Proposal and comprehensive thermodynamic performance analysis of a new geothermal combined cooling, heating and power system

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Abstract: This study proposes a new combined cooling, heating and power (CCHP) system applied for geothermal energy. In the proposed system, a water heater is introduced to utilize the exhaust geothermal water for supplying heating, and the throttled valve of the basic flash cycle is replaced by an ejector for cooling output. To evaluate the system performance, detailed mathematic models are established and validated. Based on the basic condition (170°C geothermal water), the exergy efficiency of the proposed system is 44.34% and the sum of the power, cooling output and heating output is 10,283.68 kW. Also, the exergy loss analysis demonstrates that the component of the water heater, ejector, turbine, condenser and flasher has a large improved space to reduce the exergy destructions for system performance enhancement. At last, the parametric analysis results suggest that there exists an optimal flash pressure (around 140 kPa) to maximize the exergy efficiency. Within the discussed ranges, both the increase of the outlet temperature of water heater and condenser temperature will damage the system exergy efficiency, while a higher evaporator temperature is beneficial for the system exergy efficiency.

Subjects: Renewable Energy; Engineering Economics

About the Author
Fei Yan received his Doctor degree in Power Engineering and Engineering Thermal Physics from Chongqing University. He is currently working as a research fellow and professional degree postgraduate administrator at Chongqing University. His research interests include energy conversion, utilization and storage system modelling, molecular simulation and bioinformatics. This paper is very useful in the application of geothermal energy utilization, which proposed a novel combined cooling, heating and power system based on the flash cycle. An ejector is employed for the flash cycle to reduce the pressure throttling loss and drive the ERC to produce cooling output. The proposed system has been shown to produce considerable cooling, heating and power outputs when driven by the 170°C geothermal water.

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Geothermal energy attracts lots of interest from researchers due to its advantages of large reserves and wide distribution. Commonly, geothermal energy with medium-and-high temperatures is often applied for industrial purposes. In this regard, this study aims to develop a novel energy system, which can supply power, cooling water and domestic hot water synchronously. The design purpose of proposed system has been proved according to the results of the discussion. And we hope that this research is expected to further expand the research on geothermal energy utilization methods.
**Keywords:** Combined cooling; heating and power system; geothermal energy; flasher cycle; ejector refrigeration cycle; thermodynamic and exergy analysis

1. **Introduction**

For decades, the massive use of fossil fuels brought the environmental and energy crisis to the world (Chaurasiya et al., 2021; Choudhari et al., 2021; Dasore et al., 2022; Thakur et al., 2021). In this regard, renewable energy has drawn great deals of interest (Agrawal et al., 2022; Dhanraj et al., 2021; Omar et al., 2019; J Tang et al., 2022). Geothermal energy, with a wide temperature range varying from 50 to 350°C, receives much attention among many clean and sustainable energy sources due to its large reserves and wide distribution. Generally, the lower temperature geothermal energy is used directly, for heating and bathing, while geothermal energy with medium-/high-temperature is often applied for industrial purposes, like power generation (Li et al., 2012; Parham et al., 2014; Singh et al., 2021).

Many efforts have been denoted to develop the technologies of geothermal power generation. Because of the features of simple structure, high reliability, the organic Rankine cycle (ORC) is recommended for the application of low-grade heat sources (Wang et al., 2018; Zhang et al., 2017), thus widely used for geothermal energy (Heberle & Brüggemann, 2010, 2015; Kazemi & Samadi, 2016). However, there exists large unavoidable exergy destruction when using the ORC system, which is mainly contributed by the large temperature differences during the heat absorption process of working (Wang et al., 2019; Zhou et al., 2019). In this regard, as a promising option to overcome this problem, the flash cycle has attracted lots of attention as well. Up till now, many studies are carried out on the geothermal single-flash power plant (Guzovic et al., 2012; Jalilinasrabady et al., 2012; Ozcan & Gokcen, 2009; Pambudi et al., 2014). Moreover, the single-flash power cycle still has a large room to improve, which is caused by the relatively higher temperature and pressure reinjected geofluid discharged from the flasher.

