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

Abstract

This study investigates the combustion behavior and exergy performance of a water-cooled, four-cylinder gasoline engine using pure gasoline and various gasoline–bio-alcohol blends containing ethanol, butanol, propanol, and pentanol. Experiments were conducted at engine speeds of 1000, 1500, and 2000 rpm. Exergy analysis was used to evaluate input fuel exergy, exergy losses, and exergy efficiency. Increased engine speed correlated with increased input exergy rates across all fuels, owing to greater 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 improve engine thermodynamic performance while reducing fossil fuel dependence. Among the alcohols, pentanol blends demonstrated the most favorable exergy performance at higher engine speeds.

Keywords: Bioalcohols, Exhaust, Heat, Work, Irreversibility, Engine

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 like 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, 4, 5]. Energy analysis can determine the work capacity in lost energy of internal combustion engines. Because of this, analyzing exergy analysis is crucial to figuring out the ideal operating parameters and boosting internal combustion engine efficiency. In internal combustion engines, energy analysis plays a critical role in assisting researchers and designers in making the right decisions. Adding alcohol to gasoline helps provide complete combustion conditions, increase combustion efficiency, and reduce environmental pollution because of the extra oxygen in their 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 useful 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 determine the different losses that occur while the engine runs [8-7]. While the first rule of thermodynamics is typically used to assess internal combustion engines, fuel stability and renewability are not considered. 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 energy. The most theoretically useful work that a system may produce is known as its energy. Exergy degradation, another name for irreversibility, is the primary cause of the decline in diesel engine efficiency [10]. Examples of the study work related to the exergy analysis of the gasoline engine 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 showed that increasing the amount of alcohol increased the total irreversibility in exergy and improved exergy and thermal efficiency to a partial extent. Özkan and Shakmak, 2018 [12] explored how blrnded gasoline with a 10% volume percentage of oxygenated fuel additives (solketal, methanol, and ethanol) affected the exergy parameters measured by the crank angle in a SI engine. In keeping with the study's findings, adding alcohol to gasoline raises the cylinder's peak pressure, with methanol compounds showing the most pressure increase. 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 raise exhaust exergy losses 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 significantly affect performance and emission outputs. Optimal outcomes were achieved at 9% biodiesel and 70% load, enhancing efficiency while reducing pollutant[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 much better and 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 alternate gaseous fuels, lean and low-temperature combustion, and exhaust heat recovery systems to lower energy losses and boost engine efficiency. In another study, Fu et al, 2013 [5] Exergy analysis and stability were proven by the performance and emission outcomes of ethanol and methanol added to 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, improving performance (BTE 20%, BSFC 307 g/kW·h) and reducing 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 gasoline, natural gas, and methanol fuel blends on exergy analysis in 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 (NOx) 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 that increasing the speed, exhaust losses and heat transfer increase, which leads to more energy losses. Additionally, data suggests higher energy losses with ethanol fuel compared to gasoline and methanol [15].

The continuous use of fossil fuels has reduced their resources worldwide, and for this reason, many researchers worldwide are looking for a solution to facilitate the use of fossil fuels. On the other hand, the low efficiency of biofuels compared to fossil fuels causes many researchers have done extensive research to solve these defects and improve the conditions of 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. The aim of this study is to perform a comparative exergy analysis of gasoline-alcohol blends under variable engine speeds.

2. Materials and Methods

2.1 Preparation of fuel blends

This test used gasoline from a Hamedan, Iran gas station. 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 ratio of prepared fuels is 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 gasoline, 15% by volume of pentanol, 10% by volume of ethanol, and 10% by volume of butanol, and also 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

2.2 Engine Specifications

This research used a four-cylinder gasoline engine, direct injection, water-cooled and coupled to a dynamometer, available at Bu-Ali Sina University, to conduct experiments (Figure 1). This test was conducted at a temperature of 25°C and an air pressure of 101,900 Pascal.

