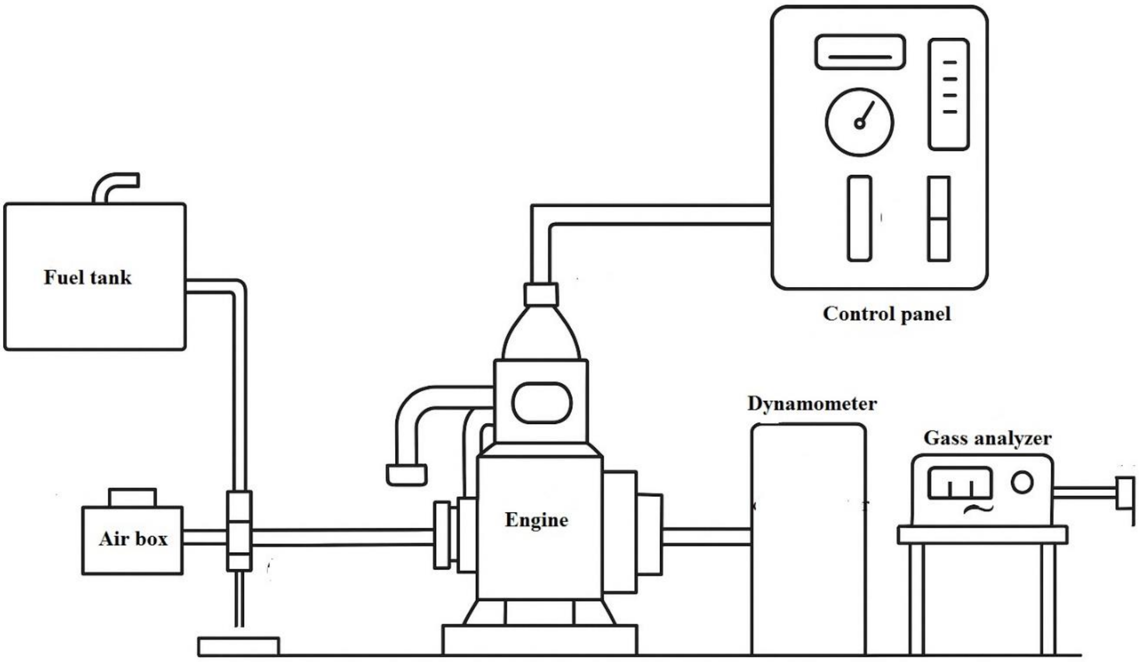

Fig. 1. Engine setup

Table 2 shows the specifications of the gasoline engine.

Table 2. Engine specifications

Model	FORD CVH414
Number of cylinders	4
Maximum power (kW)	65.5
Piston diameter (mm)	77.24
Piston stroke (mm)	74.3
Swept volume	1392
Compression ratio	9,3:1
Speed	6000 rev/min
Indicator tappings	kistler
Diameter of exhaust pipe	1/1/2" BPS
Length of exhaust pipe	1M
Oli pressure gauge	0 to 6.8 bar
Oil temperature gauge	Zeal 20 to 150

Exergy Analysis

For exergy analysis, the engine was treated as a steady-state open system, based on certain assumptions:

- The gas mixes in the intake air and exhaust gases were ideal.
- The fuel stream, exhaust gases, and intake air's kinetic and potential energy effects were disregarded [17-18].
- The temperature and reference atmosphere were taken into account. as P_0 = 1 atm and T_0 = 25 °C, respectively [18].

A diesel engine's exergy balance is expressed as [19-20]:

$$\dot{E}x_{heat} + \dot{E}x_w = \sum \dot{m}_{in}\varepsilon_{in} - \sum \dot{m}_{out}\varepsilon_{out} - \dot{E}x_{dest} \quad (1)$$

In such cases, $\dot{E}x_{heat}$ the exergy transfer rate correlates with the control volume's heat loss to the surroundings—assumedly through cooling air; $\dot{E}x_w$ denotes the shaft work exergy rate, while $\dot{E}x_{dest}$ denotes the exergy destruction rate (irreversibility). When conducting an exergy analysis, the heat-loss exergy (the amount of exergy lost as a result of heat losses from the control volume to the environment) was defined as follows:

Total heat loss from cooling water (T_C) to the environment (T_0) defined the lost-exergy rate, afterwards:

$$\dot{E}x_{heat} = \sum(1 - \frac{T_0}{T_C})\dot{Q}_{loss} \quad (2)$$

while $\dot{Q}_{loss}$ is the output heat rate that the engine's cooling water receives from the engine environment, this can be explained by

$$\dot{Q} = \dot{m}_w \times C_w \times \nabla T_w \quad (3)$$

where $\dot{m}_w$ represents mass flow rate, C_w is the cooling water's specific heat, and ∇T_w denotes the inlet-outlet cooling water temperature difference.

Net energy and net exergy work rates are identical.

$$\dot{E}x_w = \dot{W} \quad (4)$$

The fuel and input exergy rate, considering only chemical exergy, can be defined as:

$$\dot{E}x_{in} = \dot{m}_{fuel}\,\varepsilon_{fuel} \quad (5)$$

where $\dot{m}_{fuel}$ is the mass rate of fuel consumption, and ε_{fuel} is the specific exergy of the fuel that can be explained as follows:

$$\varepsilon_{fuel} = H_u\varphi \quad (6)$$

where H_u is the lower heating value of the fuel, and φ is the chemical exergy factor, which is given by:

$$\varphi = 1.0401 + 0.1728\,\frac{h}{c} + 0.0432\,\frac{o}{c} + 0.2169\,\frac{s}{c}\left(1 - 2.0628\,\frac{h}{c}\right) \quad (7)$$

where h, c, o, and s are the mass fractions of hydrogen, carbon, oxygen, and sulfur in the fuels.

