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Operation Mode and Energy Consumption Analysis of a New Energy Tower and Ground Source-Coupled Heat Pump System

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Abstract: In order to solve the problems of performance degradation in energy tower heat pump (ETHP) systems under low temperature conditions and soil heat imbalances in ground source heat pump (GSHP) systems in cold regions, a new coupled system of ETHP and GSHP systems (the ET–GSHP system) and its operating mode were proposed. The mathematical model of the system was constructed along with the system's form and operation scheme. The COP (coefficient of performance) and total energy consumption of the coupled system were then simulated and studied under a number of common operating situations. The heating season is divided into four periods based on varying outdoor ambient temperatures: the first period operates in series mode and has an average outdoor temperature of 2.38 °C; the second period operates in parallel mode and has an average outdoor temperature of −8.56 °C; the third period uses soil source heat pumps to operate separately; and the fourth period operates in series mode and has an average outdoor temperature of −11.32 °C. Operation of the coupled system in four periods was simulated and analyzed, and the operational efficiency and energy saving of the system were analyzed using an actual commercial building in a cold region as an example. The results demonstrate that the ET–GSHP system's overall energy consumption during the heating period is reduced by 4.34% when compared to the traditional GSHP systems; the system's COP can maintain a high level throughout the heating period, with an average COP of 3.315; and the soil temperature at the conclusion of the heating period is 25 °C, which is 8.89 °C higher than that of the traditional GSHP system, providing a guarantee of summer heat return. The new ET–GSHP system significantly boosts the efficiency of the system's operation, achieves effective coupling between various heat sources through multi-stage control, and offers improved energy-saving advantages.

Keywords: energy tower; ground source heat pump; coupled system; energy consumption analysis

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1. Introduction

With the continuous development of society and the improvement of people's living standards, global energy consumption is growing rapidly; the construction industry is a major user of non-renewable energy and a major source of greenhouse gas emissions [1,2]. The energy consumption of building activities accounts for 36% of total global energy consumption and 40% of total global carbon dioxide emissions [3,4]. Due to the increasingly serious problems of environmental pollution and global climate change caused by energy consumption, the Chinese government has put forward a national development strategy to achieve carbon peaking by 2030 and carbon neutrality by 2060. Therefore, renewable and clean energy [5,6], especially geothermal energy and air energy [7], should be introduced to replace traditional energy, so as to reduce the use of traditional energy sources and carbon emissions, solve the contradiction between energy demand and policy restrictions, and explore energy-saving and environmental protection building energy supply technology using renewable energy [8].

Winter heating in northern China relies heavily on the burning of fossil fuels such as coal, which has led to serious air pollution and greenhouse gas emission problems [9]. Since the dual-carbon goal was proposed, clean energy-based heating has been vigorously promoted in northern China [10]. Heat pump technology is favored by people for its high efficiency, low energy consumption, and pollution-free characteristics [11]. Air source heat pumps [12], water source heat pumps [13], GSHPs [14], and other new clean energy-based heating systems have begun to gradually replace the traditional coal heating.

Heat pumps using a single heat source have their own advantages and disadvantages [11,15]. For example, air source heat pumps directly extract heat energy from the air, and thus their heat source is the easiest to obtain. However, when operating in cold regions, when the surface temperature of the outdoor evaporator is lower than the dew point temperature of the ambient air, frosting occurs on the surface of the evaporator, thereby increasing the thermal resistance of the evaporator and the flow resistance of the air. In order to maintain stable operation, some refrigerant steam is used for frequent defrosting, thereby reducing the heating capacity and resulting in reduced operating efficiency [16–19]. Water source heat pumps use water bodies such as ponds, lakes, rivers, seawater, groundwater, or sewage as their heat source. Their operating efficiency is high, and their performance is less affected by ambient temperature changes. The only deficiency is the need for large water bodies or storage tanks near the building [13,20,21]. GSHPs are heat pump systems that use underground soil or rock as their heat source. Because the soil temperature does not change much compared to the surrounding air throughout the year [22–24], their heating efficiency is usually high and their water supply temperature is more stable. However, in cold regions, as more and more heat pumps are used for building heating, the increased heat extraction from the soil can cause a thermal imbalance problem in the soil, leading to a drop in soil temperature that affects both the heat pump unit and the environment [25–28].

In order to solve the problem of frosting in air source heat pump systems in winter, an energy tower heat pump system is proposed to replace traditional air source heat pump systems [29]. In winter, the energy tower heat pump uses ambient air as a low-grade heat source for heating, which improves facility utilization and energy efficiency [30]. In addition, the energy tower heat pump system is easy to install and is not limited by terrain conditions [31]. It has become the focus of research in recent years due to its advantages of an unrestricted heat source, anti-frosting, and a wide application area. In winter, the heating efficiency of the energy tower heat pump is 7.4% higher than that of the conventional air source heat pump system [32]. However, under low temperature conditions, the performance of the energy tower heat pump still decreases significantly, which makes it difficult to be popularized and applied in the cold regions of the north.

