Thermodynamic Analysis of a Hybrid System Coupled Cooling, Heating and Liquid Dehumidification Powered by Geothermal Energy

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Abstract: The utilization of geothermal energy is favorable for the improvement of energy efficiency. A hybrid system consisting of a seasonal heating and cooling cycle, an absorption refrigeration cycle and a liquid dehumidification cycle is proposed to meet dehumidification, space cooling and space heating demands. Geothermal energy is utilized effectively in a cascade approach. Six performance indicators, including humidity efficiency, enthalpy efficiency, moisture removal rate, coefficient of performance, cooling capacity, and heating capacity, are developed to analyze the proposed system. The effect of key design parameters in terms of desiccant concentration, air humidity, air temperature, refrigeration temperature and segment temperature on the performance indicators are investigated. The simulation results indicated that the increase of the desiccant concentration makes the enthalpy efficiency, the coefficient of performance, the moisture removal rate and the cooling capacity increase and makes the humidity efficiency decrease. With the increase of air humidity, the humidity efficiency and moisture removal rate for the segment temperatures from 100 to 130 °C are approximately invariant. The decreasing rates of the humidity efficiency and the moisture removal rate with the segment temperature of 140 °C increases respectively. Six indicators, except the cooling capacity and heating capacity, decrease with an increase of air temperature. The heating capacity decreases by 49.88% with the reinjection temperature increasing from 70 to 80 °C. This work proposed a potential system to utilize geothermal for the dehumidification, space cooling and space heating effectively.

Keywords: geothermal energy; absorption refrigeration cycle; liquid dehumidification cycle; seasonal heating and cooling cycle; enthalpy efficiency

1. Introduction

The growth of the population and the dependency of technology led to the rapid growth demand of energy [1]. In addition, the rapid growth demand of energy recognizes the challenges such as the overconsumption of fossil energy [2]. Those are disadvantageous for sustainable development. The utilization of the renewable energy and the improvement of energy utilization efficiency are two potential ways to solve the previous challenges [3]. Geothermal energy, being a renewable energy, can provide energy for the process system [4]. At the same time, the greenhouse effect intensified the requirement for air. In China, the hot summer and cold winter region is a typical climatic region. The high temperature and humidity climate makes the demands for refrigeration and dehumidification grow rapidly. The absorption refrigeration cycle (ARC) and the liquid dehumidification cycle (LDC) can be powered by the low-grade heat and are environmentally friendly [5]. Thus,
the ARC and LDC are two effective technologies to meet the demands of refrigeration and dehumidification. A review of previous literatures was conducted to indicate the objective and necessity of this work.

Geothermal energy is a renewable energy. According to the heat source temperature, geothermal energy is divided into three categories: high-temperature geothermal energy (HTGE), mid-temperature geothermal energy (MTGE) and low-temperature geothermal energy (LTGE). The temperature of the HTGE is above 150 °C and is used for the electricity generation. The electricity generation technology with dry stream is suitable for the HTGE [6]. The MTGE is geothermal energy with the temperature from 90 to 150 °C. The MTGE is used to generate electricity by the binary cycle. The Organic Rankine Cycle (ORC) and Kalina cycle are normal technologies for MTGE [7]. The LTGE is geothermal energy with a temperature lower than 90 °C. This part of energy is best suitable for direct use such as space cooling and heating, industrial processing, and greenhouse heating [8].

From the viewpoint of energy and exergy, Yazici analyzed a geothermal district heating system and found that the efficiencies of energy and exergy were 46.17% and 60.63%, respectively [9]. Kanoglu et.al., applied geothermal with a temperature of 200 °C to a hybrid system consisting of an absorption refrigeration cycle and a Claude cycle. Energy and exergy results showed that the coefficient of performance (COP) and the exergy efficiency of the proposed system are 0.162 and 67.9%, respectively [10]. Ahmadi et al. optimized and analyzed a carbon dioxide (CO₂) power cycle driven by geothermal energy. The simulation results showed that the exergy efficiency and the product cost rate of the cycle are 20.5% and USD 263,592.15/year respectively [11]. The previous research represented the reliability and sustainability of geothermal energy.