Accordingly, it is a good choice to employ some other cycle suitable for low-grade heat recovery to reuse this part of heat to propose a combined system based on the flash cycle (i.e flash-binary system), then improve the overall energy conversion efficiency. Yari (Yari, 2010) proposed a flash-ORC power plant driven by geothermal and compared its performance with other geothermal power plants. He found that the flash-ORC system with R123 owned the maximum first-law efficiency. Later, the exergoeconomic analyses were conducted for the flash-ORC as well (Shokati et al., 2015; Zhao & Wang, 2016). Meanwhile, the Kalina and transcritical CO2 cycles are also used to integrate with the flash cycle. A comprehensive thermodynamic and optimization analysis for a geothermal flash-Kalina system was conducted by Wang et al. (Wang et al., 2015). Abdolalipouradl et al. (Abdolalipouradl et al., 2019) proposed a novel system that couples transcritical CO2 and Kalina cycles with a flash cycle simultaneously. Besides combined power systems, much attention is also sight on the multi-generation systems driven by geothermal energy, like combined cooling and power (CCP) system, combined cooling, heating and power (CCHP) system, which not only can own a high efficiency but also meet the diverse energy demands. The combination of the absorption refrigeration cycle (ARC) and the flash cycle was studied by Wang et al. from a detailed thermodynamic analysis (Wang et al., 2020). Then, the proposed system was taken some modifications in another research (Ren et al., 2021). Compared to the other four flash cycles, they found that their proposed system showed the best system performance. Nami and Alviri-Moghadam (Nami & Alviri-Moghadam, 2020) evaluated a novel geothermal driven CCHP system through the exergy, economic and sustainability analysis.

The above literature reviews indicate that the flash cycle is an effective method to utilize geothermal energy. Meanwhile, the necessity and superiority of the combined system in geothermal energy utilization are proved as well. However, most of the proposed geothermal systems are
coupled with multiple single subsystems, which also means more equipment and complex structures, causing large heat loss and high investment. Moreover, it should be noted that there exist huge throttling losses caused by the valve in a flash cycle (Z Tang et al., 2021; Wu et al., 2021). In other words, reducing the throttling losses also can improve the overall system performance. To achieve that, it is a good method to use the ejector instead of the valve. The ejector is a simple device capable of restoring partially destroyed exergy. Meanwhile, it is also an important device in the ejector refrigeration cycle (ERC), which is suitable for geothermal water to produce the cooling output (Rostamnejad & Zare, 2019; Takleh & Zare, 2021; Zare & Rostamnejad Takleh, 2020). Consequently, it is of great potential to employ the ERC subsystem to form a new geothermal CCHP system, owning a high energy efficiency and supplying diverse energy simultaneously.

Thus, this paper performs a new geothermal CCHP system, which consists of the flash cycle and ERC. In the proposed system, the water heater is used for precooling the high temperature geothermal water exiting the flasher to produce heat water; and the ejector is employed to reduce the pressure throttling loss and drive the ERC to produce cooling output. The detailed thermodynamic models are established at first. Then, the thermodynamic performance and exergy loss distribution are obtained from the energy and exergy analysis under the preliminary design condition. Finally, the proposed system is studied through comprehensive parametric analyses to explore the effects of key parameters on system performance. This research is expected to further expand the research on geothermal energy utilization methods. The main innovations and contributions of this work are as follows:

1. To develop a novel CCHP system based on the single-flash cycle with a simple structure without extra flash separators and turbines.

