HYDROGEN, FUEL CELL & ENERGY STORAGE

Thermoeconomic Study of a Multigeneration System with Waste Heat Recovery from a Gas Turbine

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Abstract

The growing demand for energy and the associated crisis present a major global challenge. As a result, there is an increasing shift towards the adoption of renewable energy sources. One effective strategy to reduce fossil fuel consumption is to capture heat from the exhaust gases produced by gas turbines. This approach not only reduces fuel consumption but also lowers carbon dioxide emissions, contributing to environmental protection. A study was conducted to evaluate the energy, exergy, and exergoeconomic aspects through a simulation of a combined cycle system, which integrated a steam Rankine cycle (SRC) and an organic Rankine cycle (ORC) to utilize waste heat from gas turbines. Freshwater and hydrogen were produced using a reverse osmosis (RO) unit and a proton exchange membrane (PEM) electrolyzer. The impact of varying system parameters on performance factors was thoroughly investigated. System analysis was conducted using EES software. The results showed that the initial energy efficiency and overall exergy values were 24.92% and 15.01%, respectively, with an output work of 320,412.8 kW. Additionally, replacing the SRC and ORC condensers with two thermoelectric generator (TEG) units resulted in an additional power output of 5,046.8 kW. Exergy destruction rate assessments indicated that the gas turbine (GT) and SRC cycles exhibited the highest levels of exergy destruction.

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1 Introduction

The economic development of nations has relied on finite fossil fuels in recent years. However, it was not until the oil crisis of 1973 that the need for long-term planning and strategies to address the energy problem was truly recognized. This crisis, coupled with adverse climate changes, global warming, and ozone layer depletion caused by the greenhouse effect, underscored the urgency of revising global energy consumption policies. The impending exhaustion of fossil fuel reserves, rising costs, and stricter environmental regulations have accelerated research into the feasibility of harnessing clean and sustainable energy sources, such as geothermal, solar, biomass, and waste gas energy [1, 2]. Ultimately, the persistent reliance on fossil fuels such as oil, gas, and coal contributes to resource depletion and exacerbates environmental problems, including changes in global climate conditions [3]. A key strategy to minimize fossil fuel consumption is the recovery of heat from the combustion gases emitted by gas turbines. This approach not only reduces fuel usage but also lowers carbon dioxide emissions, thereby helping to protect the environment. In fossil fuel-powered power plants, a significant amount of energy is lost to the surrounding environment as heat. The recovery of this wasted heat is an area of active research among many scholars.

Wang et al. [4] conducted a study on a novel energy system utilizing the biogas process and performed a conceptual evaluation of the proposed configuration. This innovative energy setup integrates a multi-effect desalination (MED) unit, a Rankine cycle, and a solid oxide electrolyzer (SOE) water electrolysis cycle to produce hydrogen fuel, electricity, and freshwater. The results showed that the polygeneration system can generate approximately 1,735 kW of electrical power. Furthermore, the system has the potential to produce around 12.3 kg/h of hydrogen and 9,880 kg/h of freshwater. The energy efficiency of the system was calculated to be 36.4%, while the total exergy cost of the products was assessed to be 16.6 USD/GJ.

Zhang et al. [5] carried out a study to evaluate a new power production system based on a gas turbine. In this configuration, the combustion gases produced by the gas turbine were utilized within the Kalina cycle subsystem. The system demonstrated energy and exergy efficiencies of 45.92% and 44.47%, respectively, while the carbon dioxide emission rate was 61,563 kg/kW. Additionally, the study assessed exergy losses in the gas turbine cycle and the Kalina cycle, which were found to be 84.06% and 15.94%, respectively.

Nemati et al. [6] carried out modeling and thermodynamic optimization to evaluate the Kalina cycle and organic Rankine cycle as potential lower cycles for reusing wastewater from the CGAM gas turbine system. The organic Rankine cycle operates at a significantly lower evaporator pressure (11 bar) compared to the Kalina cycle (46 bar), which enhances its performance. Additionally, the simpler configuration of the ORC, along with its net excess power generation and superheated turbine output flow, contributes to its reliable performance in supporting the turbine.

In their research, Wang et al. [7] introduced exergy output and initial equipment cost as two key objectives for optimizing the organic Rankine cycle using R134a as the working fluid. Their focus was on improving exergy efficiency while minimizing initial costs. The study targeted design parameters such as pinch temperature difference, turbine inlet temperature and pressure, and condenser temperature difference, allowing the identification of the optimal values by fine-tuning these parameters.

Feng and their team [8] performed a parametric resolution and optimization of the ORC cycle. They used a genetic algorithm to achieve an optimal exergy efficiency of 55.97% and an optimal economic objective function of 0.142 $/kWh for the recovery cycle. These results outperformed the basic Rankine cycle by 8.1% in exergy efficiency and 21.1% in the economic objective under identical conditions.

