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Optimal Operation Control of Composite Ground Source Heat Pump System

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Abstract: During the operation of the ground source heat pump (GSHP) system, the operations of the chiller system should be controlled by adjusting the difference between water temperature and wet bulb temperature. Therefore, it is important to consider the control strategy for the switch time (ST) and wet bulb temperature difference (WBTD) of the chiller system. This paper sets up two control strategies, namely, the strategy to control the ST of system operations, and the strategy to control the WBTD. Then, theoretical modeling was carried out to compare the system energy consumption and borehole wall temperature under different strategies. The modeling results were referred to optimize the control strategy for composite GSHP systems. It was found that, under the ST control strategy, the best wet bulb temperature is 2℃, and the best chiller operation hours are 3h; under the WBTD control strategy, the best wet bulb temperature is 3.5℃, and the best WBTD is 1.5℃. In addition, the ST control strategy is superior to the WBTD control strategy, in terms of system energy consumption, borehole wall temperature and initial investment.

Keywords: Ground source heat pump (GSHP); Optimization; Switch time (ST); Wet bulb temperature difference (WBTD)

1. Introduction

In the ground source heat pump (GSHP) system, the change of mean soil temperature can be delayed by increasing the borehole depth or borehole spacing of the buried pipes. But this will widen the steady-state temperature difference, increase the area occupied by the unbalanced load, and push up the initial investment of the system. One of the feasible solutions is the composite GSHP system, which integrates a chiller system into the GSHP system. The composite system can realize cooling in summer with a limited number of buried pipes, save initial investment and provide good economy.

At present, composite GSHP systems have been explored extensively at home and abroad. Ashare [1] expounded the advantages of mixed GSHP system in public construction for the first time, and designed an auxiliary heat dissipator mainly for air conditioning Kavanaugh et al. [2, 3] recognized the high installation cost of buried pipe heat exchanger as a major reason for using the hybrid system, put forward a composite GSHP system, and found a way to determine the capacity of auxiliary heat dissipators. Yavuzturk et al. [4-6] simulated and compared different equipment combinations and operation strategies of cooling tower-assisted composite GSHP [4-6], and analyzed the influence of different control strategies on system performance, providing a reference for the design and control of auxiliary GSHP cooling system. Spitler et al. [7-9] explored the form and operation effect of the composite GSHP extensively, summarized various forms of auxiliary heat sinks, and pointed out their advantages and disadvantages. Jeon et al. [10] further improved the control strategy of cooling tower by comparing the mixed GSHP system with the traditional GSHP system. Hackel and Pertzborn [11] established an energy-based thermodynamic model for the auxiliary cooling system of the cooling tower, presented three economic optimization schemes, and compared the experimental results of various optimization schemes. Park et al. evaluated the overall performance of a composite GSHP system with underground heat exchangers and parallel auxiliary radiators, while adjusting the flow rates of the refrigerant and circulating water in the refrigeration mode

[12].

With the aim of maintaining soil heat balance, many control strategies have been developed to improve the efficiency and energy saving effect of composite GSHP systems. Relevant scholars have worked hard to improve the efficiency of these systems, and proposed different control strategies. These strategies generally fall into three categories: fixed value method, temperature difference control method, and fixed period method [13-19].

Drawing on the previous studies, this paper sets up two control strategies, namely, the strategy to control the switch time (ST) of system operations, and the strategy to control the wet bulb temperature difference (WBTD). Then, theoretical modeling was carried out to compare the system energy consumption and borehole wall temperature under different strategies. The modeling results were referred to optimize the control strategy for composite GSHP systems.

2. Theoretical Model

2.1 GSHP System

The GSHP is a highly efficient, energy-saving, and environmentally friendly air conditioning system, which uses shallow underground geothermal resources for both heating and refrigeration. With a small amount of high-grade energy (electric energy), the GSHP can switch from a low temperature heat source to a high temperature heat source. In winter, the heat is taken out from the soil, and used to warm the air supply to indoor spaces. In summer, the heat indoor is released to the soil. In this way, the system ensures the balance of underground temperature throughout the year. Figure 1 illustrates the operation of a typical GSHP system.

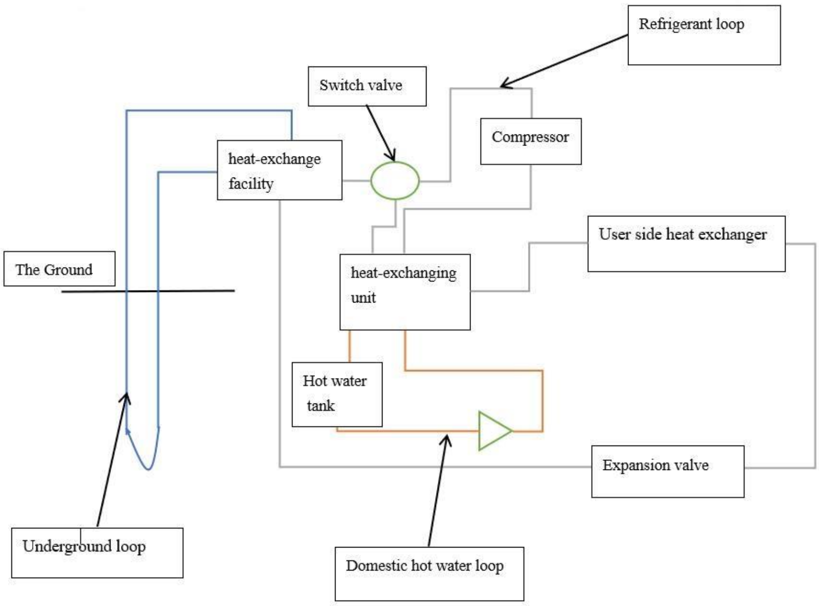

Figure 1. Operation diagram of the GSHP system

The buried pipe heat exchanger in the GSHP system can be described by a quasi-three-dimensional model, while the finite length heat source is outside the borehole. Here, the change of the fluid temperature in the depth direction is considered, along with the heat convection in the axial direction, both of which take place under the actual conditions. Meanwhile, the axial heat conduction in the borehole is ignored. On this basis, the steady-state heat transfer model of quasi-three-dimensional boreholes can be expressed by the following energy balance equations:

