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Characteristics and Application Analysis of a Novel Full Fresh Air System Using Only Geothermal Energy for Space Cooling and Dehumidification

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Abstract: To effectively reduce building energy consumption, a novel full fresh air system with a heat source tower (HST) and a borehole heat exchanger (BHE) was proposed for space cooling and dehumidification in this paper. The cooling system only adopts geothermal energy to produce dry and cold fresh air for space cooling and dehumidification through the BHE and HST, which has the advantage of non-condensate water compared to BHE systems integrated with a fan coil or chilled beam. Based on the established mathematical model of the cooling system, this paper analyzed the system characteristics, feasibility, operation strategy, energy performance, and cost-effectiveness of the proposed model in detail. The results show that the mathematical model has less than 10% error in estimating the system performance compared to the practical HST-BHE experimental set up. Under the specific boundary conditions, the cooling and dehumidification capacity of this system increases with the decrease in the air temperature, air moisture content, and inlet water temperature of the HST. The optimal cooling capacity and the system COP can be achieved when the air-water flow ratio is at 4:3. A case study was conducted in a residential building in Shenyang with an area of about 1800 m². It was found that this system can fully meet the cooling and dehumidification demand in such a residential building. The operation strategy of the cooling system can be optimized by adjusting the air-water flow ratio from 4:3 to 3:2 during the early cooling season (7 June-1 July) and end cooling season (3 August–1 September). As a result, the average COP of the cooling system during the whole cooling season can be improved from 6.1 to 8.7. Compared with the air source heat pump (ASHP) and the ground source heat pump (GSHP) for space cooling, the proposed cooling system can achieve an energy saving rate of 123% and 26%, respectively. Considering that the BHE of the GSHP can be part of the proposed HST-BHE cooling system, the integration of the HST and GHSP for space cooling (and heating) is strongly recommended in actual applications.

Keywords: hybrid space cooling system; heat source tower; borehole heat exchanger; geothermal energy; dehumidification

1. Introduction

The prevalence of high-energy-consumption buildings, constituting around 95% of the new constructions, reflects a growing demand for energy in China [1]. Among these buildings, air conditioning systems for space cooling and heating consume over 30% of the total energy [2]. Consequently, it is highly necessary to promote the application of renewable energy sources to mitigate building energy consumption.

Traditional air conditioning systems primarily rely on artificial cooling and heating sources, such as mechanical refrigeration [3,4], heat pumps [5–7], absorption refrigeration [8,9], adsorption refrigeration [10,11], boiler systems [12,13], and solar collectors [14]. While



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Copyright: © 2024 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution (CC BY) license (https:// creativecommons.org/licenses/by/ 4.0/). these systems may incorporate renewable energy utilization, they predominantly rely on electrical, thermal, and fuel energy, leading to substantial embodied or life-cycle energy consumption [15]. In response, natural heating and cooling sources have garnered attention for their potential to harness environmental energy directly or indirectly for space cooling, which is part of the regeneration of energy systems.

Natural cooling sources, including natural ventilation [16,17], sky radiation [18,19], outdoor water reservoir [20,21], and deep soil [22], offer alternatives to space cooling. However, each source presents unique challenges. For instance, while natural ventilation leverages outdoor air for cooling, its efficiency is contingent upon outdoor temperatures and air quality [23]. Similarly, sky radiation cooling relies on radiant cooling but operates solely at night [24]. An outdoor water reservoir and deep soil could offer more consistent cooling, but they are geographically constrained and may require substantial initial investments [25,26].

In terms of natural heating, solar energy and biomass are prominent options [13,14], but each has limitations. For instance, solar energy is subject to meteorological variability and spatial constraints [27]. Biomass energy poses challenges in material collection, transportation, and environmental impact [28].

The discussion above reveals that natural heating and cooling sources alone are usually insufficient for building heating and cooling needs. To enhance reliability and reduce energy consumption in building air conditioning systems, a combined approach leveraging both artificial and natural sources is imperative to maximize benefits for space cooling and heating. Currently, the integration of artificial cooling and heating sources with natural ventilation [29,30] and solar energy utilization [31,32] has been extensively adopted, yielding significant economic and environmental advantages. While much attention has been given to utilizing deep soil directly for space cooling, the integration of artificial and natural sources remains as the focus of research. Typically, the natural cooling potential of deep soil is harnessed via borehole heat exchanger (BHE) systems employing water or air as the working fluid. In water-based BHE systems, cooled water is circulated to indoor fan coils, radiant coils, or chilled beams for space cooling and dehumidification; these are known as direct ground cooling (DGC) systems [33]. Studies by Li et al. [34], Liu et al. [35], Arghand et al. [36], and Filipsson et al. [37] have demonstrated the feasibility, energy-saving benefits, and thermal comfort of integrating a BHE with fan coils or chilled beams for space cooling. Alternatively, in air-based BHE systems, cooled air is directly supplied into spaces for cooling and dehumidification, forming earth–air heat exchanger (EAHE) system [38]. Li et al. [39] investigated various parameters' impact on the thermal performance of EAHE systems, while Rodea et al. [40] confirmed the EAHE system's efficacy for cooling purposes, achieving space air temperature reductions ranging from 5.1 °C to 9.4 °C in Morelos.

When comparing the DGC system utilizing water and the EAHE system utilizing air, their respective advantages and disadvantages become apparent. First, due to water's higher specific heat compared to air, it offers a larger cooling capacity under equivalent mass flow rates. Second, the EAHE system easily provides fresh air to the room, whereas the DGC system requires a fresh air handling unit. Third, the DGC system's fan coil and chilled beam serve as indirect heat exchangers between water and air, potentially leading to the formation of condensate water and mold fungi during space dehumidification [41]. However, despite these distinctions, both systems face challenges. Achieving precise temperature and humidity control for space cooling proves difficult, and neither system accommodates space heating requirements effectively. Moreover, the utilization of a BHE entails a high initial investment, further complicating their adoption. Therefore, to meet space heating and cooling demands reliably and efficiently, it is essential to integrate deep soil natural cooling sources with artificial heating and cooling systems.

