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TECHNICAL PAPER

The investigation of an energetic and exergetic performance characteristics of a beta-type Stirling engine with a rhombic drive mechanism

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Abstract

In this study, effects of using helium, nitrogen, air, carbon dioxide and argon gases as working fluid in a beta-type Stirling engine with rhombic drive mechanism and swept volume of 365 cm^3 on the engine performance characteristics for two different stainless steel and titanium displacer pistons at charge pressures of 1–5 bar were examined. The performance characteristics of manufactured Stirling engine were investigated at 1000 K (± 10 K) hot end and 300 K (± 5 K) cold end temperatures using a specifically designed electrical heater. Energy and exergy analyses were carried out using temperature, pressure, speed and torque values measured in performance tests. As a result of the exergy analysis, helium gas performed the best in the stainless steel displacer piston at a charge pressure of 4 bar and an engine speed of 550 rpm. Under the said conditions, 0.3726 W/K entropy generation and 195.53 W destroyed exergy were calculated in thermodynamic analysis in the helium working fluid. Furthermore, under the same conditions, helium gas achieved the highest efficiency values of 48.04% for thermal efficiency, 56.54% for exergy efficiency and 69.2% for Carnot efficiency. The lowest exergetic performance was revealed in titanium displacer piston when argon was used as working fluid.

Keywords Rhombic mechanism · Working fluid · Energy · Exergy · Entropy generation

List of symbols

c_p	Gas heat capacity at constant pressure (J/kg K)
dt	Cycle time
E_{in}	Energy flow into the system (W)
Ex	Exergy rate (W)
I	Current (Amper)
$\dot{m}$	Mass flow rate (kg/s)
n	Engine speed (rpm)
P	Pressure (kPa)
P_0	Pressure of the environment (kPa)
Q_C	Cooling water heat loss (W)
Q_H	Heat transfer rate (W)
Q_{lost}	Lost heat (W)
R	Gas constant (kJ/kgK)
$\overline{R}$	Universal gas constant (8.314 J mol/K)
T_C	Cold end temperature (K)
T_H	Hot end temperature (K)
T_0	Temperature of the environment (K)
rpm	Revolutions per minute
S_{gen}	Entropy produced (W/K)
V	Voltage (Volt)
$\dot{W}$	Work (W)
η_{carnot}	Carnot efficiency (%)
η_{exergy}	Exergy efficiency (%)
η_{th}	Thermal efficiency (%

Subscripts

cw	Coolant water
dest	Destruction
heat	Heat transfer
in	Inlet
out	Outlet
sys	System

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1 Introduction

Nowadays, energy is the most important need of our modern world, and it can be used by various methods. Energy can be transformed from one form to another. Engines are machines converting heat energy into mechanical energy. Today, fossil fuels are used as energy sources in engines. Studies on developing alternative fuels have gained momentum due to the decrease in the reserve sources of these fuels and the damage they cause to our environment. Renewable and sustainable energy sources come to the forefront in alternative fuel studies. Stirling engines can easily use many renewable energy sources due to their external combustion feature. At the present day, Stirling engines have again taken their place at the top in the energy generation required due to the positive developments in manufacturing and production technology compared to other power generation systems. Stirling engines that are capable of working with all kinds of heat energy can perform energy conversion that is less harmful to the environment [1]. Stirling engines operate by displacing the working fluid between the hot space and cold space, depending on the piston movements. Although the efficiency of the theoretical Stirling cycle is close to that of the Carnot cycle, this efficiency cannot be obtained under real operating conditions [2, 3]. The real efficiency of the Stirling engine is affected by the mechanical design, the properties of the working fluid used, the cold and hot end temperature, the charge pressure of the engine, and whether it is with or without a regenerator [4]. Stirling engines are divided into two classes: kinematic and free-piston engines. Kinematic Stirling engines are divided into three classes depending on the placement order of moving parts and cylinders as alpha, beta, and gamma [5].

Gustav Schmidt performed the first mathematical analysis of the Stirling cycle in 1871 [6]. In this analysis, it is accepted that the temperatures of the expansion volume and the heater volume, the temperatures of the compression volume and the coolant volume remain the same and constant throughout the cycle. In the following years, Finkelstein developed the ideal adiabatic analysis [7]. The most significant difference of this analysis from the ideal isothermal analysis method is that the expansion (heating) and compression (cooling) volumes are considered to be adiabatic. Since there is no heat transfer in adiabatic systems, a temperature fluctuation occurs with volume change in these volumes. Babaelahi et al. recommended the ideal polytropic analysis in 2015 [8]. In this analysis, the expansion and compression volumes were considered to be polytropic instead of isothermal and adiabatic.

Martaj et al. [9] carried out a theoretical study to optimize the distribution of the total area of different heat exchangers for hot end temperatures of 600, 700, 800, and 900 K and cold end temperatures of 290, 300, 310, and 330 K using energetic, entropic, and exergetic analyses in a Stirling engine. In the theoretical analysis conducted, it was reported that the maximum exergetic efficiency value at the hot end temperature of 900 K was calculated to be 0.7118, the heat exchanger area ratio was 0.593, the maximum exergetic efficiency value at the cold end temperature of 290 K was calculated to be 0.6783, and the heat exchanger area ratio was 0.556.

Razani et al. [10] investigated the exergy flow, which includes losses in the reservoirs and regenerator and undesired heat transfer, using a thermodynamic model in a Stirling refrigerator machine. They developed a simple model for the cooling capacity and efficiency of the Stirling refrigerator machine. In this model, it was stated that the Stirling refrigerator machine contained important sources of exergy destruction. The researchers analytically determined the relationship between the cooling capacity and efficiency with the effect of the pressure ratio on the regenerator.

Stanciu et al. [11] examined the power output and thermal efficiency characteristics using a thermodynamic model established for a solar energy system utilizing a Stirling engine. It was reported that the Stirling engine used in the research was supplied by a parabolic bowl mirror operating in Bucharest to collect solar radiation as a hot end source, and electricity was generated from a special generator connected to the engine. As a result of the thermodynamic analysis performed, it was stated that the overall efficiency of the solar power system reached a maximum of approximately 35% and a minimum value of 18%. It was indicated that the highest power output was achieved at noon during the day. Moreover, the distribution among the components in which exergy losses occurred was examined in the study.

Dong et al. [12] investigated LNG cryogenic generation by the Stirling cycle method on a model. In this model, the compressor is cooled by LNG, and nitrogen gas is compressed. The turbine is heated by seawater and drives a generator for electricity generation. The constant temperature compressor performs isothermal compression, while isothermal expansion processes are realized in the constant temperature turbine. In their study, Dong et al. preferred nitrogen gas as a working fluid in the Stirling engine since it was easy to produce. As a result of the thermodynamic calculations made using isothermal compression and expansion assumptions, it was stated that the power output increased by 14.97% compared to Senboku Daini cryogenic power station in Japan. It was indicated that the seawater heat exchanger was the equipment with the highest exergy loss. It was declared that the exergy losses in these heat exchangers constituted 92.65% of the total exergy loss.

Hachem et al. [13] compared the exergy efficiency of an Ericsson engine with a maximum power of 300 W and

a gamma-type Stirling engine with a maximum engine output of 500 W. Theoretical models were established to measure the exergy destruction and efficiency of engines in the study. It was stated that the Stirling engine provided higher energy efficiency compared to the Ericsson engine due to the advantages of the regenerator. It was reported that the exergy efficiency would increase by 8.5% when a preheater was used in the Ericsson engine. When the ratio of total exergy destruction is compared for Stirling and Ericsson engines, it is observed to be close to each other. While 44% of the exergy flow from the hot source is exergy destruction in Stirling engines, 41% of it is exergy destruction in Ericsson engines.

Wang et al. [14] performed theoretical analyses by adding a Stirling engine with a regenerator and operating according to the closed cycle to the system to obtain high efficiency in a conventional cryogenic heat engine. They employed the Schmidt theory to model the temperature changes of the hot and cold zones of the Stirling engine. Since the conventional open-cycle engine depends only on the operating pressure, it was stated that as this pressure increased from 0.1013 to 3.039 MPa, the expansion energy might increase from 0 to 39% in percentage. In the study, it was indicated that higher efficiency was obtained at low pressure in the suggested closed cycle.

Szczygieł et al. [15] used a Stirling engine in a system to obtain LNG cryogenic electrical energy in a theoretical model. The effect of the design and other parameters on exergy losses was determined by the exergy analysis conducted to evaluate the thermodynamic performance of the system. It was stated that the regenerator's exergy efficiency did not depend on the compression ratio. It was expressed that exergy decreased dramatically with the increase of dead volumes in the system. In the study, the cryogenic exergy of LNG was found to be 6.8 MJ/kmol.

Hachem et al. [16] conducted energy and exergy analysis using the experimental data of a beta-type Stirling engine with a capacity of 30 W, operating at a value close to atmospheric pressure and having a maximum hot end temperature of 650 °C. The cylinder of the beta-type Stirling engine used in the research was made of glass and had a diameter of 6 cm and a height of 20 cm, and air was used as the working fluid. A copper porous regenerator was used in this engine. Insulated and non-insulated conditions were taken into consideration for the two heat sources (hot and cold end) of the Stirling engine in the study. To reduce a significant part of the heat loss in the engine, the walls were insulated from the outside, and analyses were performed. With the insulation, ~ 15% recovery of the heating power in the engine and a reduction of ~ 20% of external heat losses were provided. It was reported that the increase in heating power provided higher efficiency.

Li et al. [17] investigated the impact of exergy losses on the mechanical power obtained in a solar-powered gamma-type Stirling engine. The Finite Speed Thermodynamics and Isothermal methods were employed in the research. Heat conduction, regenerator-induced losses, piston speed, hysteresis, leakage, and friction losses were calculated in the analyses. The energy and exergy analysis indicated that the two most significant losses in this engine were regenerator defects and gap leakage. It was recommended to improve the geometry of the regenerator for the engine to operate more efficiently and to insulate the displacer and cylinder wall to decrease heat losses.