$$-\frac{\mathrm{d}\theta_{\mathrm{d}}}{\mathrm{d}Z}=\frac{\theta_{\mathrm{d}}}{S_1}+\frac{\theta_{\mathrm{d}}-\theta_{\mathrm{u}}}{S_{13}} \tag{1}$$

$$\frac{\mathrm{d}\theta_{\mathrm{u}}}{\mathrm{d}Z}=\frac{\theta_{\mathrm{u}}}{S_1}+\frac{\theta_{\mathrm{u}}-\theta_{\mathrm{d}}}{S_{13}} \tag{2}$$

where, θ_d and θ_u are the downward and upward dimensionless temperatures in the buried pipe, respectively. The two temperatures can be calculated by $\theta_d = \frac{t_{f1}(z)-t_b}{t_{f,\text{in}}-t_b}$ and $\theta_u = \frac{t_{f1}(z)-t_b}{t_{f,\text{in}}-t_b}$, with z being the depth at the calculation point (m); $t_{f1}(z)$ being the water temperature at point z (°C); $t_{f,in}$ being the inlet water temperature of the buried pipe (°C); t_b being the temperature of the borehole wall (°C); Z being the point corresponding to the dimensionless depth; S_1 and S_{13} being the dimensionless thermal resistances.

The dimensionless definite solution of the above equations is $\theta_d(0)=1$ and $\theta_d(1)=\theta_u(1)$, which satisfies the parallel arrangement of a single U-shaped buried pipe and double U-shaped buried pipes. The Laplacian transform can be performed to solve the above system of ordinary differential equations. Thus, the dimensionless solution can be obtained as:

$$\theta_{\mathrm{d}}(z) = \mathrm{ch}(\beta z) - \frac{1}{\beta S_{13}}\left[\left(\frac{S_{13}}{S_1}+1\right) - \frac{\beta S_1 \cdot \mathrm{ch}\,\beta - \mathrm{sh}\,\beta}{\beta S_1 \cdot \mathrm{ch}\,\beta + \mathrm{sh}\,\beta}\right]\mathrm{sh}(\beta z) \tag{3}$$

$$\theta_{\mathrm{u}}(z) = \frac{\beta S_1 \cdot \mathrm{ch}\,\beta - \mathrm{sh}\,\beta}{\beta S_1 \cdot \mathrm{ch}\,\beta + \mathrm{sh}\,\beta}\mathrm{ch}(\beta z) - \frac{1}{\beta S_{13}}\left[1 - \left(\frac{S_{13}}{S_1}+1\right)\frac{\beta S_1 \cdot \mathrm{ch}\,\beta - \mathrm{sh}\,\beta}{\beta S_1 \cdot \mathrm{ch}\,\beta + \mathrm{sh}\,\beta}\right]\mathrm{sh}(\beta z) \tag{4}$$

where, $\beta = \sqrt{\frac{1}{S_1^2} + \frac{2}{S_1 S_{13}}}$.

Let t_0 be the initial temperature in a semi-infinite medium. Then, the boundary temperature always remains at that level. Suppose the finite length heat source q starts to transfer heat from the vertical boundary surface, the temperature distribution in the cylindrical coordinate system will change in two dimensions, with t_0 being the zero point of excess temperature.

Following the principle of the virtual heat source method, a virtual linear heat sink is set at the position equal to the boundary of the linear heat source, with an intensity of -q. Then, the distribution of the model temperature field can be expressed as:

$$t(r,z,T) - t_0 = \frac{q_l}{4\pi k_s}\int_0^H \left\{ \frac{erfc\left[\frac{\sqrt{r^2+(z-h)^2}}{2\sqrt{aT}}\right]}{\sqrt{r^2+(z-h)^2}} - \frac{erfc\left[\frac{\sqrt{r^2+(z+h)^2}}{2\sqrt{aT}}\right]}{\sqrt{r^2+(z+h)^2}} \right\} dh \tag{5}$$

where, r is the distance from the center of the borehole (m); t is time (s); q_l is the heat flow density (W / m^2); K_s is the thermal conductivity of the rock-soil mass (W / (m·K)); a is the thermal diffusion rate of the rock-soil mass (m^2/s).

Then, the equivalent thermal resistance can be expressed as:

$$R_s(r,z,T) = \frac{t(r,z,T) - t_0}{q_l} = \frac{1}{4\pi k}\int_0^H \left\{ \frac{erfc\left[\frac{\sqrt{r^2+(z-h)^2}}{2\sqrt{aT}}\right]}{\sqrt{r^2+(z-h)^2}} - \frac{erfc\left[\frac{\sqrt{r^2+(z+h)^2}}{2\sqrt{aT}}\right]}{\sqrt{r^2+(z+h)^2}} \right\} dh \tag{6}$$

2.2 Chiller System

In the chiller system, the volume dispersion coefficient of the cooling tower packing is denoted by $\beta_{\chi v}$. This parameter measures the thermal properties, and plays an important role in computing the heat transfer of cooling tower. The parameter value is normally obtained through filler tests on cooling tower packing. The value of $\beta_{\chi v}$ can be calculated by:

$$\beta_{\chi v} = 0.57 q^{0.56} g^{0.35} \tag{7}$$

where, $\beta_{\chi v}$ is the volume dispersion coefficient, kg/(m^3·h); q is the shower density of the cooling tower, kg/(m^2·h); g is the air gravity of the cooling tower, kg/(m^2·h).

To solve the heat transfer model of the cooling tower, it is necessary to obtain the time-wise meteorological parameters of the outdoor air, including dry bulb temperature, relative humidity, water vapor separation pressure, air enthalpy, and wet bulb temperature.

3. Calculation Method and Control Strategies

3.1 Calculation Method

The composite GSHP system mainly includes two parts: the ground source and the chiller. Both parts need to consider the constant flow solution, which involves load distribution and subsystem calculation.