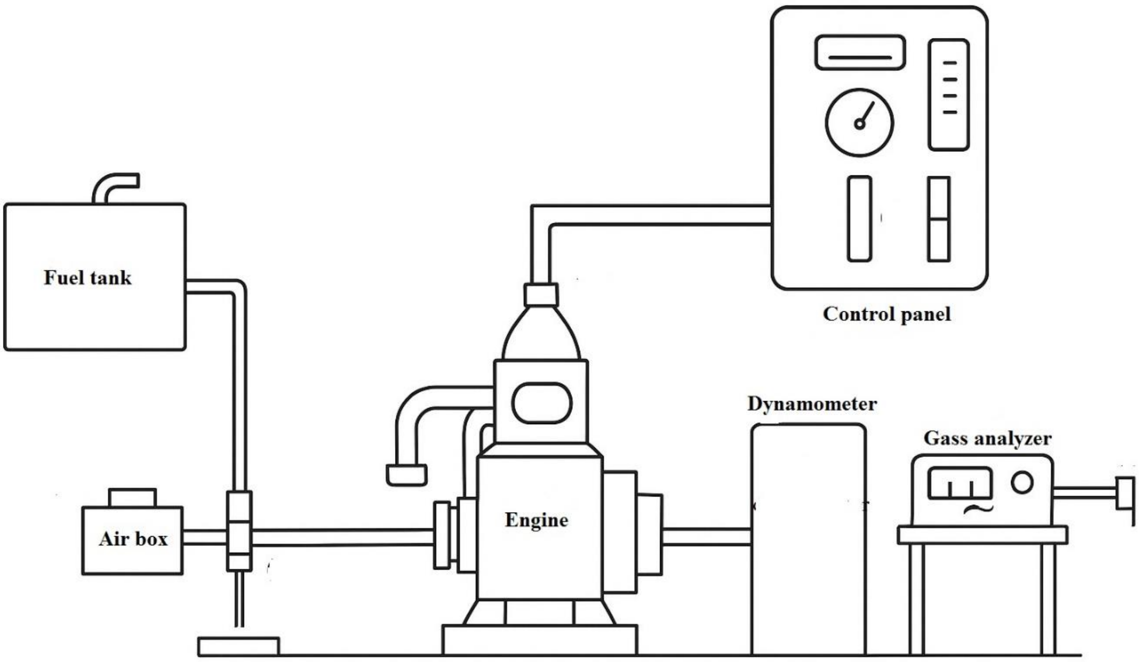

Figure 1. Engine set up.

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

2.3 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 shaft work exergy rate, while $\dot{E}x_{dest}$ denotes exergy destruction rate (irreversibility). When conducting an exergy analysis, as 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 (TC) to the environment (T0) defined the lost-exergy rate, afterwards:

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

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

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

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.

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

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

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

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:

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

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

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

Where h, c, o and s are the mass fraction of hydrogen, carbon, oxygen and sulfur contents of 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 and s is the specific entropy and the subscript "0" defined the dead state. Exhaust temperature data allows for easy determination of h and s values using refer. [21].

The concept of chemical exergy can be defined as:

$$\varepsilon_{chem} = \bar{R}T_0 \ln\frac{y}{y^e} \quad (9)$$

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, $\bar{R}$ 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:

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

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]:

$$\Psi = \frac{\dot{E}x_w}{\dot{E}x_{in}} \qquad (11)$$

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 the calculation of exergy rates using the aforementioned formulas.

3. Results and Discussion

3.1 Exergy analysis

The engine's gasoline-fueled energy flow diagrams and pentanol, butanol, propanol, and ethanol alcohols at speeds of 1000, 1500, and 2000 are shown in Figures 2,3 and 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%. 13.34%, 10.50%, and 8.74% are exergy cooling rates, respectively. Also, These results indicate that increasing the proportion of pentanol in the blend tends to reduce both the input exergy and cooling exergy rates, likely due to the higher latent heat and lower volatility of heavier alcohols like 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 rate in the fuels is equal to 40.78%, 38.09%, 44.16%, and 40.23%, and finally, the useful 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 rising speed came an increase in the rate of energy degradation in all fuels and an increase in the amount of useful work completed. The exergy rate of cooling in all fuels increased at 2000 rpm with increasing speed, but the exergy rate of exhaust gases decreased drastically with increasing engine speed from 1500 to 2000 rpm. . On the other hand, the exergy destruction rate decreased by 82.82 and 45.78 percent, 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 $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ Useful work rate decreased with increasing engine speed from 1500 to 2000 rpm in fuel $G_{60}Pe_{10}E_{10}Bu_{10}Pr_{10}$ and fuel $G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$ and $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ increased applicable work rate with increasing speed.