One solution is to optimize the ratio of each component or solution regeneration control for the ETHP itself. Xie et al. [33] proposed a framework to optimize the design parameters of ETHPs. Case studies were conducted on 21 sites in different climate regions. The results show that the optimization method has a higher economy than the traditional design parameter matrix. Zhao et al. [34] proposed a multi-mode regenerative ETHP unit and investigated its working characteristics during cooling and heating in winter and summer, as well as the operating efficiency of the solution concentration control system in winter. The results show that the ETHP system has high energy efficiency in low humidity areas. Xiao et al. [35] proposed a heat source tower heat pump with a self-regenerator for cold regions. The system uses the subcooling heat of the refrigerant at the condenser outlet as a heat source. The system achieves renewable continuous operation while also reducing the efficiency of the system by 2.5–5.1%. Based on the artificial neural network model, Huang et al. [36] optimized a case with a genetic algorithm to improve the energy saving of an ETHP system. The optimization of the inherent parameters of the ETHP heat pump system itself is helpful for the improvement of the unit's efficiency in high-efficiency working conditions, which can only play the role of icing on the cake. It is not helpful to improve the performance of the heat pump system under extremely cold conditions.

Another solution is to adopt a multi-source-coupled heat pump system in the form of a coupled system. Shen et al. [37] proposed three coupling heating methods using closed energy towers and solar collectors and built related models. The simulation results show that the heating performance of the system is greatly affected by outdoor air temperature and solar irradiance, and different coupling methods need to be adjusted according to environmental conditions. Feng et al. [38] studied the heating performance of a solar-assisted closed energy tower system under winter conditions and proposed a method to change the antifreeze solution's heat absorption from the air and the hot water tank by changing the heat transfer temperature difference between the antifreeze solution and the air and the heat collecting medium, so as to realize the complementarity of air heat energy and solar heat energy. Cheng et al. [39] coupled a closed energy tower with a cold water phase change machine. By analyzing the variation law of outdoor air dry bulb temperatures and the heating performance coefficient of the two heat pump systems, the outdoor temperature demarcation point of the switching system operation mode under winter conditions was obtained, which effectively improved the heating capacity of the energy tower system during the heating season. The coupling of energy towers and solar energy can improve the problem of low temperature performance degradation of ETHPs to a certain extent, but it is not suitable for large-scale promotion and use due to the restriction of environmental conditions.

Due to the problem of soil heat balance, ground source heat pumps are considered to be coupled with other heat pumps to ensure the heat balance of the soil throughout the year and the high efficiency of the system. Qiu et al. [40] coupled a photovoltaic collector with a ground source heat pump and solved the problem of the photovoltaic collector easily overheating in summer and the problem of unbalanced soil temperature as a result of the ground source heat pump. Zheng [41] et al. used solar energy to assist the ground source heat pump system. By controlling the heat collection of the regenerator and the graded utilization of solar energy, the utilization rate of the system was improved and the problem of soil temperature decline was eliminated. Zanetti et al. [42] used Matlab software and the finite element analysis method to simulate the performance and operation strategy of air/ground source heat pumps. The results show that the seasonal performance factor of dual source heat pumps is greatly improved compared with single heat pumps.

Among the variety of heat pump systems, GSHPs are considered for coupling with energy towers due to their stable heating temperature and soil thermal properties. Current research on the coupled system of ground sources and energy towers mainly focuses on using the energy tower to store heat in the soil across seasons. Guo et al. [43] analyzed the factors influencing the performance of the energy tower when the energy tower recharges the soil in summer in northern China. The research results show that the performance evaluation of the energy tower cannot be judged by a single index, and multiple criteria need to be considered. The optimal operating conditions of the energy tower under experimental conditions are also given. Wang et al. [44] utilized an energy tower coupled with a buried tube heat exchanger to store heat for soil across seasons in summer, and the results showed that the coupling method is feasible in cold regions. Hu et al. [45] proposed a heat source tower–ground source heat pump (HST–GSHP) integrated system and simulated the modeling and operation of several cities in cold regions. The results show that, compared with the boiler split air conditioning system and the air source heat compensator combined with the GSHP system, the energy saving rates of the HST–GSHP system are 47.6% and 31.2%, respectively, which is more economical in most cases. It can be seen that the GSHP and the ETHP have high thermal coupling, and the coupled system can greatly improve the system performance and reduce the operating cost.