The research conducted by Javadi and his team [9] explored a multi-generational system designed to produce hydrogen, freshwater, and electricity. This system incorporated a hydrogen gas turbine, a steam-methane reforming cycle, a concentrated solar power tower cycle, and a reverse osmosis desalination cycle. The proposed system was capable of generating 12.9 MW of power, producing 96.18 kg/s of freshwater, and generating 5.2 kg/s of hydrogen. Chena et al. [10] proposed a groundbreaking multi-generation plant that operates on natural gas and integrates four different cycles: steam Rankine, absorption power, ejector refrigeration, and proton exchange membrane electrolyzer. This setup is effectively paired with a gas turbine, leading to a significant improvement in overall system performance. Under standard operating conditions, the system can generate 3,896.6 kW of net power, provide a cooling load of 77.7 kW, and produce 0.22 kg/h of hydrogen. It achieves an impressive exergy efficiency of 47.1%, with the total unit cost of products and a reasonable payback period recorded at 9.16 $/GJ and 1.4 years, respectively, under baseline conditions.

Meftahpour et al. [11] introduced an innovative approach by integrating the Kalina Cycle System (KCS) into a cogeneration heat and power (CHP) setup, alongside a gas turbine (GT) cycle and a single-pressure heat recovery steam generator (HRSG). Through the development of thermodynamic and thermo-economic models, a comprehensive evaluation of the system's performance was conducted, focusing on energy, exergy, and exergoeconomic aspects. The results demonstrate a significant improvement in both the first and second law efficiencies for the GT-HRSG/KCS-11 system compared to traditional systems. The efficiency increases from 53.60% to 54.31%, while the second law efficiency rises from 50.59% to 51.27%, underscoring the substantial impact of the proposed system.

2 System Introduction

Figure 1 presents the schematic layout of the multigeneration system, which includes a steam Rankine cycle, a gas turbine cycle, an organic Rankine cycle, a domestic water heater, an RO desalination unit, and a PEM electrolyzer. The primary outputs of the system are electricity, hot water, hydrogen, and purified water.