Wills and Bello-Ochende [18] conducted the numerical analysis for different heater inlet temperatures and four different regenerator network types in an alpha-type engine model with a cylinder volume of 1.000 cm^3 and 15 kW constant energy input. The optimal dead-volume ratio, swept volume ratio, operating frequency, and phase angle are all demonstrated to decrease with increasing heater inlet temperature. It was observed that the most efficient network type as the regenerator type provided a smaller pressure decrease, had a larger cross-sectional flow area and was shorter in size.

Gholamalizadeh and Chung [19] carried out an energy and exergy analysis for a pilot parabolic solar dish-Stirling system installed in Kerman city, Iran. While the energy efficiency was very low in the months between November and February during the year, the exergy efficiency was calculated to be 13.2% maximum in July. It was indicated that the exergy losses occurred at most in the receiver, then in the Stirling engine. The exergy loss in the condenser used in the Stirling engine was high. The total exergy losses of the receiver, Stirling engine, and condenser were determined to be 43.4%, 23.3%, and 33.3%, respectively.

Entezari et al. [20] performed a study that increased efficiency by recovering high-temperature waste gases discharged from the flue in a gas turbine cycle by the Stirling engine. There were more efficient operation and a reduction in electricity cost in the combined cycle of the gas turbine and Stirling engine. It was stated that the gas turbine-Stirling engine cycle had 16.1% higher exergy efficiency and 10.3% lower electricity cost in comparison with the gas turbine cycle.

Chahartaghi and Sheykhi [21] carried out energy and exergy analyses under different operating conditions using an adiabatic numerical model that was not ideal for a beta-type Stirling engine. In the study, the researchers utilized the experimental data of the beta-type Stirling engine called GPU-3, which was designed and manufactured by General Motors Corporation, for model verification. As a result of the analyses conducted, they indicated that as the operating pressure and regenerator length increased, the exergy efficiency also increased. The researchers reported that the highest exergy efficiency was obtained in the GPU-3

Stirling engine using helium gas with the regenerator length of 35 mm, at an engine speed of 1000 rpm and operating pressure of 5.52 MPa.

Ye et al. [22] conducted an exergy analysis on a one-dimensional model in a beta-type free-piston Stirling engine. The researchers used helium gas with an average charge pressure of 3.5 MPa. The initial hot and cold end temperatures of the regenerator were 923 K and 323 K, respectively. Exergy losses due to pressure drop, regenerator, friction, and heat transfer were computed. The researchers reported that appropriate performance could be obtained when the regenerator of the Stirling engine had 0.85 porosity and a length of approximately 0.04 m.

In his study, Akbari [23] examined the Stirling power cycle, consisting of a compressor, turbine, and regenerator, the integrated transcritical carbon dioxide Rankine cycle, and the liquefied natural gas (LNG) regasification process. In the study, a comprehensive review of energy, exergy, and exergoeconomic (3E) analyses was performed. Pure nitrogen was used as the working fluid in the Stirling power cycle. In this cycle, whereas LNG in the heat exchanger was the cold flow, the working fluid heated by solar energy was the hot flow. The total exergy efficiency of the process was 64.26%.

Wang et al. [24] conducted the detailed energy and exergy analyses in terms of thermodynamics on a system consisting of a thermal-driven Stirling refrigerator and a Stirling-type thermal compressor and called the Vuilleumier refrigerator. Considering the impacts of parameters, such as the operating temperature, frequency, pressure, and stroke of the engine, it was determined in the analyses that the highest exergy loss took place in Stirling-type thermal compressor. By optimizing the external losses in the system, 60% exergy efficiency was obtained in a specific Stirling-type thermal compressor that could utilize cold energy.

Chahartaghi and Sheykhi [25] performed the energy, economic, and environmental analyses of a combined cooling, heating and power generation system that was driven by a beta-type Stirling engine with a rhombic drive mechanism using hydrogen and helium gases. In the research, two beta-type Stirling engines were utilized, and analyses were performed using a non-ideal adiabatic model. Furthermore, the energy analysis of the absorption chiller accepted that the engine used the waste heat, and thermodynamic analysis was conducted. It was indicated that as the regenerator length of the engine increased, the efficiency of the COP of the absorption chiller system decreased with the waste heat energy generated by the engine. The efficiency of the system increases with the increase in the rotation speed. It was reported that the thermal efficiency and power output of the engine depended significantly on the engine speed and regenerator length in the case of using helium as the working fluid, whereas the same was not valid for hydrogen gas. According to using helium and hydrogen as the working fluid, the values obtained in the system were as follows: electrical power: 15.24 kW, 22.52 kW, efficiency: 70%, 72.29%, COP: 0.644, 0.667, respectively.

Sheykhi et al. [26] carried out the energy, exergy, economic, and environmental analyses of the absorption chiller system with two beta-type Stirling engines with a rhombic drive mechanism, electric generator, and heat recovery. The results acquired by the theoretical analysis of the thermodynamic model established were compared with conventional energy supply systems. When the rotation speed of the Stirling engine was below 2000 rpm, it was reported that it had an appropriate exergy efficiency, but as the rotation speed increased, the efficiency performance decreased due to increased friction and thermal losses.

The region where the energy loss occurs in engines and how much of this loss can be recovered are revealed by exergy analysis. Exergy losses are observed including the regenerator, working fluid, heater, and friction in Stirling engines. It is possible to calculate how much of the lost energy can be recovered to increase engine efficiency by determining these losses by thermodynamic analysis. The optimization of the parameters affecting the mechanical power obtained by Stirling engines can be performed by exergy analysis.

In the researches in the literature, it is observed that energy and exergy analyses have been conducted for some hybrid systems that usually include a Stirling engine. There are very few studies investigating the impacts of the main operating parameters, such as regenerator characteristics, engine speed, engine charge pressure, and working fluid type that are the important performance parameters of the Stirling engine, on exergy.

In this study, the exergy analysis of the performance values obtained by using helium, nitrogen, air, carbon dioxide and argon gases and an electrical heater as a heat source under laboratory conditions in a beta-type Stirling engine with a rhombic drive mechanism was conducted. Moreover, the use of two different materials, stainless steel, and titanium in the displacer piston was investigated by exergy analysis. Energy distribution, entropy generation, destroyed exergy, thermal efficiency, Carnot efficiency, and exergy efficiency were calculated using the data acquired from the Stirling engine in the experiments performed considering the different working fluids and displacer pistons. Furthermore, the equipments with the highest exergy losses were determined and improvements in the engine design were suggested in the study.

2 Material and method

2.1 Experimental setup

As shown in Fig. 1, the experimental setup consists of a beta-type Stirling engine with a rhombic drive mechanism,

a specifically designed electrical heater, a prony-type dynamometer, ESIT BB20 model load cell, ETS1410 model digital tachometer, Schneider inductive sensor, Pakkens brand Bourdon-type manometer, VFA brand M1 series turbine-type flow meter, EMKS TCS37-100 model digital counter sensor, a panel-type boxed model of Esit brand PWI-P series load measuring device, EVO mini model temperature measuring devices, Ordel brand "N" type and "PT100" type thermocouple, data logger, and infrared thermometer. In the engine tests, the engine was first operated up to the hot end temperature of 1000 K (± 10 K), at which the study would be conducted, and the engine was expected to reach a stable state. Therefore, it was aimed to reduce the fluctuations in hot end temperature during experimental studies.

Furthermore, during the usage of helium, nitrogen, carbon dioxide, and argon gases as working fluid, the mixing of the ambient air inside the engine block with the working fluid to be used was prevented. To prevent this mixing, the working fluid to be utilized in the tests was filled into the engine block up to a pressure of 1 bar, and the working fluid was discharged back to the external environment so that the pressure inside the engine block was slightly higher compared to atmospheric pressure. This procedure was repeated three times to prevent the mixing of the ambient air inside the engine block with the working fluid to be utilized. In order for the engine to stabilize itself during the dynamometer loading, it was waited for 10–15 s after loading until the next loading. The experimental studies were performed for the same type of the working fluid with charge pressures of 1, 2, 3, 4, and 5 bar, respectively.

During the experimental studies, the engine was initially operated idle without load, and the highest engine speed was obtained. Then, the brake system loading bolt was tightened slowly and gradually, and the engine was loaded. The hydraulic braking system started to squeeze the brake disc, and the lever connected to the braking system exerted force on the load cell with the help of friction. The force value formed in the load cell and the engine speed at that moment were recorded. The torque values of the engine were calculated with mathematical expressions using the acquired load values and engine speeds. The engine performance characteristics were obtained by utilizing these values acquired at different charge pressures. The technical properties of Stirling engine with rhombic mechanism, for which thermodynamic analyses have been conducted, are presented in Table 1.

2.2 Working fluids

In the nineteenth century when Stirling engines were invented, air was utilized as a working fluid due to its direct supply under atmospheric conditions. Nowadays, gases with high heat conduction capacity, low density and viscosity properties are preferred as working fluid in Stirling engines to obtain higher efficiency during the conversion of heat energy to mechanical energy.