Among various artificial heating and cooling sources, the ground source heat pump (GSHP) emerges as the most compatible option for integration with deep soil cooling via a BHE, facilitating practical applications. During winter, the GSHP system fulfills space heating needs, while in summer, both the GSHP and DGC systems can operate

independently or together to fulfill space cooling needs, thereby reducing building energy consumption. However, as highlighted earlier, the DGC system presents drawbacks such as a lack of fresh air, issues with condensate water and mold formation during space dehumidification, and challenges in achieving precise temperature and humidity control. Consequently, it is imperative to explore innovative operational strategies that combine the GSHP and DGC systems.

In our previous studies, an integrated system involving a heat source tower (HST) and a BHE to solve the thermal imbalance caused by the GSHP in cold regions was proposed and explored extensively [42–45]. The working principle of the system (shown in Figure 1) for soil heat storage is stated as following: the low-temperature water from the BHE is sent into the HST and directly contacts the high-temperature outdoor air, thus achieving hightemperature water for soil heat storage during the non-heating season. Previous studies focused on the characteristics of the integrated system for soil heat storage, including the influences of air and water parameters, BHE structure parameters, and soil thermal conductivity on system heat storage, as well as the optimization of control strategies and the system's applicability in different cities. During the soil heat storage process, however, we found that low-temperature fresh air can also be achieved in the outlet of the HST. Especially when the temperature of water from the BHE is much lower than the dew point temperature of the outdoor air, the low-temperature fresh air can also be dehumidified in the HST. This indicates that the cooled and dehumidified fresh air can also be sent to the building for space cooling and dehumidification. This means that the proposed soil heat storage system in Figure 1 can realize both soil heat storage and space cooling at the same time. As a result, a novel hybrid cooling system of the HST and BHE (HST–BHE cooling system) is newly defined and proposed in this paper, as shown in Figure 2.



Figure 1. Principle of the HST and GSHP integrated system for soil heat storage [42].

Compared to the traditional cooling system with high energy consumption, the novel HST–BHE cooling system illustrated in Figure 2 exhibits several advantages. First, it has the potential to reach high energy efficiency for fully using renewable geothermal energy and air energy, while consuming relatively small amounts of energy only by pump and fan. Second, compared conventional DGC cooling system, it can provide fresh air and avoid the problems of condensate water and mold fungi during space dehumidification. Finally, it can easily integrate with the BHE in the existing GSHP system and reduce the initial investment cost of the HST–BHE cooling system.





In this paper, the mathematical model of the novel HST–BHE cooling system is established first; second, the cooling and dehumidification characteristics of the HST–BHE cooling system are studied and analyzed under different working conditions; third, the feasibility of the HST–BHE cooling system for space cooling and dehumidification in Shenyang is analyzed through a case study in a residential building, and its optimal operation strategy is determined; and finally, the energy saving and economy of the HST–BHE cooling system are analyzed and compared with other cooling systems. The proposed HST–BHE cooling system in this paper is a continuous investigation based on our previous research on the HST–BHE system for soil heat storage. The system design is totally different and so is the research content. The characteristics and application analysis of the HST–BHE system for space cooling and dehumidification are focused on in this paper, while our previous research primarily is focused on the soil heat storage characteristics to solve the soil thermal imbalances of the GHSP system in cold regions.

2. Methodology

2.1. Working Principle of the HST–BHE Cooling System

The working principle of the novel HST–BHE cooling system is given in Figure 2 (presented above). In fact, the novel HST–BHE cooling system is a hybrid system (in terms of direct or indirect contact with the working fluid) and is composed of a heat source tower (HST), a borehole heat exchanger (BHE), a pump, valves, and a pipeline, etc. The BHE is an indirect contact heat exchanger that transfers heat between water and soil, and the HST is a direct contact heat and mass exchanger that transfers heat and moisture between outdoor moist air and water. During the cooling season, the low-temperature circulating water is produced by the BHE and pumped into the HST. Then, heat and mass transfer occur between the circulating water and inlet air (outdoor moist air) in the HST, and cold outlet air of the HST can be achieved. Especially when the temperature of water from the BHE is much lower than the dew point temperature of the outdoor air, the outdoor air can also be dehumidified in the HST. Finally, the cold and dehumidification.

2.2. Mathematical Calculation Model of the HST–BHE Cooling System

The air handling processes of the HST–BHE cooling system is shown schematically in Figure 3.



Figure 3. Air handling processes of the HST-BHE cooling system.

In Figure 3, the state parameters of each state point (points ① to ⑥) are determined empirically based on the meteorological parameters and soil temperature in the typical cold region (climate in the case study) in China. The state parameters of state points ① and ④ are based on the outdoor moist air and indoor cooling design parameters, while those of state points ②, ⑤, and ⑥ are determined based on the heat and mass transfer processes in the HST and BHE. State point ③ is assigned a value considering an air supply temperature rise of 1 °C from state point ② through the fan. Note that points ① to ⑥ are shown in the system in Figure 2 accordingly. It can be seen from Figures 2 and 3 that the cooling and dehumidification capacity of the HST–BHE cooling system depends on the air supply temperature and moisture content (state point ③). Therefore, the state parameters of state point ③ should be determined.

The air supply temperature and moisture content are related to the heat and mass transfer processes in the HST between the air and water. According to our previous research, the air supply temperature and moisture content under different working conditions can be calculated through the numerical calculating model of the HST–BHE system. The mathematical calculation model of the HST–BHE heat storage system is obtained by combining the mathematical models of the HST and BHE when the building and city models are selected. The specific parameters of the HST and BHE model are shown in Table 1.

	Parameter	Value	
IICT	Packing size (height \times depth \times width) (m)	1.96 imes 1.2 imes 0.58	
HSI	Packing specific surface area (m^2/m^3)	200	
BHE	Pipe inner diameter of the BHE (m)	0.026	
	Number of boreholes	28	
	Borehole depth (m)	120	
	Thermal conductivity of backfill materials $(W/(m \cdot K))$	1.97	
	Soil thermal conductivity $(W/(m \cdot K))$	1.78	

Table 1. The specific parameters of the HST–BHE cooling system model.