2. The ejector is employed to reduce the pressure throttling loss and drive the ERC to produce cooling output.

3. To conduct comprehensive energy and exergy analysis for the proposed system.

2. System descriptions

Figures 1 and 2 present the schematic diagram of the basic flash and proposed CCHP systems. Compared to the basic flash cycle (Figure 1), the proposed system can both recover the waste heat and the pressure throttling loss of the single-flash cycle to improve the system performance. In the proposed system, the extracted geothermal water is delivered to the flasher, where the pressure of the geofluid decreases with a constant enthalpy, forming a two-phase flow. Then, it is separated
into steam (stream 2) and liquid (stream 4). The steam enters the turbine to produce power (state 2–3). Meanwhile, the liquid flows through the water heater for producing hot water and enters the ejector to drive the refrigeration cycle (state 4–6). In the condenser, stream 6 in company with stream 3 is condensed to the saturated liquid (state 7–8). After condensing, the liquid is divided into two streams as well. One stream flows into the valve with an expansive process, reaching the evaporation temperature. Then, stream 9 goes into the evaporator to produce cooling output and enters the ejector finally (state 8–10), while another stream is pumped to the injection well (state 11).

In this study, the following presumptions are made for the simplification of simulation.

(1) The entire system operates at a steady state;
(2) No pressure losses and heat transfer between the system and environment;
(3) The vapor and the liquid leaving the flasher are assumed as saturated (Shokati et al., 2015; Zhao & Wang, 2016);
(4) The existing streams of the condenser and evaporator are assumed as saturated (Ipakchi et al., 2019; Moghimi et al., 2018; Rostamnejad Takleh & Zare, 2019).
(5) The constant-pressuring mixing model is selected for the ejector modeling (Mosaffa & Farshi, 2018; Park et al., 2018).
3. System modeling

3.1. Ejector modeling

As shown in Figure 3, the ejector consists of suction, mixing, constant-area and diffuser sections. As said before, this paper adopts the constant-pressure mixing model for ejector modeling (Huang et al., 1999; Yen et al., 2013). Moreover, in the ejector model, the losses of fluid in the ejector are considered by using the nozzle efficiency ($\eta_n$), the mixing efficiency ($\eta_m$) and the diffuser efficiency ($\eta_d$). The fluid velocity at the inlet of the nozzle and the outlet of the diffuser chamber are ignored (Mosaffa & Farshi, 2018; Park et al., 2018).

The entrainment ratio ($\mu$) and nozzle, mixing and diffuser efficiencies are defined as,

$$\mu = \frac{m_{sf}}{m_{pf}}$$  \hspace{1cm} (1)

$$\eta_n = \frac{h_{pf,n1} - h_{pf,n2}}{h_{pf,n1} - h_{pf,n2,s}}$$  \hspace{1cm} (2)

$$\eta_m = \frac{u_{mf,m}}{u_{mf,m,s}}$$  \hspace{1cm} (3)

$$\eta_d = \frac{h_{mf,d,s} - h_{mf,d}}{h_{mf,d} - h_{mf,m}}$$  \hspace{1cm} (4)

Figure 4 shows the flowchart of ejector modeling and the governing equations. To calculate the entrainment ratio needs an iterative process, and the iteration ends when the convergence condition is satisfied.

3.2. Energy and exergy analysis

For a thermodynamic system, the equations based on the first-law thermodynamics are written as,

$$\sum m_{in} = \sum m_{out}$$  \hspace{1cm} (5)

$$\sum Q_{cv} - \sum W_{cv} = \sum m_{out}(h_{out}) - \sum m_{in}(h_{in})$$  \hspace{1cm} (6)
m presents the mass flow; W is the work; Q and h denote the work and specific enthalpy.

Except for the energy analysis, the exergy analysis is conducted as well. Following is the exergy balance equation of each component,

$$\sum E_Q - W = \sum E_{\text{out}} - \sum E_{\text{in}} + E_D$$ (7)

It should be noted that only the physics exergy term is considered here (Zare & Rostamnejad Takleh, 2020). Thus, the calculation equation is,

$$E_i = E_{\text{ph}} = m[(h_i - h_0) - T_0(s_i - s_0)]$$ (8)