In view of the existing problems of energy tower heat pumps and ground source heat pumps, the existing literature is roughly divided into two aspects. On the one hand, starting from the single heat pump itself, the performance of the heat pump unit is improved by improving adaptability across the components of the heat pump unit and adopting different optimization algorithms. This method is limited by the heat source of the heat pump itself, and the range of optimization is still relatively small. The improvement of the system efficiency can only play the role of icing on the cake, and it is not helpful to solve the problem of system performance degradation under extremely cold conditions. On the other hand, from the perspective of the coupling system, the ground source heat pump is used to operate in winter, and the energy tower and solar energy are used for heat recovery in summer, so that the soil temperature is maintained at a relatively stable level throughout the year. Although this method can theoretically balance the annual temperature of the soil, previous studies have shown that it is feasible to use the energy tower to store heat for the soil by coupling the energy tower with the soil source heat pump. However, due to the different environmental parameters in different regions, it often takes more work to achieve the simulation effect, especially in the cold regions of the north.

In order to improve the utilization rate of energy towers in high efficiency periods and solve the problem of soil thermal imbalances caused by large heat extraction in winter, this paper proposes a new type of energy tower and soil source-coupled heat pump system for the first time. The system adopts three operating modes according to the change in the average temperature during the heating season. In the series operation mode, the advantages of high heating efficiency under the conventional working conditions of the energy tower heat pump are used to heat the soil, and at the same time, the heat demand on the load side is guaranteed and the waste heat from heating is fully utilized. In the parallel operation mode, the two heat pump systems are in the range of high efficiency, which ensures stable and efficient water supply on the load side. When the performance of the energy tower heat pump drops sharply under extremely cold conditions, the ground source heat pump is used to provide thermal energy for the building alone, which ensures stable heating under low temperature conditions and meets the heating needs of the building. The operation mode of the whole system realizes the efficient coupling of the two heat sources, ensures efficient operation during the whole heating season, and has high energy saving. Taking a shopping mall heating project in Shenyang as an example, the changes in COP, total energy consumption, and average soil temperature in the heating season of the system under several typical temperature conditions are simulated and analyzed. The changes in total energy consumption and average soil temperature are compared with a traditional ground source heat pump system, and the economy and energy saving of the system are further verified. Compared with the summer heat recovery scheme proposed in the previous literature, the scheme proposed in this paper has higher economic benefits, makes full use of the waste heat, reduces the power consumption of summer heat recovery, and has strong practicability and promotion value.

2. New Coupled ET–GSHP System

The system uses a closed energy tower coupled with a ground source heat pump to supply heat, and the simplified heat pump system is shown in Figure 1. According to the outdoor air temperature, it is divided into three operation modes:

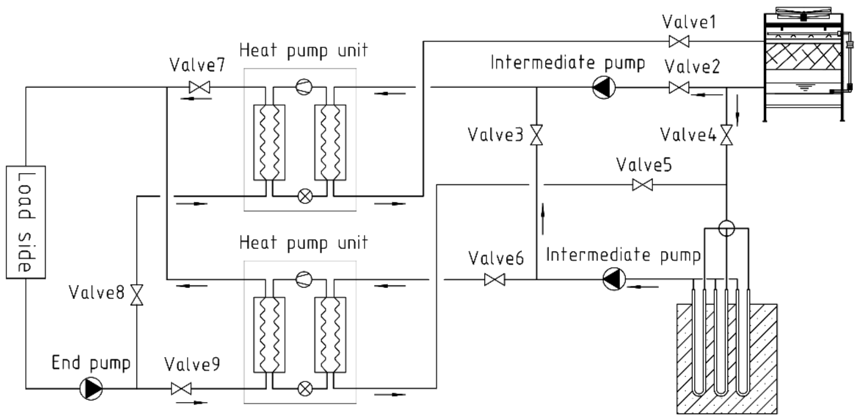

Figure 1. The total connection form of ET–GSHP system.

(1) ET–GSHP series operation mode: When the outdoor air temperature is high, the ETHP efficiency is high, and the independent operation can meet the heat demand of the end. Under the premise of ensuring the water supply temperature, the hot water flowing out of the energy tower is first sent to the soil and then sent to the heat pump unit for heating after being stored in the soil in a buried pipe. At this time, Valve 1, Valve 3, Valve 4, Valve 7, and Valve 8 open and Valve 2, Valve 5, Valve 6, and Valve 9 close. The antifreeze in the pipeline enters the energy tower to absorb the heat in the air and enters the buried pipe. After transferring part of the heat to the soil, it enters the heat pump unit and transfers the heat to the refrigerant in the evaporator of the unit. After the refrigerant absorbs heat in the evaporator, the heat is transferred to the end circulating water on the condenser side by the heat pump, and the heat is transferred to the heat user under the action of the end circulating pump to complete the cycle.
(2) ET–GSHP parallel operation mode: When the outdoor air temperature is low, the ETHP heating efficiency decreases and cannot meet the heat demand at the end alone. Therefore, the GSHP system needs to be enabled for auxiliary heating. At this time, Valve 1, Valve 2, Valve 5, Valve 6, Valve 7, Valve 8, and Valve 9 open and Valve 3 and Valve 4 close. For the energy tower side, the antifreeze in the pipeline directly enters the heat pump unit after absorbing the heat in the air in the energy tower and transmits the heat to the heat user; for the soil source side, the antifreeze in the pipeline enters the buried pipe heat exchange system, absorbs heat from the soil, enters the heat pump unit, and transmits the heat to the heat user.
(3) GSHP separate operation mode: When the outdoor air temperature is too low under extremely cold conditions, or the performance of the ETHP unit is significantly reduced due to snowfall and other reasons, the ETHP unit is closed, and only the GSHP unit is used for terminal heating. At this time, Valve 5, Valve 6, and Valve 9 open and Valve 1, Valve 2, Valve 3, Valve 4, Valve 7, and Valve 8 close. The antifreeze in the pipeline enters the buried pipe heat transfer system, absorbs heat from the soil, enters the heat pump unit, and transfers the heat to the heat user.

