

Sustainable Energy Production: Thermodynamic Evaluation of Geothermal-Based Multigeneration System

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Abstract

This study presents a financial and thermal analysis of a novel geothermal multigeneration system capable of producing electricity, cooling, fresh water, and hydrogen. The system uniquely integrates several innovative technologies as follows: Integration of Organic Rankine cycle (ORC), and Kalina cycle for efficient geothermal power generation; transcritical CO₂ cooling combined with single-effect absorption refrigeration to enable both cooling and energy generation for multi-purpose operation; regasification of Liquefied Natural Gas (LNG) as an energy reservoir of accumulating the cold energy; Proton Exchange Membrane (PEM) electrolyzer and Reverse Osmosis (RO) unit, operating the produced power for hydrogen and fresh water production, respectively. The integration of these cycles and technologies is being pioneered for the first time, a representing novel towards optimized energy and water generation from geothermal sources. The system produces a gross electrical power output of 5,962.4 kW, with an overall exergy efficiency of 32.04%, and an exergy efficiency of 11.65% specifically for electricity production. The study of exergy destruction shows turbines, compressors and heat exchangers as the primary inefficiency sources, providing space for their future optimization. The system's performance higher geothermal source temperatures; increasing the source temperature from 380 K to 415 K results in significant gains in both power and freshwater production, highlighting the importance of high-quality geothermal resources. The system's economic performance is confirmed by a total production cost of \$6559 per GJ. The potential of geothermal-powered multigeneration systems in optimizing the resource use, delivering sustainable energy, and water solutions is shown in this research.

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1 Introduction

The necessity of sustainable energy solutions arises from growing global power demands and global warming needs. The reliable operation combined with minimal environmental impact and base-load functionality makes geothermal energy a leading choice in this energy transition according to DiPippo [1]. Geothermal energy systems deliver power at a steady rate beyond wind and solar types since they supply constant electricity thus offering sustained power to meet extended power requirements. Geothermal-driven multigeneration systems generate electricity and cooling, while also providing heating, producing hydrogen fuel, and supplying fresh water – thereby enhancing both sustainability and operational efficiency. The current power grid that operates with fossil fuels produces various environmental problems through its greenhouse gas emissions and climate warming effects while it exhausts natural resources [2]. Because of growing climate change challenges worldwide, there is an urgent need to transition to renewable and clean energy sources without delay. Low-carbon energy systems must integrate renewable sources such as solar, wind, hydroelectric, biomass, and geothermal power to meet the sustainability targets outlined in the United Nations Sustainable Development Goals and the Paris Agreement. [3]. Consolidated multigeneration energy systems generate different valuable products from a single energy input. Several levels of energy utilization achieve resource optimization and create both improved renewable project financial success and decreased waste volume [4]. The geothermal-based system combines Kalina Cycle technology for power generation with energy extraction via the Organic Rankine Cycle (ORC), while managing cooling through CO₂ refrigeration integrated with single-effect absorption methods for thermal cooling. The operational adaptability of geothermal systems becomes enhanced through combining electrolytic hydrogen production with saltwater desalination [5]. Geothermal power used to operate multigeneration systems leads to fundamental developments which build resilient power systems free from decarbonization while minimizing fossil fuel usage.

Li et al. [6] introduced a geothermal power system which provides electricity generation, hydrogen production, and freshwater supply while generating thermal cooling outputs. The system daily supplies 100 MW of electrical power together with 10,000 tons of cooling and 0.5 tons of hydrogen and 50,000 m³ of freshwater. The system operates at 85% thermal efficiency while reaching an overall performance rate of 80-90% and includes electricity costs between \$50 and

\$70 per MWh. A 15-20% reduction in greenhouse gas emissions accompanies this power and resource production technology which operates as a sustainable and economical system. A new energy system by Alirahmi et al. [7] produces hydrogen energy along with electricity while supplying heating and cooling and generating freshwater at the same time. The system optimization techniques and thermodynamic analysis enable a performance level of 72.8% efficiency while lowering green gas emissions by 32% when compared with established approaches. The system generates 4.5 kilograms of hydrogen each hour combined with daily electricity production of 50 kilowatt-hours along with daily freshwater production of 500 liters. A sustainable energy solution for addressing power and water supply gaps emerges from the system with both a favorable present value and a 5.4-year return period. Research by Almutairi [8] shows a dual-purpose technique that joins LNG regasification equipment with geothermal-based dual-loop power cycle technology to generate hydrogen and water desalination products. When operating at 80%, the system achieves thermal efficiency performance that reduces carbon emissions better than conventional systems. The system generates 5 kg of hydrogen throughout each hour along with the daily output of 600 liters of desalinated water. This integrated system generates both positive net present value and pays back within five years which ensures its economic sustainability for establishing power operations and water supply while minimizing ecological risks. A system driven by GES generates electricity and hydrogen and fresh water from reverse osmosis desalination and Rankine Cycle and PEME according to Kianfard et al. [9].

An efficiency rate of 17% together with a return on investment time of 5.6 years represented the system performance. The extended timeframe of this system was accepted because its environmental operational benefits outweighed its traditional non-renewable counterparts. The research shows that geothermal power presents an efficient approach for producing sustainable energy and hydrogen alongside water supply. Zhang et al. [10] designed a power generation system that merges geothermal power with Kalina cycle technology and Double-Effect absorption cooling and LNG regasification to create electricity and produce cooling and desalinated water. The system operates at 72.3% energy-to-power efficiency combined with 48.6% exergy efficiency. This system produces 5.1 kg of hydrogen per hour together with 620 liters of freshwater every day as it minimizes LNG regasification losses. The system proves economically beneficial while also sustaining effective multi-generation operation with a positive 5.2-year return on investment. A geothermal

multigeneration system which employs LNG as a cold energy recovery agent was presented by Parikhani et al. [11]. The system accomplishes simultaneous power generation with hydrogen production as well as heating and cooling services. Researchers employed evolutionary algorithms to optimize the system through energy and exergy and exergoeconomic evaluations, achieving improved efficiency outcomes through both single- and multi-objective optimization methods. The research shows that optimization strategies lead to improved operational efficiency and reduced expenses in intricate energy systems, thereby providing a thermodynamic and economic solution.