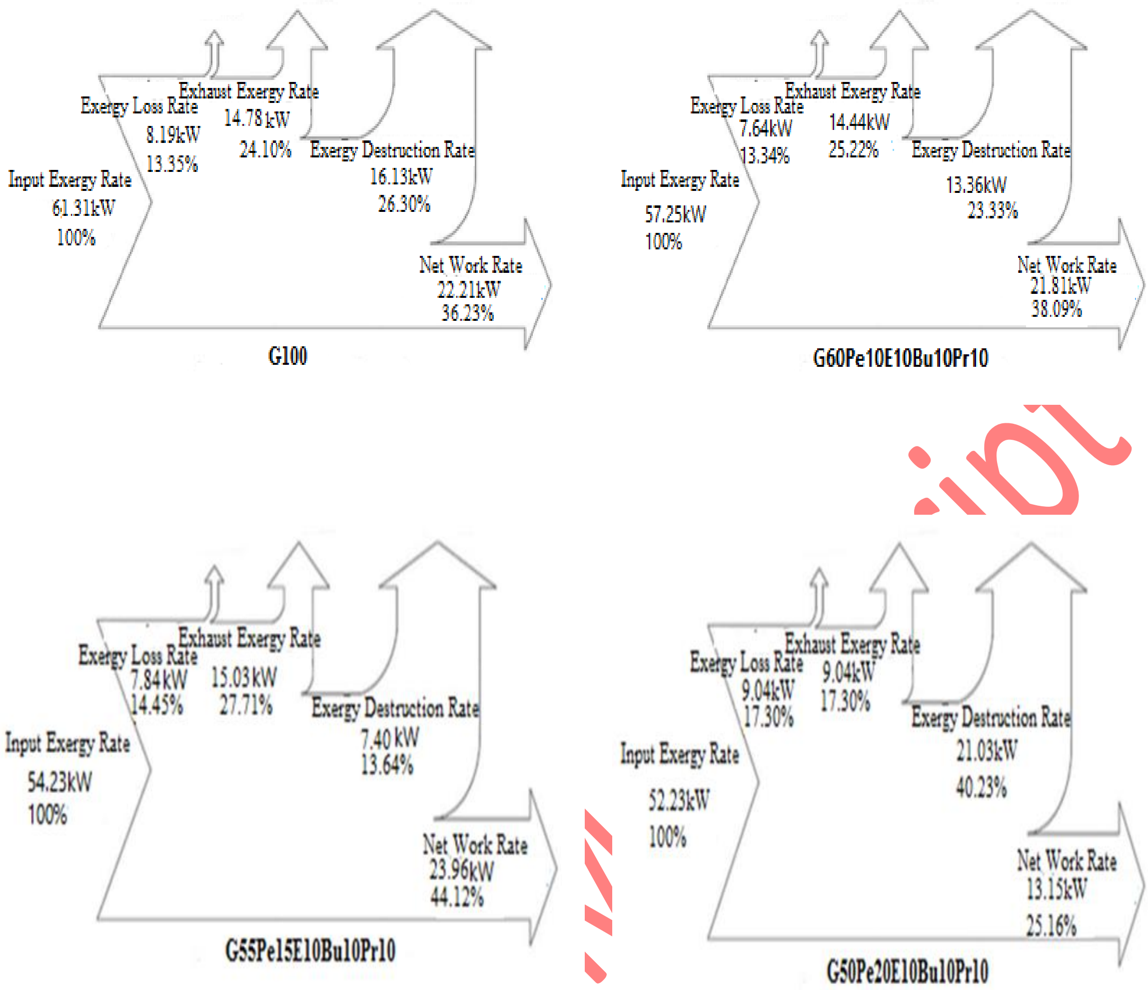

Figure 2. Exergy analysis in engine control volume at 1000 rpm.