Then, with MATLAB software (R2020a), the air supply temperature and moisture content can be calculated under different working conditions. Note that the heat and mass transfer process of the HST between air and water in the HST–BHE heat storage system

and the HST–BHE cooling system are the same. So, the calculating method of the HST proposed in our previous paper can be refined for application in this paper.

As a result, according to the air supply temperature (T_3) and moisture content (d_3), the cooling and dehumidification capacity of the HST–BHE cooling system can be calculated as follows [46]:

$$Q_{\rm c,s} = G[1.01 \times (T_4 - T_3) + 0.001 \times 1.84 \times d_3(T_4 - T_3)],\tag{1}$$

$$Q_{c,l} = G[0.001(d_4 - d_3)(2500 + 1.84T_4)],$$
⁽²⁾

$$Q_{\rm c,t} = Q_{\rm c,s} + Q_{\rm c,l},$$
(3)

$$W = 3.6G(d_4 - d_3), \tag{4}$$

where $Q_{c,s}$, $Q_{c,l}$, and $Q_{c,t}$ refer to the sensible cooling capacity, latent cooling capacity, and total cooling capacity, respectively, W; W refers to the dehumidification capacity, kg/h; *G* refers to the air mass flow rate, kg/s; T_3 and T_4 refer to the temperature of space supply air and discharge air, respectively, °C; and d_3 and d_4 refer to the moisture content of space supply air and discharge air, respectively, g/kg.

The performance coefficient (COP) is an important index to evaluate the quality of the cooling system. In the HST–BHE cooling system, the COP can be calculated as follows [43]:

$$COP = \frac{Q_{c,t}}{N_f + N_p},$$
(5)

where $Q_{c,t}$ refers to the total cooling capacity, kW; and N_p and N_f refer to the energy consumption of the water pump and fan, respectively, kW. N_p can be calculated via the following formula [44]:

$$N_{\rm p} = \frac{KHM}{\eta\rho},\tag{6}$$

where η is the efficiency of the water pump, which is assumed to be 70%; *M* is the water flow, kg/s; ρ is the density of water, kg/m³; *K* is the coefficient of motor capacity, which is set to 0.8 based on the commonly used motor capacity in practice [44]; and *H* is the pump head, kPa, which can be calculated according to *Re* [47]:

$$Re = \frac{ud}{v},\tag{7}$$

when *Re* < 4000:

$$h_{\rm f} = 9.8 \times \left(\frac{64}{Re}\right) \frac{l}{d} \frac{u^2}{2g'},\tag{8}$$

when $4000 < Re < 10^5$:

$$h_{\rm f} = 9.8 \times \left(\frac{0.3164}{Re^{0.25}}\right) \frac{l}{d} \frac{u^2}{2g'},\tag{9}$$

$$H = h + h_{\rm f} + h_{\rm j},\tag{10}$$

where *Re* is Reynolds number; *u* is the water flow velocity in the BHE, m/s; *v* is kinematic viscosity, m²/s; *d* is inner diameter of the BHE, m; *l* is the total length of the BHEs, m; *g* is 9.8 m/s²; *h* is static head, kPa; h_f is the friction head loss, kPa; and h_j is the local head loss, kPa.

As to the energy consumption of the fan (N_f) , Figure 4 shows the relationship between the energy consumption of the fan and the air flow rate of HST [48].



Figure 4. Relationship between the energy consumption of the fan and the air flow rate.

It can be seen that the cooling and dehumidification capacity of the HST–BHE cooling system is significantly influenced by the state parameters of space supply air, which are determined by the heat and mass transfer processes occurring in the BHE and HST. Consequently, a quantitative analysis of the heat and mass transfer processes between air and water in the HST, as well as the heat transfer processes between water and soil in the BHE, are necessary to gain a better understanding of the cooling and application characteristics of the HST–BHE cooling system.

2.3. Outdoor Meteorological Parameters

For the HST–BHE cooling system, the inlet air parameters of the HST and soil thermal physical parameters are selected from Shenyang city in a severe-cold region in China to perform characteristics research and application analysis. Figure 5a,b show the hourly outdoor air temperature and moisture content of Shenyang during the cooling season from 7 June to 4 September. The average and maximum outdoor air temperature is 23.8 °C and 34.1 °C, respectively, and the average and maximum outdoor air moisture content is 14.1 g/kg and 26.3 g/kg, respectively. It can be seen that the outdoor air temperature remains consistently high throughout the cooling season. On the other hand, the outdoor air moisture content is lower during the early and end stages and higher during the middle stages of the cooling season.



(a)

Figure 5. Cont.



Figure 5. Meteorological parameters of Shenyang in the cooling season: (**a**) hourly outdoor air temperature; (**b**) hourly outdoor air moisture content.

2.4. Building Model

The application analysis of the HST–BHE cooling system is based on the housing unit of the residential building model in Shenyang city. The residential building model is shown in Figure 6. The standard floor of the residential building is composed of six housing units, covering an area of 600 m². The relevant information of the residential building in Shenyang is given in Table 2. In this paper, the heat transfer coefficients of the building envelopes and indoor heat sources are determined based on current Chinese standards and specifications [49,50]. Moreover, the building cooling and heating loads are calculated via the dynamic energy simulation software DeST (20230713) based on the residential building model.



Figure 6. Housing unit of the residential building.

Area of Housing Unit (m ²)	Area of Standard Floor (m ²)	Heat Transfer Coefficient of Roof/External Wall/External Window (W/(m ² ·K))	Personal Density (Person/m ²)	Lighting/Device Power (W/m ²)
92	600	0.20/0.40/2.0	0.03	5/3.8

Table 2. Building envelope parameters and indoor heat sources.

In the following sections, the cooling and dehumidification characteristics of the HST– BHE cooling system are studied and discussed under the specific calculation conditions. Then, the application analysis of the HST–BHE cooling system in Shenyang city with a housing unit of the residential building model is conducted.