3. Methodology

3.1. Mathematical Modeling of Energy Towers

A closed energy tower is mainly composed of a heat exchanger tube and a spraying device. The antifreeze inside the heat exchanger tube exchanges heat indirectly with the air outside the tube, and when the surface temperature outside the tube is lower than 0 °C, the spraying antifreeze device sprays antifreeze on the outside surface of the tube to melt and prevent frost; at this time, the antifreeze inside the tube exchanges heat with the antifreeze sprayed outside the tube. This method is used to establish a mathematical model of the heat extraction process in the energy tower. The calculation formula of the specific heat capacity of the heat exchanger in the energy tower, c_{h}, is as follows [34]:

$$c_{\mathrm{h}} = \frac{c_{\mathrm{pa}}\rho_{\mathrm{a}}m_{\mathrm{a}}}{c_{\mathrm{pg}}\rho_{\mathrm{g}}m_{\mathrm{g}}} \tag{1}$$

where: c_{pa} is the constant pressure-specific heat capacity of air, kJ/(kg·°C); c_{pg} is the constant pressure-specific heat capacity of air, kJ/(kg·°C); ρ_{a} is the density of air, kg/m^3; ρ_{g} is the density of antifreeze, kg/m^3; m_{a} is the volume flow rate of air, m^3/s; and m_{g} is the volume flow of antifreeze, m^3/s.

The calculation formula for the number of heat transfer units of the heat exchanger, NTU_{h}, is as follows:

$$NTU_{\mathrm{h}} = \frac{\eta_{\mathrm{d}} \times NTU_{\mathrm{o}}}{1 + \frac{\eta_{\mathrm{d}} \times NTU_{\mathrm{o}}}{c_{\mathrm{h}} \times NTU_{\mathrm{i}}}} \tag{2}$$

$$NTU_{\mathrm{i}} = \frac{A_i}{\rho_{\mathrm{g}}m_{\mathrm{g}}\left(r_i + \frac{1}{h_i}\right)} \tag{3}$$

$$NTU_o = \frac{A_o h_o}{\rho_a m_a c_{pa}} \tag{4}$$

where: NTU_i is the number of heat transfer units inside the heat exchanger tube; NTU_o is the number of heat transfer units outside the heat exchanger tube; η_d is the fin efficiency of the heat exchanger tube; A_i is the total area of the inner surface of the tube, m^2; A_o is the total area of the outer surface of the tube, m^2; h_i is the heat transfer coefficient of the inner surface of the tube, W/(m^2·K); and h_o is the heat transfer coefficient of the outer surface of the tube, W/(m^2·K).

The heat transfer efficiency of the heat exchanger, ξ_g, is calculated by the formula recommended by Braun [46]:

$$\xi_g = \frac{1 - exp[-NTU_h(1 - c_h)]}{1 - c_h \times exp[-NTU_h(1 - c_h)]} \tag{5}$$

Air outlet temperature, T_{aout}:

$$T_{aout} = T_{ain} - \xi_g \times (T_{ain} - T_{gin}) \tag{6}$$

Antifreeze outlet temperature, T_{gout}:

$$T_{gout} = T_{gin} - c_g \times (T_{ain} - T_{gout}) \tag{7}$$

where T_{ain} is the air inlet temperature (°C) and T_{gin} is the antifreeze inlet temperature (°C).