$h_0$ and $s_0$ are the enthalpy and entropy at the reference state.
Table 1. Energy and exergy relations for the proposed geothermal CHP system

| Component     | Energy relations                                                                 | Exergy destruction relations |
|---------------|----------------------------------------------------------------------------------|-----------------------------|
| Flasher       | $m_1 h_1 = m_2 h_2 + m_3 h_4$                                                   | $E_{D,fs} = E_1 - E_2 - E_4$ |
| Turbine       | $W_{tur} = m_3 (h_2 - h_3)$                                                      | $E_{D,tur} = E_2 - E_3 - W_{tur}$ |
| Water heater  | $Q_{heat} = m_4 (h_4 - h_5) = m_5 (h_6 - h_1)$                                 | $E_{D,wh} = (E_4 - E_5) - (E_6 - E_1)$ |
| Ejector       | $m_1 h_6 = m_0 h_10 + m_1 h_1$                                                   | $E_{D,ej} = E_10 + E_2 - E_6$ |
| Mixer         | $m_1 h_6 = m_1 h_3 + m_2 h_5$                                                   | $E_{D,mix} = E_1 + E_4 - E_3$ |
| Condenser     | $Q_{con} = m_7 (h_7 - h_8) = m_8 (h_9 - h_3)$                                  | $E_{D,con} = E_3 - E_4$ |
| Valve         | $h_{in} = h_9$                                                                  | $E_{D,val} = E_{in} - E_3$ |
| Evaporator    | $Q_{eva} = m_3 (h_10 - h_9) = m_3 (h_11 - h_12)$                               | $E_{D,eva} = (E_10 - E_9) - (E_{12} - E_{11})$ |
| Pump          | $W_p = m_{in} (h_{11} - h_{in})$                                                 | $E_{D,p} = W_p - (E_{11} - E_{in})$ |

Table 2. Single-flash cycle model validation between present work and literature (Yari, 2010)

| Points | Present work | Ref (Yari, 2010). | RD (%) | Present work | Ref (Yari, 2010). | RD (%) | Present work | Ref (Yari, 2010). | RD (%) | Present work | Ref (Yari, 2010). | RD (%) |
|--------|--------------|-------------------|--------|--------------|-------------------|--------|--------------|-------------------|--------|--------------|-------------------|--------|
| 1      | 230.00       | 230.00            | 0      | 2797.09      | 2795              | 0.07   | 990.19       | 990.00            | 0.02   | 1.0000       | 1.0000            | 0      |
| 2      | 162.97       | 163.00            | -0.01  | 666.50       | 666.5             | 0      | 2760.67      | 2761.00           | -0.01  | 0.1456       | 0.1454            | 0.14   |
| 3      | 98.58        | 98.58             | 0      | 96.40        | 96.4              | 0      | 2531.10      | 2531.00           | 0.00   | 0.1456       | 0.1454            | 0.14   |
| 4      | 162.97       | 163.00            | -0.01  | 38.59        | 38.56             | 0.07   | 688.43       | 688.70            | -0.04  | 0.8544       | 0.8546            | -0.02  |

Table 3. ERC model validation between present work and literature (Yen et al., 2013)

| Items | $T_{gen}{^\circ}C$ | $T_{eva}{^\circ}C$ | $T_{con}{^\circ}C$ | COP | Experimental results (Yen et al., 2013) | RD/ % |
|-------|-------------------|-------------------|-------------------|-----|---------------------------------------|-------|
| $T_{gen}$ | $T_{eva}$ | $T_{con}$ | Present work | | Experimental results (Yen et al., 2013) | |
| 90    | 12    | 35.0   | 0.278 | 0.27     | 2.96 |
| 90    | 15    | 33.6   | 0.377 | 0.38     | 0.78 |
| 90    | 20    | 35.5   | 0.458 | 0.44     | 4.09 |
| 100   | 12    | 34.2   | 0.333 | 0.32     | 4.06 |
| 100   | 15    | 34.5   | 0.391 | 0.37     | 5.67 |
| 100   | 20    | 34.8   | 0.536 | 0.53     | 1.13 |
Moreover, the energy and exergy relations of the proposed system based on the above equations are summarized in Table 1.

