Thermodynamic performance analysis of comprehensive utilization system driven by geothermal energy

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Abstract. In order to improve the utilization rate of geothermal energy and realize the comprehensive utilization of geothermal energy. In this paper, a comprehensive utilization system using geothermal energy is proposed, and the thermodynamic performance under different working conditions is studied and analysed. And the Pareto method is used to optimize the calculation of different optimization options. The conclusions are as follows: The system can supply heating, cooling and electricity at the same time, and can regulate the electricity and cooling capacity according to the demand. The optimal pressure exists in the system in front of the turbine inlet to maximize the net output power of the system (when k=0). Due to heating needs, the adjustable range of k decreases with increasing of the turbine inlet pressure when the pressure of the turbine inlet is greater than 13.83 MPa. When there are differences in the importance of different output objectives, the Pareto solution is different. The importance degree of output objectives can be defined according to the demand, so as to solve the corresponding optimal solution.

1. Introduction
In recent years, global energy demand has continued to grow and will continue to grow in the future. To solve this contradiction, the development of renewable and low-carbon energy sources is an urgent task at present [1]. Geothermal energy is an important renewable and low-carbon energy source, usually in the form of low and medium grade energy (<350°C) [2]. Combined heat and power (CHP) based on ORC system has received a lot of attention from academics, and CHP will significantly improve the thermal efficiency of the system compared to ORC system [3]. However, people need not only electricity and heat, but also cold in their lives. Combined cooling, heating and power (CCHP) is an effective technology to improve the energy conversion efficiency by which electricity, heat and cold can be produced simultaneously in one unit [4]. In conventional CCHP systems, fossil fuels are usually used as the primary energy source [5]. Researchers have given more attention to CCHP systems based on renewable energy. Several research efforts have been devoted to the study of micro-CCHP systems based on the ORC or Carina cycle using energy and exergy analysis [6, 7]. Parikhani et al. [8] proposed an improved CCHP system based on Carina cycle and analyzed its feasibility based on thermodynamics and thermoeconomics. The results showed that the system had 27.68% and 198.3 $ / GJ in terms of the breakdown efficiency and overall product cost. Mohammadi and Powell [9] proposed and analyzed a new integrated cogeneration and trigeneration configuration based on a CO2 vapor compression system. Chang et al. [10] proposed and evaluated two
micro-CCHP systems based on Rankine and vapor compression cycles and high temperature PEMFC. The exergy efficiency of their proposed systems are 49.7% and 47.4%, respectively.

Existing studies of CCHP systems rarely consider heat source characteristics. At the same time, hot dry rock is different from other heat sources in that it has requirements for injection temperature [11]. In this paper, a new CCHP system is proposed based on the above-mentioned problems, which can be dynamically adjusted according to user requirements. The thermodynamic performance of the system was studied and analyzed. And the Pareto method is used to optimize calculation of all the calculated working conditions.

2. Physical model

The system diagram of CCHP is shown in Figure 1. T-s diagram of CCHP is shown in Figure 2. The geothermal water from the production well passes through the heat exchanger and returns to the injection well. The working fluid compressed by the pump absorbs heat in the heat exchanger (the process 1-2), and working fluid in front of the turbine inlet is in a supercritical state. The high-temperature and high-pressure working fluid enters the turbine to expand and do work (the process 2-3). The output mechanical work is divided into two parts, one of which drives the generator to generate electricity and the other drives the compressor to compress working fluid (the process 7-8). The compressor outlet pressure is the same as the turbine outlet pressure. The fluid coming from the compressor is mixed with the fluid coming from the turbine in the mixing chamber (the process 3/8-4). The mixed fluid releases heat in the condenser to provide heat to the heat consumer (the process 4-5). The fluid coming out of the condenser is divided into two parts, one part goes to the pump for pressurization through valve 1 (the process 5-1) and the other part goes to the expansion valve through valve 2 (the process 5-6). The working fluid from the expansion valve passes through the evaporator to supply the cold consumer (the process 6-7).

![Figure 1. System diagram of CCHP.](image1.png)

![Figure 2. T-s diagram of CCHP.](image2.png)

3. Simulation method

3.1. Thermodynamic model

The heat provided by geothermal energy can be expressed as equation (1):

\[ Q_{\text{gs}} = \dot{m}_{\text{hs}}(h_{\text{hs,in}} - h_{\text{hs,out}}) \]  

(1)

Where \( \dot{m}_{\text{hs}} \) is the heat source flow rate, \( h_{\text{hs,in}} \) and \( h_{\text{hs,out}} \) are the enthalpy of the heat source at the inlet and outlet of the heat exchanger, respectively.

The mass flow rate of the working fluid in the power cycle is calculated as equation (2):

\[ \dot{m}_{f,pc} = \frac{Q_{\text{gs}}}{\dot{h}_1 - \dot{h}_2} \]  

(2)

Where \( \dot{h}_1 \) and \( \dot{h}_2 \) are the enthalpy values at corresponding points in T-s, respectively.