Thermomechanical and chemical exergies comprise exhaust or output exergy. The definition of specific thermomechanical exergy is:

$$\varepsilon_{tm} = (h - h_0) - T_0(s - s_0) \quad (8)$$

where h is the specific enthalpy, s is the particular entropy, and the subscript "0" defines the dead state. Exhaust temperature data allows for easy determination of h and s values using reference [21].

The concept of chemical exergy can be defined as:

$$\varepsilon_{chem} = \overline{R}T_0 \ln \frac{y}{y^e} \tag{9}$$

where T_0 is the ambient temperature, y is the mol fraction of exhaust gases determined by balancing the real combustion equations of the fuels through emission measurement, $\overline{R}$ is the general gas constant, and y^e is the mole fraction of the component that falls under Table 3's definition of environment [17].

Table 3. Definition of environment [8]

Reference Component	Mol Fraction (%)
N_2	75.67
O_2	20.35
CO_2	0.03450
H_2O	3.03
CO	0.0007
SO	0.0002
H_2	0.00005
Other	0.91455

Exhaust exergy rate can be written as:

$$\dot{E}x_{ex} = \sum \dot{m}_i\,(\varepsilon_{tm} + \varepsilon_{chem})_i \tag{10}$$

where $\dot{m}_i$ is the mass rate of combustion products, ε_{tm} and ε_{chem} are the specific thermomechanical exergy and chemical exergies of the exhaust gases, respectively.

Exergetic efficiency can be illustrated as [21]:

$$\Psi = \frac{\dot{E}x_w}{\dot{E}x_{in}} \tag{11}$$

where $\dot{E}x_{in}$ is the fuel or input exergy rate, and $\dot{E}x_w$ is the net exergy work rate.

Three replications were performed for the experiments, which followed a completely randomized design (CRD). The engine's fuel tank was sequentially filled with fuel mixtures made from blended gasoline and alcohols. The engine was allowed to run for 15 minutes with each fuel blend to stabilize operating conditions before data collection began. Based on the available laboratory equipment, all necessary data—such as inlet and outlet water temperatures—were accurately recorded for calculating exergy rates using the formulas above.

Results and Discussion

Exergy Analysis

The engine's gasoline-fueled energy flow diagrams, as well as those for pentanol, butanol, propanol, and ethanol alcohols at speeds of 1000, 1500, and 2000, are shown in Figs. 2-4. The engine input exergy at 1000 rpm for gasoline and all alcohol-containing fuels is 61.31, 57.25, 54.23, and 52.2 kW, respectively, of which 13.35% is attributed to the alcohol. 13.34%, 10.50%, and 8.74% are exergy cooling rates, respectively. Additionally, these results suggest that increasing the proportion of pentanol in the blend tends to decrease both the input exergy and cooling exergy rates, likely due to the higher latent heat and lower volatility of heavier alcohols, such as pentanol, which influence combustion timing and cylinder pressure development. The exhaust exergy rates at the same speed were 24.10%, 25.22%, 27.71%, and 17.29%, respectively. The exergy destruction rates in the fuels are equal to 40.78%, 38.09%, 44.16%, and 40.23%, respectively. Finally, the valuable work done is equal to 32.04%, 23.33%, 13.65%, and 25.16%, respectively. Conversely, the cooling exergy rate has increased because the engine runs at 1500 rpm. However, the energy content of exhaust gases drastically dropped as engine speed increased. With increasing speed came an increase in the rate of energy degradation in all fuels, as well as a corresponding rise in the amount of practical work that could be completed. The exergy rate of cooling in all fuels increased with increasing speed at 2000 rpm, but the exergy rate of exhaust gases decreased drastically from 1500 to 2000 rpm. On the other hand, the exergy destruction rate decreased by 82.82% and 45.78%, respectively, with increasing engine speed in fuels $G_{55}Pe_{10}E_{10}Bu_{10}Pr_{10}$ and $G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$ compared to the base fuel, but increased in fuel $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$. The Useful work rate decreased with increasing engine speed from 1500 to 2000 rpm in fuels G60Pe10E10Bu10Pr10 and G55Pe15E10Bu10Pr10, while fuel $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ increased the applicable work rate with increasing speed.