Accordingly, to evaluate the system performance, thermal and exergy efficiencies are selected, which are defined as,

\[ \eta_{th} = \frac{W_{net} + Q_{cool} + Q_{heat}}{Q_{in}} \tag{9} \]

\[ \eta_{ex} = \frac{W_{net} + E_{cool} + E_{heat}}{E_{in}} \tag{10} \]

In Eq. (9) and Eq. (10), \( Q_{cool} \) and \( Q_{heat} \) denote the cooling capacity and heating capacity of the system. \( W_{net} \) presents the net power output. \( Q_{in} \) is the energy input for the system. \( E_{cool}, E_{heat} \) and \( E_{in} \) denote the cooling exergy, heating exergy and exergy input (Ren et al., 2021; Wang et al., 2020).

\[ W_{net} = W_f - W_p \tag{11} \]

\[ E_{cool} = E_{c2} - E_{c1} \tag{12} \]

\[ E_{heat} = E_{h2} - E_{h1} \tag{13} \]

\[ E_{in} = E_1 \tag{14} \]

3.3. Model validation
To achieve the model validation, this section compares the value obtained from mathematical models and open literature. Thus, the model of the flash cycle and ERC cycle are verified. The open data from the literature (Yari, 2010) and literature (Yen et al., 2013) are used for the flash system (Table 2) and the ERC system (Table 3), respectively. It can be seen that the results obtained from the different works have an acceptable relative error (RD), revealing that established models are of good accuracy.

4. Results and discussions
In section 3, the detailed mathematic models of the proposed system are established. This section performs a comprehensive thermodynamic performance analysis. The thermodynamic properties of working fluids can be obtained from REFPROP software provided by NIST (Lemmon et al., 2010). REFPROP is based on the most accurate pure fluid and mixture models currently available. It implements three models for the thermodynamic properties of pure fluids: equations of state explicit in Helmholtz energy, the modified Benedict-Webb-Rubin equation of state, and an extended corresponding state (ECS) model.

4.1. Preliminary analysis
To explore the feasibility of the proposed system, this section calculates the system performance under the preliminary design condition. Table 4 lists the specific setting conditions. Accordingly, the state of each point and system performance are summarized in Tables 5 and 6. Referring to Table 6, the proposed system could produce a power output of 1331.37 kW when it gets 18,390.85 kW energy input from the geothermal energy. Meanwhile, the proposed system obtains extra 5498.14 kW cooling output and 3454.17 kW heating output. Consequently, the thermal and exergy efficiencies of the system are 55.92% and 44.34%, respectively.
4.2. Exergy loss analysis

To show the exergy loss distribution of the proposed system, this section carries out an exergy loss analysis under the preliminary condition. Referring to Figure 5, the water heater has the largest exergy destruction accounting for 24.76% of the total exergy destruction ($E_{d,\text{tot}}$), because of the large temperature difference and mass flow rate in the heat exchanger. Similarly, the condenser has large exergy destruction as well, occupying the 15.53% of $E_{d,\text{tot}}$. Moreover, the high exergy destructions occurred in the ejector (19.71% of $E_{d,\text{tot}}$) and flasher (14.41% of $E_{d,\text{tot}}$) indicate that components used to separate or mix fluids are prone to exergy damage. Besides, the huge mass flow rate and the expansion process of the turbine make its

| Points | $T$ (°C) | $P$ (kPa) | $h$ (kJ/kg) | $s$ (kJ/kg·K$^{-1}$) | $m$ (kg/s) |
|--------|---------|---------|----------|----------------|-----------|
| 1      | 170.00  | 900.00  | 719.14   | 2.04           | 30.00     |
| 2      | 120.21  | 200.00  | 2706.23  | 7.13           | 2.92      |
| 3      | 25.00   | 3.17    | 2237.43  | 7.52           | 2.92      |
| 4      | 120.21  | 200.00  | 504.70   | 1.53           | 27.08     |
| 5      | 90.00   | 200.00  | 377.14   | 1.19           | 27.08     |
| 6      | 25.00   | 3.17    | 543.31   | 1.83           | 29.36     |
| 7      | 25.00   | 3.17    | 696.68   | 2.35           | 32.28     |
| 8      | 25.00   | 3.17    | 104.83   | 0.37           | 32.27     |
| 9      | 10.00   | 1.23    | 104.83   | 0.37           | 2.28      |
| 10     | 10.00   | 1.23    | 2519.21  | 8.89           | 2.28      |
| 11     | 25.11   | 900.00  | 106.11   | 0.37           | 30.00     |