3.1.1 Load distribution

Load distribution is the basic condition of the composite system. It should be designed in the light of the carrying capacity and energy efficiency of each unit in the system. Specifically, the time-by-time air conditioning load of the building is denoted as $Q_0[i]$, the rated cooling capacity of the heat pump as Q_{l1}, the rated cooling capacity of the chiller as Q_{l2}, the load assigned by the heat pump as Q_1[i], the load assigned by the chiller as $Q_2[i]$. The distribution scheme of the air-conditioning load is as follows:

(1) If $Q_0[i] <0$, i.e., during the winter heating, the chiller does not bear any load Q_2[i]=0, while the heat pump carries the load $Q_1[i]= Q_0[i]$.

(2) If $Q_0[i] \le Q_{l1}$, i.e., during the summer cooling, the cooling load does not exceed the limit of the heat pump or the chiller, and both devices can operate normally. The respective loads of the system, $Q_1[i]= Q_0[i]$ and $Q_2[i] =0$ or $Q_2[i]= Q_0[i]$ and $Q_1[i]=0$, can be determined according to the heat transfer features of the buried pipe and the operation features of the cooling tower.

(3) If $Q_{l1} \le Q_0[i] \le Q_{l2}$, i.e., during the summer cooling, the cooling load exceeds the limit of the heat pump, yet does not surpass the range of the chiller. Then, the chiller alone bears the cooling load: $Q_1[i]=0$ and $Q_2[i]= Q_0[i]$.

(4) If $Q_0[i]>Q_{l2}$, i.e., during the summer cooling, the cooling load surpasses the rated cooling capacities of both the heat pump and the chiller. Thus, the two devices must be open simultaneously. Suppose the two devices have the same temperature difference between inlet water and outlet water, and the water flow equals the rated flow rate. Then, the cooling load of the heat pump and the chiller can be distributed according to the rated cooling capacity. Thus, the cooling load of the heat pump can be expressed as:

$$Q_1[i] = \frac{Q_{l1}}{Q_{l1} + Q_{l2}} Q_0[i] \tag{8}$$

The cooling load of the chiller can be expressed as:

$$Q_2[i] = \frac{Q_{l2}}{Q_{l1} + Q_{l2}} Q_0[i] = Q_0[i] - Q_1[i] \tag{9}$$

3.1.2 Subsystem calculation

For the heat transfer model of buried pipes, the borehole in the area of the pipes is taken as the worst borehole. A total of eight boreholes, including the four boreholes opened in the recent week and the four boreholes around the borehole, are selected. Then, the joint influence of the eight boreholes on the thermal resistance of the borehole is considered, without taking account of the effect of other distant boreholes.

For the chiller model, the inlet water temperature of the cooling tower is derived from such factors as the inlet water temperature on the condensing side, the unit performance curve, as well as the performance coefficient, energy consumption, heat dissipation, and outlet capacity of the chiller. For the energy consumption of condensing side and indoor side, time-by-time statistics are used to depict the power consumption of the circulating pump.

For the heat transfer model of the cooling tower, the volume dispersion coefficient of the cooling tower packing, and the gas-water ratio are derived from the shower water density, air gravity, and wind speed. The enthalpy difference method is used to obtain the feature coefficient N_l of the cooling tower. For the outlet temperature of the cooling tower, the cooling number N' is obtained through integral approximation by Simpson's rule. When the minimum outlet water temperature of the cooling tower varies by the smallest degree, this temperature is treated as the outlet water temperature of the tower, as well as the inlet water temperature of the condensation side at the next moment of the chiller. Then, the power consumption of the fan is counted at this time.

Following the calculation strategy above, the hourly cooling and heating loads of the air conditioner can be calculated using the DeST software for energy consumption simulation, according to the device power, mean water temperature in the heat exchanger, the borehole wall temperature, the power consumption of the cooling tower, as well as the outlet water temperature.

3.2 Control Strategy

When the GSHP system is switched to the chiller, the operation time of the chiller system is controlled by the difference between the inlet temperature of the GSHP and the local wet bulb temperature. Two different control strategies are considered, when the chiller system is switched to the GSHP system. Figure 2 illustrates the control strategy of the composite GSHP system. The two strategies are implemented based on C#.

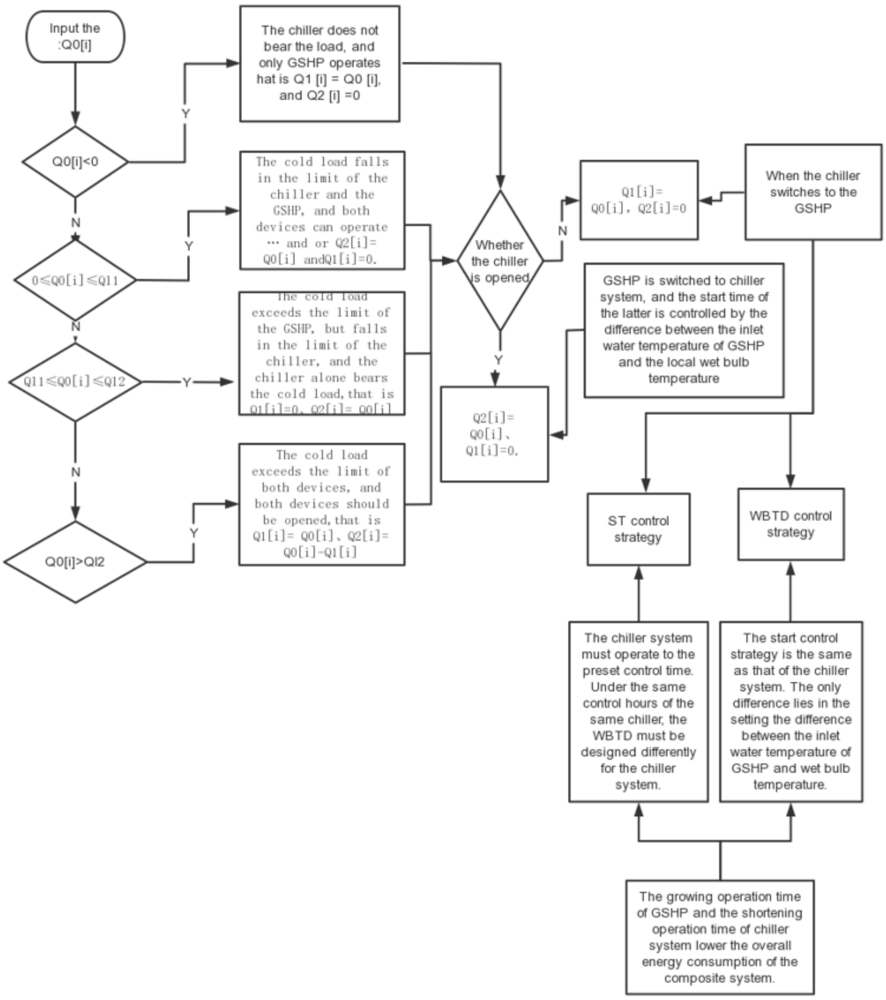

Figure 2. Schematic diagram of the control strategies of the composite GSHP system