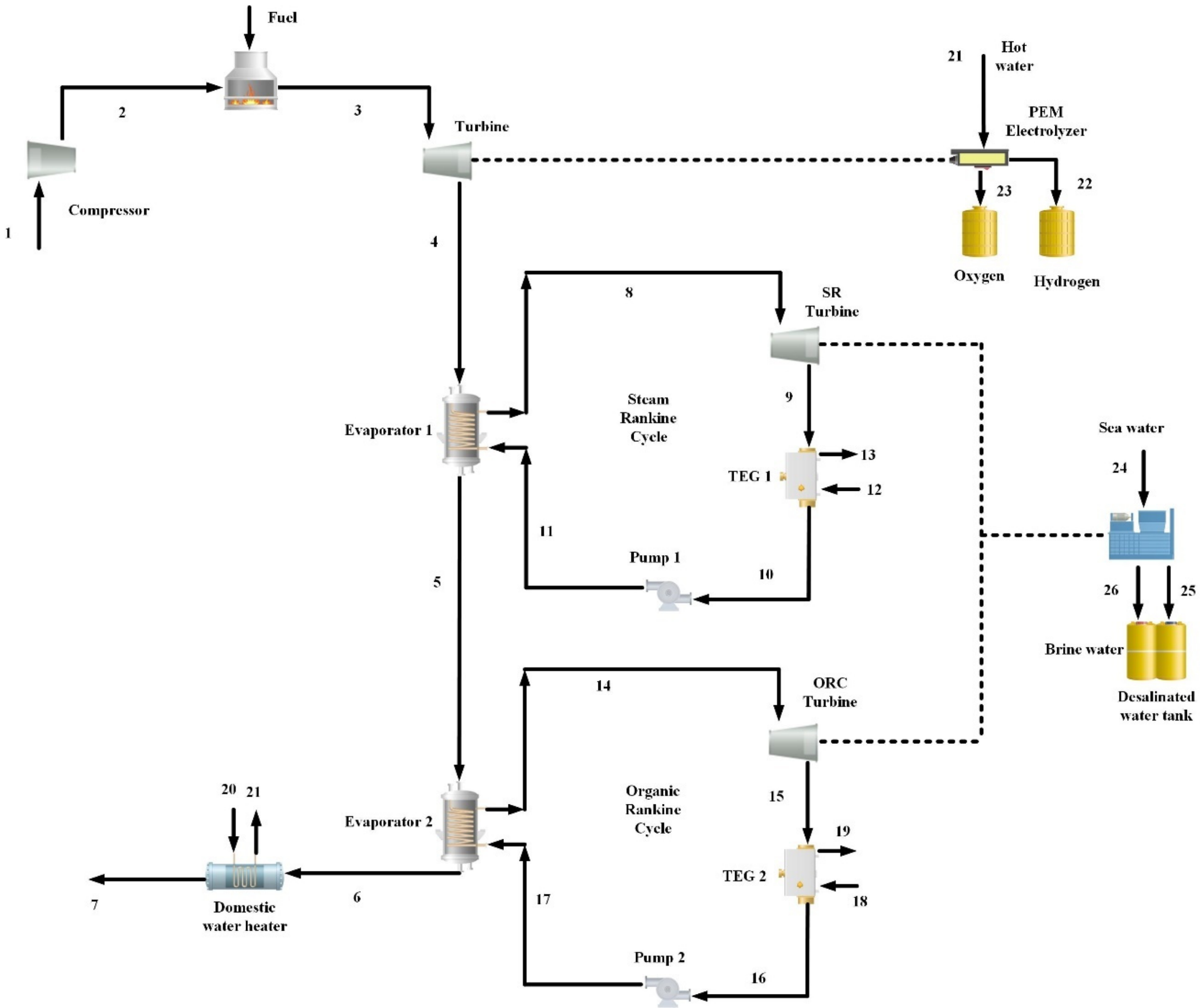

Fig. 1. The proposed multigeneration system.

Air enters the system at point 1, where its pressure is increased by the compressor. After compression, the air moves to the burner at point 2, where its temperature rises. The hot compressed air then passes through the turbine and exits at point 4. The power required for the PEM electrolyzer, which produces hydrogen in this system, is generated by the gas turbine. The hot waste gases from the gas turbine first enter the evaporator of the vapor Rankine cycle, and then, after their temperature is reduced, they flow into the evaporator of the organic Rankine cycle at point 5. Before the waste gases are released into the atmosphere, a domestic water heater is used to extract the remaining heat. In the steam Rankine cycle, water enters at point 10 and its pressure is increased. At point 11, the water enters evaporator 1, absorbs energy from the hot waste gases, and transforms into steam before passing into the turbine. After the turbine generates power, the steam is directed to the TEG unit at point 9. In this system, a thermoelectric generator (TEG) replaces the condenser to generate additional power. The same process occurs in the ORC cycle, with the key difference being the working fluid used. Water is the working fluid in the SRC cycle, while an organic fluid is used in the ORC cycle. Additionally, the working temperature in the SRC cycle is higher than in the ORC cycle. The turbines in both the SRC and ORC cycles, along with the two TEG units, supply the energy required for the RO desalination unit to produce freshwater. The principles of mass balance, energy balance, energetic and exergetic efficiency, exergy loss, and the economic assessment of the system have been thoroughly explored through thermodynamic analysis. To model the combined cycle based on the aforementioned equations, the following hypotheses are also considered: The system operates in a uniform manner. Kinetic and potential energy variations across various components are not considered.

The pressure changes in the heat exchangers and the piping that connects the components are minimal. The fluid exiting the evaporator consists of superheated steam and organic matter, with the degree of superheating (the gap between the evaporator's saturation temperature and the temperature of the outgoing fluid) remaining constant. The fluid serves as the output from the condenser and the input for the saturated liquid pump. Pumps and turbines exhibit a specific level of isentropic efficiency. Both pumps and turbines function in an adiabatic manner. For exergy analysis, the environment temperature and pressure are used as reference conditions. The heat and power transferred, as well as the exergy loss rate of each piece of equipment, can be calculated using the energy and exergy balance equations for the proposed multigeneration system, as outlined in Table 1.

Table 2 contains the essential equations for evaluating the system cost functions. Determining the cost rate of exergy flows and the exergy unit price for each flow in every piece of equipment in the system involves using cost balance equations and auxiliary equations.

Table 1. Formulations for energy conservation and the rate of exergy loss for each element of the suggested system.