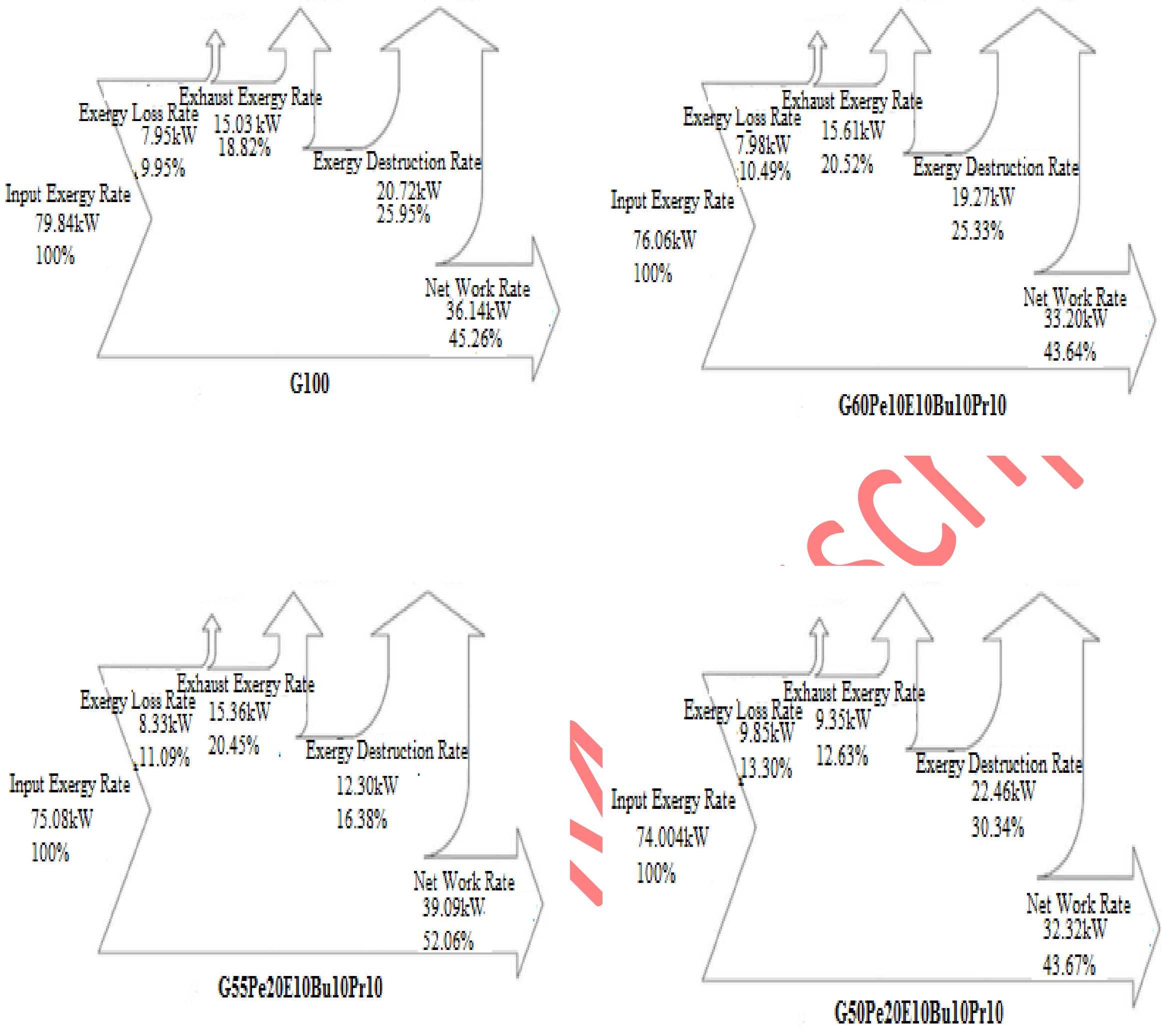

Figure 3. Exergy analysis in engine control volume at 1500rpm.