3.2.1 ST control strategy

After the GSHP system is switched to the chiller system through WBTD control, the chiller system must run to the set control time to switch back to the GSHP system, leaving enough time for the underground soil temperature of the GSHP system to recover. This also ensures that the two systems will operate stably, eliminating the need for frequent switch between them.

The range of the control temperature difference was set to 0-5℃, and the difference was increased from 0℃ to 5℃ at a step length of 0.5℃. The continuous runtime was set to 1-4h. Dependent on the time-wise load of the DeST software, the control time must be an integer. Thus, the interval of control time was set to 1h. Without changing the control hours of the chiller, the control time of the WBTD was configured for different chiller systems. Then, the total energy consumptions of the two control strategies were compared, as well as the borehole wall temperatures of the buried pipe heat exchanger. Then, the scheme with the smallest energy consumption and lowest rise of borehole wall temperature was selected as the best control strategy.

3.2.2 WBTD control strategy

The same control strategy was applied to the opening of the chiller system, but with another difference between the inlet water temperature of the GSHP and the wet bulb temperature. Using the control switch of wet ball temperature difference, the GSHP system switches to the independently running chiller system. Then, the GSHP stops running, and the inlet water temperature of the GSHP is theoretically equivalent to the borehole wall temperature. If this temperature and the outdoor wet bulb temperature drop below the set value, the chiller system should be shutdown and switched to the GSHP system.

The range of the control temperature difference was set to 0-5℃, and the difference was increased from 0℃ to

5℃ at a step length of 0.5℃. The set control temperature difference Δt_1 is greater than the opening temperature difference of the chiller. Thus, the soil temperature of the GSHP should be restored to a certain extent to run the GSHP again. This would assure the energy efficiency of the GSHP operation and stabilize the system control (to prevent frequent switching between the chiller system and the GSHP system). The control temperature difference was thus controlled to 0.5~4.5℃, and increased by a step length of 0.5℃.

Firstly, open the WBTD setting Δt_1 in the same chiller system. Next, set the closing WBTD setting Δt_2 for different chiller systems. Then, compare the total energy consumption of the system under different control strategies, and contrast the borehole wall temperature of the buried pipe heat exchanger. Finally, select the scheme with the highest energy efficiency and smallest temperature rise of the borehole wall as the best control strategy.

Once ST control is added to WBTD control, the WBTD control needs to monitor the borehole wall temperature of the underground buried tube heat exchanger. But not all GSHP projects have a borehole wall temperature monitoring system. To realize the ST control strategy, it is necessary to monitor the borehole wall temperature of the underground heat exchanger.

4. Comparative Analysis

4.1 Project Overview

The target project is the GSHP air conditioning system of a complex building in Chongqing. In the system, the buried pipe heat exchange system was configured according to the winter demand of heat exchange. Thus, the double U-type buried pipe heat exchanger was adopted in the system. The designed heat exchange per meter is 45 W/m, and the borehole depth is 100m. The maximum heat absorption of the buried tube heat exchanger was calculated based on the standard conditions of GSHP. Considering the heat and heat power differences between GSHP units, the buried tube could absorb 345.3 kW of heat from the soil in winter. Using the rich coefficient of 1.1, a total of 84 boreholes were arranged for the double U-type buried pipe heat exchanger, with a spacing of 5m. Traditionally, a GSHP system needs 198 boreholes. Thus, the initial investment of the composite GSHP system was 58% lower than that of the traditional system.

Next, the composite GSHP system was simulated under different control strategies, with a calculation period of 7 years. The operating time of the system was configured as follows: the cooling period in summer is from June 1 to September 30, and the heating period in winter is from December 1 to February 28. The initial operating time of the system was set as 00:00 on June 1.

4.2 Strategy Optimization

4.2.1 ST optimization

As shown in Figure 3, the total energy consumption of the system varies with the set values of the specific control strategy. When the running time of the chiller is fixed, as the opening WBTD set value of the chiller grows, the energy consumption of the GSHP system increases gradually, while that of the chiller system drops slowly; the total energy consumption of the composite system also undergoes a gradual decline. When the opening WBTD set value of the chiller remains unchanged, as the running time of the chiller shortens, the energy consumption of the GSHP system increases gradually, while that of the chiller system drops slowly; the total energy consumption of the composite system also undergoes a gradual decline.

The main reason is that the growing opening WBTD set value or the reducing running time of the chiller will extend the operation time of the GSHP system, and shorten that of the chiller system. In this case, the lateral condensation water temperature of the GSHP heat pumps is lower than the water temperature of the chiller. Since the GSHP operates with a higher energy efficiency than the chiller system, the extending operation time of the GSHP and the falling operation time of the chiller will reduce the total energy consumption of the entire composite system.

From the perspective of energy consumption, under the various combinations of the set values in ST, the total energy consumption of the composite system in 7 years maximized at 1,918.28mWh. This corresponds to the longest running time of the GSHP system: (0℃, 4 h). The total energy consumption of the composite system in 7 years minimized at 1,873.80mWh. This corresponds to the shortest running time of the GSHP system (5℃, 1 h). Under different control strategies, the difference between the maximum and the minimum energy consumptions of the composite system is 44.48 MWh, only 2.32% of the maximum energy consumption. This means the set value of the control strategy has little to do with the total energy consumption of the composite system. The reason is as follows:

In the building air conditioning system, the cooling load under the various control strategies is smaller than the rated cooling capacity of the GSHP system in that period. In this stage, the GSHP system only takes a small proportion of the energy consumed by the entire composite system throughout the load period. The energy consumed by the GSHP has a limited impact on the energy consumption of the entire system.