Kalem and Yilmaz [12] introduced a multi-generic geothermal facility that generates power electricity and provides heat while also producing freshwater and compressed hydrogen. The study determined a 70.1% energy efficiency and a 47.8% exergy efficiency through their thermodynamic and economic evaluations. The system generates a hydrogen output of 4.6 kg per hour along with a daily freshwater production of 560 liters while maintaining reasonable hydrogen market prices. Geothermal integration for sustainable multi-use applications demonstrates feasibility because it produces a positive net present value while requiring a 5.5-year payback period. Nasruddin et al. [13] assessed a Kalina Cycle System 11 (KCS 11) using geothermal brine at 120 °C in Indonesia. The study optimized ammonia concentration (0.85) and turbine inlet pressure (9 bar), achieving thermal efficiency of 14.45% and exergy efficiency of 44.7%. The results confirm Kalina cycle's suitability for efficient low-temperature geothermal heat recovery, outperforming conventional systems in terms of energy and exergy utilization. Bahrami and Rosen [14] designed a geothermal-powered multigeneration system integrating Kalina cycle (for generation of electricity, cooling, and hydrogen production). The system proved a multi-objective optimization approach with 46.87% and 44.13% energy and exergy efficiencies. The greatest exergy destruction was detected in the Kalina subsystem. This study proposed an effective strategy for achieving high-efficiency, zero-emission geothermal power production with valuable additional outputs.

Laghari et al. [15] analyzed a geothermal Organic Rankine Cycle (ORC) using binary working fluid mixtures, evaluating a geothermal heat source at 140 °C. The optimal R245fa/R152a blend achieved thermal and exergy efficiencies of 12.3% and 48.1%, respectively. Exergy destruction occurred mainly in the evaporator and turbine. The study emphasizes the potential of fluid mixture optimization to improve ORC performance for medium-temperature geothermal applications. Genceli et al. [16] investigated geothermal ORCs

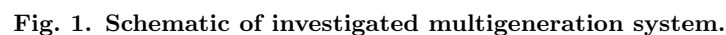
using partially evaporated working fluids. At 115 °C, using R1233zd(E), the system's exergy efficiency improved from 38.2% to 45.6%. Energy efficiency also increased by 9%, with the evaporator being a key site for exergy destruction. The results support the use of partially evaporated fluids to reduce irreversibility and enhance ORC performance in geothermal applications. Pan et al. [17] presented a hybrid cascade ORC system powered by heat from the geothermal (130 °C) and freeze energy from LNG. Thermal and exergy efficiencies of 21.2% and 54.6% were recorded in the system respectively. Integration of LNG decreased condenser exergy destruction by 18%. This hybrid system maximizes energy retrieval from the high and low enthalpy sources which provides a promising engineering for LNG-geothermal combined power generation.

Gheraout et al. [18] performed a techno-economic analysis and optimization of a geothermal-driven multigeneration system integrated with LNG regasification in their 2025 study. The system showed high performance improvements. Overall exergy efficiency of the integrated system was 32.5% as a noteworthy improvement against conventional geothermal systems. In addition, machine learning-based optimization lowered the cost of energy by 21% (LCOE) with more economical operation. The water output capacity of the system was also formidable and is estimated at 4,500 m³/day using reverse osmosis desalination. Moreover, the system integration of hydrogen production by electrolysis resulted in a daily hydrogen production rate of 1,200 kg/day, which points to the ability of this system to produce energy and fresh water simultaneously on sustainable terms. You et al. [19] proposed a novel geothermal-based multigeneration system that incorporates LNG cold energy utilization to simultaneously produce hydrogen, electricity, heating, and cooling. The performance of the system was analyzed by means of thermodynamic simulation and optimization tools. Results showed that the integrated system recorded substantial energy saving compared to conventional systems. The research shows the possibility of using combination of geothermal and LNG cold energy for sustainable and efficient production of energy in many industrial uses.

Many studies have explored geothermal power systems with multi-generation capabilities for electricity generation, heating, cooling, hydrogen production, and water management. However, to date, no studies have conducted a thermodynamic analysis of integrated cycles that combine Kalina and Organic Rankine Cycles (ORC), transcritical refrigeration, absorption refrigeration, and LNG heat sinks. This study is the first to investigate the thermodynamic performance of this novel combination of cycles, which has not been explored pre-

sustainable solutions for energy and water needs.

The schematic diagram of the studied multigeneration system configuration can be seen in [Figure 1](#).



transcritical CO₂ refrigeration and single-effect absorption refrigeration cycle. The PEM electrolyzer obtains its power supply from the ORC and the Kalina turbine supplies the necessary energy for reverse osmosis unit desalination operations to produce freshwater. A transcritical CO₂ refrigeration cycle acts as the cooling

mechanism to boost overall performance of the system. Multiple outputs from this system combine electricity production with heating and cooling as well as the generation of hydrogen and freshwater. Thermodynamic analyses confirm that the integrated system demonstrates excellent operational efficiency and a minimal ecological footprint, ensuring effective resource management and sustainable operation for the combined generation of electricity, cooling, hydrogen, and water desalination.