Equipment	Energy balance equations	Exergy destruction rate equations
Compressor	$\dot{W}_{\text{comp}} = \dot{m}_1(h_2 - h_1)$	$\dot{\text{E}}\text{x}_{D,\text{comp}} = \dot{W}_{\text{comp}} + \dot{\text{E}}\text{x}_1 - \dot{\text{E}}\text{x}_2$
Gas Turbine	$\dot{W}_{\text{GT}} = \dot{m}_3(h_3 - h_4)$	$\dot{\text{E}}\text{x}_{D,\text{GT}} = \dot{\text{E}}\text{x}_3 - \dot{W}_{\text{GT}} - \dot{\text{E}}\text{x}_4$
DWH	$\dot{Q}_{\text{DWH}} = \dot{m}_6(h_6 - h_7) = \dot{m}_20(h_{21} - h_20)$	$\dot{\text{E}}\text{x}_{D,\text{DWH}} = \dot{\text{E}}\text{x}_6 + \dot{\text{E}}\text{x}_20 - \dot{\text{E}}\text{x}_7 - \dot{\text{E}}\text{x}_{21}$
SRC evaporator	$\dot{Q}_{\text{eva,SRC}} = \dot{m}_4(h_4 - h_5) = \dot{m}_8(h_8 - h_{11})$	$\dot{\text{E}}\text{x}_{D,\text{eva,SRC}} = \dot{\text{E}}\text{x}_{11} + \dot{\text{E}}\text{x}_4 - \dot{\text{E}}\text{x}_8 - \dot{\text{E}}\text{x}_5$
ORC evaporator	$\dot{Q}_{\text{eva,ORC}} = \dot{m}_5(h_5 - h_6) = \dot{m}_{14}(h_{14} - h_17)$	$\dot{\text{E}}\text{x}_{D,\text{eva,ORC}} = \dot{\text{E}}\text{x}_5 + \dot{\text{E}}\text{x}_17 - \dot{\text{E}}\text{x}_6 - \dot{\text{E}}\text{x}_{14}$
SRC turbine	$\dot{W}_{t,\text{SRC}} = \dot{m}_8(h_8 - h_9)$	$\dot{\text{E}}\text{x}_{D,t,\text{SRC}} = \dot{\text{E}}\text{x}_8 - \dot{W}_{t,\text{SRC}} - \dot{\text{E}}\text{x}_9$
ORC turbine	$\dot{W}_{t,\text{ORC}} = \dot{m}_{14}(h_{14} - h_{15})$	$\dot{\text{E}}\text{x}_{D,t,\text{ORC}} = \dot{\text{E}}\text{x}_{14} - \dot{W}_{t,\text{ORC}} - \dot{\text{E}}\text{x}_{15}$
SRC TEG	$\dot{Q}_{\text{TEG,SRC}} = \dot{m}_9(h_9 - h_{10}) = \dot{m}_{12}(h_{13} - h_{12})$	$\dot{\text{E}}\text{x}_{D,\text{TEG,SRC}} = \dot{\text{E}}\text{x}_9 + \dot{\text{E}}\text{x}_{12} - \dot{\text{E}}\text{x}_{10} - \dot{\text{E}}\text{x}_{13}$
ORC TEG	$\dot{Q}_{\text{TEG,ORC}} = \dot{m}_{15}(h_{15} - h_{16}) = \dot{m}_{18}(h_{19} - h_{18})$	$\dot{\text{E}}\text{x}_{D,\text{TEG,ORC}} = \dot{\text{E}}\text{x}_{18} + \dot{\text{E}}\text{x}_{15} - \dot{\text{E}}\text{x}_{16} - \dot{\text{E}}\text{x}_{19}$
SRC Pump	$\dot{W}_{P,\text{SRC}} = \dot{m}_{10}(h_{11} - h_{10})$	$\dot{\text{E}}\text{x}_{D,P,\text{SRC}} = \dot{W}_{P,\text{SRC}} - \dot{\text{E}}\text{x}_{10} + \dot{\text{E}}\text{x}_{11}$
ORC Pump	$\dot{W}_{P,\text{ORC}} = \dot{m}_{16}(h_{17} - h_{16})$	$\dot{\text{E}}\text{x}_{D,P,\text{ORC}} = \dot{W}_{P,\text{ORC}} - \dot{\text{E}}\text{x}_{16} + \dot{\text{E}}\text{x}_{17}$
PEM	$\dot{W}_{\text{PEM}} = (\dot{m}_{21}h_{21} - \dot{m}_{22}h_{22} - \dot{m}_23h_23)$	$\dot{\text{E}}\text{x}_{D,\text{PEM}} = \dot{\text{E}}\text{x}_{21} + \dot{W}_{\text{PEM}} - \dot{\text{E}}\text{x}_{22} - \dot{\text{E}}\text{x}_{23}$
RO	$\dot{W}_{\text{RO}} = (\dot{m}_{24}h_{24} - \dot{m}_{25}h_{25} - \dot{m}_{26}h_{26})$	$\dot{\text{E}}\text{x}_{D,\text{RO}} = \dot{\text{E}}\text{x}_{24} - \dot{\text{E}}\text{x}_{25} - \dot{\text{E}}\text{x}_{26}$

Table 2. Cost functions for each system component

Equipment	Purchase cost ($)
Compressor	$Z_{\text{comp}} = \left(\frac{75\dot{m}_1}{0.9-\eta_{\text{comp}}}\right)\left(\frac{P_2}{P_1}\right) \times \ln\left(\frac{P_2}{P_1}\right)$
Heat exchangers	$Z_{\text{HX}} = 130\left(\frac{A}{0.093}\right)^{0.78}$
Evaporator	$Z_{\text{eva}} = 1397\left(A_{\text{eva}}\right)^{0.89}$
Turbine	$Z_t = 4405\left(\dot{W}_t\right)^{0.7}$
Pump	$Z_p = 1120\left(\dot{W}_p\right)^{0.8}$
TEG	$Z_{\text{TEG}} = 1500\,\dot{W}_{\text{TEG}}$
PEM	$Z_{\text{PEM}} = 1000\,\dot{W}_{\text{PEM}}$
RO	$Z_{\text{RO}} = n_e n_p c_k + n_p c_p + 996\left(\dot{m}_{25}\right)^{0.8}$