Then, the antifreeze absorbs heat, Q_g:

$$Q_g = c_{pg} \rho_g m_g \left(T_{gout} - T_{gin}\right) \tag{8}$$

3.2. Mathematical Modeling of Buried Pipe Systems

GSHPs are heat pump systems that utilize underground soil as the heat source of the heat pump, the core of which is the buried pipe heat exchanger. In winter heating, heat is extracted from the soil through the buried pipe heat exchanger and supplied to the end user after the heat pump has elevated the heat. The outlet temperature of the buried pipe can be calculated by Equation (9). When the fluid velocity is close to zero, the outlet temperature of the buried pipe heat exchanger is close to the surrounding soil temperature [47].

$$T_{GHE,\,out} = \beta T_{GHE,\,in} + (1 - \beta) T_{soil} \tag{9}$$

Here, $T_{GHE,\,in}$ is the inlet temperature of the ground heat exchanger, °C; $T_{GHE,\,out}$ is the outlet temperature of the ground heat exchanger, °C; T_{soil} is the soil temperature; and β is the damping coefficient, which can be expressed by Equation (10):

$$\beta = \exp\left(-\frac{k_{soil} V_{soil}}{c_f m_f}\right) = \exp\left(-\frac{k_{GHE} L_{GHE}}{c_f m_f}\right) \tag{10}$$

where: k_{soil} is the thermal conductivity of the soil, W/(m^3·K); k_{GHE} is the thermal conductivity of the ground heat exchanger, W/(m·K); V_{soil} is the volume of soil, m^3; L_{GHE} is the length of the ground heat exchanger, m; c_f is the specific heat capacity of the fluid, J/(kg·°C); and m_f is the mass flow rate of the fluid, kg/s.

4. Numerical Simulation

4.1. Architectural Overview

In order to determine the optimal operating conditions and economy of the ET–GSHP system, this paper selects a four-story shopping mall building in Shenyang as the research object. The total construction area of the building is 7625 m^2, and the heating time is from 8:00 a.m. to 10:00 p.m., a total of 14 h per day. Thermal disturbance factors such as building envelope structure, fresh air demand, and internal personnel lighting are in accordance with the provisions of GB 50736-2012 [48] Code for Design of Heating, Ventilation and Air Conditioning of Civil Buildings. The specific design parameters are shown in Table 1. The building model and load simulation are carried out by DeST 3.0 simulation software. The changes in ambient temperature and building heat load throughout the year are shown in Figure 2. The coldest outdoor temperature is −24.5 °C, and the peak building heat load is 808.9 kW.

Table 1. Building heat load design parameters.

Design Parameters	
Heating method	Air conditioning and heating
Supply and return water temperature/°C	45/40
Indoor design temperature/°C	18
Relative humidity/%	40
Building envelope	50% energy efficiency design standard for public buildings
Per capita area (m^2/person)	8
Room occupancy rate	0.8

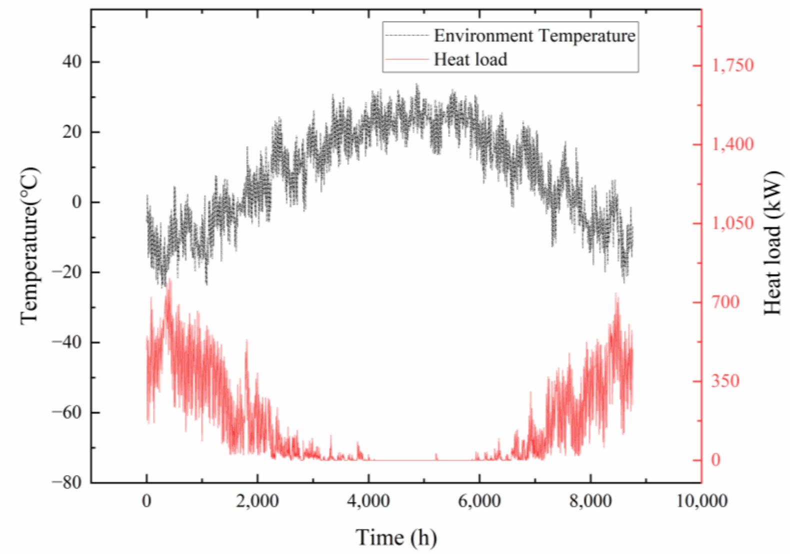

Figure 2. Urban ambient temperature and building heat load.

4.2. System Operation Mode and Simulation Model Construction

According to the actual equipment required by the coupled heat pump system, the relevant components are selected in the TRNSYS platform to build the system model. TRNSYS is a completely and extensively transient simulation tool [49,50] in HVAC areas. The following components were used to build the model on the TRNSYS 16.0 software simulation platform: meteorological parameters (Type15-2), pump (Type114), heat storage tank (Type4c), GSHP (Type227), diverter (Diverter), tee piece, equation, time controller (Typel4h), ETHP (Type225), integrator (Type24), graphic output unit (Type65c), and data output unit (Type25c). The selection of each piece of equipment needs to be determined according to the maximum heating load. For the ETHP system, its heating capacity only needs to meet the building heat load when the outdoor temperature is high in the early stage of heating. The ETHP unit selects a heat pump unit with a rated heating capacity of 490 kW from Halidom. With the decrease in outdoor environment temperature, the efficiency of the ETHP is reduced, and the GSHP is used to heat the building with its parallel transportation behavior. When the outdoor environment temperature drops to −15 °C, the heat transfer capacity of the energy tower drops sharply, which is extremely unfavorable to the application of the ETHP [51]. At this time, the ETHP unit is closed, and the GSHP unit is used for heating alone. The GSHP system, because it needs to bear the heating needs alone, should be selected according to the maximum heating load during the heating period, and a heat pump unit with a rated heating capacity of 890 kW from Climaveneta should be selected. For the ground heat exchanger, a parallel vertical single U-shaped heat exchanger is used. The buried pipe depth of a single well is 100 m, and 100 wells are designed. The design value of the heat exchange per unit well depth is 33 W/m.