In the experimental studies, the impacts of using helium, nitrogen, air, carbon dioxide and argon gases as the working fluid with the properties given in Table 2 in a beta-type Stirling engine with a rhombic drive mechanism shown in Fig. 1 on the engine performance characteristics were examined for

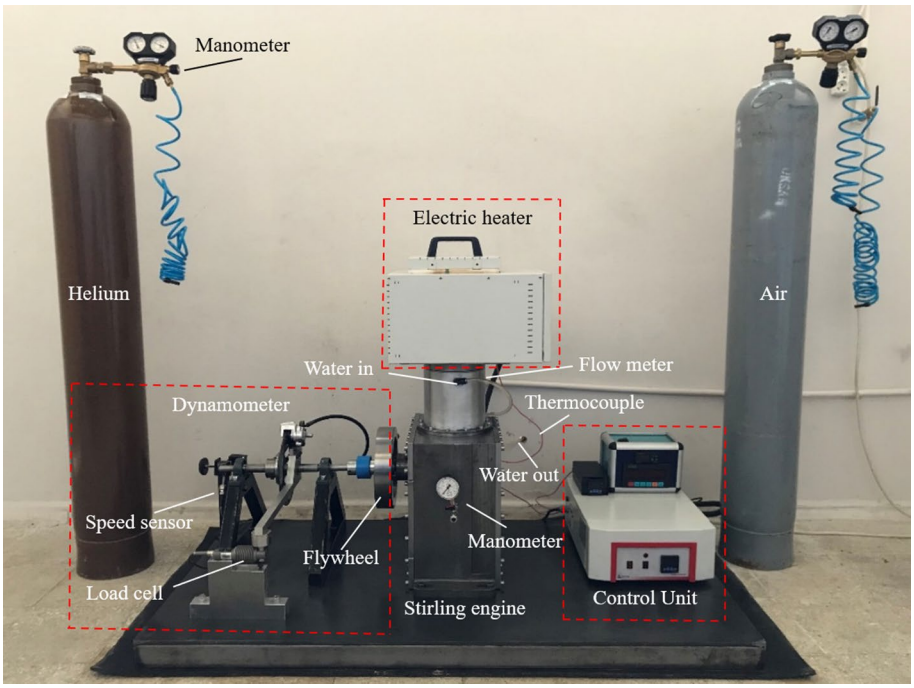


Fig. 1 Schematic layout of the experimental apparatus

Table 1 Technical properties of prototype Stirling engine with rhombic drive mechanism

Engine type	Beta
Swept volume (cm^3)	365
Displacer cylinder bore, Dd (mm)	88
Displacer piston bore (mm)	87
Displacer piston length, hd (mm)	200
Displacer piston stroke (mm)	60
Power cylinder bore, Dp (mm)	88
Power piston bore (mm)	87.97
Power piston length, hp (mm)	130
Power piston stroke (mm)	60
Cylinder length, Ht (mm)	517
Distance between gear axes, *L* (mm)	132.5
Displacer rod length, ldr (mm)	340
Displacer rod bore, Dr (mm)	16
Connecting rod length, lr (mm)	66.23
Crank radius, Rc (mm)	24.838

two different displacer pistons made of stainless steel and titanium material at charge pressures of 1, 2, 3, 4, and 5 bar.

2.3 Displacer piston

In beta-type Stirling engines, the cycle is realized by the power piston operating concentrically in the same cylinder and the displacer pistons. The cylinder volume in this type of engine is divided into two parts as hot and cold spaces by a displacer piston. The hot volume space is in the upper part of the displacer piston. In this space, the expansion process is carried out by heating the working fluid. Thus, the hot volume space is also called the expansion space. The cold volume space where compression is performed is the volume remaining between the top point of the power piston and the lower point of the displacer piston. This space is called the compression space. In an engine without a regenerator, the working fluid passes in two directions between the hot and cold volumes through the space between the displacer piston and the cylinder wall. In the present research, as seen in Fig. 2, two different displacer pistons were manufactured from AISI/SAE 303 (DIN X8CrNiS18-9 stainless steel) and Ti 6Al 4V (GR5 Titanium) materials due to their easy machinability, high corrosion resistance and mechanical properties, and their impacts on the engine performance were examined experimentally. According to the properties of the material utilized in the displacer piston, the caps made of stainless steel and titanium were welded to both open ends in a way that the working fluid would not leak inside it. The lower end of the displacer piston was specially threaded to facilitate the connection of the displacer rod.

Table 2 Thermodynamic properties of working fluids

Working fluids	Properties	Temperature (K)		
		250	500	1000
Air ($P=1$ atm) M: 28.975	C_p (kJ/kg K)	1.003	1.029	1.141
	C_v (kJ/kg K)	0.716	0.742	0.855
	ρ (kg/m^3)	1.412	0.706	0.353
	$\mu\times10^6$ (kg/ms)	15.99	26.70	41.53
	$k\times10^6$ (kW/mK)	22.27	40.41	67.54
Argon ($P=1$ atm) M: 39.95	C_p (kJ/kg K)	0.5201	0.5357	0.5892
	C_v (kJ/kg K)	0.3119	0.3213	0.3534
	ρ (kg/m^3)	1.69	0.811	0.381
	$\mu\times10^6$ (kg/ms)	19.39	34.2	55.6
	$k\times10^6$ (kW/mK)	15.15	26.8	49.3
Carbon dioxide ($P=1$ atm) M: 44.01	C_p (kJ/kg K)	0.791	1.014	1.234
	C_v (kJ/kg K)	0.605	0.786	0.957
	ρ (kg/m^3)	2.145	1.073	0.536
	$\mu\times10^6$ (kg/ms)	12.60	23.67	39.51
	$k\times10^6$ (kW/mK)	12.90	32.74	67.52
Helium ($P=1$ atm) M: 4	C_p (kJ/kg K)	5.19	5.25	5.33
	C_v (kJ/kg K)	3.1156	3.149	3.197
	ρ (kg/m^3)	0.195	0.097	0.048
	$\mu\times10^6$ (kg/ms)	18.40	29.30	46.70
	$k\times10^6$ (kW/mK)	134	202.6	405.2
Nitrogen ($P=1$ atm) M: 28.01	C_p (kJ/kg K)	1.039	1.056	1.167
	C_v (kJ/kg K)	0.742	0.759	0.870
	ρ (kg/m^3)	1.251	0.650	0.318
	$\mu\times10^6$ (kg/ms)	15.4	26.1	41.8
	$k\times10^6$ (kW/mK)	22.35	38.3	62.6



(a) Stainless steel displacer **(b)** Titanium displacer

Fig. 2 Pictorial view of displacer pistons with stainless steel and titanium material

3 Analyses

The energy or exergy criteria are commonly utilized in the investigation and evaluation of energy conversion systems. Exergy analysis reveals the work potential and quality of the energy and enables the determination of the amount, types, and places of the unusable part of the energy and the losses caused by irreversibilities. Therefore, the losses in the system and the efficiency of the system can be determined in detail, the parts operating with low efficiency can be determined and improved, and different systems can be compared with each other. From this aspect, exergy analysis is very useful for the efficient use of energy resources, the design of more efficient energy conversion systems, and the improvement of the performance of an existing system. In the analyses conducted in the present research, as seen in Fig. 3, the exergetic performance of a beta-type Stirling engine with rhombic drive mechanism was investigated.

In the study, the following assumptions were made when performing the thermodynamic analysis.

- The mass of the working fluid utilized throughout the cycle is constant.
- Working fluids are perfect gases.
- The thermophysical properties of working fluids are constant.
- Heating and cooling are performed instantly in a constant volume.
- Pumping and flow losses are neglected.
- The cylinder wall and piston temperatures are constant.

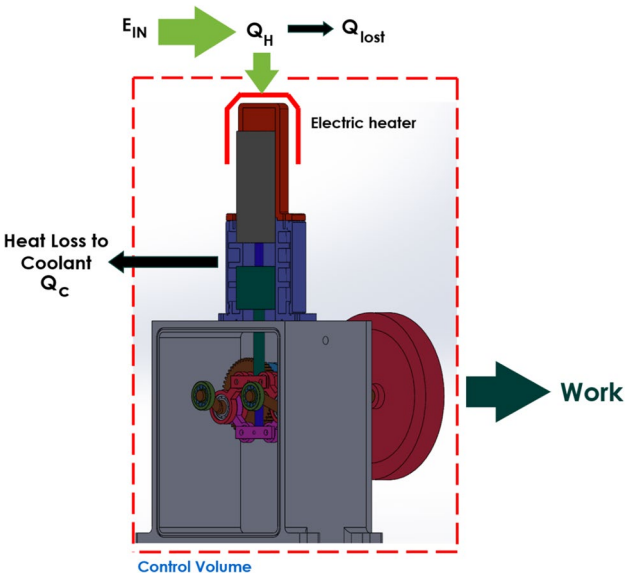


Fig. 3 Schematic view of the applied beta-type Stirling engine as an open thermodynamic system with input and output terms

3.1 Energy analysis

The energy analysis in Stirling engines is performed according to the first law of thermodynamics. In this study, the energy entering the engine is provided by the electrical heater, which is an external heat source. The heat entering the system is calculated using Eq. (1). Here, V is the voltage, I is the current, and dt indicates the cycle time [27].

$$E_{\text{in}} = V \cdot I \cdot \text{dt} \tag{1}$$

The energy balance in the engine is calculated using Eq. (2). Here, Q_H indicates the heat received by the working fluid, Q_C indicates the heat amount of the cooling water, and W indicates the work taken from the engine. The heat amount of the cooling water can be determined using Eq. (3). In this equation, $\dot{m}$, c_p, ΔT and dt are the mass flow rate of the cooling water, heat capacity, temperature difference, and cycle time, respectively [28].

$$W = Q_{\text{H}} - Q_{\text{C}} \tag{2}$$

$$Q_{\text{C}} = \dot{m} c_{\text{p}} \Delta T \cdot \text{dt} \tag{3}$$

While the part of the energy given to the Stirling engine is taken by the working fluid, the other part is given to the environment, causing a certain part of the energy to be unused. Increasing the amount of heat lost that is calculated using Eq. (4) adversely affects the engine efficiency. Hence, insulation and design measures must be taken to reduce the amount Q_{lost} in the design of the Stirling engine receiving heat from an external source [29].

$$Q_{\text{lost}} = E_{\text{in}} - Q_{\text{H}} \tag{4}$$

According to the 2nd law of thermodynamics, the presence of irreversibilities reduces the efficiency of Stirling engines. Entropy generation is a measure of the irreversibilities observed during the change of state. Entropy generation is also used as a criterion to reveal the efficiency of Stirling engines. Entropy balance is presented in Eq. (5). Since the Stirling engine represents a closed system and does not change over time, entropy generation can be found using Eq. (6). Here, T_H and T_C denote the hot and cold end temperatures, respectively [14].

$$S_{\text{in}} - S_{\text{out}} + S_{\text{gen}} = \Delta S_{\text{sys}} \tag{5}$$

$$\frac{Q_{\text{H}}}{T_{\text{H}}} - \frac{Q_{\text{C}}}{T_{\text{C}}} + S_{\text{gen}} = 0 \tag{6}$$