The absorption refrigeration cycle (ARC) is usually applied in low-grade waste heat recovery [12] or the recovery of other sustainable energy [13]. The advantages of environmental friendliness and low cost cause ARC to attract increased attention. Liu et al. proposed a hybrid system combining an ARC and a Kalina cycle. The simulation results indicated that the hybrid system could improve the net output power by 45% [14]. Subsequently, the author optimized the hybrid system based on the results of economic analysis, energy analysis and the life cycle assessment. According to the optimization results, the optimal energy efficiency and the relative minimum cost were 16.74% and USD 1.85 \times 10^5 [15]. Geothermal energy is a common power for ARC. Khosravi et al. optimized a hybrid system combined with an ARC, a solar heating collection system and a geothermal energy system. The optimized results indicated that the payback time of the system was about 8 years [16]. Geothermal water out of the flash-binary geothermal power plant was used to drive the ARC and desalination by Gnaifaid et.al. The exergy efficiency and the cost rate with the optimization results were 58% and USD 242/h respectively [17]. El Haj Assad et al. applied geothermal energy to produce power by a single-flash geothermal power plant and then used the water to drive the ARC. The results showed that COP of the system could get 0.87 with the water temperature of 120 °C and the solution heat exchange effectiveness of 0.9 [18]. Al-Hamed et al. developed a hybrid system driven by geothermal energy to produce electric power, space heating and cooling, and domestic hot water. The energetic and exergetic efficiencies were 53.33% and 37.07%, respectively, and the energetic efficiency with the optimization results could increase to 63.60% [19]. From the viewpoint of energy, exergy and exergoeconomy, Ghaebi et al. proposed and analyzed an integrated system driven by geothermal energy. The system consisted of an ARC and a liquefied natural gas cold energy recovery system and the thermal efficiency, exergy efficiency and sum unit cost of the product of the system were 85.92%, 18.52%, and USD 68.76/GJ respectively [20]. These previous literatures indicated that geothermal energy had excellent ability to drive the ARC.

The dehumidification process is significant for hot summer and cold winter regions in China. At present, the main dehumidification systems are based on condensation dehumidification technology [21]. The traditional condensation system reduces the air temperature into the dew point temperature to separate water vapor [22]. This process
aggravated energy consumption [23]. In order to reduce energy consumption, the liquid dehumidification cycle (LDC) is regarded as the alternative technology to the traditional dehumidification technology [24]. Low et al. designed a LDC driven by solar energy and found that the efficiency of the system and the minimum area of solar heating collector were 78.8% and 59.83 m$^2$, respectively [25]. Guan et al. determined the optimal mass flow rate of the desiccant and indicated that the optimal efficiency of the heat exchanger was 0.56 with the optimal ratio of desiccant-to-air being 1.5 [26]. The authors also modified the analytical solutions of the dehumidifier and the deviation between the modified results and the experimental results is less than 10% [27]. Zhang et al. optimized the LDC from the viewpoint of exergy destruction. The optimal exergy efficiency and COP of the LDC were 25% and 7.4, respectively [28]. Chen et al. developed a novel LDC driven by solar energy and optimized the area of solar heating collector [29]. Su et al. combined the ARC, vapor compression cycle with the LDC, and indicated that the efficiency of the proposed system was 34.97% higher than a traditional air conditioning system [30]. The advantageous dehumidification performance and the reliability of the LDC testified the feasibility of the LDC in hot summer and cold winter regions of China.

In this work, a hybrid system composed of an absorption refrigeration cycle, a liquid dehumidification cycle and seasonal heating and cooling cycles is developed. The power of the proposed system is geothermal energy. This hybrid system can meet the dehumidification demand, the space cooling demand, and the space heating demand, concurrently. Six indicators, including the moisture removal rate (MRR), the humidity efficiency (HE), the enthalpy efficiency (EE), COP, the cooling capacity and the heating capacity are proposed to evaluate the hybrid system. This work provides an effective solution to utilize geothermal energy.

2. System Description and Design Process

2.1. Description of System

The hybrid system is provided in Figure 1. The heat source of the proposed system is from a geothermal well. The proposed system consists of four cycles, including an absorption refrigeration cycle (ARC), a liquid dehumidification cycle (LDC) and two heat exchangers. The working pairs of ARC and LDC are lithium bromide solution (LiBr-H$_2$O) and lithium chloride solution (LiCl-H$_2$O), respectively. The mentioned cycles are combined to meet dehumidification, refrigeration, cooling, and heating demands.

2.1.1. The Geothermal Water

Geothermal water is the heat source of the proposed system. First, geothermal water is pumped from G1 with the mass flow rate of 5 kg/s and the temperature of 150 °C. The high-temperature fluid is introduced into the regenerator (G1) of LDC to provide the heat for the regeneration process of the desiccant. Then, the fluid enters the generator of the ARC to provide the heat for the separation of the vapor. Afterword, depending on the season, the mid-temperature fluid flows into the heat exchanger 3 (HX3) for heating purposes during winter. Finally, the low-temperature fluid returns to the reinjection well (G4).