The heating period in Shenyang begins on November 1 and lasts until March 31 of the following year. The change in outdoor environment temperature throughout the year is shown in Figure 2. Because the outdoor ambient temperature has a great influence on the performance of the ETHP, the whole heating period is divided into four stages according to the change in the outdoor ambient temperature. The first stage is the early stage of heating. From 1 November to 24 November, the average outdoor temperature is 2.38 °C, the heat efficiency of the ETHP is high, and the coupling system is operated in series mode. The second stage is the middle heating period and the end heating period. The middle heating period is from 24 November to 12 January, and the average outdoor temperature is −8.56 °C. The end heating period is from 17 February to 31 March, and the average outdoor temperature is −0.7 °C. In these two periods, the coupling system is operated in parallel mode. The third stage is from 12 January to 17 February, which is the date when the outdoor environment temperature is mostly below −15 °C. The average outdoor temperature in this stage is −13.32 °C, and the GSHP is used to heat the building alone.

Figure 3 shows the series operation mode of the coupled system. At this time, it is located in the early stage of heating, and the outdoor temperature is high. At night, only the ambient temperature during operation is considered, that is, after sunset to 10 pm. During this period, the temperature at night is 1~8 °C, and it can reach 10~17 °C during the daytime, meaning the heating effect of the ETHP system is better. The design water supply temperature is 45 °C, the design water outlet temperature of the ETHP unit is 50 °C, and the design temperature difference of the ground heat exchanger is 5 °C.