The ratio of the heat given from an external source to work in Stirling engines is calculated as thermal efficiency using Eq. (7). The Carnot efficiency in these engines can

be found from Eq. (8) using the cold and hot end temperatures [30].

$$\eta_{\text{th}} = \frac{W}{Q_{\text{H}}} \tag{7}$$

$$\eta_{\text{Carnot}} = \frac{T_{\text{H}} - T_{\text{L}}}{T_{\text{H}}} = 1 - \frac{T_{\text{L}}}{T_{\text{H}}} \tag{8}$$

3.2 Exergy analysis

The exergy balance of the Stirling engine operating in the closed cycle is presented in Eq. (9). The exergy transfer flow (Ex_{heat}), exergetic power (W), and exergy destruction (Ex_{dest}) formed with heat transfer are used in the exergy calculations of the engine. Heat and other losses from the engine to the environment along the cooling water in the engine are also included in the exergy analysis [31].

$$\Delta\text{Ex} = \sum \text{Ex}_{\text{in}} - \sum \text{Ex}_{\text{out}} - \text{Ex}_{\text{dest}} \tag{9}$$

$$\text{Ex}_{\text{w}} = W \tag{10}$$

The exergy transfer formed with heat transfer can be calculated using Eqs. (11) and (12). Here, T_0 is taken as the ambient temperature [32].

$$\text{Ex}_{\text{heat}} = Q_{\text{H}}\left(1 - \frac{T_0}{T_{\text{H}}}\right) \tag{11}$$

$$\text{Ex}_{\text{heat}} = W + \text{Ex}_{\text{dest}} \tag{12}$$

As given in Eq. (13), exergy efficiency is the ratio of the energy given by the power obtained from the engine. Exergetic efficiency is a measure of the decrease in the exergy of the energy that is transferred to the engine [33].

$$\eta_{\text{exergy}} = \frac{W}{\text{Ex}_{\text{in}}} \tag{13}$$

3.3 Error and uncertainty analysis

Although careful studies are carried out during the tests, some unexpected errors occur in the results obtained. In this study, the accuracy of the measuring devices used in experimental studies using uncertainty analysis was determined. In this analysis method, if the desired size to be calculated is R and n independent variables ($x_1, x_2, x_3, \ldots x_n$) that affect this size are expressed as $R = (x_1, x_2, x_3, \ldots x_n)$. The error amount of each argument is ($W_1, W_2, W_3, \ldots W_n$) and the error amount of R is calculated from W_R Eq. (14) [36]. Measuring instruments used in experimental studies and calculated. The uncertainties of the sizes are given in Table 3.

$$w_{\text{R}} = \left[\left(\frac{\partial R}{\partial x_1} w_1\right)^2 + \left(\frac{\partial R}{\partial x_2} w_2\right)^2 + \cdots + \left(\frac{\partial R}{\partial x_n} w_n\right)^2\right]^{1/2} \tag{14}$$

4 Results and discussion

In this research, two different displacer pistons made of stainless steel and titanium material were used in a beta-type Stirling engine. The performance tests of helium, nitrogen, air, carbon dioxide and argon gases as the working fluid were conducted depending on the engine speed at 1000 K (±10 K) hot end and 300 K (±5 K) cold end temperatures using the specifically designed electrical heater. In the said tests, charging pressures were changed at 1-bar intervals between 1 and 5 bar charge pressures for each working fluid. All tests were repeated in both the stainless steel and titanium displacer piston.

Due to being an external combustion engine, Stirling engines operate with different energy sources. To obtain more power and torque from the heat energy given in these engines, the energy must be transferred to the working fluid faster. The physical, chemical, and thermal properties of the working fluid utilized in Stirling engines are important for mechanical power generation.

The type of the working fluid used and charge pressure influence the power of Stirling engines. Because of the

Table 3 Measuring instruments and calculated uncertainties

	Parameters	Unit	Measuring instruments	Uncertainty value
Uncertainties of measuring instruments	Engine speed	rpm	Speed sensor	±1
	Temperature	℃	Thermocouple	±0.1
	Pressure	Bar	Manometer	±0.25
	Mass flow rate	l/d	Flow meter	±0.1
	Weight	kg	Load cell	±0.001
Uncertainties of calculated	Engine torque	Nm	3,741	±0.0025
	Engine power	W	215,48	±0.0185

insufficient working fluid in the system at low charge pressures in the tests, the maximum power could not be obtained, as given in Fig. 4. In the experiments performed by using air, carbon dioxide, helium, and nitrogen gases as the working fluid with a stainless steel and titanium displacer piston in the test engine, the maximum engine power occurred at a charge pressure of 4 bar. However, in the experiments conducted with argon gas and titanium displacer piston, it was determined that the maximum engine power was generated at a charge pressure of 3 bar. The power obtained from the Stirling engine varies depending on the engine speed and engine torque. The decrease in the heat exchange time and the increase in friction losses due to the increase in the engine speed cause the engine power to decrease.

As seen in Fig. 5, the engine torque values are low due to the insufficient amount of the working fluid in the system at low charge pressures. Another reason for the decrease in the engine torque values with the increasing charge pressure is the high friction that emerges as a result of the contact of the increased amount of the working fluid in the cylinder with the cylinder walls and piston surfaces. As the engine speed increases at high pressures, friction, heat leakage, and mechanical losses become more effective. These losses decrease engine power.

As can be seen in Fig. 6 and Table 4, the maximum power values acquired from different working fluids differ according to the charge pressure, engine speed, and the material of the displacer piston. The stainless steel displacer piston performed better compared to the titanium one in all working fluids, except for nitrogen. As seen in Table 4, in the experiments performed with the stainless steel and titanium displacer piston and by using air, carbon dioxide, helium, and nitrogen gases, the maximum engine power output occurred at a charge pressure of 4 bar. However, in the experiments performed with argon gas and titanium displacer piston, it was revealed that the maximum engine power occurred at a charge pressure of 3 bar.

By conducting the energy analysis, it was determined how much of the energy given to the Stirling engine from an external source was converted into work. Heat losses are high due to the insufficient working fluid and incomplete heat transfer at low pressure and speed values. With the increasing speed in the engine, the heat transfer time decreases and the energy transferred to the working fluid decreases. Moreover, the amount of heat transferred to the cooling water in the engine was found.

The charge pressure at which the maximum power outputs are obtained varies in each working fluid. In the study, energy distributions were found by considering the maximum engine power values obtained from different working fluids. These energy distributions are presented according to the different material of the displacer piston in Figs. 7 and 8. The furnace, which utilizes electrical energy as a heat source in the test engine, is the energy that enters the system. In the research, the energy lost from the cooling water was calculated, and friction and heat leaks (heat energy lost from the heating furnace and the outer surface of the engine) were identified as other losses. It is observed in the tests that the working fluids reached the maximum power at different pressure and different speed values. The best mechanical