The chiller still needs to run in other load periods, and adds to the heat dissipation. Under different set values of the control strategy, the inlet water temperature and performance of the chiller and the GSHP system have little difference, and thus consume largely the same amount of energy. Therefore, there is no significant variation in the total energy consumed by the composite system, despite the change between control strategies.

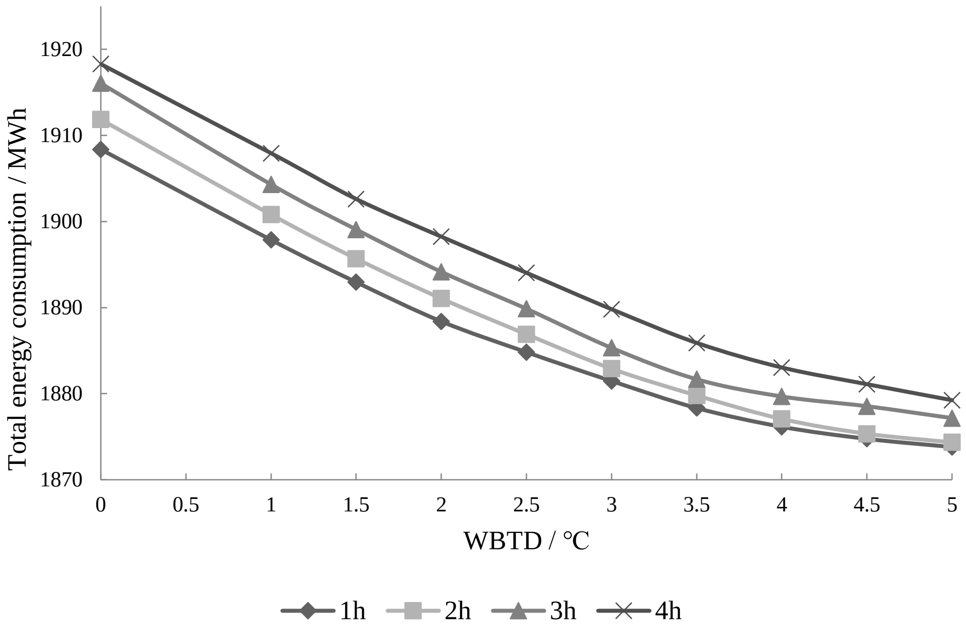

Figure 3. Total energy consumption of the composite system in 7 years

Figure 4 shows the borehole wall temperature of the composite system after 7 years. When the running hours of the chiller remains the same, as the set value of WBTD for chiller opening, the borehole wall temperature gradually rises. When the set value of WBTD for chiller opening remains unchanged, that temperature increases with the reducing running hours of the chiller. The main reason lies in the growth of the set value of WBTD for chiller opening, or the reduction in the set running hours of the chiller. Thus, the GSHP system needs to run for more hours, releasing more heat into the soil. The soil temperature thus increases, and so does the borehole wall. As a result, the GSHP system consumes more energy.

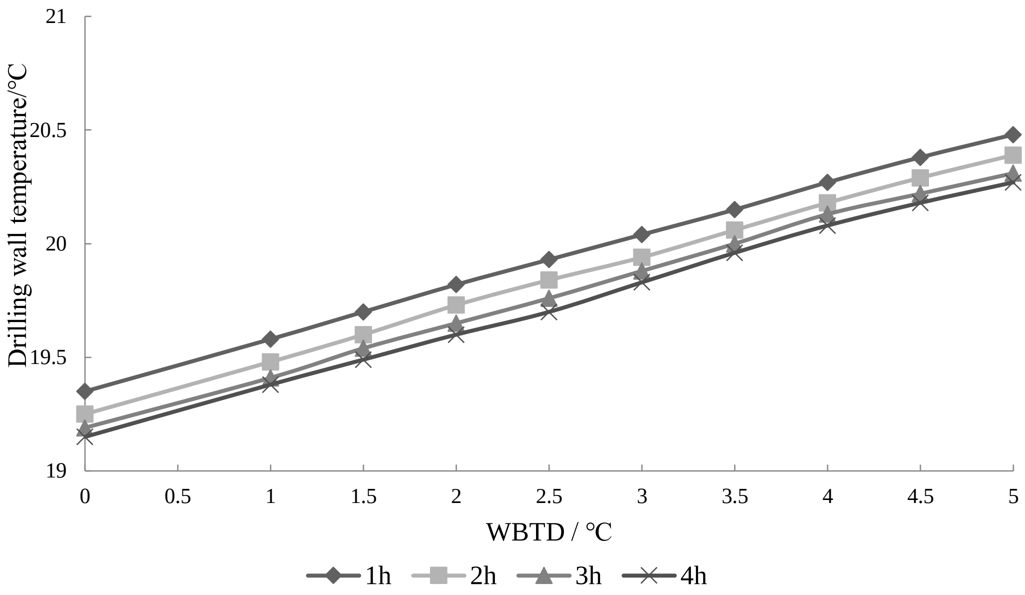

Figure 4. Borehole wall temperature at the end of 7 years

By increasing the WBTD set value of the chiller opening or reducing the running hours of the chiller, the GSHP system will run for a longer time, while the chiller system will operate for a shorter time. Then, the entire composite system will consume fewer energy, and the borehole wall temperature of the buried pipe heat exchanger will gradually increase.

The best control strategy should minimize the total energy consumed by the composite system, without changing the borehole wall temperature. Following this principle, the total energy consumption of the composite system and the borehole wall temperature were compared, under different running hours of the chiller. When the chiller operates for different time, the total energy consumption of the composite system always intersects the borehole wall temperature, when the WBTD of the chiller is set at 2℃. When the WBTD of the chiller is below that level, the total energy consumption of the composite system would rise, while the borehole wall temperature would fall. When the WBTD of the chiller is above that level, the total energy consumption of the composite system would fall, while the borehole wall temperature would rise. Overall, when the chiller operates for different time the optimal control strategy is to set the WBTD of the chiller to 2℃, which optimizes the total energy consumption of the composite system and the borehole wall temperature.