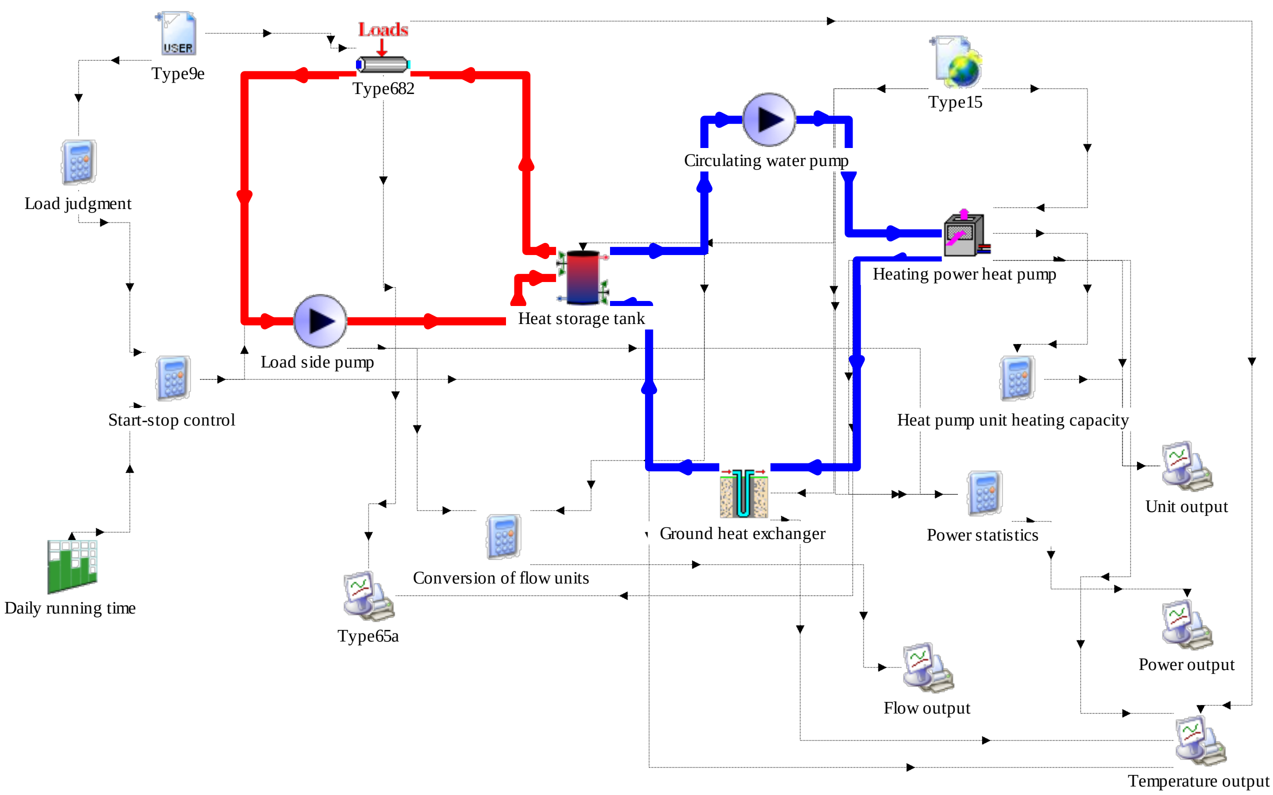

Figure 3. Series operation mode of the coupled system.

Figure 4 shows the parallel operation mode of the coupled system. At this time, it is located in the middle and late stages of heating. The outdoor temperature is relatively low, as low as −14.45 °C at night and up to 5 °C during the day. The heating efficiency of the ETHP decreases and the COP decreases and can no longer meet the heat demand on the load side alone. At this time, the heat storage in the soil is stopped, the GSHP unit is turned on, and the design water supply temperature is still maintained at 45 °C.

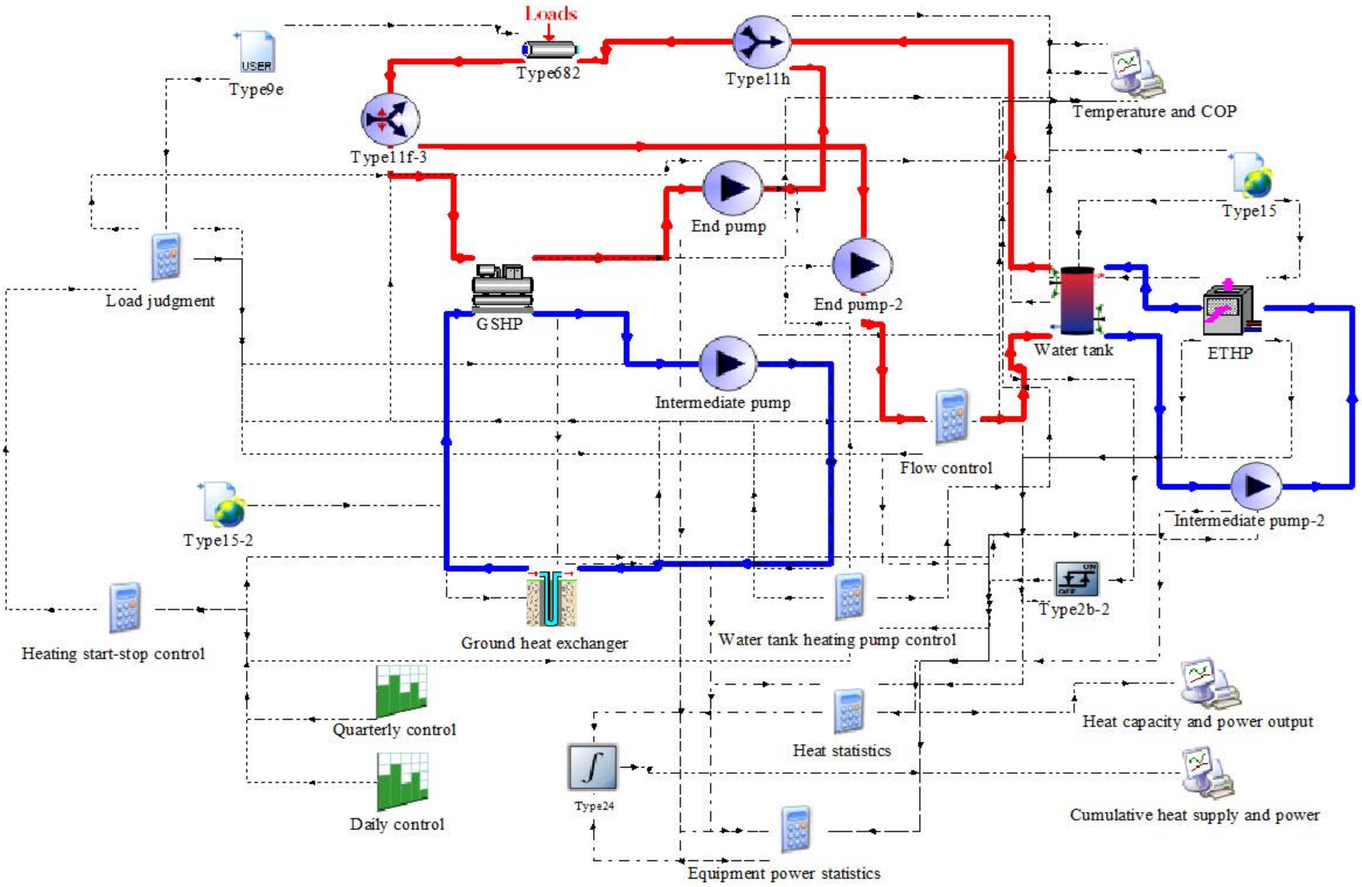

Figure 4. Parallel operation mode of the coupled system.