The output power of turbine can be expressed as equation (3):

\[ W_{\text{tur}} = \dot{m}_{f,pc} (\dot{h}_2 - \dot{h}_3) \]  

(3)

Where \( \dot{h}_3 \) is the enthalpy of the working fluid at the outlet of the turbine.

The power consumed by compressor can be expressed as equation (4):

\[ W_{\text{comp}} = kgW_{\text{tur}} \]  

(4)
Where \( k \) is the ratio of compressor power consumption to turbine output power. The mass flow rate of the working fluid in the refrigeration cycle is calculated as equation (5):

\[
m_{f,\text{ref}} = \frac{W_{\text{com}}}{h_7 - h_8}
\]

(5)

Where \( h_7 \) and \( h_8 \) are the enthalpy of the working fluid at the inlet and outlet of the compressor, respectively.

The power consumed by pumps can be expressed as equation (6):

\[
W_p = m_{f,pc}(h_5 - h_6) + (m_{he} + m_{co})gH
\]

(6)

Where \( h_5 \) is the enthalpy of the working fluid at the inlet of the pump, \( m_{co} \) and \( m_{he} \) are the mass flow rate of water for cooling and heating, respectively. And \( g \) is the gravitational acceleration, \( H \) is pump head.

The net output power of system can be expressed as equation (7):

\[
W_{\text{net}} = (W_{\text{tur}} - W_{\text{com}})\eta_{\text{gen}} - W_p
\]

(7)

Where \( \eta_{\text{gen}} \) is the generator efficiency.

The cooling capacity provided by system can be expressed as equation (8):

\[
Q_{\text{cool}} = m_{f,\text{ref}}(h_7 - h_8)
\]

(8)

Where \( h_8 \) is the enthalpy of the working fluid at the outlet of the expansion valve.

The heating capacity provided by system can be expressed as equation (9):

\[
Q_{\text{heat}} = (m_{f,pc} + m_{f,\text{ref}})(h_4 - h_5)
\]

(9)

\[
h_4 = \frac{(m_{f,pc}g_7 + m_{f,\text{ref}}g_8)}{(m_{f,pc} + m_{f,\text{ref}})}
\]

(10)

The isentropic efficiency of compressor can be expressed as equation (10) [12]:

\[
\eta_c = 1.003 - 0.121(P_7/P_8)
\]

(11)

Where \( P_7 \) and \( P_8 \) are the pressure of the working fluid at the inlet and outlet of the compressor, respectively.

3.2. Simulation details

The simulation calculation is based on MATLAB and Refprop. In order to simplify the simulation, some factors that have little effect on the actual operation are usually ignored [13]. Detailed assumptions are as follows: (1) Stable operation of the system; (2) Ignore heat loss and pressure drop in the pipe. As a working fluid in energy conversion systems, CO2 has unique characteristics such as low viscosity, high heat transfer coefficient, non-toxicity and non-flammability [14]. Boundary conditions of simulation are shown in Table 1.

| Parameters                          | Symbol      | Values | Unit |
|-------------------------------------|-------------|--------|------|
| Temperature of heat source          | \( T_{hs,in} \) | 150    | °C   |
| Temperature of heat source at outlet| \( T_{hs,out} \) | 60     | °C   |
| Pressure of heat source             | \( P_{hs} \) | 10     | MPa  |
| Mass flow rate of heat source       | \( m_{hs} \) | 1      | kg/s |
| Environment temperature             | \( T_0 \)   | 20     | °C   |
| Isentropic efficiency of turbine    | \( \eta_{tur} \) | 0.8    |      |
| Isentropic efficiency of pump       | \( \eta_{p} \) | 0.75   |      |
| Temperature of the state 2          | \( T_2 \)   | 145    | °C   |
| Pump head                           | \( H \)     | 10     | m    |
| Condensation temperature            | \( T_5 \)   | 25     | °C   |

Table 1. Boundary conditions of simulation.
4. Results and discussion

4.1. Simulation results of thermodynamic performance

The variation of the thermodynamic performance of the system with the pressure of turbine inlet is shown in Figure 3. When \( k = 0 \), variations of the power consumption of pump, output power of generator and the net output power of system with the pressure of turbine inlet is shown in Figure 3 (a). As the turbine inlet pressure increases, the power consumption of pump and output power of generator both increase, but due to the difference in the magnitude of the increase, the net output power of the system has a maximum value (\( P_f = 18.83 \) MPa, \( W_{net} = 33.913 \) kW). The variation of cooling capacity and heating capacity of the system with the turbine inlet pressure is shown in Figure 3 (b). When \( k = 0 \), the heating capacity decreases gradually with the increase of turbine inlet pressure, and the cooling capacity is 0. The reason is that the heat absorbed by system is a constant value, the mass flow rate of working fluid in power cycle is definite value. With \( k \) increases, mass flow rate of the working fluid in the cooling cycle increases, so the cooling capacity and heating capacity of system increases.