The thermodynamic evaluation of the integrated system relies on balance equations for mass flow and energy and entropy and exergy. Mass conservation occurs through control volumes by mass balance principles while energy conservation follows the fundamental thermodynamic law. System irreversibilities produce entropy through the evaluation process of the entropy balance which includes both internal and external sources of entropy generation. The thermodynamic system optimization process can be achieved through exergy analysis which uses the principles of the second law of thermodynamics to detect lost energy and inefficiencies. A significant metric known as exergy destruction enables the measurement of resource losses in order to identify optimization possibilities. The thermodynamic performance of a system reflects in exergy efficiency which describes the extent to which fuel transforms into operational project. The equations applied for modeling the system are [20,21]:

$$\sum \dot{m}_{in} - \sum \dot{m}_{out} = \frac{dm_{cv}}{dt}, \quad (1)$$

$$\dot{Q} - \dot{W} + \sum_{in} \dot{m}_{in} \left(h_{in} + \frac{V_{in}^2}{2} + gZ_{in} \right) - \sum_{out} \dot{m}_{out} \left(h_{out} + \frac{V_{out}^2}{2} + gZ_{out} \right) = \frac{dE_{cv}}{dt}, \quad (2)$$

$$\dot{S}_{gen} = \sum_{out} \dot{m}_{out} ex_{out} - \sum_{in} \dot{m}_{in} ex_{in} - \sum \frac{\dot{Q}}{T} + \frac{dS_{cv}}{dt}, \quad (3)$$

$$\dot{E}_{x_{in}} + \dot{E}_{x_Q} = \dot{E}_{x_{out}} + \dot{E}_{x_W} + \dot{E}_{x_D}, \quad (4)$$

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

$$\dot{E}_{x_Q} = \left(1 - \frac{T_0}{T_i} \right) \dot{Q}_i, \quad (6)$$

$$\dot{E}_{x_{out}} = \dot{m}_{out} ex_{out}, \quad (7)$$

$$\dot{E}_{x_W} = \dot{W}, \quad (8)$$

$$\dot{E}_{x_D} = T_0 \dot{S}_{gen}, \quad (9)$$

$$ex = ex_{ph} + ex_{ch}, \quad (10)$$

$$\eta_{ex} = \frac{\dot{E}_{x_P}}{\dot{E}_{x_F}} = 1 - \frac{\dot{E}_{x_D}}{\dot{E}_{x_F}}. \quad (11)$$

The multigeneration system receives evaluation through energy and exergy analysis to trace energy movements and detect exergy destruction according to Table 1. System performance reaches its maximum potential through this method by identifying causes of irreversibility which leads to better efficiency and sustainability.

3 Results and Discussion

The following assumptions were applied in the simulation of the combined cycle to simplify the analysis and ensure consistency:

- The saturated liquid condition is assumed for both the R123 fluid and the ammonia-water mixture at the outlet of Heat Exchanger 3, simplifying the modeling of phase change processes.
- The Kalina cycle separator is assumed to produce saturated steam from the vapor outlet and saturated liquid from the liquid outlet.
- Thermodynamic equilibrium is assumed among fluid components in the absorption cycle, with the absorber and generator exit streams at the same temperatures as their corresponding sections.
- The absorption chiller operates with a saturated liquid refrigerant exiting the condenser and saturated vapor at the outlet of Evaporator 1.
- To ensure consistency in evaluating the absorption cycle, the solution exiting both the generator and absorber is assumed to be in a saturated liquid state.
- In the CO₂ transcritical refrigeration cycle, it is assumed that Evaporator 2 produces saturated vapor at its outlet to maintain simplicity in the cycle modeling.

Simulation models undergo validation procedures to confirm that their predictions of thermodynamic cycle performance are both reliable and accurate. The simulation results from this research correspond to the findings of Wu et al. [22], regarding thermal efficiency and pressure performance of transcritical CO₂ cycles. Working fluid properties together with heat exchanger design led to minor discrepancies which mainly appeared at high temperatures. The simulation software shows sufficient capability to model transcritical CO₂ cycles based on validated results. The supplied data encompasses multiple points in Table 2.

The simulation results from EES software indicated that the combined system generated a total of 5,962.4 kW of power, with 3,289.6 kW produced by the Kalina cycle and 2,672.8 kW by the ORC cycle, establishing the Kalina cycle as the primary power source. The designed system reached an exergy efficiency peak of

32.04% coupled with an electrical exergy efficiency of 11.65% which falls within standard numbers for integrated thermal power systems. The system operators should focus on optimization since the total exergy destruction reached 24,401.63 kW. The cogeneration unit operated as a cooling facility that generated 25,549.12 kW of cooling power. The system's electricity produc-

tion cost was estimated at \$90.6/GJ, while the overall combined unit cost of all outputs was calculated at \$65.59/GJ. The system simultaneously generated 7.998 kg/s of freshwater and 0.009 kg/s of hydrogen while raising the environmental quality and economic return which enhances the operational sustainability of integrated energy systems.