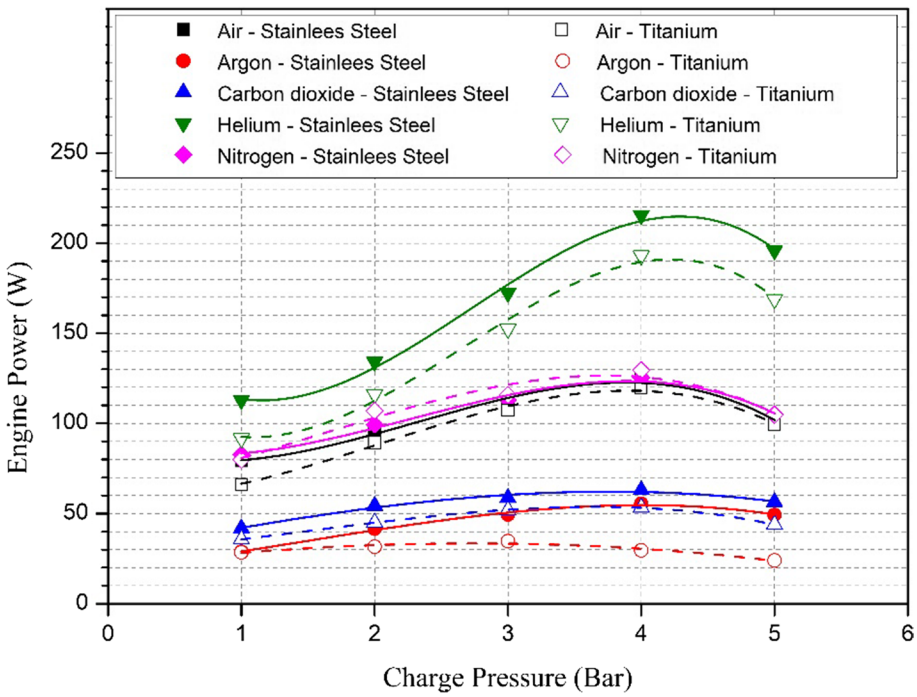


Fig. 4 The comparison of maximum engine powers depending on charge pressure

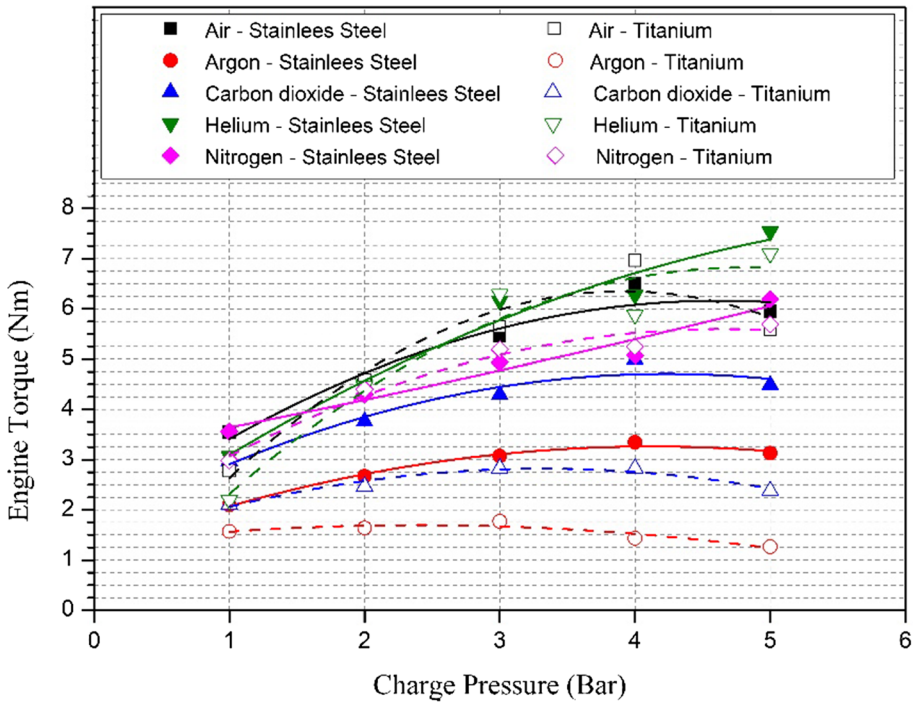


Fig. 5 The comparison of maximum engine torques depending on charge pressure

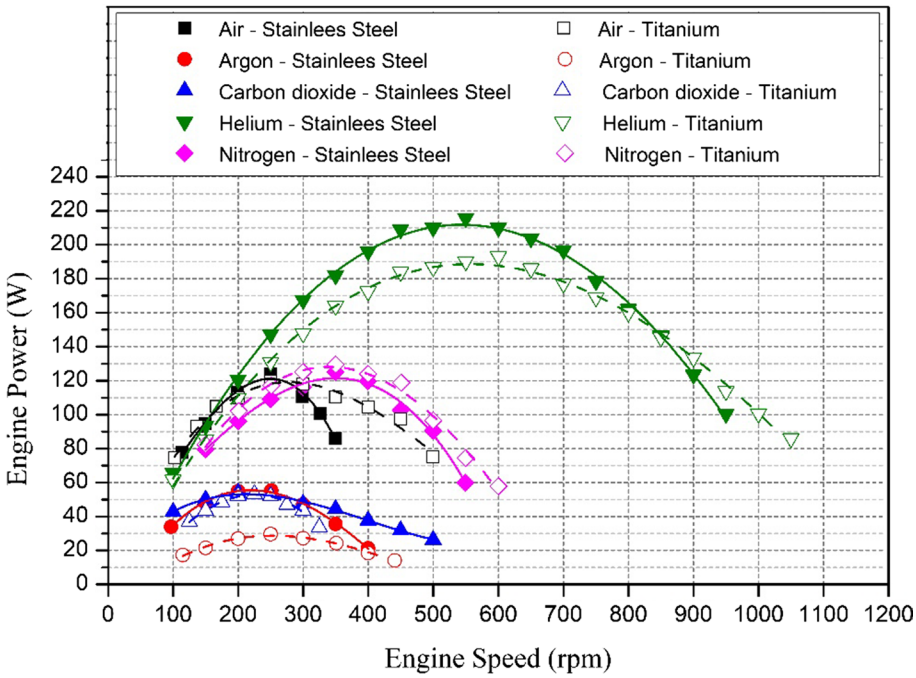


Fig. 6 The comparison of engine powers depending on different engine speeds at 1000 K heater temperature (4 bar charge pressure)

and thermal performance is achieved when helium gas is used as the working fluid in both displacer pistons. In the studies performed with Stirling engines in the literature [34, 35], hydrogen gas is observed to be the only working fluid that works more efficiently than helium gas. Although the performance of hydrogen gas in Stirling engines is good, it was not preferred in the current research due to its explosion limit being in a wide scale of 4.1–74.2%.

Because the current, voltage, and cycle time are effective in the calculations of the amount of energy entering the system, the values of the working fluids at a charge pressure of 3–4 bar are close to each other. The viscosity, density, and thermal conductivity properties of working fluids are important in terms of heat transfer between hot and cold ends. In the case of using the stainless steel displacer piston and helium gas when the maximum power output was acquired,

Table 4 The comparison of maximum engine powers depending on displacer piston material

Working fluid	Displacer piston with stainless steel material			Displacer piston with titanium material		
	Charge pressure (bar)	Engine speed (rpm)	Max. power (W)	Charge pressure (bar)	Engine speed (rpm)	Max. power (W)
Helium	4	550	215.48	4	600	193.12
Nitrogen	4	350	125.19	4	350	129.52
Air	4	250	124.06	4	250	119.58
Carbon dioxide	4	250	63.1	4	225	53.3
Argon	4	250	55.42	3	300	34.66

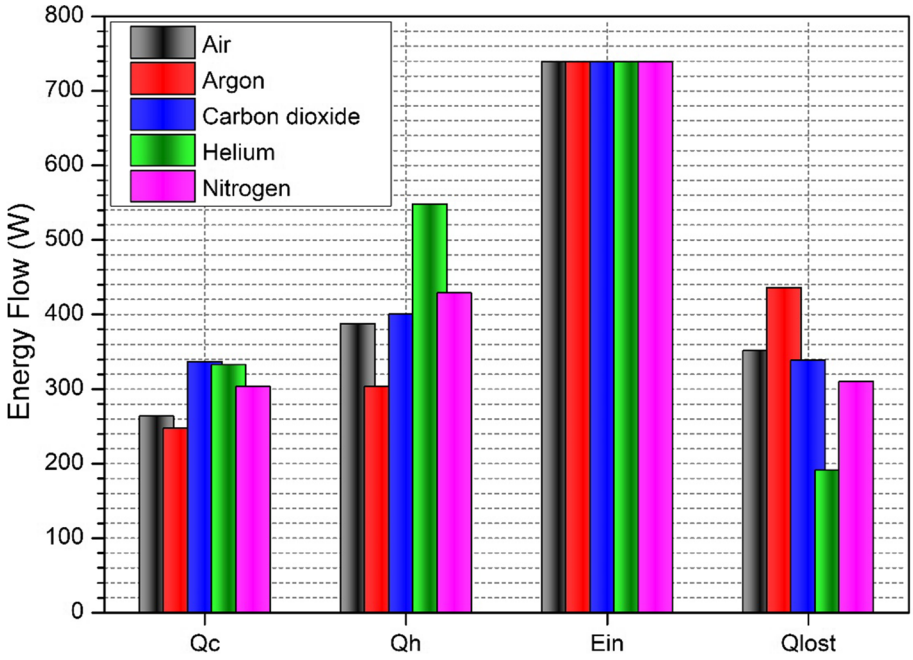


Fig. 7 The variation of energy distributions in using displacer piston with stainless steel material

the heat entering the system was calculated to be 739.2 W, the heat received by the working fluid was 548.3 W, the heat lost to the environment was 190.9 W, and the heat spent on cooling was calculated to be 332.818 W. The highest heat loss occurred in both displacer pistons when the argon working fluid was utilized.

Since Stirling engines operate according to the hot and cold end temperature difference, this temperature difference should be high in order to increase performance and efficiency. Necessary precautions must be taken at the engine design stage to reduce losses in the amount of heat transferred to the working fluid from the hot source.

Exergy losses occurring due to events such as heat transfer, flow, and friction under various operating conditions of the test engine were determined. In the tests, the engine power increases up to a certain value depending on the increase in the engine speed and then decreases. In Stirling engines, charge pressure for operating is provided from the crankcase, and counter-force is formed in the lower part of the power piston. This counter-force has an adverse effect on power. At high pressures, as the engine speed increases, friction, heat leakage, and mechanical losses become more effective. These losses reduce the power of the engine. Moreover, as the engine speed increases, the heating and cooling times required for the working fluid shorten, and engine power values decrease. Another reason for the decrease in engine power values with the increasing charge pressure is high friction occurring as a result of the contact of the increased amount of the working fluid inside the cylinder with the cylinder walls and piston surfaces. These losses reduce the power of the engine.

Figures 9 and 10 show the exergy calculated by considering the situation when the highest power outputs of the working fluids according to the charge pressure and engine speed were obtained and destroyed by the system. The exergy destroyed represents the lost work potential. The

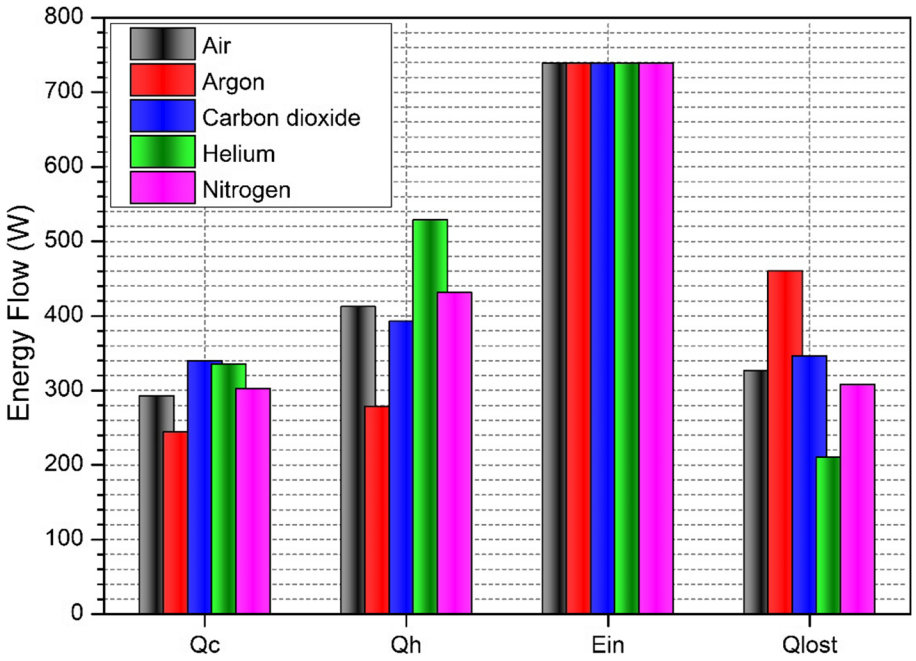


Fig. 8 The variation of energy distributions in using displacer piston with titanium material

amount of the exergy destroyed in all working fluids at low engine speeds is considerably high. As given in Table 2, the working fluids reach the maximum power at various charge pressures and engine speeds. Under the conditions when this maximum power is reached, the exergy destroyed takes the lowest value in all working fluids. The exergy destroyed increases when moving away from the conditions (pressure and engine speed) of this maximum power value. When helium gas was used as the working fluid and charge pressure was 4 bar, the amount of the exergy destroyed was calculated to be 195.53 W at 550 rpm for the stainless steel displacer piston.