3. Results and Discussion

3.1. Experimental Validation of the HST–BHE Simulation Model

For the validation of the HST–BHE system simulation model, the HST and BHE models were validated by the author's group [42] and by Diao et al. [51], respectively, and the HST–BHE cooling system model was validated through experiments in Zhenjiang city in China. The experimental system is shown in Figure 7, which includes essential components such as a water pump, two electromagnetic flowmeters, two frequency converters, temperature sensors, temperature and humidity sensors, pipeline wind speed sensors, etc. Notably, the system's pump and fan are controlled by a frequency converter to adjust the operation parameters, facilitating the adjustments to the air and water flow rates under different experimental conditions.



Figure 7. Real images of experimental equipment and system: (a) HST; (b) BHE; (c) HST–BHE system.

During the experiment, the outlet air temperature and moisture content of the HST (referred to as state point (2) in Figure 3) are measured, and the cooling capacity and dehumidification capacity are calculated according to the outlet air state of the HST. Therefore, the mathematical model of the HST–BHE system is verified by comparing the experimental and calculated values of the outlet air state of the HST.

The validation results of the HST–BHE mathematical model are shown in Table 3. The experimental example of the case is shown in Figure 8 under both the air and water mass flow rate of 0.8 kg/s. It can be seen from Table 3 and Figure 8 that the calculated values of outlet air temperature and moisture content of the HST exhibit an average relative error of -0.8% and 2.84%, respectively, and a maximum relative error of -4.85% and 8.41%, respectively.

Table 3. Errors of calculated parameters under four working conditions.

Air and Water Flow Rate (kg/s)	Average/Maximum Relative Error of Outlet Air Temperature of HST (°C)	Average/Maximum Relative Error of Outlet Air Moisture Content of HST (g/kg)
0.4/0.8	-1.22%/-4.63%	2.36%/4.87%
0.8/0.8	-0.58%/-4.05%	3.14%/4.87%
1/1.2	-0.51%/-3.27%	2.55%/5.19%
1/1.6	-0.89%/-4.85%	3.29%/8.41%



Figure 8. Comparison of experimental and calculated values when both air and water flow is 0.8 kg/s: (a) outlet air temperature of the HST; (b) outlet air moisture content of the HST.

3.2. Characteristics Research of HST–BHE Cooling System

3.2.1. Specific Calculation Conditions

When the structure parameters of the HST and BHE are fixed, the cooling and dehumidification characteristics of the HST–BHE cooling system are mainly affected by water parameters (mass flow rate, inlet temperature of the HST), outdoor air parameters (temperature, moisture content, mass flow rate), and soil parameters (temperature, thermal properties etc.). This section aims to investigate the characteristics of the HST–BHE cooling system by analyzing the impact of different parameters mentioned above. During the calculations, the air temperature, air moisture content, air relative humidity, air mass flow rate, water mass flow rate, and water inlet temperature are separately changed. The selected specific calculation conditions are shown in Table 4.

 Table 4. Selected specific calculation conditions.

147		Outdoo	Water in HST			
Condition	Temperature (°C)	Moisture Content (g/kg)	Relative Humidity	Flow Rate (kg/s)	Inlet Temperature (° C)	Flow Rate (kg/s)
1	24	14	74%	3–8	13	3
2	20-34	14	95-41%	4	13	3
3	24	6-18	32-95%	4	13	3
4	24	14	74%	4	13	2-8
5	24	14	74%	4	8–16	3

The outdoor air parameters are determined by the meteorological parameters of the Shenyang city model, while the inlet water parameters of the HST are determined by the BHE structure and soil thermal properties. In Table 4, the selected outdoor air parameters and their respective ranges are based on the average outdoor air temperature of 24 °C and moisture content of 14 g/kg during the cooling season in Shenyang. The average inlet water temperature of 13 °C in the HST is based on the deep soil temperature of 8 °C in Shenyang and the average heat transfer temperature differences between water and soil of 5 °C in the BHE [42]. The variation ranges of the air and water mass flow rate are based on the reasonable range of the water-spraying density $(10-40 \text{ t/(m^2 \cdot h)})$ and air speed (1.0-3.0 m/s) of the cross-flow direct contact heat exchanger [52].

In Table 4, the flow rate of air and water is determined based on the calculations and analysis of the system COP and total cooling capacity. First, Figure 9 shows the HST–BHE system COP and total cooling capacity under different air and water mass flow rates, while the dotted line in Figure 9a represents the average COP of 7.0 of the GSHP system for space cooling in Shenyang [53]. The results show that when the HST–BHE system COP needs to be higher than that of the GSHP system, the water mass flow rate should be less than

6.0 kg/s under different air mass flow rates. On the other hand, when considering that the cooling capacity-to-initial-investment ratio of the HST–BHE cooling system is higher than that of the GSHP system (about 1.8 kW/RMB10,000 [54]), the cooling capacity of the HST–BHE cooling system should be higher than 52 kW (calculated in Section 3.3.4) under the different water flow rate. According to the total cooling capacity under different air and water flow rates shown in Figure 9b, it can be concluded that when the cooling capacity of the HST–BHE cooling system needs to be higher than the GHSP system under the same cooling capacity-to-initial-investment ratio, the water and air mass flow rate should be higher than 2.0 kg/s and 3.0 kg/s, respectively. Based on the comparison of COP and the cooling capacity between HST–BHE cooling system and the GSHP system, it is evident that a water flow rate range of 3–5 kg/s and an air flow rate range of 4–8 kg/s should be selected. Note that the calculation results in Figure 7 are based on the specific parameters of the HST–BHE standard model shown in Table 1.



Figure 9. System COP and total cooling capacity under different air and water flow rates: (**a**) COP; (**b**) total cooling capacity.

Further, to determine the optimal air and water flow rate ratio, Table 5 presents the COP ratio (i.e., the ratio of COP at different air and water flow rates to maximum COP) and the total cooling capacity ratio (i.e., the ratio of cooling capacity at different air and water flow rates to maximum cooling capacity) for the optimal air and water flow rates. Note that the sum of the COP ratio and total cooling capacity ratio provides an overall perspective of the ability of the HST–BHE cooling system to balance COP and space cooling. According to Table 5, the maximum sum of the COP ratio and total cooling capacity ratio (1.60) is achieved with an air–water flow ratio of 4:3. Therefore, the optimal air–water flow ratio of 4:3 is selected as the standard working condition.