In addition, under different WBTD set values, the total energy consumption of the composite system always intersects the borehole wall temperature, when the chiller runs for 2h and 3h. Since the total energy consumption does not change, when the wet bulb temperature of the chiller varies, the borehole wall temperature is lower, and the air conditioning system operation is more stable, when the chiller operates for 3h. Thus, the optimal control strategy should set the chiller running hours to 3h.

In summary, the optimal ST control strategy of the chiller is as follows: the WBTD of chiller opening is 2℃ and the running hours of chiller is 3h. Under the set values (2℃, 3 h), the composite system consumes a total amount of the energy of 1,894.15 MWh, and the borehole wall temperature is 19.65℃, at the end of the 7 years. Compared with the initial soil temperature of 19.05℃, the borehole wall temperature at the end of the period rises by 0.6℃, and remains basically stable throughout the period.

4.2.2 WBTD optimization

Figure 5 illustrates how the total energy consumption of the composite system varies with the set values of each control strategy. When the WBTD set value remains the same for chiller closing, as the WBTD of chiller opening increases, the GSHP system consumes more and more energy in operation, while the chiller system consumes fewer and fewer energy. Then, the composite system consumes less and less energy in operation. When the WBTD set value remains the same for chiller opening, as the WBTD of chiller closing increases, the GSHP system consumes more and more energy in operation, while the chiller system consumes fewer and fewer energy. Then, the composite system consumes less and less energy in operation. The main reason is as follows:

Whether the WBTD value for chiller opening or closing increases, the GSHP system will run for more hours, and the chiller system will run for less hours. The water temperature entering the GSHP heat pumps is below that of the chiller, and the GSHP system is more energy efficient than the chiller. Therefore, the extension of GSHP running hours and the reduction of chiller running hours will reduce the total energy consumption of the whole composite system.

Under various combinations of the ST set values, the composite system consumes a total energy up to 1,901.30MWh in 7 years, where the chiller runs for the longest possible time and the GSHP runs for the shortest possible time, corresponding to the smallest WBTD between chiller opening and closing. The minimum total energy consumption of the composite system in 7 years stands at 1,874.40mWh, where the chiller runs for the shortest possible time and the GSHP runs for the longest possible time, corresponding to the largest WBTD between chiller opening and closing.

Under different control strategies, the maximum and minimum energy consumptions of the composite system differ by is 26.90mWh, only 1.41% of the maximum energy consumption. Therefore, the set values of the control strategy have little impact on the total energy consumed by the composite system.

Figure 6 shows the borehole wall temperature of the composite system after 7 years. By increasing the WBTD set value of chiller opening or closing, the GSHP system will run for longer hours, while the chiller system will run for less hours. Then, the entire composite system will consume less energy. The borehole wall temperature of the buried tube heat exchanger will gradually increase.

The best control strategy should minimize the total energy consumed by the composite system, without changing the borehole wall temperature. Following this principle, the total energy consumption of the composite system and the borehole wall temperature were compared, with the WBTD below 2.5℃ for chiller closing. Under different WBTD set values, the total energy consumption of the composite system always intersects the borehole wall temperature, when the WBTD of the chiller is set at 3.5℃. When the WBTD of the chiller is below that level, the total energy consumption of the composite system would rise, while the borehole wall temperature would fall. When the WBTD of the chiller is above that level, the total energy consumption of the composite system would fall, while the borehole wall temperature would rise. Overall, the optimal control strategy is to set the WBTD of chiller closing to 3.5℃.

In addition, under different WBTD set values, the total energy consumption of the composite system always intersects the borehole wall temperature, when the WBTD of chiller closing is at 1.5℃. Since the total energy consumption does not change, when the WBTD of the chiller varies, the borehole wall temperature is lower, and the air conditioning system operation is more stable, if the WBTD of chiller closing stays at 1.5℃. Hence, the optimal control strategy should set the chiller closing wet bulb temperature to 1.5℃.