5. Conclusions

In this study, the CCHP system was designed and optimized to achieve the best performance under the preliminarily set conditions. The geothermal water has the highest temperature, which can be used as a renewable energy source for the proposed CCHP system. The system was designed to have a total exergy efficiency of 48.89%, with a high exergy efficiency of the turbine (80.34%) and the water heater (84.25%) contributing to this value. The exergy analysis showed that the components prone to exergy damage are the water heater, condenser, ejector, and flasher. The high exergy destruction in the water heater and condenser indicates that the temperature difference and mass flow rate are the most significant factors affecting the exergy destruction in these components. The high exergy destruction in the ejector and flasher indicates that the components used to separate or mix fluids are prone to exergy damage. The results of this study can be used to optimize the design of the CCHP system and improve its performance.

| Parameters | Value |
|------------|-------|
| Geothermal water temperature (°C) | 170   |
| Geothermal water pressure (kPa) | 900   |
| Geothermal water mass flow rate (kg/s) | 30    |
| Ambient temperature (°C) | 20    |
| Ambient pressure (kPa) | 101.325 |
| Domestic hot water temperature (°C) | 70    |
| Heater outlet temperature (°C) | 90    |
| Flasher pressure (kPa) | 200   |
| Evaporation temperature (°C) | 5     |
| Condenser temperature (°C) | 25    |
| Pump isentropic efficiency (%) | 70    |
| Turbine isentropic efficiency (%) | 80    |
| Nozzle efficiency (%) | 85    |
| Mixing efficiency (%) | 95    |
| Diffuser efficiency (%) | 85    |

Table 4. The basic setting conditions for the proposed CCHP system (Z Tang et al., 2021; Wang et al., 2020; Zare & Rostamnejad Takleh, 2020)

Table 5. Thermodynamic properties of each point under basic condition
exergy destructions reach 16.32% of $E_{d, tot}$. Accordingly, more efforts should be denoted on the above components to reduce exergy losses. There is little space to reduce these exergy losses, as the remaining components have far fewer exergy losses than the aforementioned components.

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Table 6. Thermodynamic performance of the proposed geothermal CCHP system

| Items                        | Valve       |
|------------------------------|-------------|
| $W_{\text{net}}$ (kW)        | 1369.91     |
| $W_p$ (kW)                   | 38.54       |
| $W_{\text{net}}$ (kW)        | 1331.37     |
| $Q_{\text{heat}}$ (kW)       | 3454.17     |
| $Q_{\text{cool}}$ (kW)       | 5498.14     |
| $Q_{\text{in}}$ (kW)         | 18,390.85   |
| $\eta_{\text{th}}$ (%)       | 55.92       |
| $\eta_{\text{ex}}$ (%)       | 44.34       |

Figure 5. Exergy losses analysis of the proposed geothermal CCHP system.
4.3. Thermodynamic parametric analysis

This section performs the parametric analysis to determine the effects of key parameters on system thermodynamic performance. Based on Figure 2, it should be noted that the flash pressure ($P_{fls}$) determines the thermodynamic states of the points at the turbine inlet and water heater inlet, which will affect the system power and heating outputs. The condenser temperature ($T_{con}$) affects the condenser pressure (i.e. the turbine back pressure) and then directly affects the turbine power output. The above two parameters mainly affect the power output, while the parameters that affect heating and cooling output also should be discussed. Thus, the evaporation temperature ($T_e$) and heater outlet temperature ($T_{wh,out}$) are selected as well. It should be noted that the other parameters keep unchanged when one parameter is discussed.