2.1.2. The Absorption Refrigeration Cycle

The absorption refrigeration cycle (ARC) is used to produce the cooling energy for the proposed system. In this cycle, the LiBr-H$_2$O is chosen as the working pair. The strong LiBr-H$_2$O solution is heated by geothermal water. Due to the difference of boiling point, the vapor is produced in the generator. The vapor enters the condenser (A1). In the condenser, the cooling water of 33 °C is used to cool the vapor into saturated water. After flowing through the valve 1 (A3), the saturated water is introduced into the evaporator to produce the cooling energy. In the evaporator, the flash makes the saturated water become vapor. This process produces cooling energy. Then, depending on the season, the vapor from evaporator (A4) flows into the HX4 for the cooling purpose during summer or flows into the absorber to continue the cycle. In the absorber, the rich solution contacts with the vapor
directly. The rich solution absorbs the vapor and becomes a weak solution. In the generator, the solution is divided as rich solution and vapor. The vapor is used for refrigeration and the rich solution is used to absorb the vapor from evaporator. The rich solution first flows into the HX1 (A5) to preheat the new solution and then enters the absorber (A7) after flowing through the valve 2 (A6). The weak solution from the absorber, pumped by pump 1 (A8), is introduced into the HX1 (A9) and then enters into the generator (A10) to continue the cycle. Generally, the ARC is used to produce cooling energy.

2.1.3. The Liquid Dehumidification Cycle

The liquid dehumidification cycle (LDC) is applied to remove moisture in the air. This cycle selected the LiCl-H₂O as the working pair. There are five main units for the LDC, including a dehumidifier, a regenerator, a heat exchanger and two pumps. In the dehumidifier, the outside air (1) contacts with the liquid desiccant directly. The mass and heat transformation makes the wet air become dry air. Then, the dry air enters the room (2). The solution with low concentration from the dehumidifier flows into pump 2 (D1) and then enters into HX2 for preheating (D2). In the regenerator, the weak solution (D3) contacts with the air from room (4) directly. The heat for regeneration is provided by geothermal water. In this unit, the weak solution becomes the rich solution and then flows into the HX2 (D4) to preheat the weak solution. The rich solution from D5 is pumped by pump 3 (D5) and then enters into the condenser (D6). The rich solution is cooled by the refrigerant of ARC then comes into the dehumidifier to continue the cycle consequently. The LDC meets the dehumidification demand by recovering geothermal effectively.

2.1.4. Seasonal Cooling and Heating Cycles

The seasonal cooling and heating cycles are composed of two heat exchangers. For the cooling purpose of summer, the HX4 keeps working during summer to provide cooling energy. In addition, the air from outside is dehumidified and cooled simultaneously. The HX3 is used for the heating purpose during winter. Geothermal water flows through the HX3 to release heat during winter and then flows into the reinjection well.

Figure 1. Schematic diagram for the coupling system.
2.2. The Mathematical Equations for Two Subsystems

Based on the conservation of mass and energy, the LiBr-H\textsubscript{2}O ARC and the LiCl-H\textsubscript{2}O LDC are designed and analyzed. From the Figure 1, the main units of LiBr-H\textsubscript{2}O ARC and LiCl-H\textsubscript{2}O LDC can be classified as: pumps, valves, the heat exchangers, and packed towers. The mathematical equations can be calculated as follows and the detailed equations are introduced in Appendix A.

Mass balance: \[ \sum m = 0 \]  
(1)

Material balance: \[ \sum mx = 0 \]  
(2)

Energy balance: \[ \sum Q + \sum W + \sum mh = 0 \]  
(3)

where \( m \) represents the mass flow rate; \( h \) and \( Q \) are the enthalpy and heat of streams respectively; \( W \) is the power consumed by pumps.

2.3. The Performance Indicators for the Coupling System

(1) Moisture removal rate (MRR): The MRR represents the moisture removal ability of the coupling systems. It can be defined as [31]:

\[ \Delta \omega = \omega_{a,in} - \omega_{a,out} \]  
(4)

where \( \omega_{a,in} \) and \( \omega_{a,out} \) represent the inlet air humidity and outlet air humidity respectively.

(2) Humidity efficiency (HE): The HE is an index to evaluate the performance of the dehumidifier. It can reveal the heat and mass transformation between air and desiccant in the dehumidifier. The HE is calculated as [32–34]:

\[ \eta_\omega = \frac{\omega_{a,in} - \omega_{a,out}}{\omega_{a,in} - \omega_{Ts,equ}} \]  
(5)

where the \( \omega_{Ts,equ} \) is the equilibrium-humidity between the desiccant and air.

(3) COP: COP, as a usual performance index, can represent the utilization efficiency of a geothermal source. COP for the coupling system is defined as [35]:

\[ \text{COP} = \frac{m_a \times (h_{a,in} - h_{a,out})}{Q_T} \]  
(6)

where the \( m_a \) is the mass flow rate of inlet air. \( h_{a,in} \) and \( h_{a,out} \) represent the enthalpy of the inlet air and outlet air respectively. \( Q_T \) is the heat duty consumed by the coupling system.

(4) Enthalpy efficiency (EE): The EE can evaluate the performance of the dehumidifier from the viewpoint of thermodynamics. It can be calculated as [36]:

\[ \eta_h = \frac{h_{a,in} - h_{a,out}}{h_{a,in} - h_C} \]  
(7)

where \( h_C \) is the enthalpy of the cooling pair.