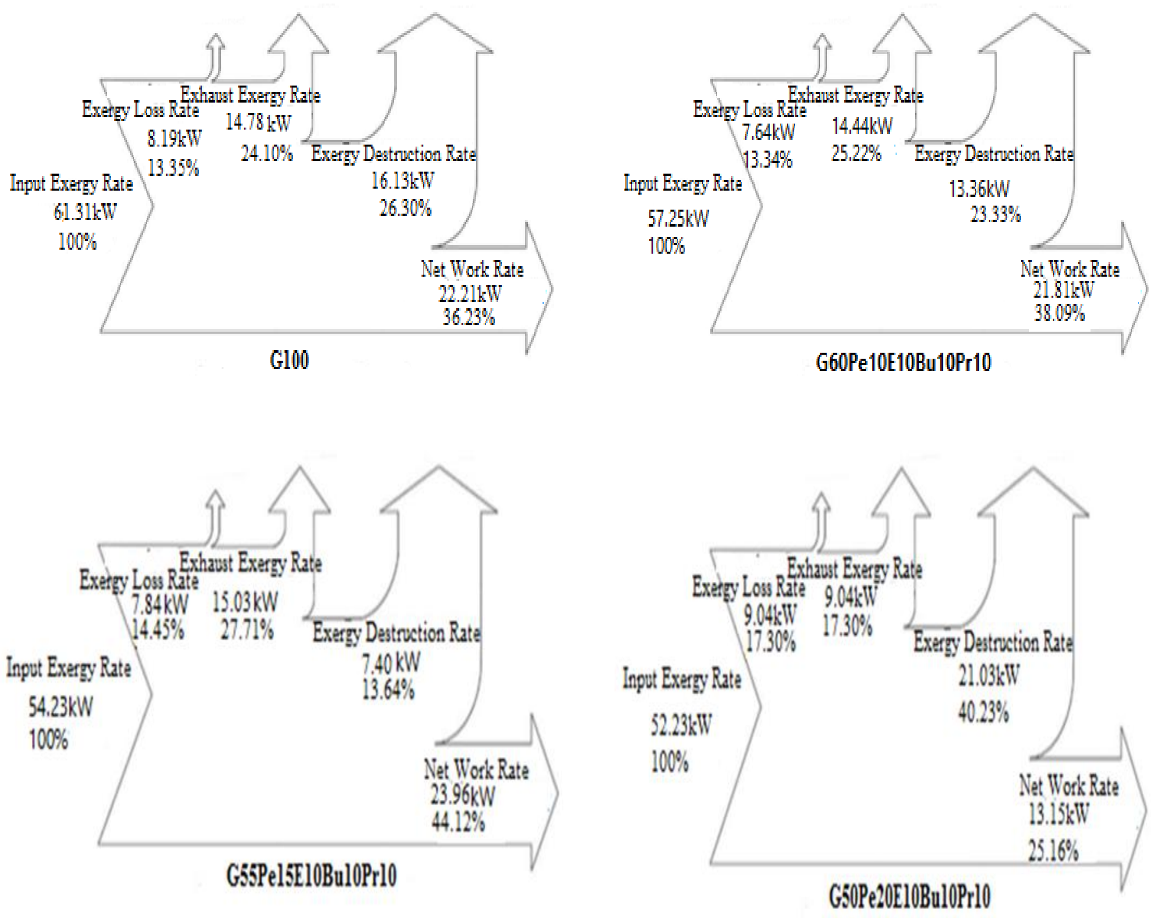

Fig. 2. Exergy analysis in engine control volume at 1000 rpm

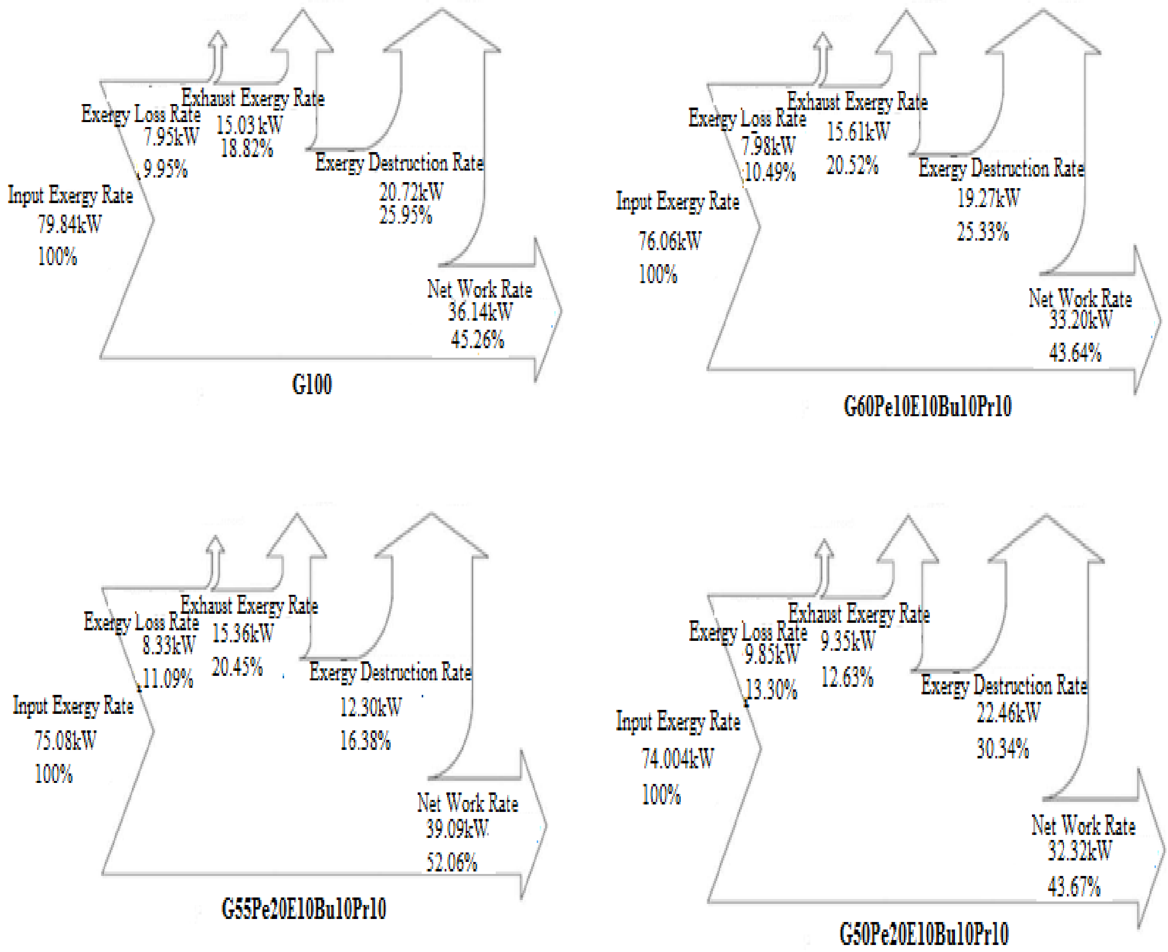

Fig. 3. Exergy analysis in engine control volume at 1500 rpm

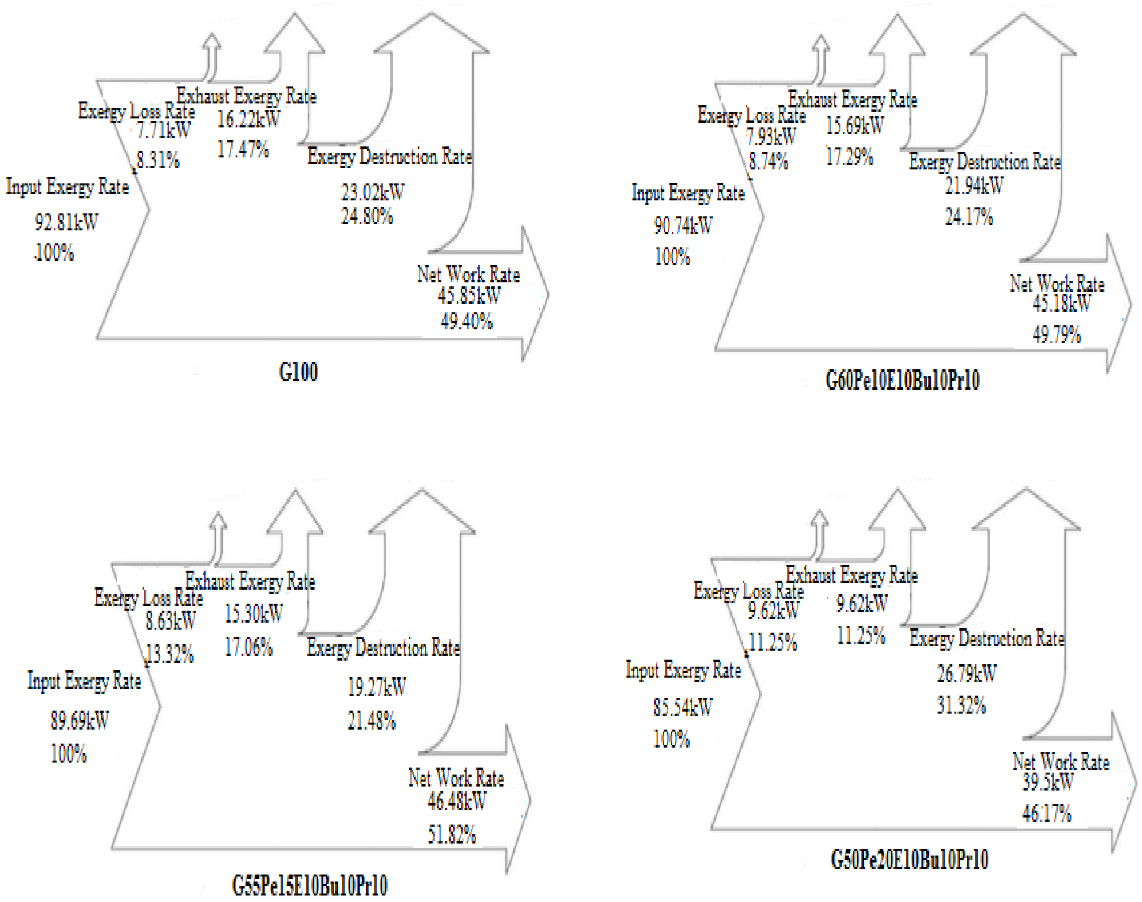

Fig. 4. Exergy analysis in engine control volume at 2000 rpm

Exergy analysis was conducted based on experimental data at various engine speeds and with different volumetric ratios of bio-alcohols. The results for the distribution of exergy components—including input fuel exergy, exergy through heat transfer, exhaust exergy, exergy work, and exergy destruction—at 1000, 1500, and 2000 rpm are presented in Figs. 5-9.