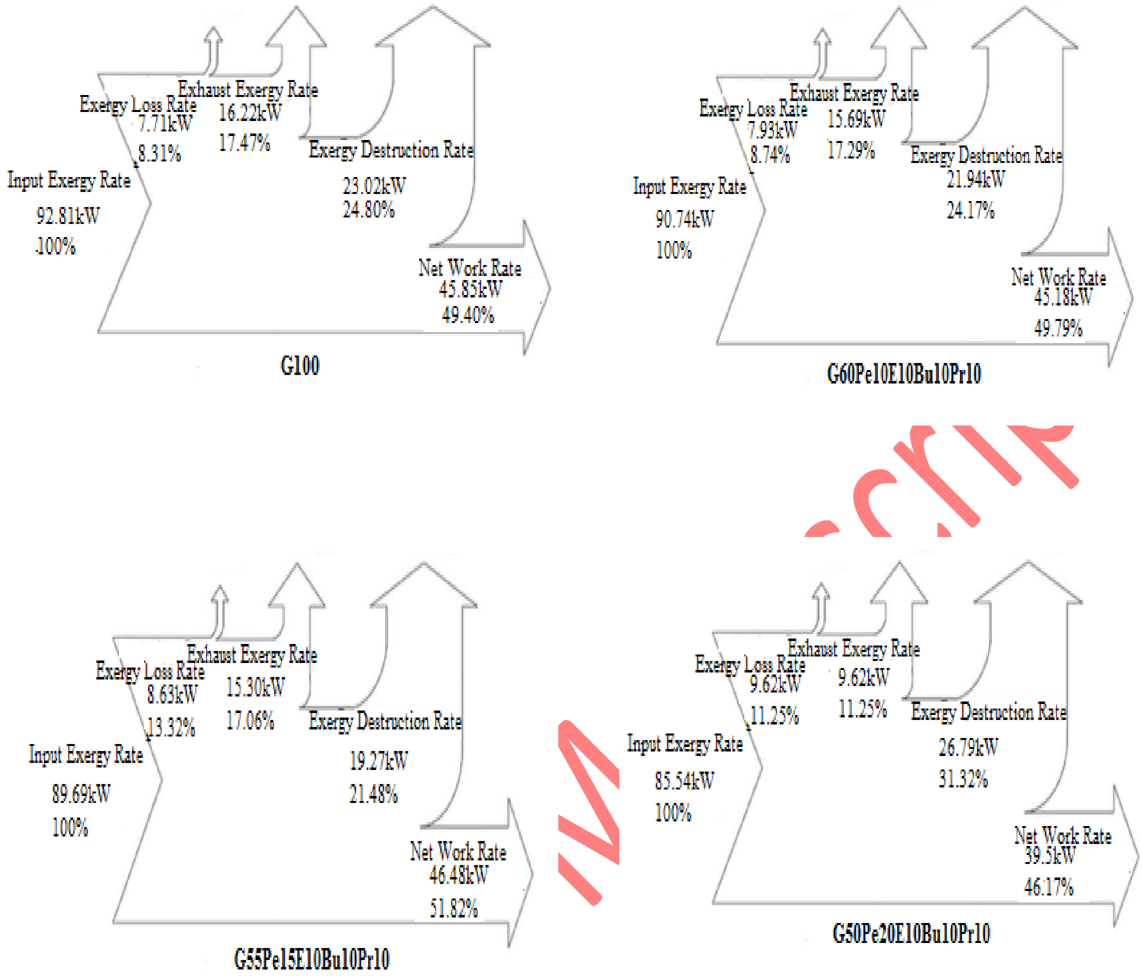

Figure 4. Exergy analysis in engine control volume at 2000 rpm.

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 Figures 5 through 9.

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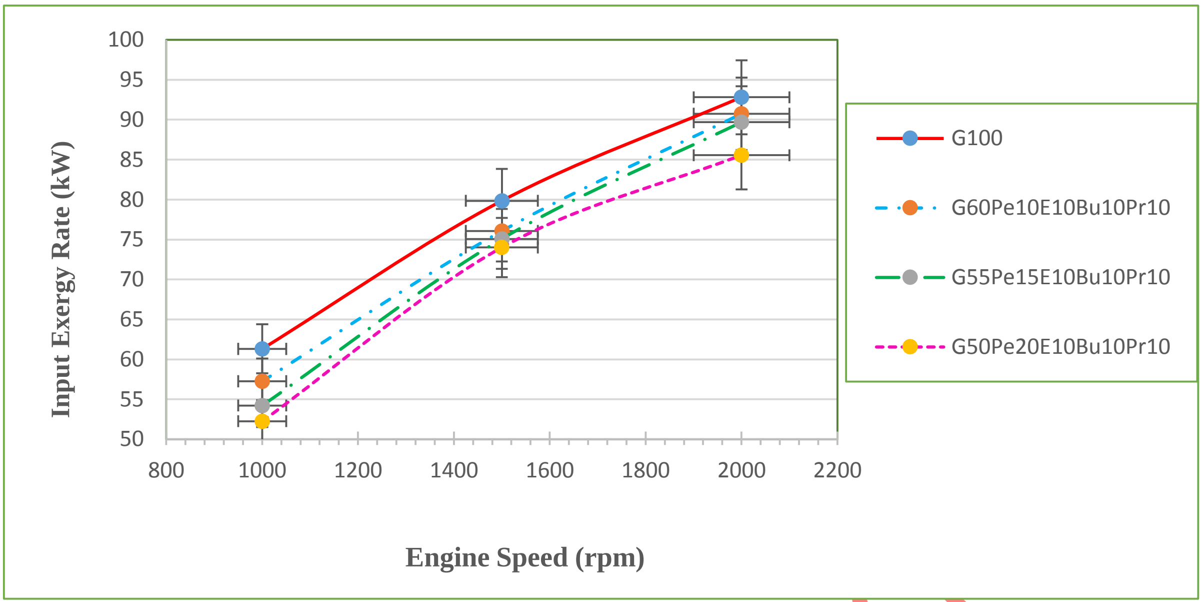

Figure 5. Input exergy rate at different engine speeds.

According to Figure 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 rises with engine speed, which causes an increase in engine input energy rate. Additionally, when engine speed increases, all fuel blends have an increased ignition pressure. The combustion chamber receives extra fuel to fulfill the present power need, 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, alcohol fuels ignite more slowly than gasoline, creating greater pressure inside the cylinder. Furthermore, Yoon et al.'s research revealed that the pressure inside an engine cylinder varies when alcoholic fuels are added to gasoline [15].