Table 3. Cost equilibrium and supporting equations for every element of the suggested system

Component	Cost balance equation	Auxiliary equation
Compressor	$\dot{C}_1 + \dot{Z}_{\text{comp}} + \dot{C}_{w,\text{comp}} = \dot{C}_2$	$c_1 = 0$
Gas Turbine	$\dot{C}_3 + \dot{Z}_G T = \dot{C}_4 + \dot{C}_{w,\text{GT}}$	$c_3 = c_4$
SRC Evaporator	$\dot{C}_4 + \dot{C}_{11} + \dot{Z}_{\text{eva,SRC}} = \dot{C}_5 + \dot{C}_8$	$c_6 = c_9$
ORC Evaporator	$\dot{C}_5 + \dot{C}_{17} + \dot{Z}_{\text{eva,ORC}} = \dot{C}_6 + \dot{C}_{14}$	$c_6 = c_9$
SRC turbine	$\dot{C}_8 + \dot{Z}_{t,\text{SRC}} = \dot{C}_9 + \dot{C}_{w,t,\text{SRC}}$	$c_8 = c_9$
ORC turbine	$\dot{C}_{14} + \dot{Z}_{t,\text{ORC}} = \dot{C}_{15} + \dot{C}_{w,t,\text{ORC}}$	$c_{14} = c_{15}$
SRC TEG	$\dot{C}_9 + \dot{C}_{12} + \dot{Z}_{\text{TEG,SRC}} = \dot{C}_{10} + \dot{C}_{13}$	$c_9 = c_{10}$, $c_{12} = 0$
ORC TEG	$\dot{C}_{15} + \dot{C}_{18} + \dot{Z}_{\text{TEG,ORC}} = \dot{C}_{16} + \dot{C}_{19}$	$c_{15} = c_{16}$, $c_{18} = 0$
PEM	$\dot{C}_{21} + \dot{C}_{w,\text{PEM}} + \dot{Z}_{\text{PEM}} = \dot{C}_{22} + \dot{C}_{23}$	$c_{w,\text{PEM}} = c_{w,t,\text{SRC}}$, $c_{23} = 0$
RO	$\dot{C}_{24} + \dot{Z}_{\text{RO}} + \dot{C}_{w,\text{RO}} = \dot{C}_{25} + \dot{C}_{26}$	$c_{w,\text{RO}} = c_{w,t,\text{ORC}}$, $c_{25} = c_{26} = 0$

3 Results and Discussions

Simulating the multigeneration system requires the selection of specific input parameters. Table 4 provides a list of the parameters needed for system modeling. It is crucial to define these parameters before proceeding with further calculations.

The system modeling was conducted using EES software with the provided initial data. Table 5 presents a comparison of the operational features of the proposed multigeneration system.

Figure 2 illustrates the relationship between the GT turbine inlet temperature and the system's energy and exergy efficiency. The graphs show that increasing the GT turbine inlet temperature from 1200 K to 1500 K results in a decline in both energy and exergy efficiency. Specifically, as the inlet temperature of the GT rises, energy efficiency decreases from 30.16% to 15.01%, while exergy efficiency drops from 18.1% to 9.165%. This represents a 50% reduction in the system's efficiency. The efficiency decline is attributed to the decrease in power output from the turbine as its inlet temperature increases. Since the GT turbine is the primary power source in the system, the reduction in its power generation leads to an overall drop in total efficiency.

Figure 3 shows the effect of the GT turbine inlet temperature on hydrogen and freshwater production. The results indicate that as the turbine inlet temperature increases, freshwater production rises, while hydrogen production decreases. Specifically, when the GT turbine temperature is increased from 1200 K to 1500 K, freshwater output increases from 17.31 kg/s to 20.07 kg/s, while hydrogen output decreases from 0.6938 kg/s to 0.2976 kg/s. According to the system schematic, the GT turbine supplies power to the PEM electrolyzer, while the SRC and ORC turbines and TEG units provide power to the RO unit. As the GT turbine inlet temperature increases, less power is generated by the turbine, leading to a reduction in hydrogen production. On the other hand, a higher GT turbine inlet temperature directs more power to the SRC and ORC subsystems. This increase in power availability enhances the performance of the turbines and TEG units, ultimately boosting freshwater production by the system.