Table 1. Energy balance equations and exergy destruction rate expressions for individual system components [23–26]

Component	Energy balance equations	Exergy destruction rate equations
Kalina cycle		
Heat exchanger 1	$\dot{Q}_{HX1} = \dot{m}_1(h_1 - h_2) = \dot{m}_{10}(h_{11} - h_{10})$	$\dot{E}x_{D,HX1} = \dot{E}x_1 + \dot{E}x_{10} - \dot{E}x_2 - \dot{E}x_{11}$
Separator	$\dot{m}_{11}h_{11} = \dot{m}_5h_5 + \dot{m}_{12}h_{12}$	$\dot{E}x_{D,sep} = \dot{E}x_{11} - \dot{E}x_5 - \dot{E}x_{12}$
Turbine 1	$\dot{W}_{turb1} = \dot{m}_5h_5 - \dot{m}_6h_6$	$\dot{E}x_{D,turb1} = \dot{E}x_5 - \dot{W}_{turb1} - \dot{E}x_6$
Expander 1	$\dot{W}_{Exp1} = \dot{m}_{12}h_{12} - \dot{m}_{13}h_{13}$	$\dot{E}x_{Exp1} = \dot{E}x_{12} - \dot{W}_{Exp1} - \dot{E}x_{13}$
Heat exchanger 2	$\dot{Q}_{HX2} = \dot{m}_{13}(h_{13} - h_{14}) = \dot{m}_{10}(h_{10} - h_9)$	$\dot{E}x_{D,HX2} = \dot{E}x_1 + \dot{E}x_{10} - \dot{E}x_2 - \dot{E}x_{11}$
Mixer	$\dot{m}_7h_7 = \dot{m}_6h_6 + \dot{m}_{14}h_{14}$	$\dot{E}x_{D,mix} = \dot{E}x_6 + \dot{E}x_{14} - \dot{E}x_7$
Heat exchanger 3	$\dot{Q}_{HX3} = \dot{m}_7(h_7 - h_8) = \dot{m}_{14}(h_{15} - h_{14})$	$\dot{E}x_{D,HX3} = \dot{E}x_7 + \dot{E}x_{18} - \dot{E}x_8 - \dot{E}x_{15}$
Pump 1	$\dot{W}_{pump1} = \dot{m}_8(h_9 - h_8)$	$\dot{E}x_{D,pump1} = \dot{W}_{pump1} - \dot{E}x_9 + \dot{E}x_8$
ORC cycle		
Pump 2	$\dot{W}_{pump2} = \dot{m}_{17}(h_{18} - h_{17})$	$\dot{E}x_{D,pump2} = \dot{W}_{pump2} - \dot{E}x_{18} + \dot{E}x_{17}$
Expander 2	$\dot{W}_{Exp2} = \dot{m}_{15}h_{15} - \dot{m}_{16}h_{16}$	$\dot{E}x_{D,Exp2} = \dot{E}x_{15} - \dot{W}_{Exp2} - \dot{E}x_{16}$
Heat exchanger 4	$\dot{Q}_{HX4} = \dot{m}_{16}(h_{16} - h_{17}) = \dot{m}_{20}(h_{21} - h_{20})$	$\dot{E}x_{D,HX4} = \dot{E}x_{16} + \dot{E}x_{20} - \dot{E}x_{17} - \dot{E}x_{21}$
LNG system		
Pump 3	$\dot{W}_{pump3} = \dot{m}_{19}(h_{20} - h_{19})$	$\dot{E}x_{D,pump3} = \dot{W}_{pump3} - \dot{E}x_{20} + \dot{E}x_{19}$
Heat exchanger 5	$\dot{Q}_{HX5} = \dot{m}_{21}(h_{21} - h_{22}) = \dot{m}_{55}(h_{56} - h_{55})$	$\dot{E}x_{D,HX5} = \dot{E}x_{21} + \dot{E}x_{55} - \dot{E}x_{22} - \dot{E}x_{56}$
Turbine 2	$\dot{W}_{turb2} = \dot{m}_{22}h_{22} - \dot{m}_{23}h_{23}$	$\dot{E}x_{D,turb2} = \dot{E}x_{22} - \dot{W}_{turb2} - \dot{E}x_{23}$
Heat exchanger 6	$\dot{Q}_{HX6} = \dot{m}_{23}(h_{23} - h_{24}) = \dot{m}_3(h_3 - h_4)$	$\dot{E}x_{D,HX5} = \dot{E}x_{23} + \dot{E}x_3 - \dot{E}x_{24} - \dot{E}x_4$
Transcritical refrigeration system		
Evaporator 2	$\dot{Q}_{eva2} = \dot{m}_{39}(h_{39} - h_{44}) = \dot{m}_{47}(h_{47} - h_{48})$	$\dot{E}x_{eva2} = \dot{E}x_{44} + \dot{E}x_{47} - \dot{E}x_{39} - \dot{E}x_{48}$
Compressor	$\dot{W}_{comp} = \dot{m}_{40}h_{40} - \dot{m}_{39}h_{39}$	$\dot{E}x_{D,comp} = \dot{E}x_{39} + \dot{W}_{comp} - \dot{E}x_{40}$
Gas cooler	$\dot{Q}_{GC} = \dot{m}_{41}(h_{41} - h_{42}) = \dot{m}_{45}(h_{46} - h_{45})$	$\dot{E}x_{GC} = \dot{E}x_{41} + \dot{E}x_{45} - \dot{E}x_{42} - \dot{E}x_{46}$
Evaporator 1	$\dot{Q}_{eva1} = \dot{m}_{33}(h_{34} - h_{33}) = \dot{m}_{42}(h_{42} - h_{43})$	$\dot{E}x_{eva1} = \dot{E}x_{33} + \dot{E}x_{42} - \dot{E}x_{34} - \dot{E}x_{43}$
Expansion valve 1	$\dot{m}_{43}h_{43} = \dot{m}_{44}h_{44}$	$\dot{E}x_{D,EXV1} = \dot{E}x_{44} - \dot{E}x_{43}$
Absorption refrigeration system		
Absorber	$\dot{Q}_{Abs} = \dot{m}_{25}h_{25} + \dot{m}_{34}h_{34} - \dot{m}_{30}h_{30} = \dot{m}_{37}(\dot{m}_{38} - \dot{m}_{37})$	$\dot{E}x_{Abs} = \dot{E}x_{34} + \dot{E}x_{30} + \dot{E}x_{37} - \dot{E}x_{25} - \dot{E}x_{38}$
Generator	$\dot{Q}_{gen} = \dot{m}_{27}h_{27} - \dot{m}_{28}h_{28} - \dot{m}_{31}h_{31} = \dot{m}_{40}(\dot{m}_{40} - \dot{m}_{41})$	$\dot{E}x_{gen} = \dot{E}x_{40} + \dot{E}x_{27} - \dot{E}x_{28} - \dot{E}x_{31} - \dot{E}x_{41}$
Condenser	$\dot{Q}_{cond} = \dot{m}_{31}(\dot{m}_{31} - \dot{m}_{32}) = \dot{m}_{35}(\dot{m}_{36} - \dot{m}_{35})$	$\dot{E}x_{cond} = \dot{E}x_{31} + \dot{E}x_{35} - \dot{E}x_{32} - \dot{E}x_{36}$
SHX	$\dot{Q}_{SHX} = \dot{m}_{26}(\dot{m}_{27} - \dot{m}_{26}) = \dot{m}_{28}(\dot{m}_{29} - \dot{m}_{28})$	$\dot{E}x_{SHX} = \dot{E}x_{26} + \dot{E}x_{28} - \dot{E}x_{27} - \dot{E}x_{29}$
Expansion valve 2	$\dot{m}_{29}h_{29} = \dot{m}_{30}h_{30}$	$\dot{E}x_{D,EXV2} = \dot{E}x_{29} - \dot{E}x_{30}$
Pump 4	$\dot{W}_{pump4} = \dot{m}_{25}(h_2 - h_{25})$	$\dot{E}x_{D,pump4} = \dot{W}_{pump4} - \dot{E}x_{26} + \dot{E}x_{25}$
Expansion valve 3	$\dot{m}_{32}h_{32} = \dot{m}_{33}h_{33}$	$\dot{E}x_{D,EXV3} = \dot{E}x_{32} - \dot{E}x_{33}$
RO	$\dot{W}_{RO} = \dot{m}_{49}h_{49} - \dot{m}_{50}h_{50} - \dot{m}_{51}h_{51}$	$\dot{E}x_{D,RO} = \dot{E}x_{49} - \dot{E}x_{50} - \dot{E}x_{51}$
PEM	$\dot{W}_{PEM} = \dot{m}_{52}h_{52} - \dot{m}_{53}h_{53} - \dot{m}_{54}h_{54}$	$\dot{E}x_{D,PEM} = \dot{E}x_{52} + \dot{W}_{PEM} - \dot{E}x_{53} - \dot{E}x_{54}$