2.4. Model Validations and the Design of the Coupling System

The LiBr-H\textsubscript{2}O ARC and the LiCl-H\textsubscript{2}O LDC are simulated in Aspen Plus. The ELECNRTL method is selected as the basic property method. In Table 1, the specification for the models is listed in detail. The generator is simulated by the Heater model and the Flash2 model. The Heater model and the ABSBR1 model are applied to simulate the regenerator. The condensation pressure, decided by the condensation temperature, is 7.38 kPa. According to the other references [14,15,37], the most optimal temperature of the generator for LiBr-H\textsubscript{2}O ARC is 90 °C. In this work, the generator temperature is set as 80 °C due to a temperature decrease of 10 °C. In our previous work [12,35], the reliabilities of LiBr-H\textsubscript{2}O ARC and LiCl-H\textsubscript{2}O LDC have been validated. The comparison results
are listed in Tables 2 and 3. The deviations of 0.4% and 0.2% represent the reliabilities of the LiBr-H₂O ARC and LiCl-H₂O LDC. The design procedure is introduced in Figure 2. The first step of the design process is to determine the refrigeration temperature. After the assumption of the segment temperature, the LiBr-H₂O ARC and LiCl-H₂O LDC are designed, respectively. In the coupling system, geothermal source is divided into two different parts. The high-temperature part is introduced to regenerate the desiccant and the low-temperature part is used to heat the LiBr-H₂O solution. In addition, the cooling energy produced by the LiBr-H₂O ARC is used for the refrigeration of desiccant and room respectively. Finally, the data is recorded to evaluate the coupling system. Before the simulation of the system, the design parameters are listed in Table 4. For the accuracy of the simulation, necessary assumptions are as follows [38–40]:

1. The system is stable;
2. The heat loss and the drop loss of pipes are negligible;
3. The system is simulated with no kinetic energy and potential energy;
4. The units of the coupling system have no heat loss;
5. The stream and the liquid refrigeration are assumed as saturated.

Table 1. The detailed specification of the models.

| Equipment                  | Types                                   | Specification                                                                 |
|----------------------------|-----------------------------------------|-------------------------------------------------------------------------------|
| LiBr/H₂O absorption refrigeration system | Generator, Flash2, Condenser, Absorber, Evaporator, Heat exchanger, Pump, Valve | Pressure drop: 0 kPa; duty: 0 kW; pressure drop: 0 kPa; temperature: 80 °C |
|                            |                                        | Condenser heater, Absorber heater, Evaporator heater, Heat exchanger heater, Pump, Valve | Vapor fraction: 0, pressure: 7.38 kPa; Vapor fraction: 0, Pressure drop: 0 kPa; Vapor fraction: 1, Pressure drop: 0 kPa; Heat outlet-cold inlet temperature approach: 10 °C; Outlet pressure: specified |

Table 2. The comparison results of absorption refrigeration cycle.

| Parameters               | Units | Reference [41] | This Work | Data Type | Deviation |
|--------------------------|-------|----------------|-----------|-----------|-----------|
| Generator temperature    | °C    | 84.8           | 84.8      | Input     |           |
| Condensation temperature | °C    | 39.8           | 39.8      | Input     |           |
| Absorber temperature     | °C    | 35.5           | 35.5      | Input     |           |
| Refrigeration temperature| °C    | 8.6            | 8.6       | Input     |           |
| COP                       |       | 0.7701         | 0.7734    | Output    | 0.4%      |
Table 3. The comparison results of liquid dehumidification cycle.

| Parameters                | Units | Reference [42] | This Work | Data Type | Deviation |
|---------------------------|-------|----------------|-----------|-----------|-----------|
| Desiccant temperature     | °C    | 25.2           | 25.2      | Input     |           |
| Solution concentration   | %     | 38.5           | 38.5      | Input     |           |
| Air temperature           | °C    | 33.2           | 33.2      | Input     |           |
| Air humidity              | kg/kg dry air | 0.0175       | 0.0175    | Input     |           |
| Humidity efficiency       | /     | 0.530          | 0.532     | Output    | 0.38%     |

Figure 2. The design procedure for the coupling system.

Table 4. The detailed parameters for the simulation of system.

| Parameters                                      | Unit     | Value     |
|-------------------------------------------------|----------|-----------|
| Mass flow rate of inlet air                     | kg/s     | 1.5       |
| Air humidity                                    | g/kg     | 10.82–22.12 |
| Air temperature                                 | °C       | 18.4–35.8 |
| Concentration of LiCl-H₂O                      | %        | 30–40     |
| Generator temperature                           | °C       | 80        |
| Concentration of LiBr-H₂O                      | %        | 56        |
| Geothermal water temperature                    | °C       | 150       |
| Geothermal water reinjection temperature        | °C       | 70        |
| Mass flow rate of geothermal water              | kg/s     | 5         |
3. Results and Discussions

Based on the simulation results, the influence of design parameters on the indicators are studied in this section. The relationship between the design parameters and the indicators are shown in vivid pictures.