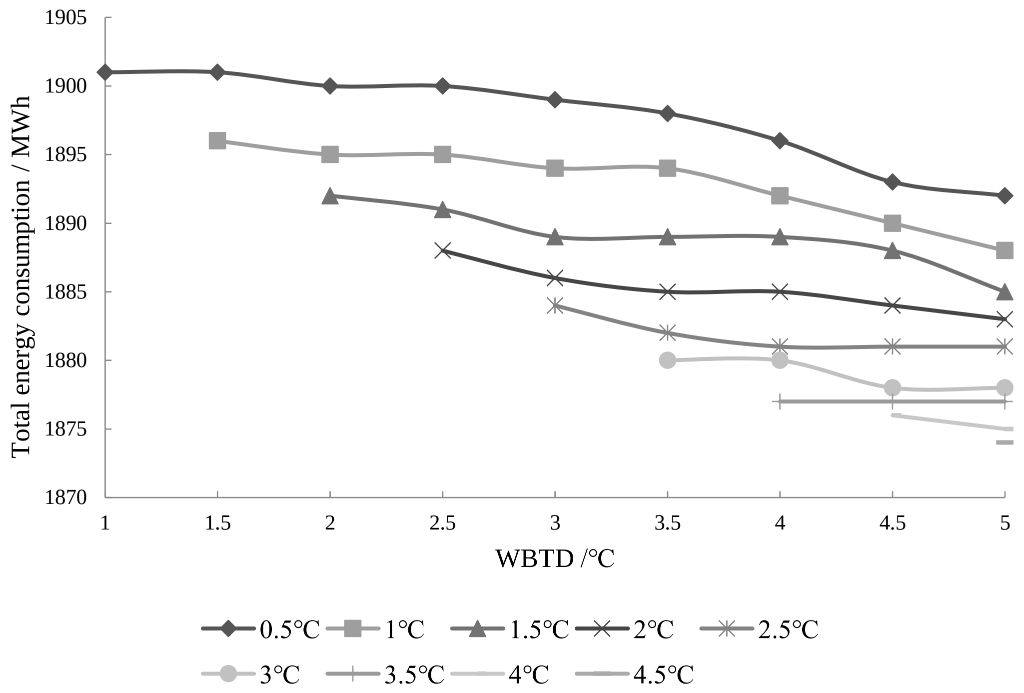

Figure 5. Total energy consumption of the system in 7 years

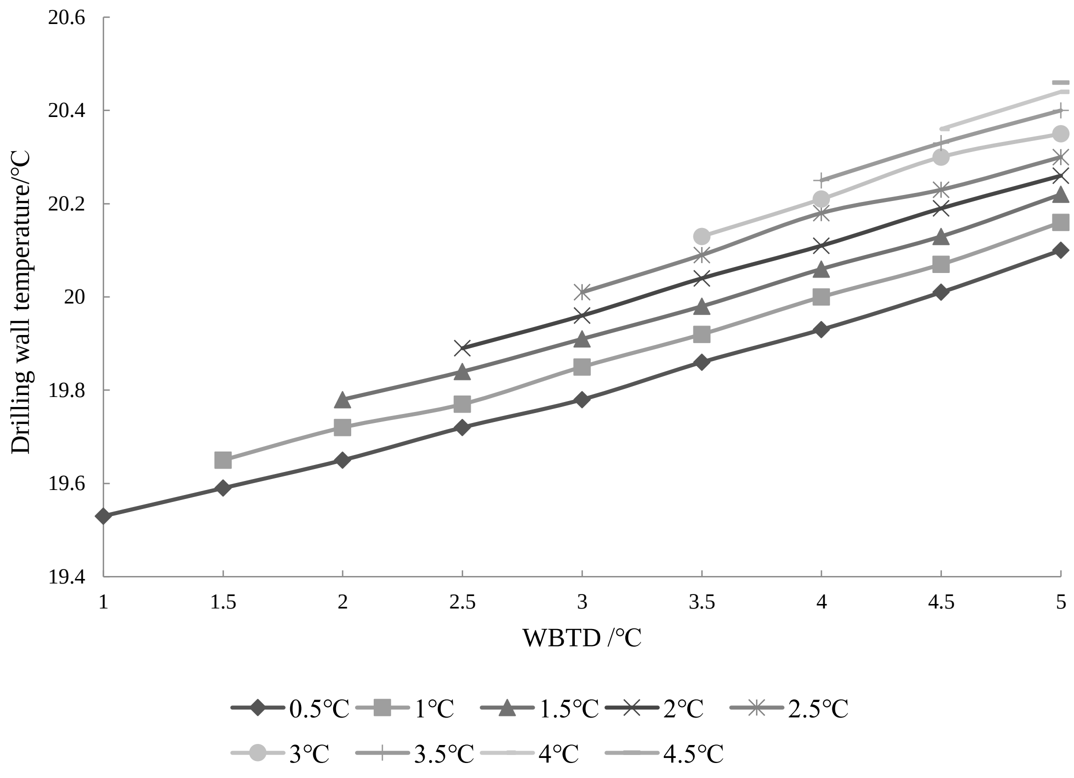

Figure 6. Borehole wall temperature at the end of system 7

The previous analyses have concluded that the set values of different control strategies exert little impact on the energy consumption of the system. Since the borehole wall temperature should not far surpass the initial soil temperature, the temperature rise is controlled to 1℃. Therefore, the set values corresponding to the borehole wall temperature above 20.05℃ are removed.

In summary, the optimal WBTD control strategy of the chiller is as follows: the WBTD of chiller opening is 3.5℃, and that of chiller closing is 1.5℃. Under the set values (3.5℃, 1.5h), the composite system consumes a total amount of the energy of 1,888.67 MWh, and the borehole wall temperature is 19.98℃, at the end of the 7 years. Compared with the initial soil temperature of 19.05℃, the borehole wall temperature at the end of the period rises by 0.93℃, and remains basically stable throughout the period.

4.2.3 Further comparison

The composite system was compared with the traditional GSHP scheme under controlled and uncontrolled conditions (Table 1).

For the composite GSHP system, the total energy consumption of the system has not much difference. It is only necessary to focus on borehole wall temperature. The borehole wall temperature of the ST is below that of the WBTD. When the chiller switches to the GSHP, the ST can be realized by regulating the running hours of the chiller, eliminating the need to monitor the underground soil temperature. During the switch, the GSHP system does not operate at the last moment, the underground borehole wall temperature must be monitored to replace the heat pump inlet temperature, and the underground soil temperature must also be monitored to increase the initial investment of the system. In this case, the ST is better than the WBTD.

Table 1. Operation comparison

Operation mode		Total energy consumption of the system in 7 years / MWh	Borehole wall temperature at the end of the 7 years is /℃	Soil temperature rise /℃
Composite system (uncontrolled)	Only the heat pump runs	1,877.48	21.03	1.98
	Only the chiller runs	1,995.81	18.31	-0.74
Composite system (control)	Optimal ST scheme	1,894.15	19.65	0.60
	Optimal WBTD scheme	1,888.67	19.98	0.93
Traditional GSHP system	Running through the summer season	1,858.07	22.15	3.10

When it comes to the total energy consumption of the composite system in the 7 years, the chiller operation is inefficient, dragging down the performance of the entire system. Hence, the total energy consumption of the composite system is larger than that of the traditional GSHP system, but the energy consumed by each part is similar. The traditional GSHP system requires 198 100m-deep vertical double U-shaped buried pipes, while the composite system only needs 84. The initial investment of the buried pipes falls by 58%. Besides, the composite GSHP system can maintain a constant underground soil temperature, balance underground soil heat exchange, and ensure the efficient operation of the GSHP system. Therefore, the composite GSHP saves the initial investment, and stabilizes the underground soil temperature, compared with the traditional GSHP system, without consuming too much more energy.

5. Conclusions

(1) When the ST strategy is adopted for the chiller, the WBTD of chiller opening is 2℃ and the running hours of chiller is 3h. Under the set values (2℃, 3 h), the composite system consumes a total amount of the energy of 1,894.15 MWh, and the borehole wall temperature is 19.65℃, at the end of the 7 years. Compared with the initial soil temperature of 19.05℃, the borehole wall temperature at the end of the period rises by 0.6℃, and remains basically stable throughout the period.