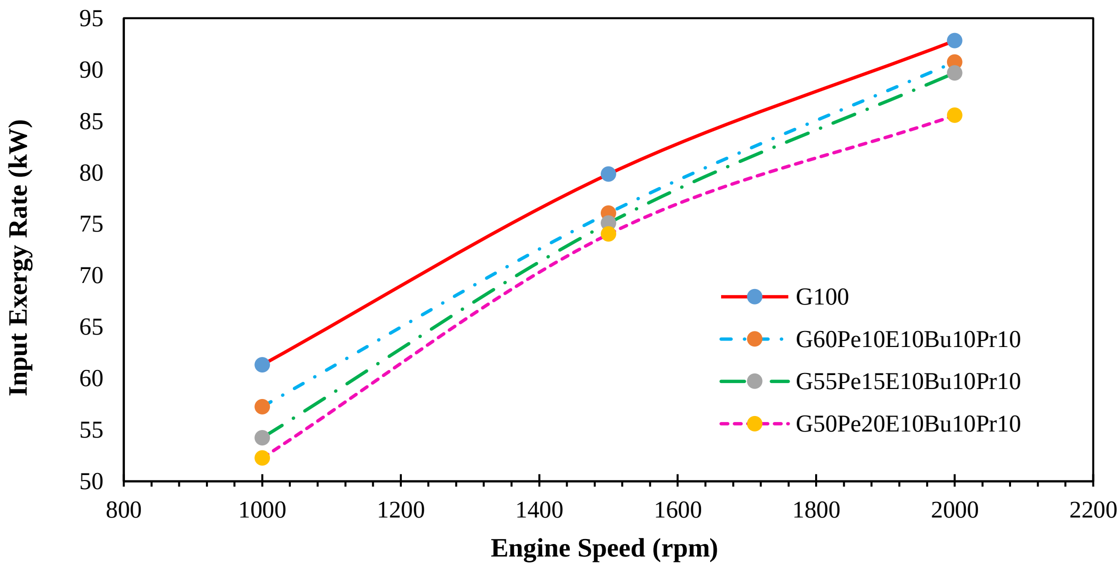

Fig. 5. Input exergy rate at different engine speeds

According to Fig. 5, the engine input exergy increases with the increase in engine speed. Among the experimental fuels, G_{100} fuel (pure gasoline) has the highest input exergy at all speeds. Due to the rise in the percentage of pentanol in the fuels, the amount of exergy input of the engine decreases, respectively, and the trend of the input exergy at 1000 to 2000 rpm was the same for all fuel blends. Fuel consumption increases with engine speed, resulting in a rise in the engine's power output. Additionally, when engine speed increases, all fuel blends have an increased ignition pressure. The combustion chamber receives extra fuel to meet the current power demand, which is the cause.

Additionally, two reasons raise cylinder pressure while using alcohol fuels. The first factor is that alcohol fuels have a higher Latent Heat Evaporation Rate (LHER) than gasoline. One of the key elements in improving performance is the second factor, which involves adding more alcohol to gasoline to increase volumetric efficiency [9]. According to a study, because alcohol fuels have a higher volume efficiency and latent heat evaporation rate (LHER) than gasoline, adding methanol and ethanol to gasoline raises the combustion pressure in the engine's combustion chamber by 36.61%, because alcohol fuels have a higher volume efficiency and latent heat evaporation rate (LHER) than petrol. Additionally, compared to petrol, the sluggish flame speed of ethanol and methanol fuels is higher [22]. According to Vancoillie et al. [23], research on engine efficiency using gasoline and alcohol blends suggests that alcohol fuels ignite more slowly than gasoline, resulting in higher cylinder pressure. Furthermore, Yoon et al.'s research revealed that the pressure inside an engine cylinder varies when alcoholic fuels are added to gasoline [15].