![Figure 3](image1.png)
(a) Power consumption, output power and net output power (\( k = 0 \))

![Figure 3](image2.png)
(b) Cooling capacity and heating capacity

**Figure 3.** Variation of thermodynamic performance with the pressure of turbine inlet.

Variation of thermodynamic performance with the \( k \) is shown in Figure 4. As shown in Figure 4 (a), since the power consumption of compressor increases as \( k \) increases, the output power of generator and the net output power of the system decrease, and when \( k \) is greater than 0.6, the net output power of the system is negative (the system requires additional input power). When \( k \) is equal to 0, the output power of generator is equal to 0. As shown in Figure 4 (b), when the pressure of turbine inlet is a constant value, the mass flow rate of working fluid in power cycle is definite value. With \( k \) increases, \( m_{ref} \) in refrigeration cycle increases. So, the cooling and heating capacity increase.

![Figure 4](image3.png)
(a) Output power and \( W_{net} \)

![Figure 4](image4.png)
(b) Cooling and heating capacity

**Figure 4.** Variation of thermodynamic performance with the \( k \) (\( P_f = 13.93 \) MPa).
To ensure the temperature of the heating, the temperature of point 4 is limited. The maximum value of $k$ varies with the pressure of turbine inlet as shown in Figure 5. As the pressure of turbine inlet increases, the temperature of point 3 decreases. Meanwhile, as $k$ increases, the mass flow rate of working fluid in refrigeration cycle increases. Due to the temperature of point 8 in the cooling cycle is lower, the temperature at point 4 decreases as the mass flow rate of the working fluid in the cooling cycle increases. As a result, there is a limit to the maximum value of $k$. When $P_2$ is less than 13.83 MPa, the maximum value of $k$ is equal to 1. When $P_2$ is greater than 13.83 MPa, the maximum value of $k$ gradually decreases, and when $P_2$ is equal to 18.83 MPa, the system can obtain the maximum net output power when $k$ is equal to 0.

![Figure 5](image)

**Figure 5.** Variation of the maximum value of $k$ with the pressure of turbine inlet.

### 4.2. Optimization calculation

In the CCHP system, the system can provide cooling, heating and net output power. However, the cooling capacity and heating capacity of the system vary in the same trend, but the net output power is opposite to them. Therefore, it is very important to determine the suitable working condition. The Pareto method was used to optimize the different working conditions and two different optimization options were calculated. The detailed optimization options are as follows:

1) The importance degree of net output power (a) is greater than that of cooling capacity (b) and heating capacity (c), and the importance degree of cooling capacity and heating capacity is equal (a>b=c).

2) The importance degree of $W_{net}$, cooling and heating capacity are equal (a=b=c).

Firstly, the decision problem has three objectives:

$$F_y(y = 1, 2, 3)$$

Assume that there are $n$ solutions, and the $x$ solution corresponds to the $y$ objective as follows:

$$F_{xy}(x = 1, 2, \cdots, n; y = 1, 2, 3)$$

The ideal solution is:

$$A^* = \{F_{xy}^+, F_{xy}^+, F_{xy}^+\} = \{\max F_{xy} \mid x = 1, 2, \cdots, n; y = 1, 2, 3\}$$

The worst solution is:

$$B^* = \{F_{xy}^-, F_{xy}^-, F_{xy}^-\} = \{\min F_{xy} \mid x = 1, 2, \cdots, n; y = 1, 2, 3\}$$

The normalized value of the solution is

$$A_{xy}(x = 1, 2, \cdots, n; y = 1, 2, 3)$$

Finally, using the Euclidean method as a measure of distance, and when distance is minimized, the solution is optimal. The distance formula is

$$d_y = \sqrt{a((A_x - A_x^y)/(A_x^+ - A_x^-))^2 + b((A_y - A_y^y)/(A_y^+ - A_y^-))^2 + c((A_z - A_z^y)/(A_z^+ - A_z^-))^2}$$

(y = 1, 2, 3; x = 1, 2, \cdots n)
The optimization results are shown in Figure 6 (a), and the comparison of different optimization options is shown in Figure 6 (b). Figure 6(a) contains the data points of all the working conditions, and the optimal working conditions obtained by optimizing the two different weights are marked in the figure. When a=b=c, the net output power is negative in the optimized solution, so additional input power is required. And when a>b=c, the net output power is positive in the optimized solution. The electric power can be output while supplying cooling and heating at the same time. However, the cooling capacity and heating capacity of the optimized option with a>b=c option will decrease compared to the optimized option with a=b=c.

![Figure 6](image_url)

(a) Optimization results  
(b) Comparison of different optimization options

Figure 6. Different optimization options and comparison.

5. Conclusions
In this paper, a comprehensive utilization system (CCHP) using geothermal energy is proposed, and the thermodynamic performance under different working conditions is studied and analysed. And the Pareto method is used to optimize the calculation of different optimization options. The conclusions are as follows:

1. The system can supply heating, cooling and electricity at the same time, and can regulate the electricity and cooling capacity according to the demand.
2. The optimal pressure exists in the system in front of the turbine inlet to maximize the net output power of the system (when \(k=0\)).
3. Due to heating needs, the adjustable range of \(k\) decreases with increasing of the turbine inlet pressure when the pressure of the turbine inlet is greater than 13.83 MPa.
4. When there are differences in the importance of different output objectives, the Pareto solution is different. The importance degree of output objectives can be defined according to the demand, so as to solve the corresponding optimal solution.

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