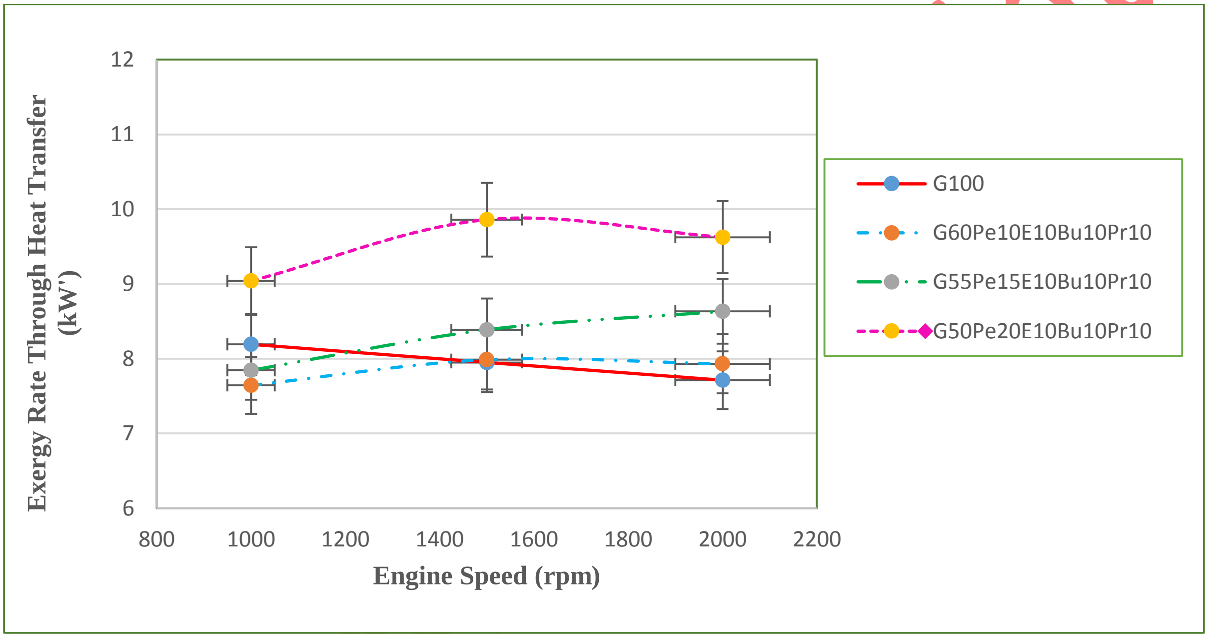

Figure 6. Exergy rate through heat transfer at different engine speeds

Figure 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, with the increase in engine speed from 1000 to 1500 rpm, this trend is increasing, but from 1500 to 2000 rpm, it decreases slightly. As shown in Figure 6, the exergy rate through heat transfer is also strongly affected by engine speed and fuel type. Figure 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 and increases due to the engine cycle increasing at the same high speeds, which decreases 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.