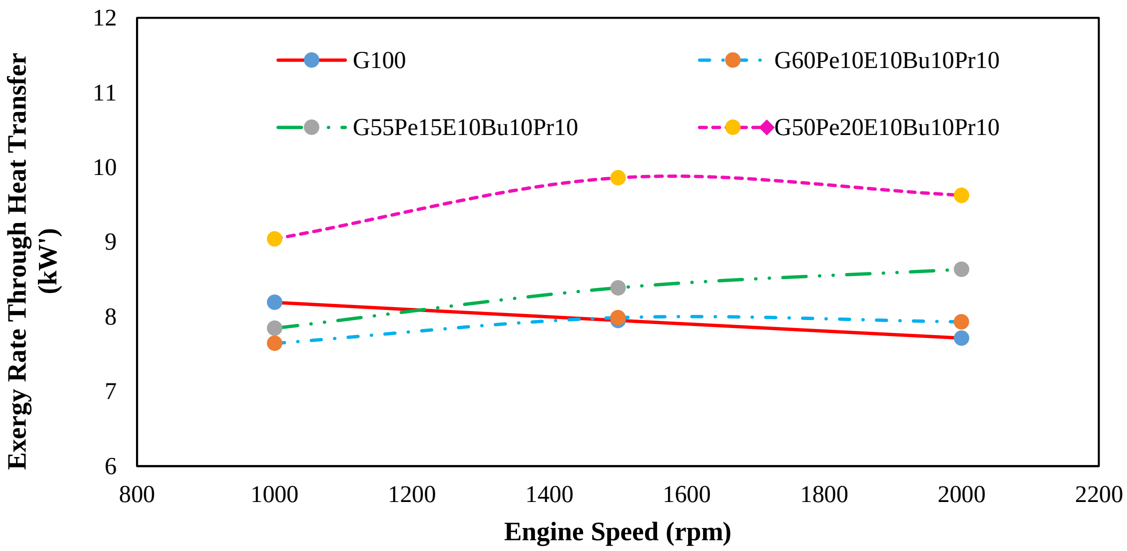

Fig. 6. Exergy rate through heat transfer at different engine speeds

Fig. 6 displays the cooling system's exergy; as engine speed increases, the exergy rate through heat transfer in G_{100} fuel falls. However, in other fuel blends, the exergy rate through heat transfer increases with the volume percentage of pentanol in the fuel blends. The highest exergy rate through heat transfer related to $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ fuel at 1500 rpm is 9.58 kW. However, in the mentioned fuel blend, the trend increases with the increase in engine speed from 1000 to 1500 rpm. However, from 1500 to 2000 rpm, it decreases slightly. As shown in Fig. 6, the exergy rate through heat transfer is also strongly affected by engine speed and fuel type. Fig. 6 shows that the exergy rate through heat transfer increases when engine speed increases in fuels containing bio-alcohols, even at low speeds. This tendency can be explained by the fact that heat transfer is limited at high engine speeds, but increases due to the engine cycle becoming more efficient at the same high speeds, which in turn decreases the heat transfer time. The engine generates the maximum exergy rate through heat transfer when the fuel's alcohol content rises [24]. This behavior can be explained by the fact that, compared to other blends, the exergy released by heat transfer increased with an increase in gasoline's pentanol concentration. This is because fuel blends with higher pentanol content have higher octane numbers and calorific values. As a result, when the volume percentage of pentanol increases, so does the combustion temperature.

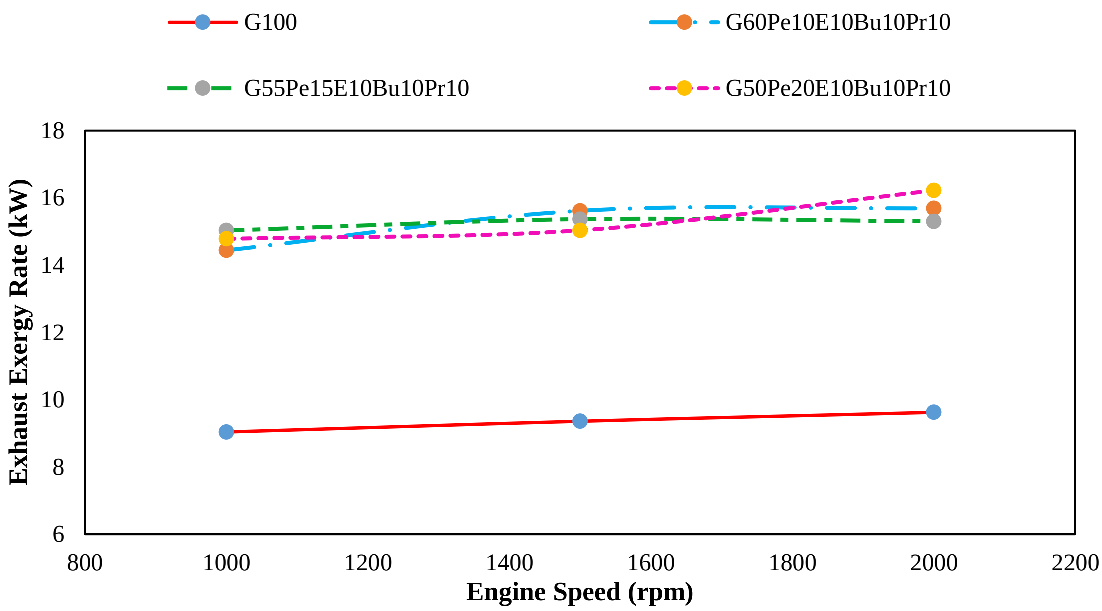

Fig. 7. Exhaust exergy rate at different engine speeds

Fig. 7 displays the rate of exhaust exergy. The engine's exhaust gases release part of the fuel energy into the atmosphere. The temperature and mass flow rate of exhaust gases determine the exhaust exergy rate. Pure gasoline fuel increases slowly with increasing speed from 1000 to 2000 rpm. This fuel has the lowest exhaust exergy compared to other fuels. The exhaust exergy rate in $G_{60}Pe_{10}E_{10}Bu_{10}Pr_{10}$ and $G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$ fuel blends increases with increasing speed from 1000 to 1500 rpm, and decreases from 1500 to 2000 rpm. In the $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ fuel blend, the exhaust exergy rate decreases with increasing speed from 1000 to 1500 rpm and then increases. It should be noted that all alcoholic fuel blends fall within a specific range on the diagram and are very close to one another. High levels of exhaust gas exergy are observed at high engine speeds, as the exergy rate of the exhaust increases with speed due to an increase in combustion temperature. This can be explained by the fact that the test engine's combustion cycles were increased while operating at the same low speeds. This raised the temperature of combustion and, consequently, the energy of the exhaust gases [25]. The exhaust temperature and the molar ratios of constituents in the exhaust gas influence the exhaust exergy rate. The exergy calculation also includes physical and chemical exergy components. Due to comparable dominating effects, the variance in exhaust energy and exhaust exergy rates with respect to fuel type and engine load is similar [26]. As shown in Fig. 7, alcohol-added fuels exhibit greater exhaust exergy rates compared to gasoline, especially at lower engine speeds. Higher alcohol content leads to increased exhaust energy. However, at high speeds, fuels containing a larger volume percentage of pentanol exhibit a higher exhaust exergy rate than gasoline. Particularly at high engine speeds, a greater amount of alcohol in the fuel causes the exhaust's exergy rate to climb more sharply.