4.3.1. Effects of the flash pressure ($P_{fls}$)

Figure 6 shows the effects of the flash pressure ($P_{fls}$) on the proposed system performance. Referring to the figure, the exergy efficiency ($\eta_{ex}$) shows a parabolic trend during the discussed range of flash pressure (shown in Figure 6 (b)), and Figure 6 (a) can be used to explain this phenomenon. When $P_{fls}$ increases, the temperature at the flasher outlet increases ($T_{fls}$) as well (from 99.61°C to 133.52°C), then contributing to the increasing enthalpy drop of turbine. Meanwhile, the vapor exiting the flasher drops from 4.00 kg/s to 2.18 kg/s, which means the working fluid flowing into the turbine is reduced. The positive effect brought by the increasing enthalpy drop cannot offset the negative effect brought by the decreasing mass flow of vapor.
Consequently, the power output of turbine gets worse, then reducing the net power output of the system ($W_{\text{net}}$) (shown in Figure 6 (a)). However, the high $T_{\text{fls}}$ also means the high temperature of the stream flowing into the water heater, then leading to the increment of the heating output ($Q_{\text{heat}}$). Meanwhile, due to the more working fluids entering the evaporator caused by the large entrainment ratio ($\mu$), the cooling output ($Q_{\text{cool}}$) increases. According to the above discussion, it can be concluded that an increase $P_{\text{fls}}$ can improve $Q_{\text{heat}}$ and $Q_{\text{cool}}$ but decrease $W_{\text{net}}$. Hence, when the increase $Q_{\text{heat}}$ and $Q_{\text{cool}}$ dominate the change of $\eta_{\text{ex}}$, $\eta_{\text{ex}}$ increases; while when the decline of $W_{\text{net}}$ plays the domination role, $\eta_{\text{ex}}$ begins to decrease.

### 4.3.2. Effects of condenser temperature ($T_{\text{con}}$)

Figure 7 shows the variation trend of system performance under the change of condenser temperature ($T_{\text{con}}$). Referring to Figure 7, as $T_{\text{con}}$ rises, $\eta_{\text{ex}}$ decreases linearly due to the decline of $W_{\text{net}}$ and $Q_{\text{cool}}$. As shown in Figure 7 (a), since the temperature of the exiting flasher stream is not affected, $Q_{\text{heat}}$ keeps unchanged. When $T_{\text{con}}$ rises, the turbine back pressure increases ($T_{\text{fls}}$) as well, leading to a decreasing enthalpy drop of the turbine. Meanwhile, there is no change occurred in the mass flow rate of working entering the turbine, so the power output of the turbine decreases. Thus, $W_{\text{net}}$ shows the same downward trend. It also can be seen that the value of $\mu$ drops from 0.08 to 0.04 when $T_{\text{con}}$ arises (as shown in Figure 7 (b)). In other words, the mass flow rate of the
refrigerant stream flowing into the evaporator decreases, leading to a decline for \( Q_{\text{cool}} \). As a result, \( \eta_{\text{ex}} \) decreases with an increasing \( T_{\text{con}} \) under the combined effects of the decreasing \( W_{\text{net}} \) and \( Q_{\text{cool}} \).

4.3.3. Effects of the evaporator temperature (\( T_{e} \))

The effects of the evaporator temperature (\( T_{e} \)) on the proposed system performance are illustrated in Figure 8. It is worth noting that both \( W_{\text{net}} \) and \( Q_{\text{heat}} \) are kept constant when \( T_{e} \) rises 5 to 10°C, because the change of \( T_{e} \) does not affect the state point of power and heat subsystems. In other words, the change of \( T_{e} \) will bring a huge effect on the system cooling output. Referring to Figure 8 (b), a higher \( T_{e} \) brings a higher outlet pressure to the evaporator, leading to an increase in \( \mu \). An increasing \( \mu \) also improves the mass flow rate of the refrigerant stream entering the evaporator, which makes the \( Q_{\text{cool}} \) increase from 3522.00 kW to 5498.14 kW. As a result, the \( \eta_{\text{ex}} \) grows up with the increasing \( T_{e} \) due to the huge improvement of the \( Q_{\text{cool}} \).