3.1. The Effect of Desiccant Concentration on the Coupling System

The effect of desiccant concentration on performance indicators is provided in Figures 3–5. With an increase of the desiccant concentration, COP, $\Delta \omega$, $\eta_h$ and cooling capacity increase while the $\eta_\omega$ decreases, generally.

![Figure 3. The effect of desiccant concentration on the performance indicators with the refrigeration temperature of 2 °C.](image)

(a) COP and EE; (b) HE and MRR; (c) cooling capacity.
Figure 4. The effect of desiccant concentration on the performance indicators with the refrigeration temperature of 5 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.

Figure 5. The effect of desiccant concentration on the performance indicators with the refrigeration temperature of 10 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.
Figure 5. The effect of desiccant concentration on the performance indicators with the refrigeration temperature of 10 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.

Figure 3 is the performance indicators with the refrigeration temperature of 2 °C. For the $\eta_\omega$, $\eta_h$, and COP, the increase of segment temperature has little effect on the indicators. But the $\Delta \omega$ with the segment temperature of 140 °C is lower than that of 100 to 130 °C. As the desiccant concentration increases from 30% to 40%, the $\eta_h$ and COP rise with the increased rate of 5.78% and 5.79%, respectively. On the contrary, the $\eta_\omega$ decreases by 1.21% with the desiccant concentration increasing from 30% to 40%. However, the $\Delta \omega$ increases by 5.06%. The last index is the cooling capacity. Figure 3c indicates that the desiccant concentration and segment temperature have serious influence on the cooling capacity. With the increase of desiccant concentration, the cooling capacity for different segment temperature rises, respectively. As the segment temperature increases, the cooling capacity for the same desiccant concentration increases concurrently. In addition, the cooling capacity with the segment temperature of 100 °C is lower than that of 110 to 140 °C.

Figure 4a,c shows the performance indicators with the refrigeration temperature of 5 °C. As shown in Figure 3, the increase of desiccant concentration leads to an increase of COP, $\Delta \omega$, $\eta_h$ and cooling capacity and a decrease of $\eta_\omega$. Compared with the results with the refrigeration temperature of 2 °C, the values of COP, $\Delta \omega$, $\eta_h$ and $\eta_\omega$ move increasingly closer. As the desiccant concentration rises from 30% to 40%, the $\eta_h$ and COP for different segment temperature increase by 7.29% and 7.28%, respectively. The $\Delta \omega$ increases by 6.43% while the $\eta_\omega$ decreases by 1.27%. For the index of cooling capacity, the effect of desiccant concentration and segment temperature is obvious. The tendency of the cooling capacity with the refrigeration temperature of 5 °C is similar to that of 2 °C. The cooling capacity with the segment temperature of 100 °C is lower than that of 110 to 140 °C. However, the cooling capacity with the segment temperature of 120 °C is close to that of 130 and 140 °C.

The performance indicators with the refrigeration temperature of 10 °C are introduced in Figure 5. Figure 5a is the $\eta_h$ and COP of the proposed system. Similar to the previous results, the values of COP and $\eta_h$ with different refrigeration temperature are almost equal. With the increase of desiccant concentration from 30% to 40%, the increase rate of $\eta_h$ and COP are 10.84% and 10.83%, respectively. The $\Delta \omega$ decreases by 9.69%. The $\eta_\omega$ with the refrigeration temperature of 140 °C is lower than that of 100 to 130 °C as the desiccant concentration increases from 30% to 35%. However, as the concentration increases to 40%, the $\eta_\omega$ with different segment temperatures are increasingly close. Figure 5c shows the cooling capacity with the refrigeration temperature of 10 °C. As the desiccant concentration increases, the cooling capacity of the proposed system increases sluggishly. With the same desiccant concentration, the increase of the segment temperatures from 110 to 140 °C leads to an almost homogeneous increase for the cooling capacity.

In general, a higher desiccant concentration represents a bigger COP, $\eta_h$, $\eta_\omega$ and cooling capacity, while lowering $\Delta \omega$. As the refrigeration temperature increases from 2
to 10 °C, the $\Delta \omega$, $\eta_\omega$, COP and cooling capacity decreases, while the $\eta_h$ increases. From the viewpoint of the change of desiccant concentration, higher concentration means better performance for the hybrid system. From the viewpoint of the refrigeration temperature, lower refrigeration temperature means better performance for the hybrid system.