Table 4. Initial considered parameters for cycle analysis

Parameters	Unit	Value
SRC turbine inlet temperature	°C	450
ORC turbine inlet temperature	°C	120
The air inlet mass flow rate of	kg/s	500
Isentropic efficiency of GT Compressor	%	85
Isentropic efficiency GT Turbine	%	85
PEM		
P_{H_2}, P_{O_2}	atm	1
T_{PEM}	°C	80
$E_{\mathrm{act},a}$	kJ/mol	76
$E_{\mathrm{act},c}$	kJ/mol	18
λ_a	–	14
λ_c	–	10
D	mm	50
J_a^{ref}	A/m^2	1.7×10^5
J_c^{ref}	A/m^2	4.6×10^3
RO		
Recovery ratio, RR	–	0.3
Number of elements, n_e	–	7
Number of pressure vessels, n_v	–	42
Seawater salinity, X_f	g/kg	43

Table 5. The simulation results for the suggested system in general

Parameters	Unit	Value
η_{en}	%	24.92
η_{ex}	%	15.01
$\dot{W}_{\mathrm{GT},t}$	kW	299021
$\dot{W}_{\mathrm{SRC},t}$	kW	10809
$\dot{W}_{\mathrm{ORC},t}$	kW	5536
$\dot{W}_{\mathrm{TEG1}}$	kW	4696
$\dot{W}_{\mathrm{TEG2}}$	kW	350.8
$\dot{W}_{\mathrm{net}}$	kW	320412.8
$\dot{m}_{H_2}$	kg/s	0.556
$\dot{m}_{\mathrm{Freshwater}}$	kg/s	18.31
Z	$	6821000

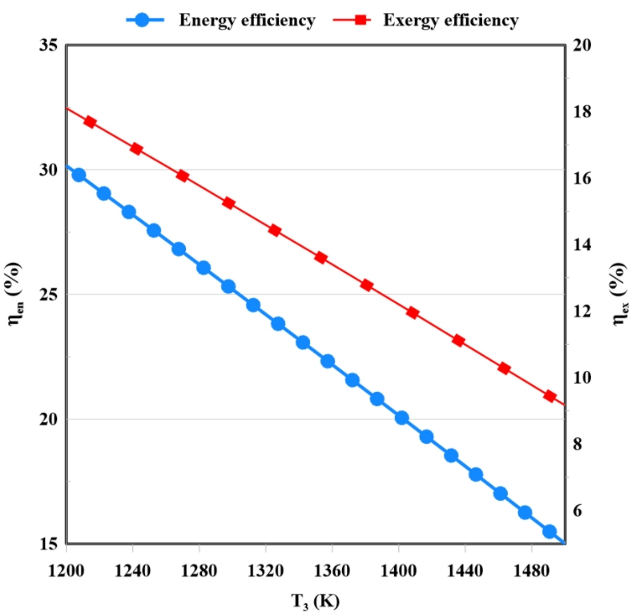

Fig. 2. The impression of the GT turbine inlet temperature on the energy and exergy efficiency of the system.

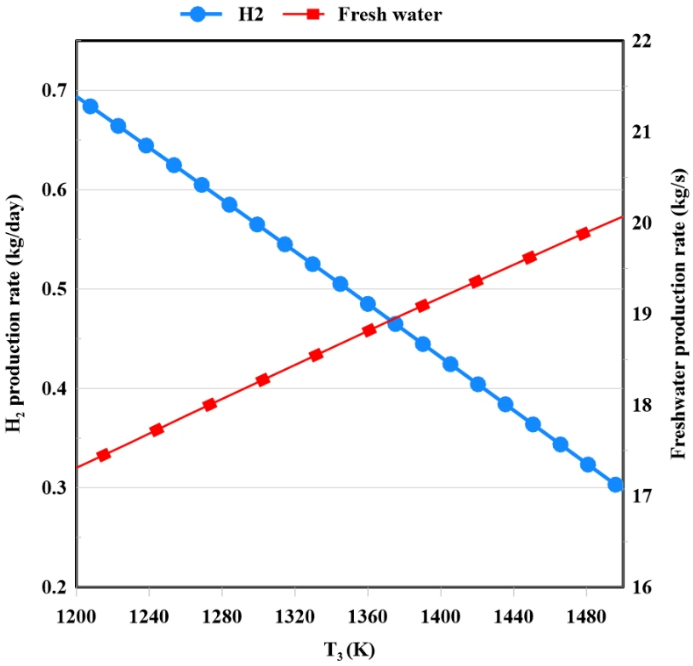

Fig. 3. The influence of the GT turbine inlet temperature on the quantity of freshwater and hydrogen production rates.

Figure 4 shows the variation in the energy and exergy performance of the system as the temperature of the steam Rankine cycle (SRC) evaporator increases from 720 K to 820 K. Both energy and exergy efficiencies exhibit an upward trend as the SRC evaporator temperature rises. Specifically, with a 100 K increase in evaporator temperature, energy efficiency increases from 25.54% to 25.87%, while exergy efficiency rises from 15.44% to 15.67%. The increase in SRC evaporator temperature leads to higher power generation by the SRC turbine and TEG 1, resulting in improved system efficiency.