Table 2. Transcritical cycle validation with the results of Wu et al.'s work [22].

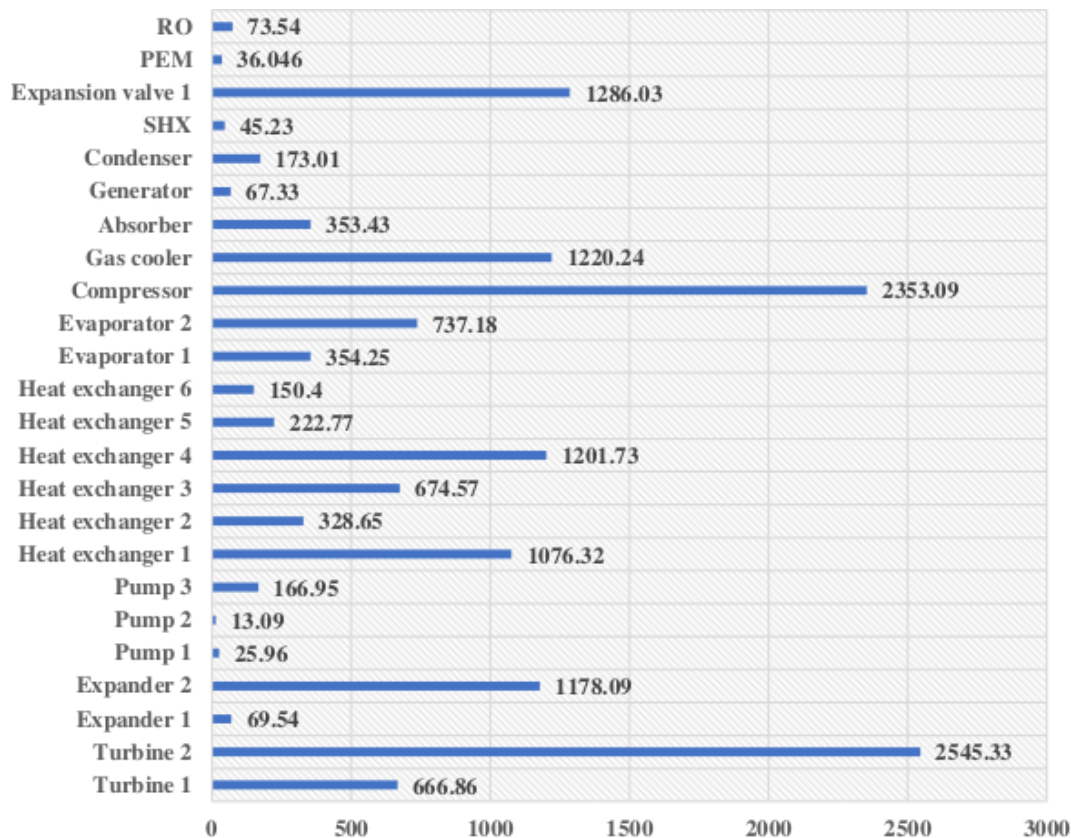
Parameter	Present work	Ref. [22]
$\dot{Q}_{LTR}$ (kW)	453.83	436.36
$\dot{Q}_{HTR}$ (kW)	571.67	571.4
$\dot{W}_{mc}$ (kW)	76.13	76.09
$\dot{W}_{RC}$ (kW)	73.28	73.53
$\dot{W}_{turb}$ (kW)	393.18	393.24