3.2. The Effect of Air Humidity on the Coupling System

The performance indicators are introduced with different air humidity and refrigeration temperature from Figures 6–8. Figure 6 shows the indicators with a refrigeration temperature of 2 °C. The results in Figure 6a indicate that the increase of segment temperature has a little influence on COP and $\eta_h$. With the rising of the air humidity, the $\eta_h$ and COP increase by 24.66% and 187.37%, respectively. It means that the increase of air humidity has a great effect on COP. The increase of air humidity makes the $\Delta \omega$ and $\eta_\omega$ with the segment temperatures from 100 to 130 °C, respectively, change slightly. But the $\Delta \omega$ and $\eta_\omega$, with a segment temperature of 140 °C, decreases by 3.54% and 3.55%, respectively. The decrease rate of $\Delta \omega$ increases from 0.16% to 3.39%. The decrease rate of $\eta_\omega$ increases from 0.15% to 3.38%. The increase of air humidity makes the cooling capacity rise. The cooling capacity with the segment temperature of 100 °C is smaller than the cooling capacities with the segment temperatures from 110 to 140 °C.

Figure 6. The effect of air humidity on the performance indicators with a refrigeration temperature of 2 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.
Figures 7 and 8 show the performance indicators with the refrigeration temperature of 5 and 10 °C. Similar to the results in Figure 6, the $\eta_h$ and COP with different segment temperature are almost equal, and the segment temperature has a great effect on the $\Delta \omega$ and $\eta_\omega$, with the segment temperature of 140 °C, and has little influence on the $\Delta \omega$ and $\eta_\omega$ with the segment temperatures from 100 to 130 °C. As the air humidity increases, the $\eta_h$ and COP with a refrigeration of 5 °C, increase by 20.43% and 195.07% respectively. The $\eta_h$ and COP with the refrigeration of 10 °C increase by 14.37% and 217.90%, respectively. The results of COP represent that the rise of refrigeration temperature has great effect on the increase rate of COP. The influence of refrigeration temperature on the increase rate of $\eta_h$ becomes smaller. For the refrigeration temperature of 5 °C, the decrease rate of the $\Delta \omega$ and $\eta_\omega$ with the segment temperature of 140 °C are 2.69% and 2.7%, respectively. For the refrigeration temperature of 5 °C, the decrease rate of the $\Delta \omega$ and $\eta_\omega$ with the segment temperature of 140 °C are 2.09% and 2.08%, respectively. The cooling capacities with the refrigeration temperature of 5 and 10 °C increase with the increase of air humidity, respectively.

(a) (b) (c)

Figure 7. The effect of air humidity on the performance indicators with the refrigeration temperature of 5 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.
As the refrigeration temperature increasing from 2 to 10 °C, the $\Delta \omega$, $\eta_\omega$, and COP decrease, while the $\eta_h$ increases. The cooling capacities with the segment temperatures from 110 to 140 °C has little change.

3.3. The Effect of Air Temperature on the Coupling System

The results of the performance indicators for different air temperature are shown in Figures 9–11. The increase of air temperature makes the $\Delta \omega$, $\eta_\omega$, $\eta_h$ and COP decrease, generally. The rise of air temperature has great effect on the indicators with different segment temperatures. The results in Figure 9 indicate that COP with the segment temperatures from 100 to 120 °C are almost equal. The $\eta_h$ has the same tendency. The decrease rates of $\eta_h$ for different segment temperatures increase from 12.18% to 25.62% and the decrease rates of COP for different segment temperatures rise from 23.41% to 35.14%. The $\eta_\omega$ with the segment temperature of 100 °C is equal to that of 110 °C. As the air temperature increases, the decrease rate of $\eta_\omega$ increases to 17.06%, and the increase rate of $\Delta \omega$ increases to 17.06%. The cooling capacities for different segment temperatures increase with the rise of air temperature. With the rise of segment temperature, the cooling capacity with the same air temperature increases.

Figure 8. The effect of air humidity on the performance indicators with the refrigeration temperature of 10 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.
110 to 140 °C for different refrigeration temperatures are within the range of 200 to 280 kW.

(a) COP and EE; (b) HE and MRR; (c) cooling capacity.

Figure 9. The effect of air temperature on the performance indicators with the refrigeration temperature of 2 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.

Figure 10. Cont.
Figure 9. The effect of air temperature on the performance indicators with the refrigeration temperature of 2 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.

Figure 10. The effect of air temperature on the performance indicators with the refrigeration temperature of 5 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.

Figure 11. The effect of air temperature on the performance indicators with the refrigeration temperature of 10 °C. (a) COP and EE; (b) HE and MRR; (c) cooling capacity.
Figures 10 and 11 are the indicators with the refrigeration temperatures of 5 and 10 °C, respectively. For the refrigeration temperature of 5 °C, the decrease rates of $\eta_h$ with different segment temperatures increase from 11.65% to 25.85%. The decrease rates of $\eta_\omega$ with different segment temperatures increase to 17.90%. For the refrigeration temperature of 10 °C, the decrease rates of $\eta_h$ with different segment temperatures increase from 10.64% to 23.87%. The decrease rates of $\eta_\omega$ with different segment temperatures increase to 16.47%.