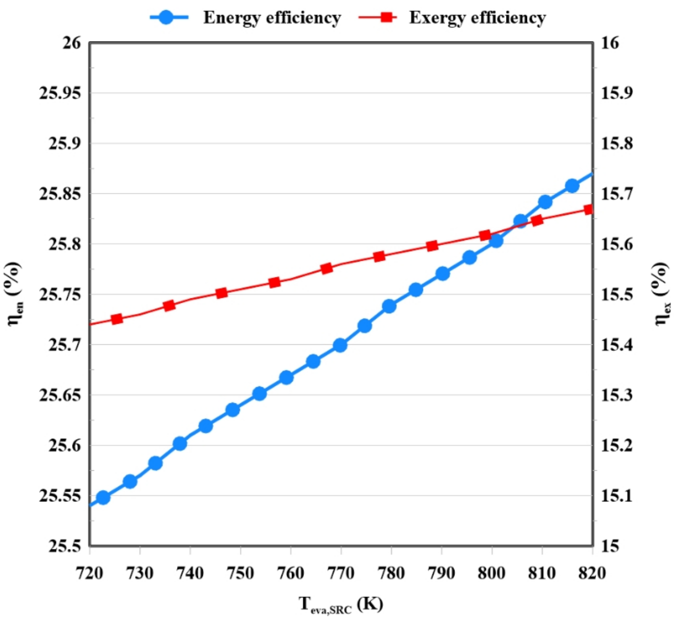

Fig. 4. The impact of steam Rankine cycle evaporator temperature changes on the energy and exergy efficiencies.

Figure 5 illustrates the changes in system power generation and the rate of exergy destruction as the temperature of the steam Rankine cycle (SRC) evaporator increases from 720 K to 820 K. As the evaporator temperature rises, both power output and the rate of exergy destruction within the system show an upward trend. Specifically, system power generation increases from 138,400 kW to 141,137 kW, while the exergy destruction rate rises from 646,889 kW to 739,080 kW. An increase in evaporator temperature results in a greater enthalpy difference, while keeping the inlet drive temperature and pinch temperature difference constant. This change leads to a higher output drive temperature and a reduction in the enthalpy difference of the drive gas, which consequently lowers the steam flow rate in the steam Rankine cycle. Additionally, the higher steam temperatures entering the turbine result in a larger turbine enthalpy difference, which compensates for the reduced steam flow rate and significantly enhances the work output of the steam turbine. The maintained pinch temperature difference, along with the increased temperature of the drive output from the steam cycle's evaporator to the organic cycle's evaporator inlet, increases the enthalpy difference of the waste gas flow entering and exiting the organic cycle evaporator. This, coupled with the constant enthalpy difference within the organic cycle evaporator, leads to an increase in the flow rate of the organic fluid in the evaporator, thereby boosting the work output of the organic cycle turbine. As a result, the total work output – combining the work of both the organic and steam cycles – shows a significant increase.

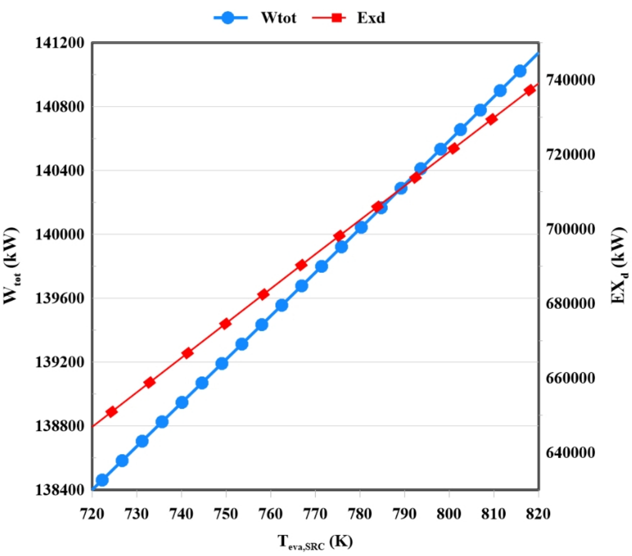

Fig. 5. The influence of steam Rankine cycle evaporator temperature changes on the power generated by the system and the rate of exergy destruction of the system.

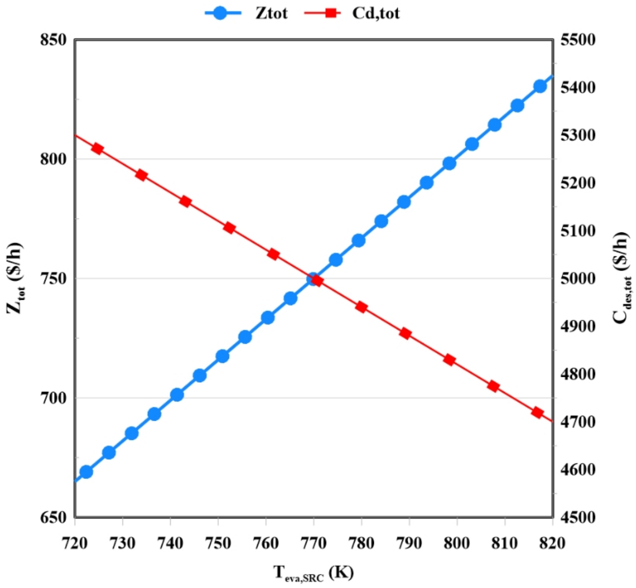

Fig. 6. The influence of steam Rankine cycle evaporator temperature changes on the overall initial cost rate and the overall cost of exergy destruction.