Since nitrogen and air working fluids reach their maximum power values when the charge pressure is 4 bar, as in helium gas, and at lower speeds compared to helium, the amount of the exergy destroyed is lower. When high engine

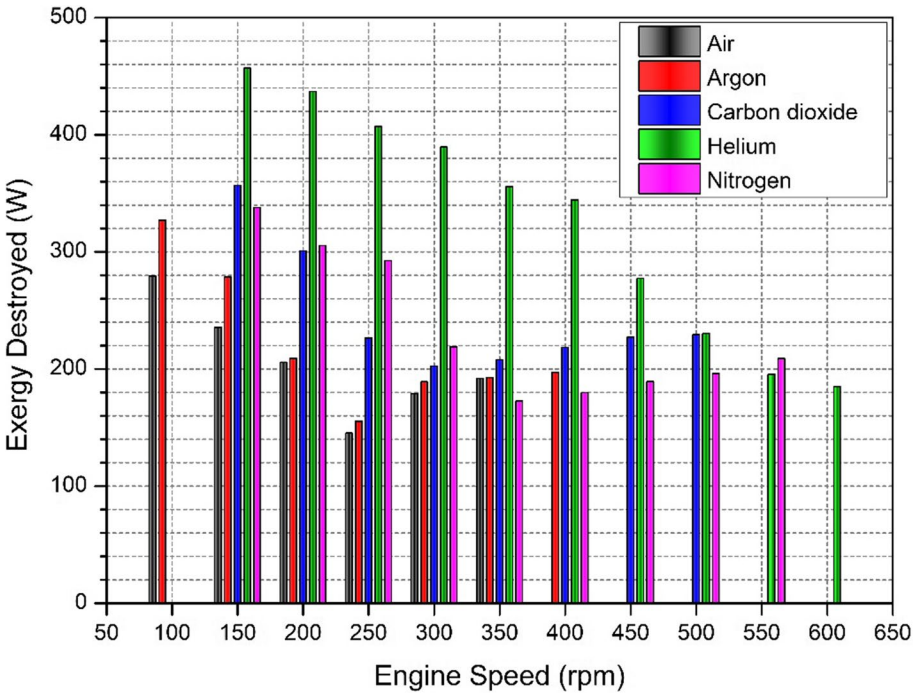


Fig. 9 The variation of exergy destroyed in using displacer piston with stainless steel material

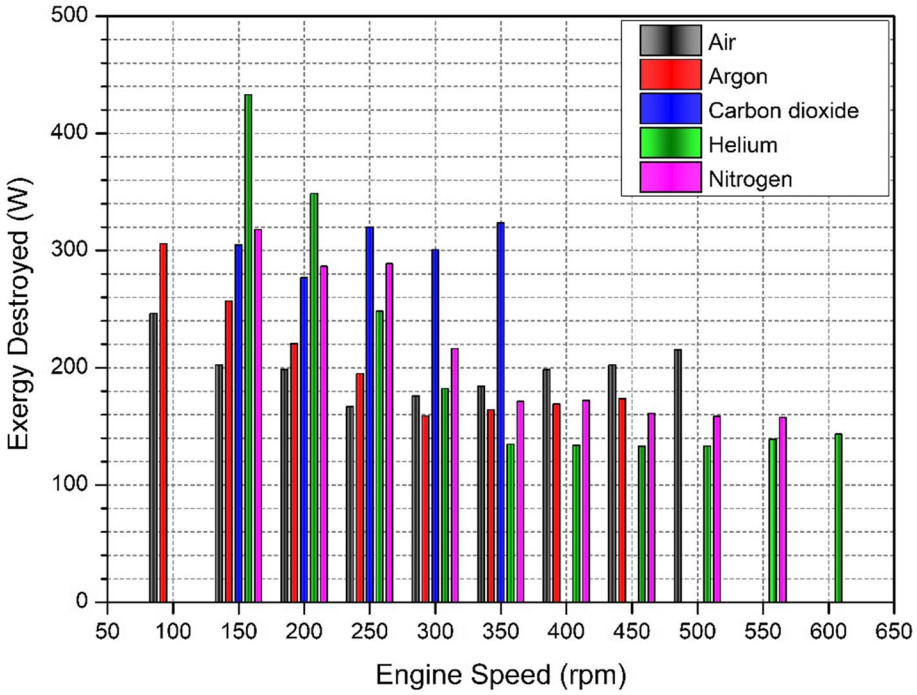


Fig. 10 The variation of exergy destroyed in using displacer piston with titanium material

speeds and high charge pressures are desired in Stirling engines, exergy is an important parameter in terms of the selection of a working fluid.

Furthermore, as seen in Fig. 6, the types of the working fluid to be preferred in Stirling engines have a significant effect on the engine operating speed ranges. While high engine speeds can be reached in some working fluids, they could not have been reached at all in other working fluids. Therefore, exergy analysis calculations could not be made for the relevant working fluid at some engine speeds.

The irreversibilities occurring in Stirling engines that operate with heat energy always produce entropy. All machines that produce entropy always destroy exergy. Figures 11 and 12 demonstrate the entropy changes produced by the system and calculated by considering the material of the displacer piston at which the highest power outputs of the working fluids were obtained according to the charge pressure and engine speed. In the material of both displacer pistons, helium appeared to be the working fluid with the lowest entropy generation and the highest power generation. In all tests in the study, the entropy generation decreases until the engine speeds at which the maximum power is obtained, and increases after the value at which the maximum power is obtained. The output power generated in the engine varies according to the working fluid type and charge pressure. The parameters of heat received by the working fluid (Q_H), the amount of heat lost by cooling water (Q_C), and the generated output power (W) affect entropy generation. The losses occurring in the engine lead to entropy generation. The amount of heat lost by cooling water (Q_C) is an important parameter in entropy generation. In the study, entropy generation at 3–4 bar charge pressure of the working fluids in the titanium displacer piston is presented in Fig. 12. In the experiments, while 193.2 W output power was generated at 600 rpm when nitrogen gas was used as the working fluid at a charge pressure of 4 bar, 119.58 W output power was generated at 250 rpm when air was used at the same charge pressure, which affects entropy generation.

Table 5 presents exergy flows at the engine speeds at which the highest engine power is obtained at charge pressure of 1–5 bar when different working fluids and displacer pistons are used in the test engine. In the conversion of thermal energy at high temperature into work, the exergy flow expresses the quality as well as the quantity of energy. Stirling engines operate between hot and cold end temperature differences. The reference temperature is the most important factor in the calculation of energy losses in these engines. Working fluids reach the maximum power at different engine speeds. According to the second law of thermodynamics, work is more valuable than heat because work is completely converted into heat, while heat cannot be converted into work completely and continuously. The maximum amount of useful work that can be obtained from a system by reducing it to the environmental conditions is a feature of the system and the environment, and it is defined as exergy. The conversion of the energy given to the engine into useful work is determined by the exergy flow. The highest exergy flow performance was obtained when helium gas was utilized as the working fluid in the stainless steel displacer piston and at a charge pressure of 4 bar.

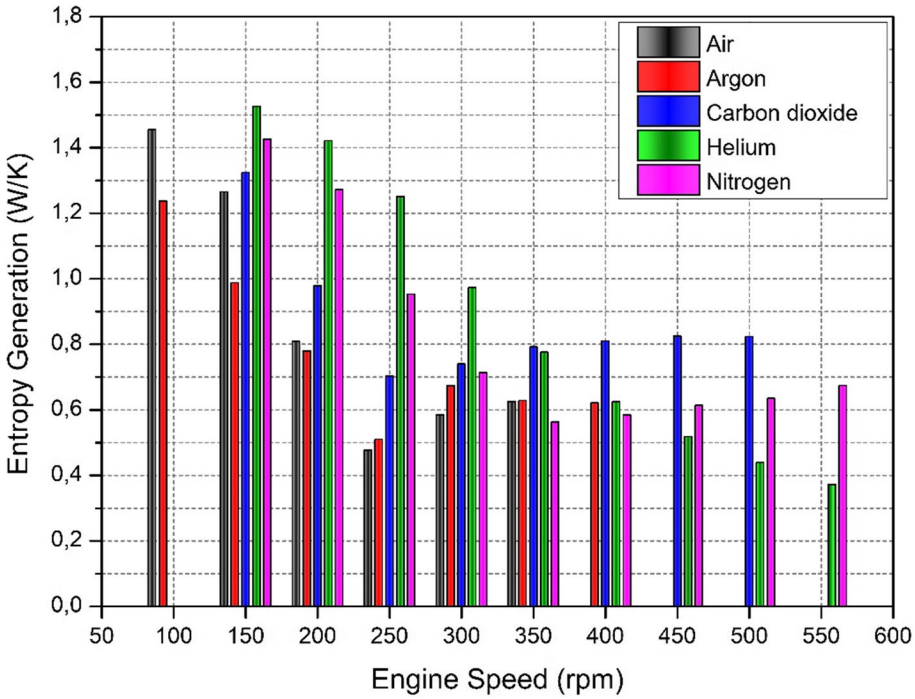


Fig. 11 The variation of entropy generation in using displacer piston with stainless steel material