As the refrigeration temperature increases, the $\Delta \omega$ and COP with same air temperature and segment temperature have little change. The $\eta_\omega$ and $\eta_h$ increase with a rise of refrigeration temperature. The cooling capacities with the segment temperatures from 110 to 140 °C for different refrigeration temperatures are within the range of 200 to 280 kW.

3.4. The Heating Capacity of the Proposed System

Figure 12 shows the results of the heating capacity of the proposed system. The heating capacity decreases by 49.88% with the increase of reinjection temperature. The space heating energy is collected from the reinjection water. The rise of reinjection temperature leads to the increase of constant-pressure specific heat. However, the increase rate of the constant-pressure specific heat is smaller than the increase rate of the difference for reinjection temperature. Thus, the heating capacity decreases with an increase of reinjection temperature.

![Graph showing heating capacity vs. temperature](image)

Figure 12. The space heating energy of the proposed system.

4. Conclusions

This work proposed a hybrid system combined dehumidification, space heating and space cooling. The humidity efficiency ($\eta_\omega$), the enthalpy efficiency ($\eta_h$), the moisture removal rate ($\Delta \omega$), the coefficient of performance (COP), the cooling capacity and the heating capacity for the proposed system were analyzed. The effect of design parameters, including the desiccant concentration, air temperature and air humidity, and refrigeration temperature on the performance indicators were conducted. The conclusions are summarized as follows.

1. The increase of the desiccant concentration leads to the increase of COP, $\eta_h$, $\Delta \omega$ and cooling capacity. The $\eta_\omega$ decreases with an increase of desiccant concentration. The increase of refrigeration temperature makes the $\eta_\omega$, COP, $\Delta \omega$ and cooling capacity decrease and makes the $\eta_h$ increase.

2. The increase of the air humidity makes the COP, $\eta_h$, and cooling capacity increase. For the same refrigeration temperature, the increase of segment temperature from 100 to 130 °C has a small effect on $\eta_\omega$. The decrease rates of $\eta_\omega$ and $\Delta \omega$ for the segment temperature of 140 °C increase with the increase of air humidity.
(3) The increase of air temperature leads to a decrease of COP, $\eta_h$, $\Delta \omega$, and $\eta_\omega$. As the refrigeration temperature rises, the $\Delta \omega$ with the same segment temperature and air temperature are almost equal. COP has the same tendency, but the $\eta_\omega$ and $\eta_h$ increase respectively with the cooling capacities for the segment temperature from 110 to 140 $^\circ$C, which are within the range of 200 and 280 kW.

(4) The heating capacity decreases by 49.88% with the reinjection temperature increasing from 70 to 80 $^\circ$C.

Author Contributions: Conceptualization, investigation, writing—review and editing, and supervision, A.X.; methodology, investigation, writing—original draft, and writing—review and editing, M.X.; writing—review and editing and software, N.X.; investigation and validation, Y.X.; visualization, J.H.; investigation and software, Y.C.; supervision and writing—review and editing, Z.L.; conceptualization, project administration, S.Y. All authors have read and agreed to the published version of the manuscript.

Funding: This research was founded by the National Natural Science Foundation of China project (grant numbers: 22008265), the Research Foundation of Education Bureau of Hunan Province, China Project (grant numbers: 19B148) and College Students’ Innovative Entrepreneurial Training Plan Program (grant numbers: S202111535010).

Institutional Review Board Statement: Not applicable.

Informed Consent Statement: Not applicable.

Data Availability Statement: Not applicable.

Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

- $m$: mass flow rate, [kg s$^{-1}$]
- $Q_L$: heat duty for absorption refrigeration cycle, [kW]
- $h$: enthalpy, [kJ kg$^{-1}$]
- $x$: concentration, [%]
- $Q_H$: heat duty for liquid dehumidification cycle, [kW]
- $W$: power of pump, [kW]
- $Q_T$: heat duty consumed by the hybrid system, [kW]
- $Q_{CW}$: heat duty absorbed by the cooling water, [kW]
- $A$: absorption refrigeration cycle
- $D$: liquid dehumidification cycle
- $a_{,in}$: inlet air
- $a_{,out}$: outlet air
- $T_{s,\text{equ}}$: equilibrium humidity
- $C$: cooling pair
- $T$: total energy consumption
- $\text{evap}$: evaporator
- $\text{con}$: condenser
- $\text{pump}$: pump
- $\text{ARC}$: absorption refrigeration cycle
- $\text{COP}$: coefficient of performance of the coupling system
- $\text{LDC}$: liquid dehumidification cycle
- $\text{HTGE}$: high-temperature geothermal energy
- $\text{MTGE}$: mid-temperature geothermal energy
- $\text{LTGE}$: low-temperature geothermal energy
- $\text{ORC}$: organic Rankine Cycle
- $\text{HE}$: humidity efficiency
- $\text{MRR}$: moisture removal rate
- $\text{EE}$: enthalpy efficiency
- $\eta$: efficiency
- $\omega$: humidity
Appendix A