Table 5. Ratio of COP and total cooling capacity under the superior air and water flow rates.

Air Flow Rate (kg/s)	Water Flow Rate (kg/s)	Ratio of COP	Ratio of Total Cooling Capacity	Sum of Ratio	Air Flow Rate (kg/s)	Water Flow Rate (kg/s)	Ratio of COP	Ratio of Total Cooling Capacity	Sum of Ratio
4	3	0.60	1.00	1.60	6	5	0.87	0.43	1.30
4	4	0.66	0.60	1.26	7	3	0.76	0.69	1.45
4	5	0.69	0.38	1.07	7	4	0.87	0.56	1.43

Air Flow Rate (kg/s)	Water Flow Rate (kg/s)	Ratio of COP	Ratio of Total Cooling Capacity	Sum of Ratio	Air Flow Rate (kg/s)	Water Flow Rate (kg/s)	Ratio of COP	Ratio of Total Cooling Capacity	Sum of Ratio
5	3	0.67	0.90	1.57	7	5	0.94	0.42	1.36
5	4	0.74	0.62	1.36	8	3	0.78	0.58	1.37
5	5	0.79	0.42	1.20	8	4	0.91	0.51	1.42
6	3	0.72	0.81	1.52	8	5	1.00	0.40	1.40
6	4	0.81	0.60	1.41					

Table 5. Cont.

In the following sections, the influences of different parameters on the cooling and dehumidification characteristics of the HST–BHE cooling system are analyzed based on the selected specific calculation conditions in Table 4.

3.2.2. Variation in Air Mass Flow Rate

The cooling characteristics of the HST–BHE cooling system under the change in air mass flow rate are shown in Figure 10. The specific calculation parameters are shown in Table 4 (working condition 1).



Figure 10. Cooling characteristics of the HST–BHE cooling system under different air mass flow rates: (a) difference in air temperature and moisture content; (b) cooling and dehumidification capacity.

Figure 10a shows that increasing the air flow rate from 3 kg/s to 8 kg/s reduces the air temperature difference from 10.6 °C to 7.7 °C and the moisture content difference from 2.0 g/kg to 0.4 g/kg. Additionally, Figure 10b indicates that with the increasing air flow rate, the total cooling capacity of the HST–BHE cooling system increases from 18.0 kW to 72.9 kW, and the sensible cooling capacity increases from 32.7 kW to 63.8 kW, while the latent cooling and dehumidification capacity increase first and then decrease. The maximum latent cooling and dehumidification capacity of 15.9 kW and 22.5 kg/h can be achieved at an air flow rate of 4 kg/s.

According to the heat and mass transfer processes in the HST, the increasing air flow rate enhances the heat and mass transfer coefficient of air and water, consequently increasing the heat transfer quantity of the HST. However, on the air side, the sensible heat quantity is determined by the air flow rate and air temperature difference, while the latent heat quantity is determined by the difference between the air flow rate and the air moisture content. Although the heat quantity proportionally increases with the air flow rate, the increment is smaller compared to the increase in the air flow rate; thus, the temperature difference and moisture content difference decrease accordingly. Consequently, the outlet air temperature and moisture content of the HST decrease with the increase in the air flow rate, as shown in Figure 10a.

In addition, formulae (1) to (4) show that the sensible cooling capacity is determined by the air temperature difference between the space supply air, discharge air, and air mass flow rate, while latent cooling capacity and dehumidification capacity are determined by the moisture content difference and air flow rate. An increase in the air flow rate leads to the temperature and moisture content of outdoor air not being reduced by low-temperature water in a timely manner, subsequently increasing the temperature and moisture content of space supply air. Moreover, the variation trends of the cooling and dehumidification capacity depend on the change in air flow rate, and the difference in air temperature and moisture content between the space supply air and discharge air. The combined effect of these factors leads to the cooling and dehumidification capacity change trends shown in Figure 10b.

3.2.3. Variation in Outdoor Air Temperature

The cooling characteristics of the HST–BHE cooling system under the change in outdoor air temperature are shown in Figure 11. The specific calculation parameters are shown in Table 4 (working condition 2).



Figure 11. Cooling characteristics of the HST–BHE cooling system under different outdoor air temperatures: (a) difference in air temperature and moisture content; (b) cooling and dehumidification capacity.

Figure 11a shows that the air temperature difference decreases from 10.6 °C to 7.8 °C and the moisture content difference decreases from 1.7 g/kg to 1.0 g/kg when the outdoor air temperature increases from 20 °C to 34 °C. Figure 11b indicates that when the outdoor air temperature increases from 20 °C to 34 °C, sensible cooling, latent cooling, total cooling capacity, and dehumidification capacity decrease from 43.5 kW, 17.7 kW, 61.3 kW, and 25.1 kg/h to 32.2 kW, 10.4 kW, 42.5 kW, and 14.6 kg/h, respectively.

When the outdoor air temperature increases, the temperature difference between the outdoor air and water also increases, which accelerates water evaporation and subsequently increases the temperature and moisture content of space supply air. As a result, the difference in air temperature and moisture content between the space supply air and discharge air decreases. Accordingly, the cooling and dehumidification capacity decreases with the increasing outdoor air temperature; the variation trend of the cooling and dehumidification capacity depends on the difference in air temperature and moisture content.

3.2.4. Variation in Outdoor Air Moisture Content

The cooling characteristics of the HST–BHE cooling system under the change in outdoor air moisture content are shown in Figure 12. The specific calculation parameters are shown in Table 4 (working condition 3).



Figure 12. Cooling characteristics of the HST–BHE cooling system under different outdoor air moisture contents: (a) difference in air temperature and moisture content; (b) cooling and dehumidification capacity.