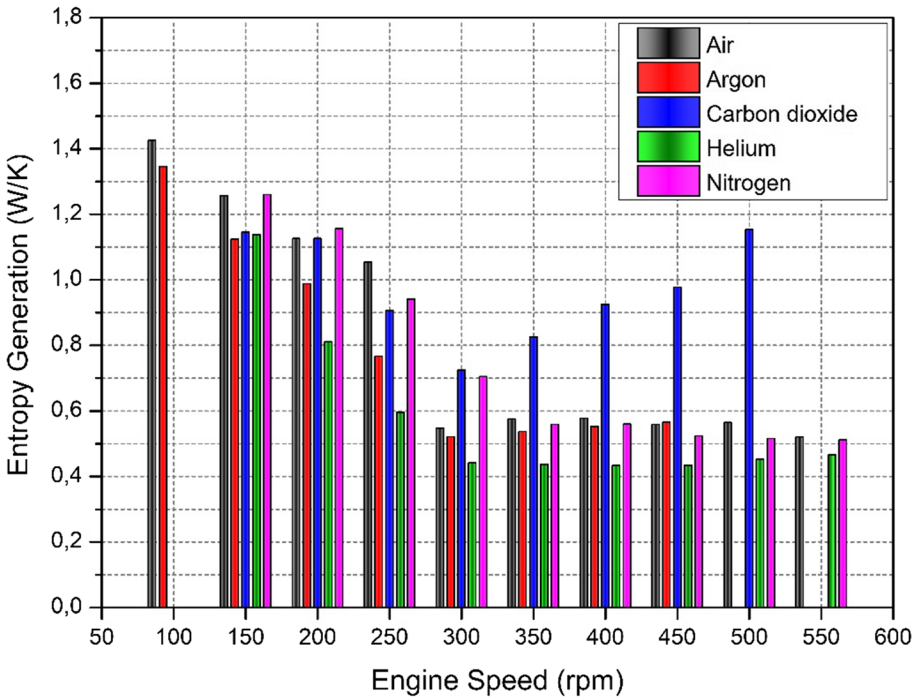


Fig. 12 The variation of entropy generation in using displacer piston with titanium material

Exergy efficiency is defined as the ratio of the useful exergy output to the spent exergy input. The exergy efficiencies calculated according to the charge pressure at which the maximum power is obtained according to the data acquired from the test engine are presented in Figs. 13 and 14. As seen in Table 4, in the experiments performed by using air, carbon dioxide, helium and nitrogen gases, while the maximum engine power occurred at a charge pressure of 4 bar, it occurred at a charge pressure of 3 bar in the experiments in which argon gas was used. Exergy efficiency increased when irreversibility decreased in thermodynamic calculations made at different charge pressures. When helium was used as the working fluid, the exergy efficiency of 56.54% was obtained in the stainless steel displacer piston

Table 5 The comparison of exergy flows depending on displacer piston material

Charge pressure (bar)	Air		Argon		Carbon dioxide		Helium		Nitrogen	
	Engine speed (rpm)	Exqh (W)	Engine speed (rpm)	Exqh (W)	Engine speed (rpm)	Exqh (W)	Engine speed (rpm)	Exqh (W)	Engine speed (rpm)	Exqh (W)
Displacer piston with stainless steel material										
1	300	223.526	200	274.515	300	199.845	550	337.497	350	286.219
2	300	234.84	200	283.343	300	249.849	500	352.760	300	323.374
3	300	263.701	250	185.523	300	261.354	550	368.516	350	318.273
4	250	269.451	250	210.82	300	278.066	550	381.112	350	297.866
5	250	272.457	200	288.732	250	328.245	400	414.094	250	371.827
Displacer piston with titanium material										
1	350	231.456	250	142.547	300	192.146	350	299.933	350	237.41
2	400	243.654	300	165.215	300	242.738	400	316.851	450	293.951
3	350	272.874	300	193.746	250	280.381	400	332.456	350	320.417
4	250	286.69	300	239.488	250	272.851	350	367.521	350	299.649
5	250	271.038	250	261.961	200	323.118	300	375.125	300	349.933

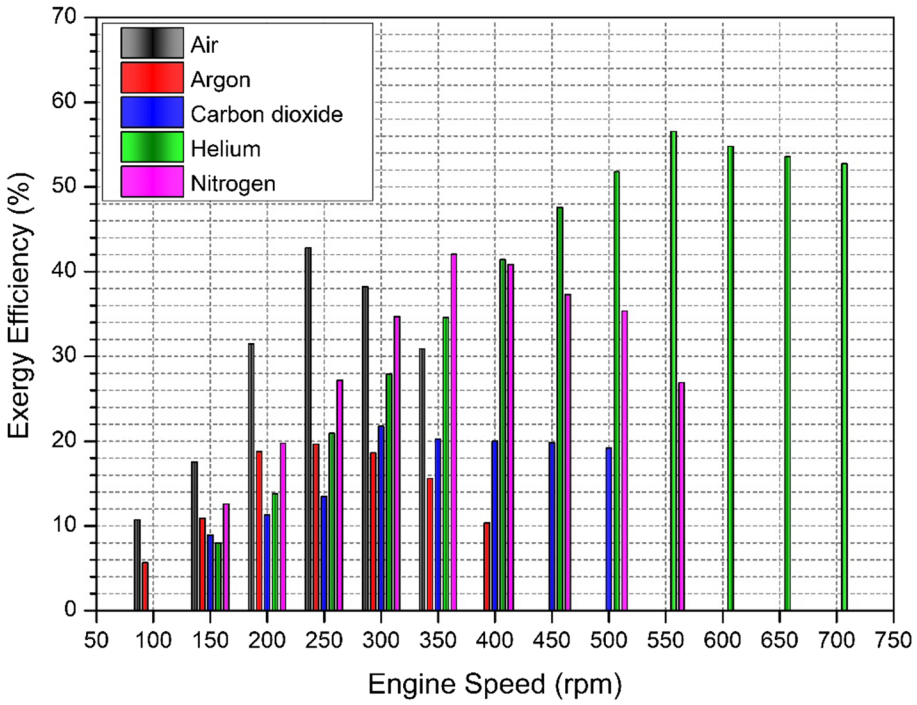


Fig. 13 The variation of exergy efficiency in using displacer piston with stainless steel material

at a charge pressure of 4 bar. The lowest exergy efficiency is 2.13% at an engine speed of 100 rpm when argon is used in the titanium displacer piston. The engine power output acquired under these conditions is 17.33 W.

The working fluid inside the Stirling engine continuously gains heat from the hot source or loses heat to the cold source by moving back and forth at a high speed between the hot and cold volumes, and as a result of the repetition of these processes, the engine power is generated. Some of the energy taken from an external source in these engines cannot be used due to losses. In experimental studies, the thermal efficiency of the working fluids at the charge pressures at which the maximum engine power output values were acquired in the tests conducted with the stainless steel and titanium displacer pistons was calculated and given according to the material of the displacer piston in Figs. 15 and 16. The high thermal efficiency depends on the power obtained during the cycle. The thermal efficiency reached the highest value when helium was used in the test engine. Thermal efficiencies are obtained to be close to each other in case of

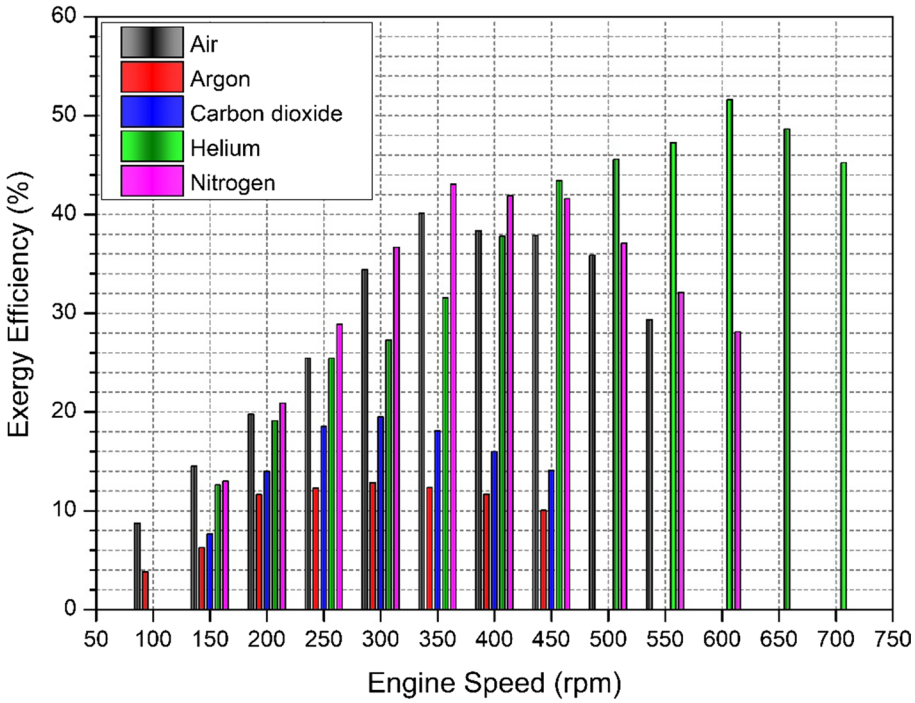


Fig. 14 The variation of exergy efficiency in using displacer piston with titanium material

using nitrogen and air. When evaluated in terms of thermal efficiency in Stirling engines, air may be preferred instead of nitrogen since it is easier to supply. However, when nitrogen gas is utilized as the working fluid, the engine speed operating range is better compared to air. When argon and carbon dioxide gases are used as the working fluid, the thermal efficiency decreases significantly. When calculating the thermal efficiency, the heat (Q_H) taken by the working fluid is decisive. In engine tests, hot-end and cold-end temperatures varied between approximately 990–1010 K and 295–305 K, respectively, depending on operating conditions.