As shown in Figure 6, the economic factors of the system change with an increase in the steam evaporator temperature. It is evident that raising the steam evaporator's temperature results in a higher overall initial cost rate for the system. The initial cost associated with the steam cycle increases due to the interactions between its components, leading to a higher overall initial cost for the combined cycle. At the same time, the cost rate related to exergy destruction diminishes as the evaporator temperature rises. Specifically, when the evaporator temperature rises from 720 K to 820 K, the initial cost rate increases from 665 \$/h to 835 \$/h, while the total cost of exergy destruction decreases from 5300 \$/h to 4700 \$/h.

Figure 7 illustrates the effect of varying the temperature difference in the steam cycle's pinch evaporator on its economic factors. As shown in the figure, as the temperature difference in the steam evaporator pinch increases from 5 °C to 25 °C, the initial cost rate rises from 702 \$/h to 959.6 \$/h. This increase in costs affects all components of the cycle, leading to a reduction in the turbine's output work and the steam flow rate. As a result, the increase in the cost rate within the organic cycle becomes significant, contributing to an overall rise in initial expenditure. Furthermore, Figure 7 shows that the cost rate associated with exergy destruction increases as the pinch temperature difference in the evaporator rises. This increase in the organic cycle's flow rate enhances irreversibility and raises the associated costs in the cycle. In contrast, within the steam cycle, the higher temperature difference in the evaporator, and the resulting rise in irreversibility for this component, lead to a higher cost of exergy destruction. This trend dominates in the steam cycle, where, although other components experience a reduction in costs due to the decreased steam flow rate, there is a slight increase in costs related to exergy destruction within the steam cycle.

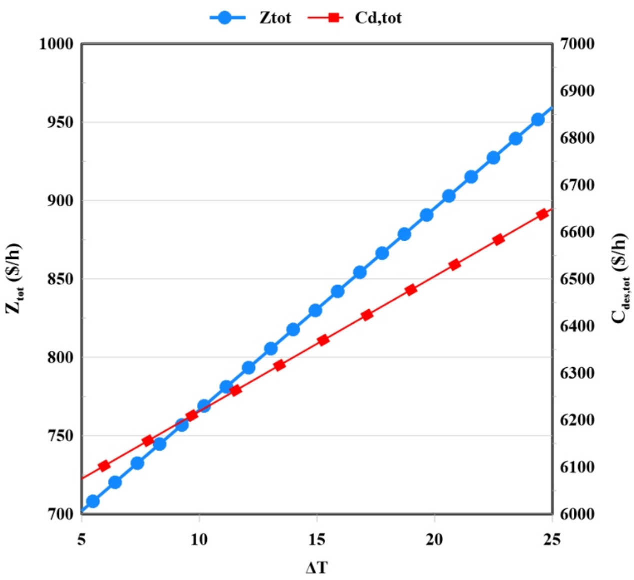

Fig. 7. The influence of changes in the temperature difference of steam Rankine pinch evaporator on the total initial cost rate and the total cost of exergy destruction.

4 Conclusions

An assessment has been conducted on a proposed cycle designed to generate power, hot water, freshwater, and hydrogen by utilizing the waste heat from a gas turbine. The cycle integrates a gas turbine, steam Rankine cycle, organic Rankine cycle, PEM electrolyzer, RO unit, and a domestic water heater (DWH). The primary outputs of the system include power, hydrogen, hot water, and freshwater. The research employed thermodynamic and thermoeconomic analyses to thoroughly evaluate the performance of the cycle. A detailed thermodynamic simulation was conducted using EES software, accompanied by a parametric study to demonstrate the cycle's behavior under varying input conditions. The main conclusions drawn from this investigation are as follows.

- The system demonstrates energy and exergy outcomes of 24.92% and 15.01%, respectively.
- The overall power generated by the system from three turbines and two TEGs is 320412.8 kW.
- This system generates a total of 0.556 kg/s of hydrogen.
- The system provides a total of 18.31 kg/s of freshwater.
- Augmenting the GT turbine inlet temperature results in a decrease in both energetic and exergetic efficiency, also the hydrogen production rates, although the freshwater production rate rises.
- Enhancements in the SRC evaporator contribute to increases in energy and exergy efficiency, total power output, exergy destruction rate, and initial cost rate.

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