The exergy destruction rates of key system components are shown in Figure 2. Turbine 2 had the highest rate at 2545.33 kW, followed by Turbine 1 at 666.86 kW. Expander 2 exceeded Expander 1 by 69.54

kW, with rates of 1178.09 kW and 1108.55 kW, respectively. Pump 3 had the highest destruction at 166.95 kW, while Pump 2 had the lowest at 13.09 kW.

Heat exchanger units showed varied rates, with Heat Exchanger 4 at 1201.73 kW, Heat Exchanger 1 at 1076.32 kW, and Heat Exchanger 3 at 674.57 kW. The gas cooler (1220.24 kW) and compressor (2353.09 kW) were the largest contributors to exergy destruction. Evaporator 2 (737.18 kW) surpassed Evaporator 1 (354.25 kW), and the condenser had 173.01 kW.

The total system exergy destruction was 17,373.64 kW, with turbines, compressor, expansion valve, and heat exchangers identified as the main areas for potential improvement.

**Fig. 2.** Exergy destruction rate of all the component of the studied system.

There is a direct relationship between the geothermal source temperature and performance characteristics, including both power output and freshwater production as presented in Figure 3. A dramatic increase in system output is observed as the geothermal source temperature rises from 380 K to 415 K. Specifically, the power output of the Kalina cycle turbine increases from 1,885.71 kW to 10,286.2 kW, significantly enhanc-

ing the operational capacity of the Reverse Osmosis (RO) unit. This increase in turbine power yields a corresponding increase in the capacity of freshwater generation capacity, from 5.553 kg/s to 12.966 kg/s. Such numbers emphasize the significance of geothermal source temperature for system efficiency optimization. Higher temperatures not only enhance the Kalina's cycle thermodynamic performance, but also increase the

available mechanical power for the RO unit. As the turbine output increases, a corresponding increase in the RO unit's freshwater production capacity is observed, proving the effectiveness of the integrated system at producing energy and water at maximum efficiency. The direct relationship between the quality of the geothermal energy and system output indicates the importance of maximizing geothermal source temperatures to achieve optimal performance in both energy generation and freshwater production.

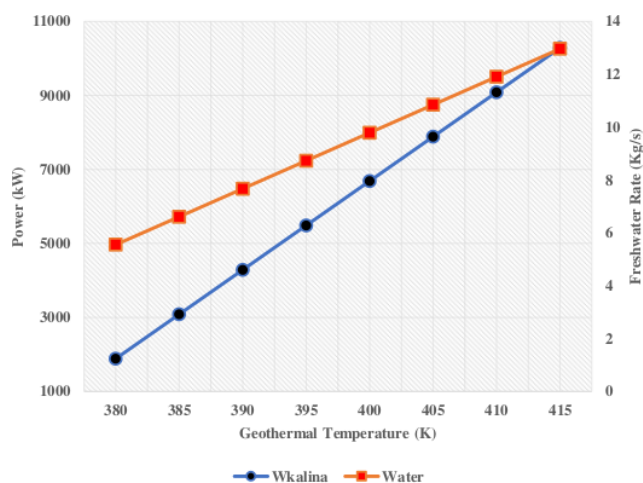


Fig. 3. The effect of geothermal temperature on the amount of power generation in Kalina cycle and freshwater production rate.

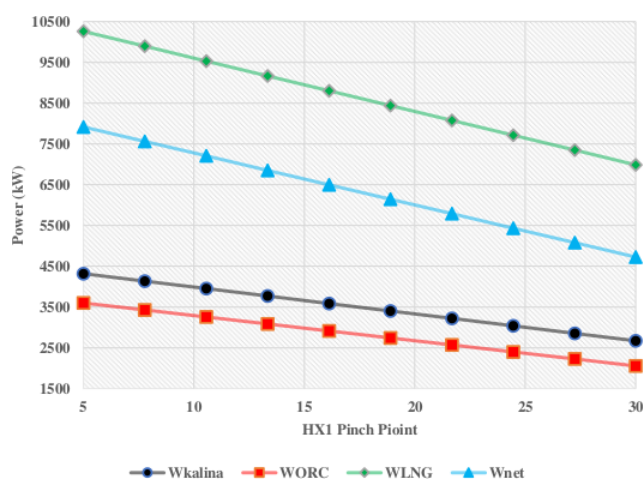


Fig. 4. The effect of heat exchanger 1 pinch point on the amount of power generation in the system.

Thermo-energy is absorbed by the working fluid in Kalina cycle from the geothermal fluid through Heat Exchanger 1. From performance data in Figure 4, significant variations in power output throughout the system are established with the variation of the pinch temperature difference ($\Delta T_{\text{pinchHX1}}$) in Heat Exchanger 1.