The influences of the outdoor air moisture content in Figure 12 are similar to those of the outdoor air temperature shown in Figure 11. With the increasing outdoor air moisture content from 6 g/kg to 18 g/kg, the air temperature difference decreases from 11.4 $^{\circ}$ C to 9.0 $^{\circ}$ C and the moisture content difference decreases from 3.7 g/kg to 0.4 g/kg. Additionally, the sensible cooling, latent cooling, total cooling capacity, and dehumidification capacity have decrease from 46.7 kW, 37.4 kW, 84 kW, and 52.8 kg/h to 37 kW, 4.4 kW, 41.4 kW, and 6.3 kg/h, respectively.

For the specific water parameters, the increase in the outdoor air moisture content will inevitably increase the moisture content of space supply air, which reduces the latent cooling and dehumidification capacity of the HST–BHE cooling system. Moreover, a high outdoor air moisture content increases the moisture condensation when the air makes contact with water in the HST, which increases the temperature of the space supply air and slightly weakens the sensible cooling capacity.

3.2.5. Variation in Water Mass Flow Rate

The cooling characteristics of the HST–BHE cooling system under the change in water mass flow rate are shown in Figure 13. The specific calculation parameters are shown in Table 4 (working condition 4).

Figure 13a shows that increasing the water mass flow rate from 2 kg/s to 8 kg/s increases the air temperature difference from 9.0 °C to 10.9 °C and the air moisture content difference from 1.0 g/kg to 2.3 g/kg. Additionally, Figure 13b shows that when the water mass flow rate increases, sensible cooling, latent cooling, total cooling capacity, and dehumidification capacity increase from 37.2 kW, 10.1 kW, 47.3 kW, and 14.3 kg/h to 44.8 kW, 23.8 kW, 68.6 kW, and 33.7 kg/h, respectively.

The increase in water flow rate enhances the heat and mass transfer process in the HST, leading to the air extracting more of the cold quality from the water, thus decreasing the temperature and moisture content of the space supply air. As a result, the differences in air temperature and moisture content between the space supply air and discharge air are increased. In addition, the water temperature is limited by the soil temperature, and the

change range is also small in handling the outdoor air, which leads to the differences in air temperature and moisture content increasing slowly with the increasing water mass flow rate. Theoretically, the lowest temperature of the space supply air can be equal to the water temperature, and the lowest moisture content can be equal to that of saturated air under the water temperature. Accordingly, the cooling and dehumidification capacity increase with the increasing water flow rate; the variation trend of cooling and dehumidification capacity depends on the difference in air temperature and moisture content under the specific air mass flow rate.



Figure 13. Cooling characteristics of the HST–BHE cooling system under different water flow rates: (a) difference in air temperature and moisture content; (b) cooling and dehumidification capacity.

3.2.6. Variation in Inlet Water Temperature of the HST

The cooling characteristics of the HST–BHE cooling system under the change in the inlet water temperature of the HST are shown in Figure 14. The specific calculation parameters are shown in Table 4 (working condition 5).



Figure 14. Cooling characteristics of the HST–BHE cooling system under different inlet water temperatures of the HST: (a) difference in air temperature and moisture content; (b) cooling and dehumidification capacity.

Figure 14a shows that increasing the inlet water temperature from 8 °C to 16 °C reduces the air temperature difference from 13.1 °C to 7.8 °C and the moisture content difference from 3.6 g/kg to 0.1 g/kg. Additionally, Figure 14b shows that as the inlet water temperature increases, the sensible cooling, latent cooling, total cooling capacity, and dehumidification capacity exhibit noticeable decreases, decreasing from 54.0 kW, 37.0 kW, 91.0 kW, and 52.3 kg/h to 32.3 kW, 1.3 kW, 33.7 kW, and 1.88 kg/h, respectively.

The increased inlet water temperature of the HST decreases the temperature difference between water and air in the HST and the moisture content difference between the outdoor air and the saturated air under the water temperature. Furthermore, the temperature and moisture content of space supply air are increased. As a result, the cooling and dehumidification capacities decrease because of the increased temperature and moisture content of space supply air.

3.2.7. Characteristic Summary

Based on the analysis of Figures 10–14, several conclusions can be drawn related to the space cooling and dehumidification of the HST–BHE cooling system: (1) an increase in the air flow rate leads to an increased sensible and total cooling capacity, but the latent cooling and dehumidification capacity initially increase and then decrease; (2) an increase in the water flow rate leads to an increased cooling (including sensible, latent, and total cooling capacity) and dehumidification capacity; (3) an increase in the inlet air temperature, moisture content, and water temperature of the HST results in a decreased cooling and dehumidification capacity; (4) the low-temperature water has a limit when it comes to dealing with outdoor air; specifically, the lowest temperature of space supply air equals the water temperature; (5) under the specific calculation conditions, the optimal cooling capacity and COP of the cooling system can be achieved when the air–water flow ratio is 4:3.

3.3. Application Analysis of HST-BHE Cooling System

It can be seen from the above section that the HST–BHE cooling system demonstrates good cooling and dehumidification capacity under different outdoor air temperatures and moisture content. In this section, a city model and building model are selected, and the hourly cooling and dehumidification capacities of the HST–BHE cooling system are calculated based on an air–water flow ratio of 4:3 to analyze the application characteristics of the HST–BHE cooling system.

3.3.1. Hourly Cooling and Dehumidification Characteristics of the HST-BHE System

Figure 15 shows the hourly temperature and moisture content of the space supply air in Shenyang. Note that the time span from 0 to 2160 h in Figure 15 corresponds to the cooling season from 7 June to 4 September in Shenyang. The average temperature and moisture content of the space supply air are 16.1 °C and 11.1 g/kg, respectively, which are calculated according to the average values of the hourly air supply temperature and moisture content throughout the cooling season. It can be seen from Figure 15 that the temperature and moisture content of the space supply air are relatively low during the early and end stages and high during the middle stages of the cooling season, and the variation trends of the temperature and moisture content of the space supply air are influenced by outdoor meteorological parameters. Note that the indoor design temperature and moisture content are set to 26 °C and 14.79 g/kg, respectively, and the HST–BHE cooling system operates to cool and dehumidify the space only when the temperature and moisture content of the space supply air are below the indoor design temperature and moisture content, respectively.