(2) When the WBTD strategy is adopted for the chiller switching, the optimal control strategy is to set the WBTD of chiller opening is 3.5℃, and that of chiller closing is 1.5℃. Under the set values (3.5℃, 1.5h), the composite system consumes a total amount of the energy of 1,888.67 MWh, and the borehole wall temperature is 19.98℃, at the end of the 7 years. Compared with the initial soil temperature of 19.05℃, the borehole wall temperature at the end of the period rises by 0.93℃, and remains basically stable throughout the period.

(3) Considering both borehole wall temperature and initial system investment, the ST strategy is better than the WBTD strategy. In the actual situation, the WBTD fluctuates sharply from time to time, adding to the difficulty of WBTD control. It is also difficult to measure the control accuracy. Therefore, it is recommended to adopt the ST to control the chiller system when switching to GSHP system.

Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

Conflicts of Interest

The authors declare no conflict of interest.

References

[1] B. Ashrae, "Commercial/institutional ground-source heat pumps engineering manual," In *Atlanta: American Society of Heating, Refrigerating and Air-Conditioning Engineers*, ASHRAEAtlanta, 1995.

[2] S. P. Kavanaugh and K. Rafferty, *Ground-Source Heat Pumps: Design of Geothermal Systems for Commercial and Institutional Buildings*, American Society of Heating, Refrigerating and Air-Conditioning Engineers, 1997.

[3] S. P. Kavanaugh, "A design method for hybrid ground-source heat pumps," *ASHRAE Tran.*, vol. 104, pp. 691-691, 1998.

[4] C. Yavuzturk and J. D. Spitler, "Comparative study of operating and control strategies for hybrid ground-source heat pump systems using a short time step simulation model," *ASHRAE Tran.*, vol. 106, pp. 192-192, 2000.

[5] M. H. Khan, A. Varanasi, J. D. Spitler, D. E. Fisher, and R. D. Delahoussaye, "Hybrid ground source heat pump system simulation using visual modeling tool for HVACSIM+," *Building Simulat.*, vol. 8, pp. 641-648, 2003.

[6] H. Esen, M. Inalli, and M. Esen, "Numerical and experimental analysis of a horizontal ground-coupled heat pump system," *Build. Environ.*, vol. 42, no. 3, pp. 1126-1134, 2007. https://doi.org/10.1016/j.buildenv.2005.11.027.

[7] J. D. Spitler, "Research planning for the HVAC&R industry," *HVAC&R Res.*, vol. 13, no. 5, pp. 681-682, 2007.

[8] S. J. Rees, J. D. Spitler, Z. Deng, C. D. Orio, and C. N. Johnson, "A Study of Geothermal Heat Pump and Standing Column Well Performance," *ASHRAE Tran.*, vol. 110, no. 1, pp. 3-13, 2004.

[9] P. Cui, H. Yang, J. D. Spitler, and Z. Fang, "Simulation of hybrid ground-coupled heat pump with domestic hot water heating systems using HVACSIM+," *Energ. Buildings*, vol. 40, no. 9, pp. 1731-1736, 2008. https://doi.org/10.1016/j.enbuild.2008.03.001.

[10] J. Jeon, S. Lee, D. Hong, and Y. Kim, "Performance evaluation and modeling of a hybrid cooling system combining a screw water chiller with a ground source heat pump in a building," *Energy*, vol. 35, no. 5, pp. 2006-2012, 2010.

[11] S. Hackel and A. Pertzborn, "Effective design and operation of hybrid ground-source heat pumps: three case studies," *Energ. Buildings*, vol. 43, no. 12, pp. 3497-3504, 2011. https://doi.org/10.1016/j.enbuild.2011.09.014.

[12] H. Park, J. S. Lee, W. Kim, and Y. Kim, "Performance optimization of a hybrid ground source heat pump with the parallel configuration of a ground heat exchanger and a supplemental heat rejecter in the cooling mode," *Int. J. Refrig.*, vol. 35, no. 6, pp. 1537-1546, 2012. https://doi.org/10.1016/j.ijrefrig.2012.05.002.

[13] Y. Man, H. Yang, and J. Wang, "Study on hybrid ground-coupled heat pump system for air-conditioning in hot-weather areas like Hong Kong," *Appl. Energ.*, vol. 87, no. 9, pp. 2826-2833, 2010. https://doi.org/10.1016/j.apenergy.2009.04.044.

[14] Y. Yuan, X. Cao, L. Sun, B. Lei, and N. Yu, "Ground source heat pump system: A review of simulation in China," *Renewable and Sustainable Energy Reviews*, vol. 16, no. 9, pp. 6814-6822, 2012. https://doi.org/10.1016/j.rser.2012.07.025.

[15] K. Mensah, Y. S. Jang, and J. M. Choi, "Assessment of design strategies in a ground source heat pump system," *Energ. Buildings*, vol. 138, pp. 301-308, 2017. https://doi.org/10.1016/j.enbuild.2016.12.055.

[16] Y. Shang, M. Dong, and S. Li, "Intermittent experimental study of a vertical ground source heat pump system," *Appl. Energ.*, vol. 136, pp. 628-635, 2014. https://doi.org/10.1016/j.apenergy.2014.09.072.

[17] L. Xu, L. Pu, S. Zhang, and Y. Li, "Hybrid ground source heat pump system for overcoming soil thermal imbalance: A review," *Sustain. Energy. Techn.*, vol. 44, Article ID: 101098, 2021. https://doi.org/10.1016/j.seta.2021.101098.

[18] A. Nguyen, "Determination of the ground source heat pump system capacity that ensures the longevity of a specified ground heat exchanger field," *Renew. Energ.*, vol. 169, pp. 799-808, 2021. https://doi.org/10.1016/j.renene.2021.01.035.

[19] C. Roselli, M. Sasso, and F. Tariello, “Dynamic simulation of a solar electric driven heat pump for an office building located in Southern Italy,” *Int. J. Heat Technol.*, vol. 34, pp. S496-S504, 2016. https://doi.org/10.18280/ijht.34S243.