For LiBr-H\(_2\)O ARC, the main units can be classified as: four heat exchangers, four valves, two packed towers and a pump. The mathematical equations of the units can be calculated as following:

**Generator:**

\[
m_{A10} = m_{A5} + m_{A1} \tag{A1}
\]

\[
m_{A10}x_{A10} = m_{A5}x_{A5} \tag{A2}
\]

\[
Q_L = m_{A5}h_{A5} + m_{A1}h_{A1} - m_{A10}h_{A10} \tag{A3}
\]

**Evaporator:**

\[
m_{A3} = m_{A4} \tag{A4}
\]

\[
Q = m_{\text{evap}} \times (h_{A4} - h_{A3}) \tag{A5}
\]

**Condenser:**

\[
m_{A1} = m_{A2} \tag{A6}
\]

\[
Q_{CW} = m_{\text{con}} \times (h_{A1} - h_{A2}) \tag{A7}
\]

**Absorber:**

\[
m_{A8} = m_{A4} + m_{A7} \tag{A8}
\]

\[
m_{A8}x_{A8} = m_{A7}x_{A7} \tag{A9}
\]

\[
Q_{CW} = m_{A8}h_{A8} - (m_{A4}h_{A4} + m_{A7}h_{A7}) \tag{A10}
\]

**Heat exchanger:**

\[
m_{A5} = m_{A6} \tag{A11}
\]

\[
x_{A5} = x_{A6} \tag{A12}
\]

\[
m_{A9} = m_{A10} \tag{A13}
\]

\[
x_{A9} = x_{A10} \tag{A14}
\]

\[
m_{A5}h_{A5} - m_{A6}h_{A6} = m_{A9}h_{A9} - m_{A10}h_{A10} \tag{A15}
\]

**Valves:**

\[
m_{A2} = m_{A3} \tag{A16}
\]

\[
h_{A2} = h_{A3} \tag{A17}
\]

\[
m_{A6} = m_{A7} \tag{A18}
\]

\[
x_{A6} = x_{A7} \tag{A19}
\]

\[
h_{A6} = h_{A7} \tag{A20}
\]

**Pump 1:**

\[
m_{A8} = m_{A9} \tag{A21}
\]

\[
x_{A8} = x_{A9} \tag{A22}
\]

\[
W_1 = m_{\text{pump}} \times (h_{A9} - h_{A8}) \tag{A23}
\]

For LiCl-H\(_2\)O LDC, the main units can be classified as: a heat exchanger, two packed towers and two pumps. The mathematical equations of the units can be calculated as following:

**Dehumidifier:**

\[
m_1 + m_{D7} = m_2 + m_{D1} \tag{A24}
\]

\[
m_1h_1 - m_2h_2 = m_{D7}h_{D7} - m_{D1}h_{D1} \tag{A25}
\]

**Regenerator:**

\[
m_3 + m_{D3} = m_4 + m_{D4} \tag{A26}
\]
\[ Q_H = (m_4 h_4 + m_4 h_4) - (m_3 h_3 + m_3 h_3) \]  \hspace{1cm} (A27)

LiCl condenser:
\[ m_{D6} = m_{D7} \]  \hspace{1cm} (A28)
\[ x_{D6} = x_{D7} \]  \hspace{1cm} (A29)
\[ Q = m_{\text{con}} \times (h_{D6} - h_{D7}) \]  \hspace{1cm} (A30)

Pump 2:
\[ m_{D1} = m_{D2} \]  \hspace{1cm} (A31)
\[ x_{D1} = x_{D2} \]  \hspace{1cm} (A32)
\[ W_2 = m_{\text{pump}} \times (h_{D2} - h_{D1}) \]  \hspace{1cm} (A33)

Pump 3:
\[ m_{D5} = m_{D6} \]  \hspace{1cm} (A34)
\[ x_{D5} = x_{D6} \]  \hspace{1cm} (A35)
\[ W_3 = m_{\text{pump}} \times (h_{D6} - h_{D5}) \]  \hspace{1cm} (A36)

Heat exchanger:
\[ m_{D3} = m_{D2} \]  \hspace{1cm} (A37)
\[ x_{D3} = x_{D2} \]  \hspace{1cm} (A38)
\[ m_{D4} = m_{D5} \]  \hspace{1cm} (A39)
\[ x_{D4} = x_{D5} \]  \hspace{1cm} (A40)
\[ m_{D2} h_{D2} - m_{D3} h_{D3} = m_{D4} h_{D4} - m_{D5} h_{D5} \]  \hspace{1cm} (A41)

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