With increase in the pinch temperature difference from 5 °C to 30 °C, the power outputs realized from all subsystems are significantly reduced. The ORC subsystem loses 670.74 kW, the LNG subsystem loses 3598 kW and the power output of the Kalina cycle falls by 4317.62 kW. This overall decrease from 7915.6 kW to 4723.66 kW is mainly caused by diminishing rates of working fluid flows. Despite staying at the same components, the efficiency of Turbine 1 and Expander 1 are adversely affected by increasing in pinch temperature difference. The cause of energy losses at reduced flow rates is system wide and does not vary across the ORC subsystem which reflects a limitation inherent to the entire system. In the LNG subsystem, the major reason for the decreased power output from Turbine 3 is the reduction of flow rates although the enthalpy difference records a slight increment. This outcome draws attention to the importance of sustaining appropriate condition matching for Heat Exchanger 1. As the pinch temperature gap increases the efficiency of the system decreases, which is evidence of the fact that accurate thermal administration is necessary for maximizing the overall potency creation and sustaining system performance in individual complexes.

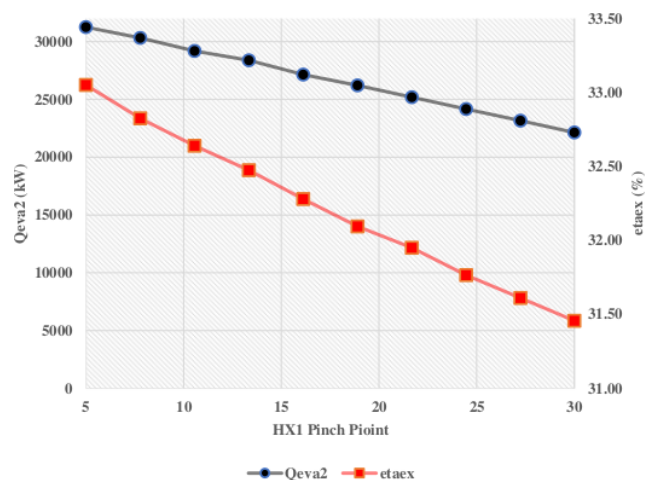


Fig. 5. The effect of heat exchanger 1 pinch point on the amount of cooling output rate and exergy efficiency of the system.

The effect of the changes in Heat Exchanger 1's pinch temperature difference ($\Delta T_{\text{pinchHX1}}$) on the output and system efficiency in multigeneration cooling can be reviewed in the figure below (Figure 5). With the variation in pinch temperature difference of 5 °C and 30 °C, both the system output and exergy efficiency report substantial reductions. In particular, the amount of cooling output falls by 8240.36 kW- from 30,373.2 to 22,132.84 kW- while exergy efficiency rates reduce by 1.59% from 33. Such reductions demonstrate the pivotal role of thermal performance of Heat

Exchanger 1 in maximizing system performance. The lower working fluid flow rate in the ORC subsystem, resulting from larger pinch temperature differences, is the primary cause of the system's decreased performance. This leads to reduced power output and a reduced cooling output as well. Although higher pinch temperature differences are standard for exergy conservation in the geothermal and natural gas subsystems, they are counterproductive at elevated levels. In particular, both cooling output and power generation are dropped, thus decreasing the total system efficiency. Thus a low pinch temperature difference for Heat exchanger1 is recommended to enable maximum optimization where maximum cooling output is realized and maximum exergy efficiency attained.

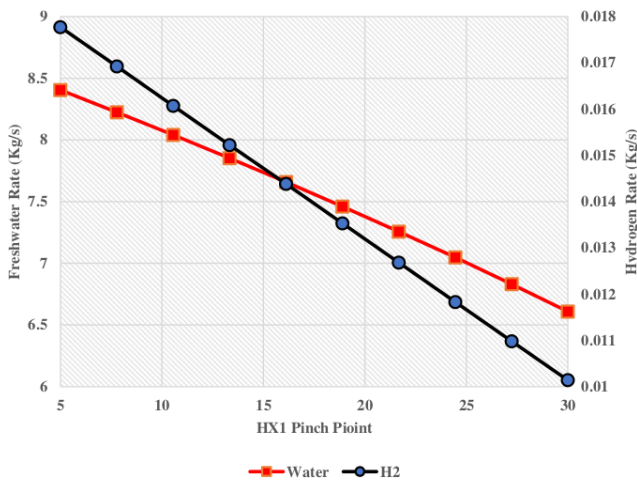


Fig. 6. The effect of heat exchanger 1 pinch point on the amount of hydrogen and freshwater production rates.

As shown in Figure 6, freshwater and hydrogen production rates resulting from Heat Exchanger 1's pinch temperature difference ($\Delta T_{\text{pinchHX1}}$) are illustrated. With an increase in pinch temperature difference from 5 °C to 30 °C, both production rates decline significantly. Specifically, hydrogen production falls from 0.01777 kg/s to 0.01014 kg/s and the freshwater decreases from 8.404 kg/s to 6.609 kg/s. These reductions are mostly as a result of direct effect of the differences in the pinch temperature on important system components. Freshwater output is particularly important and the cycle turbine of the Kalina drives the reverse osmosis (RO) unit. With increasing differences of pinch temperature, turbine output reduces directly reducing the RO capacity. Additionally, the PEM electrolyzer whose heat requirement is supplied for by the Kalina cycle through Heat Exchanger 3 experiences the impact of higher pinch temperature. This leads to lower energy available for the ORC subsystem, leading to a reduction in turbine power output

and the electricity available for hydrogen production. Such changes reveal how thermal interferences to the pinch temperature difference throw off the system's energy distribution. By decreasing the power applied to a turbine and the heat supplied to critical subsystems, greater differences in temperatures at the pinch lead to lower hydrogen and freshwater generation. Optimization of the rates of production requires that pinch temperature differences are carefully managed to maintain the efficiency of the system over all the subsystems.