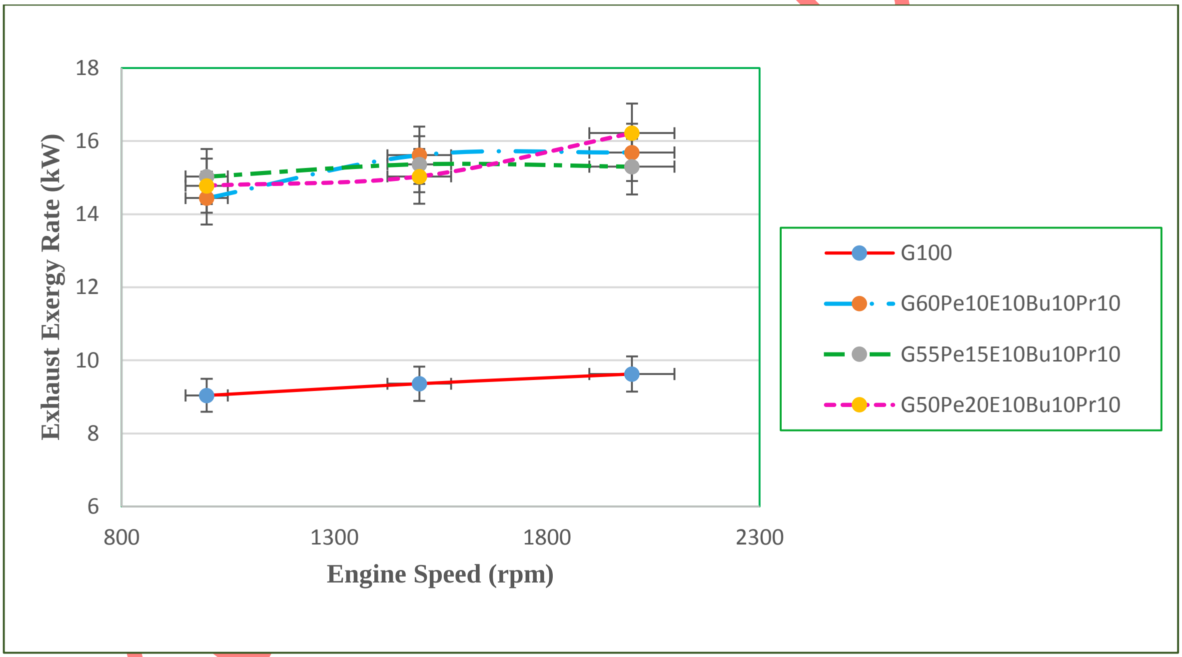

Figure 7. Exhaust exergy rate at different engine speeds

Figure 7 displays the rate of exhaust exergy. The engine's exhaust gases release part of the fuel energy into the atmosphere. Temperature and the 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 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 are in a particular range in the diagram and are very close to each other. High levels of exhaust gas exergy can be seen 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 exhaust gases' energy [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. Because of comparable dominating effects, the variance in exhaust energy and exhaust exergy rates concerning fuel type and engine load are similar [26]. As shown in Figure 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.

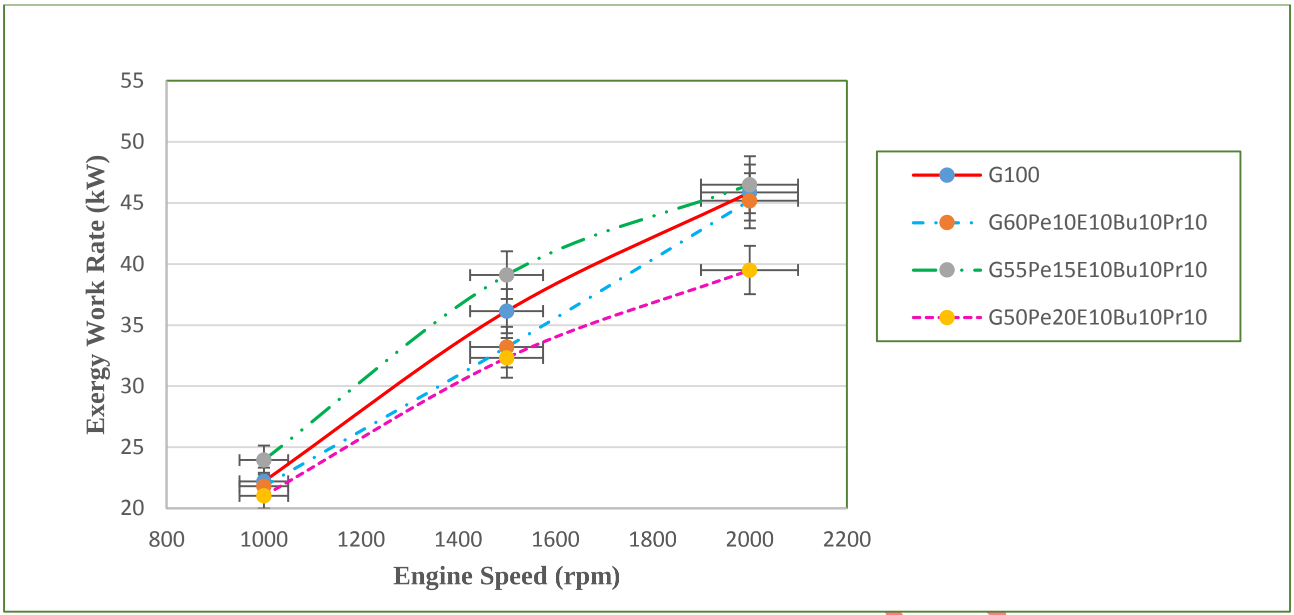

Figure 8. Exergy work rate at different engine speeds.

Figure 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 has a similar trend and increases with increasing 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. Also, the lowest exergy work rate related to 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. In addition, the $G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$ fuel blend has 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.