4.3.4. Effects of outlet temperature of the water heater (\( T_{\text{wh,out}} \))

Figure 9 shows the effects of outlet temperature of the water heater (\( T_{\text{wh,out}} \)) on the proposed system performance. As can be seen from Figure 9 (a), the \( W_{\text{net}} \) is not affected by the change of \( T_{\text{wh,out}} \), due to the constant pressure drop of the turbine and mass flow rate entering the turbine. While \( Q_{\text{cool}} \) and \( Q_{\text{heat}} \) show opposite variation trends within the discussion range. When \( T_{\text{wh,out}} \) rises, the temperature drop in the water heater (i.e. \( T_{5}-T_{4} \)) will decrease. So the heat release in the heat exchanger also decreases because of the constant mass flow rate of the working fluid,
resulting in the reduction of $Q_{\text{heat}}$ (from). Meanwhile, the increase of $T_{\text{wh,out}}$ increases the value of $\mu$ (from 0.07 to 0.09) and then leads to more working fluid flows into the evaporator. Thus, there is an increment of $Q_{\text{cool}}$ with the change of $T_{\text{wh,out}}$. But the improvement of $Q_{\text{cool}}$ cannot play the leading role in the variation of $\eta_{\text{ex}}$, so it can be seen that $\eta_{\text{ex}}$ gets worse as $T_{\text{wh,out}}$ increases.

5. Conclusions
This study proposes a new geothermal energy system to supply power, cooling water and domestic hot water synchronously. In the proposed system, the flash cycle is used to generate power, and the water heater and ejector refrigeration cycle are used to reduce the exergy losses, and then produce heating and cooling outputs. The preliminary analysis is conducted to evaluate the system performance under the basic condition. Then, an exergy loss analysis is carried out to determine the exergy destruction distribution of the design condition. Finally, the effects of four key parameters on system performance are obtained with parameter analysis. The following are the main conclusions:

(1) The introduction of the ERC and water heater can not only improve the energy efficiency of the single-flash cycle but also meet diverse energy demands from users. When driven by the 170°C geothermal water, the proposed system owns the exergy efficiency with 44.34%.

(2) The exergy loss analysis shows that the largest exergy destruction occurs in the water heater with the proportion of 24.76% of the total system exergy destruction. Meanwhile,
the ejector, turbine, condenser and flasher also have high exergy destruction, occupying the proportion of 19.71%, 16.32%, 15.53% and 14.41%, respectively. More efforts should be denoted on the above components to reduce exergy losses.

(3) For the proposed system, there exists an optimal flash pressure to reach the maximum exergy efficiency. A higher evaporator temperature is beneficial for the system exergy efficiency. Meanwhile, the system exergy efficiency will get worse when the outlet temperature of water heater and condenser temperature increase.

The present work only studies proposed system performance from the thermodynamic aspect. In the future work, two main aspects could be studied. One is to study the performance of the system from more perspectives, such as economic and environmental aspects. Another is to develop more efficient energy systems driven by geothermal energy that can be proposed through highly efficient subcycles.

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Nomenclature

| Subscripts and abbreviations |
|------------------------------|
| $\varepsilon$ | exergy rate (kW) |
| $h$ | specific enthalpy (kJ·kg$^{-1}$) |
| $m$ | mass flow rate (kg·s$^{-1}$) |
| $p$ | pressure (kPa) |
| $q$ | heat transfer rate (kW) |
| $s$ | entropy (kJ·kg$^{-1}$·K$^{-1}$) |
| $T$ | temperature (°C) |
| $x$ | Quality |
| $\alpha$ | inlet/ambient |
| $\beta$ | outlet |
| Greek letters |
| $\eta$ | efficiency (%) |
| $\phi$ | power |
| $\psi$ | efficiency |

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