During the tests, cooling water inlet and outlet temperatures, cooling water flow rate and engine power were measured. The amount of heat lost by cooling water was

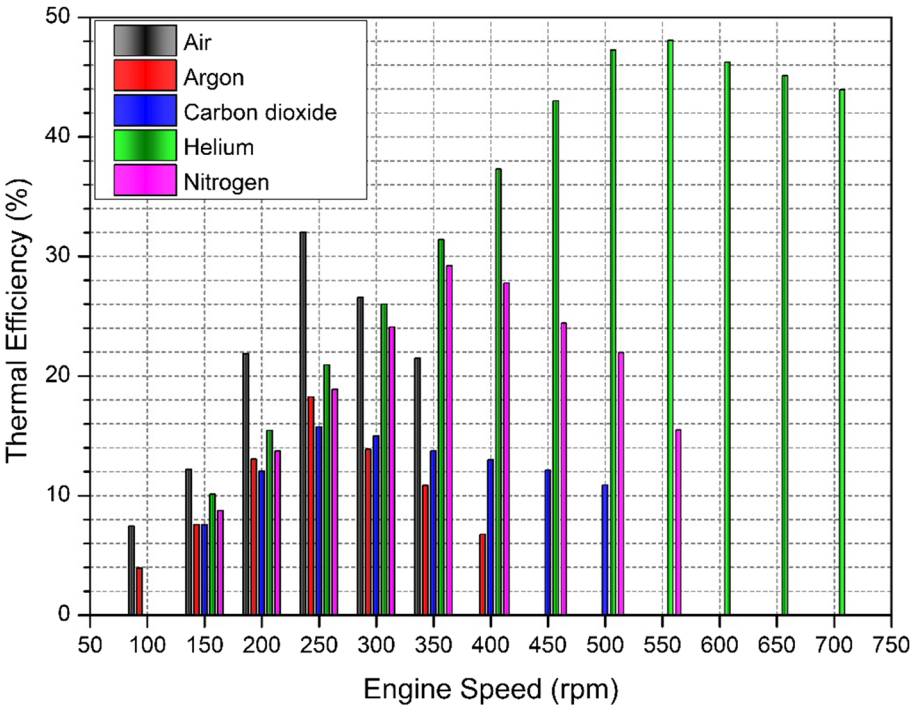


Fig. 15 The variation of thermal efficiency in using displacer piston with stainless steel material

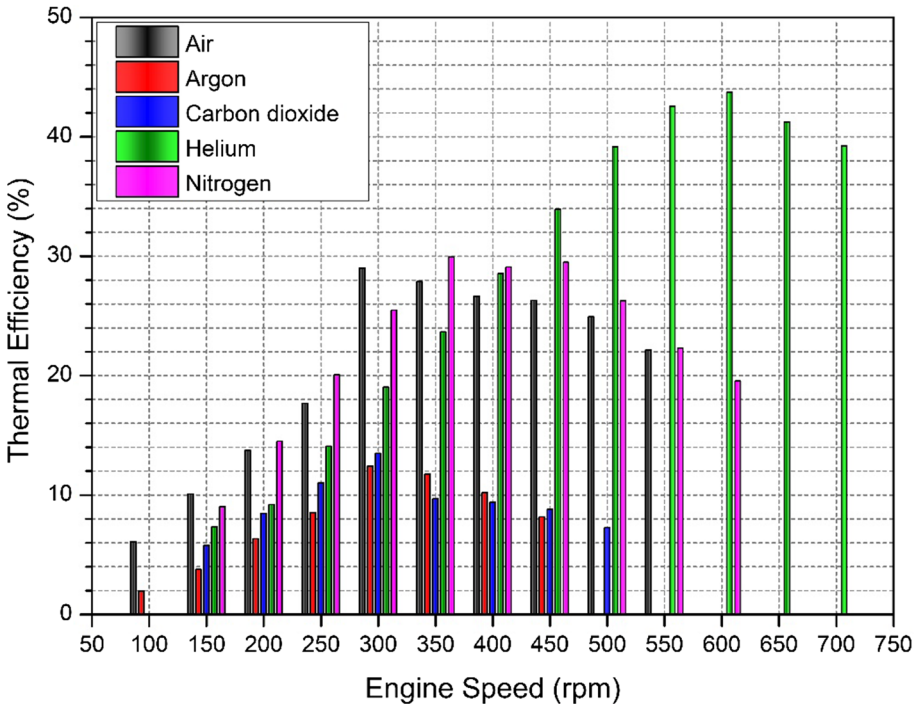


Fig. 16 The variation of thermal efficiency in using displacer piston with titanium material

calculated. The amount of heat received by the working fluid was determined using the amount of heat lost by cooling water and engine power. Thermal efficiency was found using Eq. (7). The Carnot cycle has the highest efficiency according to the expression given in Eq. (8) among heat engines operating between the same heat sources temperature limits. The Carnot efficiency is accepted as the maximum thermal efficiency in Stirling engines. This efficiency increases with the increasing difference between the hot and cold end temperatures according to the Carnot cycle. The efficiency of the theoretical Stirling cycle is equal to the efficiency of the Carnot cycle. At the same time, the theoretical efficiency of this type engines does not depend on the type of gases, but it depends on temperatures of hot source from which heat is taken and temperatures of cold sources to which the heat is given. The Carnot, thermal, and exergy efficiencies at different charge pressures were calculated for each working fluid in this study. Since the maximum engine power in all experimental studies exhibited the best performance at a charge pressure of 4 bar when the stainless steel displacer piston and helium gas as the working fluid were utilized, efficiencies under these conditions are shown in Fig. 17.

5 Conclusions

In the present research, the performance tests of a beta-type Stirling engine with a rhombic drive mechanism and a swept volume of 365 cm^3 were conducted. Five different working fluids, including helium, nitrogen, air, carbon dioxide and argon were used in the experiments. Furthermore, the effect of the displacer piston made of stainless steel and titanium material on the engine performance was investigated. The tests were carried out in 5 stages at a charge pressure varying between 1 and 5 bar for all working fluids and different displacer pistons.

- It was observed that the thermal conductivity performance of the working fluids affects the power output of the engine positively and negatively.
- In the tests carried out in the study, the highest power output was obtained to be 215.48 W in case of using helium as the working fluid at a pressure of 4 bar and an engine speed of 550 rpm in the stainless steel displacer piston.
- When the titanium displacer piston was utilized in the engine, the highest power output of 193.12 W was obtained at a pressure of 4 bar and a speed of 600 rpm when helium gas was used as the working fluid.
- In the exergy flow, which reveals the quality and the quantity of the engine power output, it was found to be 381.112 W at 550 rpm when helium was used as the working fluid at a pressure of 4 bar in the stainless steel displacer piston.
- The increase in exergy efficiency is provided by the decrease in irreversibilities. The maximum exergy efficiency value was obtained to be 56.54% at a charge pressure of 4 bar in the stainless steel displacer piston when helium was used as the working fluid at which the highest power was obtained in all tests.

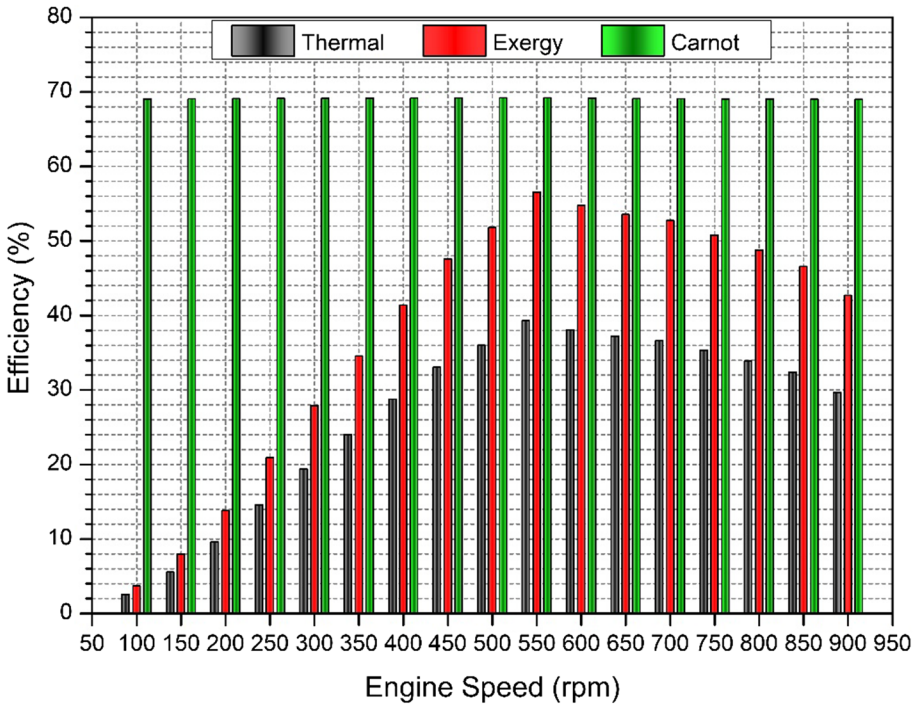


Fig. 17 The variation of thermal, Carnot and exergy efficiencies in using displacer piston with stainless steel material and helium working fluid

- The lowest exergy efficiency was obtained to be 2.13% when argon was used as the working fluid at a charge pressure of 1 bar and an engine speed of 100 rpm in the titanium displacer piston.
- The increase in the charge pressure in Stirling engines causes more working fluid to be used. In return to the increasing mass amount, the heat energy transferred to the working fluid decreases. Moreover, the increase in pressure in the engine also increases the losses.
- The highest thermal efficiency was obtained to be 48.04% in the helium working fluid under the highest power output conditions. The decrease in the loss of the amount of heat transferred to the engine body and working fluid in the heater will reduce entropy generation in Stirling engines.
- In all tests in the study, entropy generation decreases up to the pressure and engine speed at which the maximum power is obtained and increases after the value at which this maximum power is obtained. In the stainless steel material displacer piston, the entropy produced at a charge pressure of 4 bar and engine speed of 550 rpm is 0.3726 W/K when helium gas is used as the working fluid.
- Exergy losses, including from the working fluid, heater, and friction, are observed in Stirling engine. Exergy losses are high at low charge pressures and low engine speeds. All working fluids performed the best at charge pressures of 3–4 in both displacer pistons used in the research. It was observed that the exergy losses were low under conditions with the increased engine power.
- The thermal, exergy and Carnot efficiencies were calculated under different pressure conditions for all working fluids and displacer pistons. In the thermal, exergy and Carnot efficiency calculations performed using the values obtained from the tests, when helium was used as the working fluid at a charge pressure of 4 bar in the stainless steel displacer piston, 48.06%, 56.54%, and 69.2% were obtained, respectively.
- It is suggested to investigate the impacts of different working fluids (especially hydrogen) and displacer pistons and cylinders made of different materials by using the energy and exergy analysis methods in the following periods. The advantages of Stirling engines over internal combustion engines can be explained in more detail by conducting exergoeconomic and environmental analyses.

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