Figure 15. Hourly temperature and moisture content of space supply air in Shenyang (from 7 June to 4 September): (**a**) hourly temperature of space supply air; (**b**) hourly moisture content of space supply air.

Figure 16 shows the hourly cooling and dehumidification capacity of the HST–BHE cooling system in Shenyang. As shown in Figure 16a, the sensible cooling capacity remains stable throughout the operation period, while the latent cooling capacity initially decreases before increasing. The total cooling capacity is the algebraic sum of the sensible and latent cooling capacity, leading to a similar trend to that of the latent cooling capacity. The hourly dehumidification capacity, as shown in Figure 16b, shows the same trend as the latent cooling capacity. The reduction in the latent cooling and dehumidification capacity can be attributed to the increased temperature and moisture content of the space supply air. Moreover, the latent cooling and dehumidification capacity values from 2 August to 3 August, indicating that the system is not suitable for space cooling and dehumidification from 2 August to 3 August. Therefore, it can be seen that the hourly cooling characteristics of the HST–BHE cooling system are significantly affected by meteorological parameters.



Figure 16. Hourly cooling and dehumidification capacity of the HST–BHE cooling system in Shenyang (from 7 June to 4 September): (**a**) hourly cooling capacity; (**b**) hourly dehumidification capacity.

3.3.2. Capacity Matching between the Cooling System and Building Cooling Load

According to the hourly cooling and dehumidification characteristics, the total cooling and dehumidification capacity of the HST–BHE cooling system exhibits high values during the early and end stages but low values during the middle of the cooling season. However, the building cooling and dehumidification load typically exhibits an opposite trend to the cooling and dehumidification capacity of the HST–BHE cooling system. Therefore, it is essential to match the cooling and dehumidification capacity of the HST–BHE cooling system with the building cooling and dehumidification load in actual engineering applications. Generally, simultaneously meeting both cooling and dehumidification loads is a challenge for the HST–BHE cooling system. Considering that meeting the building cooling load is more important than the dehumidification load in acquire control of building environment, this paper proposes to prioritize the building cooling load followed by the building dehumidification load when matching the HST–BHE cooling system with the building.

Figure 17 exhibits the matching between the cooling capacity of the HST–BHE cooling system and the building cooling load under different residential building areas in Shenyang. Note that the building cooling load with different building areas is calculated by increasing

the standard floor number of the residential building, and that non-guaranteed hours represent the time in which the cooling system cannot satisfy the building cooling load. As shown in Figure 17, the results indicate an increase in the building cooling load and the non-guaranteed hours of the HST–BHE cooling system with an increase in building areas. Moreover, during the early period (7 June–1 July) and the end period (3 August–1 September), the hourly cooling capacity of the HST–BHE cooling system is much higher than the building cooling load. The ratios of the hourly cooling load to the building cooling load are 5182%, 2541%, 1661%, and 1220% for the buildings with areas of 600, 1200, 1800, and 2400 m², respectively.



Figure 17. Match between the cooling capacity and the building cooling load in Shenyang (from 7 June to 4 September).

In order to determine the maximum building area that the HST–BHE cooling system could satisfy under the standard model, Table 6 shows the non-guaranteed hours of the HST–BHE cooling system under different residential building areas. The non-guaranteed hours of the cooling system should be controlled for under 50 h [55]. It can be concluded from Table 6 that when the residential building area is lower than 1800 m², the HST–BHE cooling system could satisfy the cooling load under the standard model.

 Table 6.
 Non-guaranteed hours of the HST-BHE cooling system under different residential building areas.

Residential Building Area (m ²)	Accumulated Building Cooling Load (MWh)	Non-Guaranteed Hours (h)
600	8.6	4
1200	17.2	18
1800	25.7	46
2400	34.3	73

Figure 18 shows the matching between the dehumidification capacity of the HST–BHE cooling system and the residential building with an area of 1800 m². The results reveal that the HST–BHE cooling system cannot satisfy the building dehumidification load for only 2.7% of the time (59 h), which can be considered sufficient to satisfy the building with an area of 1800 m².



Figure 18. Matching between the dehumidification capacity and the building dehumidification load for a residential building with an area of 1800 m² (from 7 June to 4 September).

As a result, based on the standard model of the HST–BHE cooling system, the cooling and dehumidification capacity of the HST–BHE cooling system can satisfy the cooling and dehumidification load of a residential building with a maximum area of 1800 m². However, during the early period (7 June–1 July) and the later period (3 August–1 September), the hourly cooling capacity of the HST–BHE cooling system is 1661% higher than the building cooling load, which can only be discharged to the outside, resulting in a waste of the cooling and dehumidification capacity of the system. Therefore, the operation of the HST–BHE cooling system during the early and late stages of the cooling season should be optimized to reduce cooling and dehumidification capacity and better match the building cooling load.

3.3.3. System Operation Optimization

The outdoor air temperature and moisture content are determined by the city model, while the inlet water temperature of HST is determined by both the standard model of the HST–BHE cooling system and the city model. Consequently, adjusting the flow rates of the air or water is the primary means of optimizing the system operation in Shenyang. In addition, the index used to evaluate the optimization effect is the matching degree of cooling capacity and building cooling load during the early and end periods. When the cooling capacity of the HST–BHE cooling system exceeds the building cooling load, it is recommended to reduce the cooling capacity to approach the building cooling load to avoid wasting the cooling capacity and improve the system COP. Figure 19 shows the optimization result of the HST–BHE cooling system when the air flow rate decreases from 4 kg/s to 3 kg/s and water flow rate decreases from 3 kg/s to 2 kg/s, respectively.

As shown in Figure 19a, reducing the air and water flow rates during the two optimization periods can avoid more waste in system cooling capacity in the early and end periods of the cooling season. Figure 19b indicates that the dehumidification capacity of the optimized system decreases less than the cooling capacity. Figure 19c shows that the COP of the optimized system increases with the decrease in air and water flow. Furthermore, it is observed that adjusting the air–water flow rate ratio to 3:2 during the two optimization periods results in the least amount of cooling capacity waste and the highest increase in COP. Therefore, the optimal optimization result can be determined at an air–water flow rate of 3 kg/s and 2 kg/s, respectively.