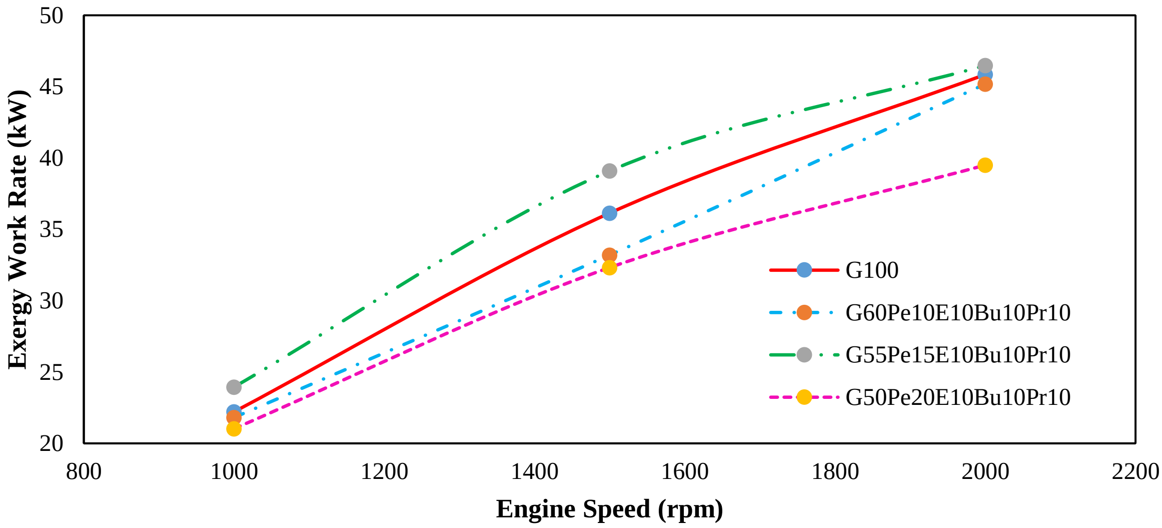

Fig. 8. Exergy work rate at different engine speeds

Fig. 8 shows the exergy work rate. The Exergy work rate is the same as the energy of the motor axis because the power energy is equivalent to exergy [26]. The Exergy work rate in all fuel blends exhibits a similar trend, increasing with speed from 1000 to 2000 rpm. The highest exergy work rate related to the $G_{60}Pe_{10}E_{10}Bu_{10}Pr_{10}$ fuel blend at 2000 rpm is 46.48 kW. Additionally, the lowest exergy work rate associated with the $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ fuel is 21.03 kW. When engine speed increases, the exergy work rate increases because bio-alcohols have a lower calorific value than conventional fuel blends, which lowers shaft energy. The low calorific value of pentanol explains the increase in exergy work rate. This outcome also aligns with a prior study showing a clear correlation between fuel exergy and LHV [24]. The reasons for this behavior are that the $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ fuel blend has a lower heat of vaporization and a higher octane number than the $G_{60}Pe_{10}E_{10}Bu_{10}Pr_{10}$ and $G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$ fuel blends, which leads to an increase in the combustion process. Additionally, the G50Pe20E10Bu10Pr10 fuel blend exhibits moderate heating values and octane numbers. Hence, the $G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$ fuel blend provides a moderate exergy work rate.