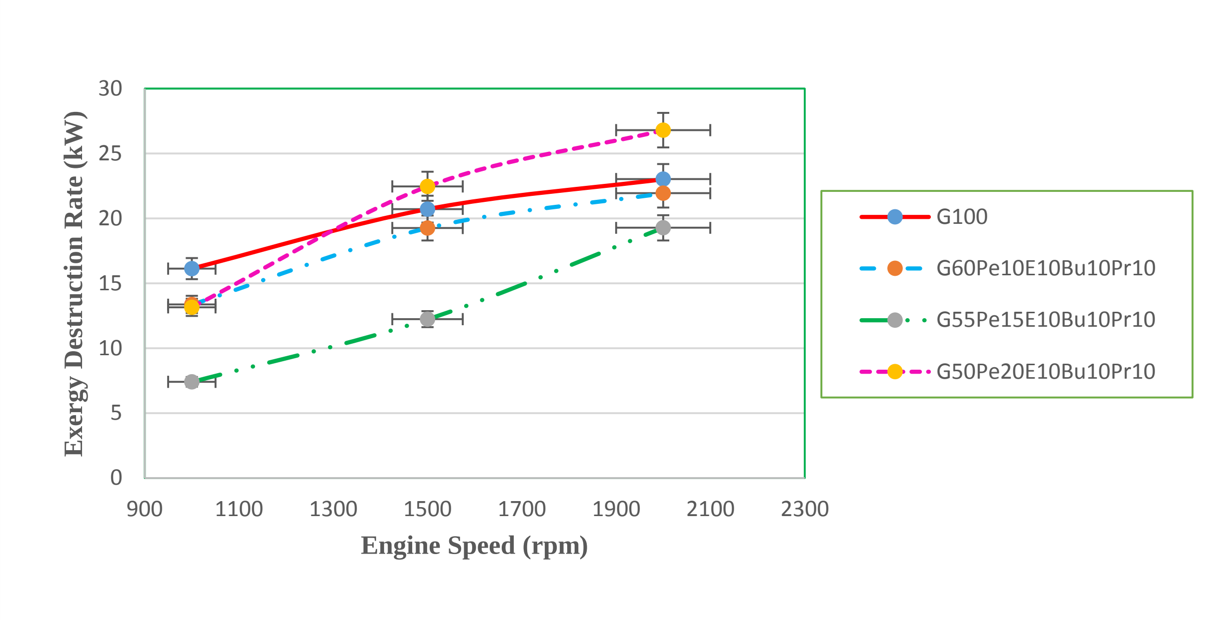

Figure 9. Exergy destruction rate at different engine speeds.

The rate of energy destruction at various engine speeds is displayed in Figure 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 in all fuel blends is the highest. 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]. Figure 9 shows that as engine speed increases, the rate of energy destruction also increases. This could be because of the piston's increased average movement at high engine speeds, which increases combustion at high engine speeds and increases engine speed and friction between engine parts. Additionally, the in-cylinder pressure impacts friction based on the piston speed, eventually increasing exergy degradation.

5. Conclusions

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, although this rate increased with engine speed due to higher fuel consumption. Alcohol-based fuels produced greater in-cylinder pressures owing to 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 was 21.03 kW for G50Pe20E10Bu10Pr10. 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.

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Conflicts of Interest: The authors declare no conflicts of interest.

Nomenclature		*Greek symbol*	
G_{100}	Gasoline100%	ε	Specific flow exergy (kJ/kg)
$G_{60}Pe_{10}E_{10}Bu_{10}Pr_{10}$	Gasoline 60%+ Pentanol 10% + ethanol 10%+ butanol 10% +propanol 10%	a	air
$G_{55}Pe_{15}E_{10}Bu_{10}Pr_{10}$	Gasoline 55%+ Pentanol 15% + ethanol 10%+ butanol 10% + propanol 10%	φ	Chemical exergy factor
$G_{50}Pe_{20}E_{10}Bu_{10}Pr_{10}$	Gasoline 50%+ Pentanol 20% + ethanol 10%+ butanol 10% + propanol 10%	ch	chemical
$\dot{\mathbf{E}}\mathbf{x}$	exergy rate (kW)	ψ	Exergy efficiency (%)
$\dot{\mathbf{m}}$	mass flow rate (kg/s)	*Subscripts*	
$\mathbf{H_u}$	lower heating value (kJ/kg)	0	dead state
$\dot{\mathbf{Q}}$	heat transfer rate (kW)	c	cooling air
$\dot{\mathbf{W}}$	work rate (kW)	dest	destruction
$\overline{\mathbf{R}}$	universal ideal gas constant (kJ/kgK)	ex	exhaust
H	specific enthalpy (kJ/kg)	in	input
S	specific entropy (kJ/kgK)	out	output
T	temperature (K)	i	numerator
		w	work

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