Figure 19. Optimization results of the HST–BHE cooling system when the air flow and water flow decrease, respectively (from 7 June to 4 September): (**a**) total cooling capacity; (**b**) dehumidification capacity; (**c**) COP.

Table 7 shows the cooling characteristics of the cooling system before and after optimization in Shenyang throughout the cooling season. The results indicate that after optimization, the 22.2 MWh waste amount of accumulated cooling capacity can be avoided and the average COP of the cooling system can be improved by 42.6%, while the nonguaranteed rates of cooling and dehumidification remain unchanged.

Before Optimization	After Optimization
4:3 (7 June to 4 September)	4:3 (1 July–3 August, 1–4 September) 3:2 (7–30 June, 4–31 August)
170.1	147.9
53.6	46.4
6.1	8.7
46	46
59	59
	Before Optimization 4:3 (7 June to 4 September) 170.1 53.6 6.1 46 59

Table 7. Cooling characteristics before and after optimization in Shenyang.

3.3.4. Energy Consumption and Economic Analysis

In order to investigate the energy saving potential of the HST–BHE cooling system compared to traditional cooling systems, the energy consumption of different cooling systems is calculated and shown in Table 8. Considering coal accounts for 70% of China's energy consumption, electricity is converted to coal as the primary energy for a fair comparison, and the power generation efficiency of coal is considered to be 0.33 [56]. As shown in Table 8, the energy saving rate of the HST–BHE system is 123% and 26% higher than that of the air source heat pump (ASHP) system and the GSHP system, respectively.

Table 8. Primary energy efficiency of different cooling systems.

	Cooling Capacity (MWh)	System COP	Primary Energy Consumption (kW)	Primary Energy Efficiency
ASHP system [57]	147.9	4.0	112.0	1.3
GSHP system [53]	147.9	7.0	64.0	2.3
HST-BHE cooling system	147.9	8.7	51.5	2.9

Table 9 shows the primary equipment investment of the HST–BHE cooling system, with the manufacturer providing the unit prices of each piece of equipment. It can be observed that the borehole represents the highest cost of the HST–BHE cooling system. Additionally, when the HST–BHE cooling system is integrated with the existing GSHP system, only the HST and water pump costs are necessary, representing only 5.4% of the total investment of the HST–BHE cooling system.

Table 9. Primary equipment investment of the HST-BHE cooling system.

Unit	Capacity	Unit Price (CNY per Capacity)	Cost (CNY)
Borehole	28	9600	268,800
HST (m ²)	273	35	9555
Water pump (m^3/h)	28.8	200	5760
Initial investment	-	-	284,115

Figure 20 compares the operating cost and total cost (initial investment and operating costs) of the HST–BHE cooling system with other systems over 20 years. The results show that the operating cost of the HST–BHE cooling system is 52.6% and 73.0% lower than that of the GSHP and ASHP systems over 20 years, respectively. By integrating HST with the existing BHE in the GSHP, the total cost of the HST–BHE system can be reduced by

60.6% over 20 years. The initial investment of the HST–BHE system with the new BHE is 198.0% higher than that of the ASHP, but the total cost is lower than that of the AHSP starting from the 9th year and 35.3% lower in the 20th year. While the initial investment cost of the HST–BHE system with an existing BHE in the GSHP system is 83.9% lower than that of the ASHP, and the total cost is 74.5% lower than that of the AHSP over 20 years. Consequently, it is best to integrate the HST with the existing BHE in the GSHP, which can not only reduce operating costs for space cooling but also reduce the initial investment of the HST–BHE system. In actual application, whether or not it is integrated with GSHP to form the HST–BHE system, the HST–BHE system always has better economy over a 20-year lifecycle than the AHSP system.



Figure 20. Economic comparison of the HST-BHE system with other systems over 20 years.

4. Conclusions

In this paper, a novel hybrid HST–BHE cooling system is proposed to facilitate the use of natural cooling sources for space cooling. Based on the developed mathematical HST–BHE cooling system model, city model, and residential building model, the cooling characteristics, operation strategy, energy saving rate, and economy of the HST–BHE cooling system were studied. The main conclusions are as follows.

The error rate between the mathematical model and the experimental set up of the HST–BHE system is less than 10%. The cooling and dehumidification capacity of the HST–BHE cooling system decreases with the increase in the inlet air temperature, moisture content, and water temperature of the HST. The cooling capacity and COP of the HST–BHE cooling system can be adjusted by changing the air and water flow rate. Under the specific calculation conditions, when the air and water flow rates are 4 and 3 kg/s, the optimal system COP of about 15.0 and the total cooling capacity of 55 kW of the HST–BHE cooling system can be achieved.

Based on the residential building models in Shenyang city, the hourly cooling and dehumidification capacity of the HST–BHE cooling system initially decreases and then increases during the cooling season and can meet the demands of a residential building with an area 1800 m^2 . By optimizing the air flow rates from 4 kg/s to/3 kg/s and the water flow rates from 3 kg/s to 2 kg/s, the 22.2 MWh waste amount of accumulated cooling capacity can be avoided, the average COP of the cooling system can be improved by 42.6%, and the non-guaranteed rates of cooling and dehumidification are maintained for 59 h during the early and end periods of the cooling season.

Compared to the ASHP and GSHP systems, the HST–BHE cooling system has energy saving rates of 123% and 26% and operating cost saving rates of 52.6% and 73.0% over 20 years, respectively. The initial investment of the HST–BHE system with the new BHE is 198.0% higher than that of the ASHP, but the total cost is lower than that of the AHSP

starting from the 9th year and 35.3% lower in the 20th year. When integrating the HST with the existing BHE in the GSHP, 60.6% of the total cost of the HST–BHE system can be further reduced over 20 years.

It is important to recognize that the proposed HST–BHE system is suitable only for space cooling and dehumidification in cold and severe-cold regions. Moreover, several implementation details require attention in practical applications, such as the distribution of the BHE when integrating the HST with the GSHP and the packing adjustment and modularization of the HST when a single HST is insufficient to meet the building cooling load.

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