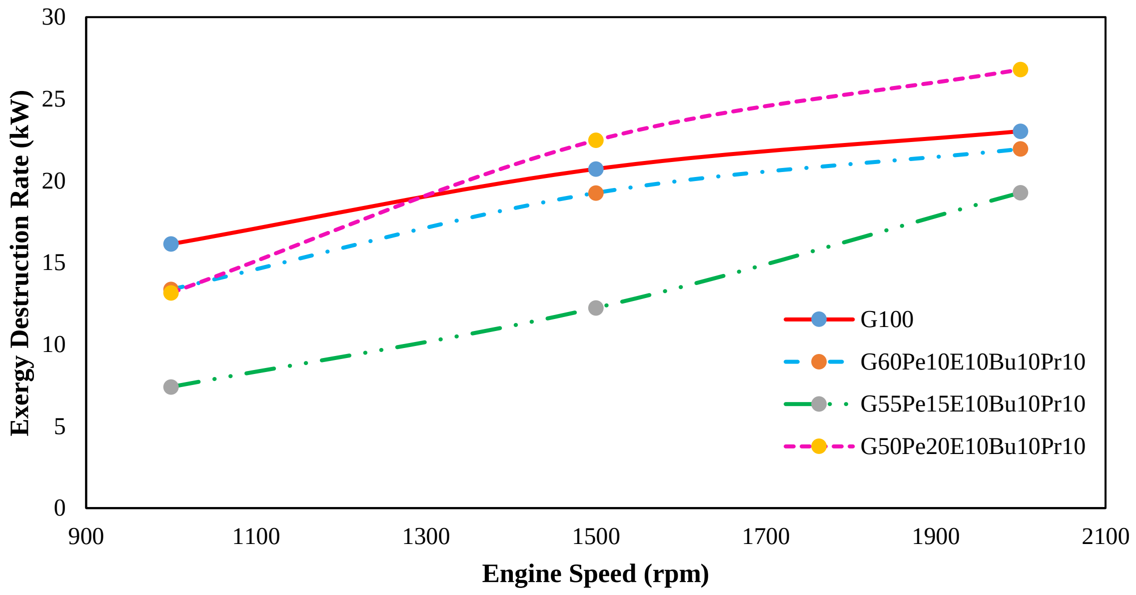

Fig. 9. Exergy destruction rate at different engine speeds

The rate of energy destruction at various engine speeds is displayed in Fig. 9. Because of the higher friction losses, the rate of energy destruction reaches its maximum at the fastest speed. The rate of energy destruction reflects the amount of energy that is irreversibly lost and cannot be recovered within the system cycle. This loss cannot be identified through conventional energy analysis. Unlike other loss rates, this type of exercise cannot be turned into productive work since it occurs in the engine as a result of irreversible processes, including strong heat transfer, turbulent flow in the combustion chamber, mixing of the air-fuel blend, flow friction pressure decrease, gas expansion, etc. [28]. Compared to other fuel blends at lower speeds, the exergy destruction rate at 2000 rpm is the highest in all fuel blends. The rate of energy destruction increases when bio-alcohols are added to gasoline in the $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ fuel blend at 1500 and 2000 rpm compared to pure gasoline. It is evident that adding more alcohol to the blends increases the irreversibility effect and produces entropy. Analogous research also indicates that [29] exergy destruction is considered the highest loss from input exergy, directly affecting brake power due to its effect on fuel energy converted to output [26]. Fig. 9 shows that as engine speed increases, the rate of energy destruction also increases. This could be due to the piston's increased average movement at high engine speeds, which enhances combustion and increases friction between engine parts. Additionally, the in-cylinder pressure affects friction based on piston speed, ultimately increasing exergy degradation.

Conclusion

This study comprehensively analyzed the exergy losses in a four-cylinder, water-cooled gasoline engine fueled with various blends of gasoline, ethanol, butanol, pentanol, and propanol at engine speeds of 1000, 1500, and 2000 rpm. The findings revealed that increasing the pentanol concentration in the fuel blends led to a decrease in the engine’s input exergy rate. However, this rate increased with engine speed due to higher fuel consumption. Alcohol-based fuels produce greater in-cylinder pressures due to their delayed spark and combustion characteristics compared to pure gasoline. Unlike pure gasoline, where heat transfer exergy declined with speed, blends with higher pentanol content exhibited increasing heat transfer exergy, reaching a maximum of 9.85 kW for the G50Pe20E10Bu10Pr10 blend at 1500 rpm. Exhaust exergy, governed by exhaust gas temperature and mass flow rate, also rose with engine speed, with the lowest values observed for pure gasoline. The exergy work rate showed a consistent upward trend across all blends with increasing speed, peaking at 48.46 kW for the

G60Pe10E10Bu10Pr10 blend at 2000 rpm, while the lowest value was 21.03 kW for the G50Pe20E10Bu10Pr10 blend. Moreover, exergy destruction was found to be highest at 2000 rpm across all fuel blends, especially in alcohol-rich mixtures, indicating increased irreversibility and entropy generation with higher bio-alcohol content.

Funding

This research received no external funding.

Acknowledgments

Data was collected from the Renewable Energy Lab at Bu-Ali Sina University. The authors thank Bu-Ali Sina University for providing the materials used to collect the data.

Conflicts of Interest

The authors declare that they have no conflicts of interest.

Nomenclature

G_{100}	Gasoline100%
$G_{60}Pe_{10}E_{10}Bu_{10}Pr_{10}$	Gasoline 60%+ Pentanol 10% + ethanol 10%+ butanol 10% +propanol 10%
$G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$	Gasoline 55%+ Pentanol 15% + ethanol 10%+ butanol 10% + propanol 10%
$G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$	Gasoline 50%+ Pentanol 20% + ethanol 10%+ butanol 10% + propanol 10%
$\dot{\mathbf{E}}\mathbf{x}$	Exergy rate (kW)
$\dot{\mathbf{m}}$	Mass flow rate (kg/s)
$\mathbf{H_u}$	Lower heating value (kJ/kg)
$\dot{\mathbf{Q}}$	Heat transfer rate (kW)
$\dot{\mathbf{W}}$	Work rate (kW)
$\overline{\mathbf{R}}$	Universal ideal gas constant (kJ/kgK)
H	Specific enthalpy (kJ/kg)
S	Specific entropy (kJ/kgK)
T	Temperature (K)

Greek symbol

$\boldsymbol{\varepsilon}$	Specific flow exergy (kJ/kg)
a	Air
$\boldsymbol{\varphi}$	Chemical exergy factor
ch	Chemical
$\boldsymbol{\psi}$	Exergy efficiency (%)

Subscripts

0	Dead state
C	Cooling air
Dest	Destruction
Ex	Exhaust
in	Input
out	Output
i	Numerator
w	Work

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How to cite: Khoshkname D, Shadidi B, Haji Agha Alizade H. Thermodynamic Performance and Exergy Analysis of a Four-Cylinder Gasoline Engine Fueled with Gasoline-Alcohol Blends: The Role of Pentanol, Butanol, Ethanol, and Propanol. Journal of Chemical and Petroleum Engineering 2025; 59(2): 387-403.