The maximum pressure in the Kalina cycle is a major determinant influencing the production rates of both hydrogen and freshwater. When the pressure in the Kalina cycle increases from 2500 kPa to 5000 kPa, power generation initially rises, with hydrogen production increasing from 0.0091 kg/s to 0.0234 kg/s and ORC turbine power output rising from 1,850 kW. Nonetheless, with further increasing pressure, especially above 5000kPa, performance shows a decreasing trend from the consequence of decreased flow rate of turbine one. Consequently, both ORC power generation and freshwater production decrease. At 3750 kPa, the system records its maximum rate of freshwater production of 7.392 kg/s. However, when the pressure increases to 5000 kPa, freshwater production is modified to 6.75 kg/s. Although LNG subsystem works at best at higher pressures, its higher operational pressure reduces system efficiency even though freshwater output improves marginally. These results emphasize the need to optimize system pressure in order to attain a balanced performance. Determining the best pressure value is a must to achieve simultaneous production of Hydrogen and freshwater with the highest efficiency of possible system.

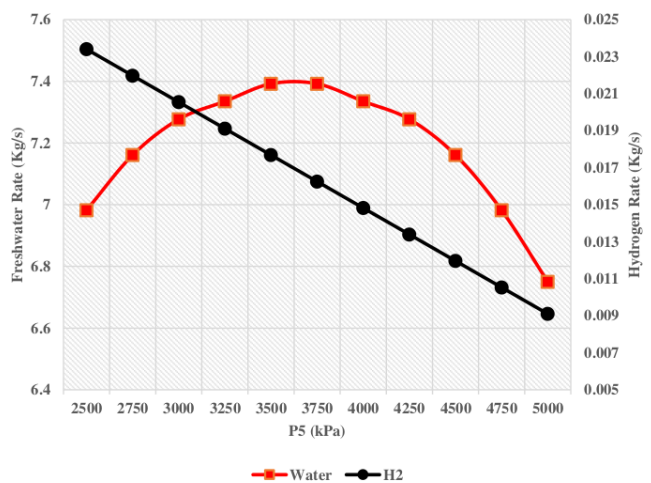


Fig. 7. The effect of Kalina cycle high pressure on the amount of hydrogen and freshwater production rates.

As Figure 8 illustrates, the coefficient of perfor-

mance (COP) for the single-effect absorption system rises rapidly with increasing generator temperature to reach maximum values at 348 K before dropping at further temperatures. Namely, the COP increases a lot from 303 K to 348 K attaining the maximum level of 0.87. However, with increasing generator temperature, the COP begins to decline to 0.77 at 383 K. This pattern illustrates the impact of generator temperature on system performance, where the coefficient of performance (COP) improves at lower temperatures but deteriorates as temperatures rise. Similar trends are found when the radiant temperature of the absorber is varied over different absorber temperature ranges with a lower maximal COP achieved. At an absorber temperature (T_{abs}) of 308 K, the system achieves its maximum COP of 0.77 at a generator temperature of 358 K; however, this value decreases to 0.73 when the generator temperature rises to 383 K. At T_{abs} equal to 313 K, the system's peak COP attains its maximum at 368 K (0.70), then decreases to 0.67 at 383 K. Most of the observed reduction in COP with increased absorber temperatures arises from increased irreversibilities in the process as generator temperatures increases. These irreversibilities cause inefficiency that derails total system performance. Interestingly, a reduced solution circulation rate increases the COP on a temporary basis since the heating demands over the three pressure levels are lowered. Nonetheless, this superiority is undone by the demerits of an increased temperature of the generator, which reduces COP.

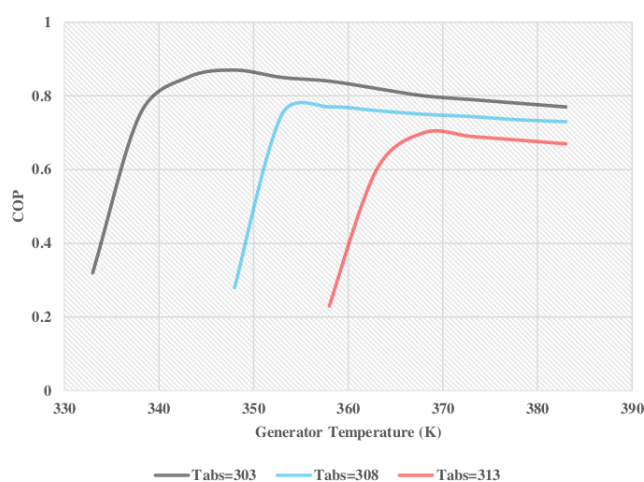


Fig. 8. Variation of COP with generator temperature at different absorber temperatures

4 Conclusions

This study proves the viability of a new multigeneration system based on geothermal energy in conjunction with Kalina and Organic Rankine cycles accompa-

nied by novel refrigeration and desalination technologies. Such a unique composition enables the generation of electricity, cooling, hydrogen, and fresh water, and demonstrates the versatility of the geothermal energy and its various applications. The system shows excellent thermodynamic potential with exergy efficiencies of 32.04% and 11.65% with a total exergy waste of 24,401.63 kW. The major inefficiencies are produced by Turbine 2 and its related compression systems. In future research, these components should be optimized to improve the overall system performance. In addition, further investigation is needed to support the scalability and practicality of this system in the real world, so that it may make its contribution to sustainable energy and water solutions in resource-rich areas. With ongoing research and development, this multigeneration approach holds great promise as a key contributor to addressing global challenges in energy efficiency and water sustainability.

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