Figure 5 shows the separate operation mode of the GSHP. At this time, the outdoor temperature is mostly below −15 °C, and the minimum temperature reaches −24.5 °C. Under this condition, the heat transfer capacity of the energy tower drops sharply, and the heating efficiency of the unit decreases seriously. At this time, the ETHP unit is closed, and the GSHP with stable heating is used to heat the building alone.

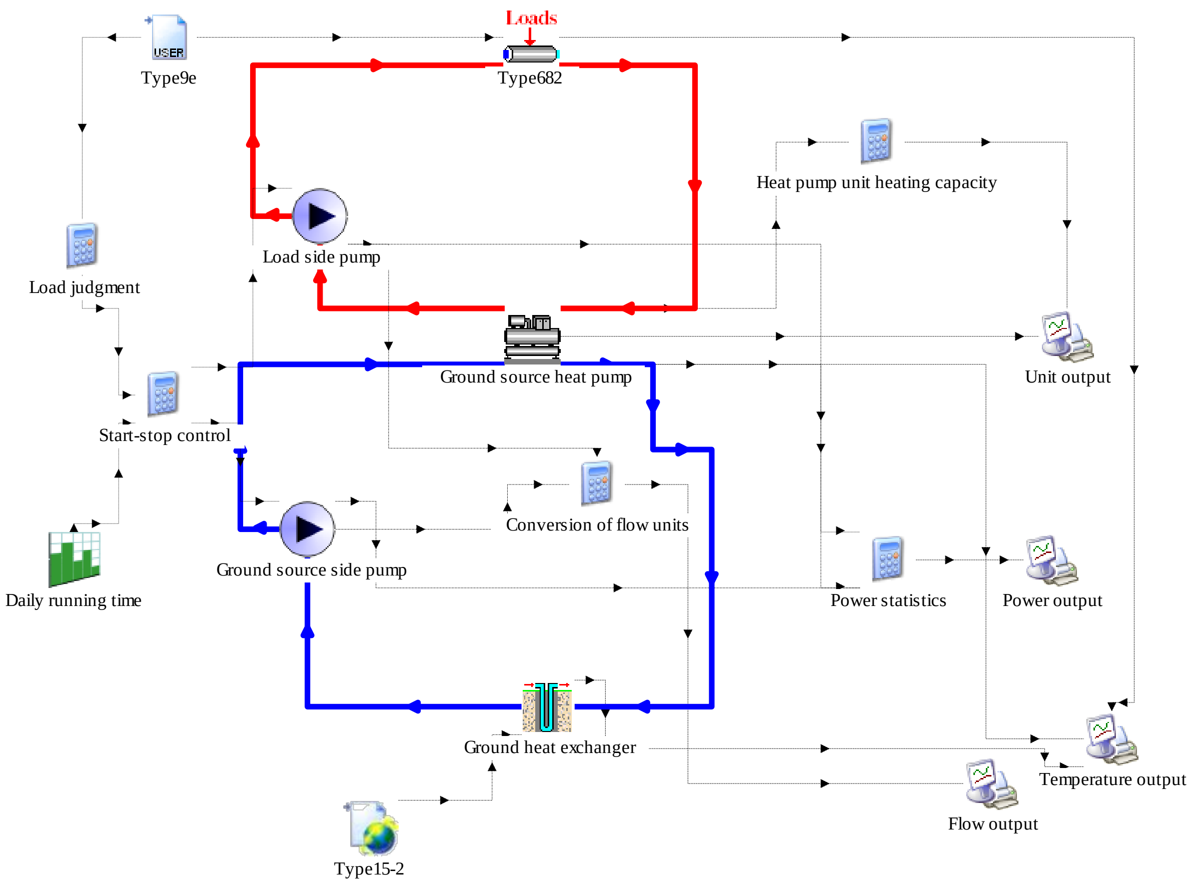

Figure 5. GSHP system individual heating mode.

5. Results and Discussion

The research content is mainly divided into three parts, namely, the time ratio of the operation mode of the coupled system during the heating period, the analysis of the system COP and energy consumption, and the comparison of the average soil temperature changes. The operation mode of the system is different due to the change in ambient temperature. Three different operation modes are adopted throughout the heating period, so it is necessary to allocate reasonably to maintain the high energy efficiency of the system. The goal of optimization is to maximize the use of each heat source under the premise of satisfying the set water supply temperature, so that the COP of the system can be maintained at a high level and the total energy consumption of the system can be reduced. The feasibility of the system operation mode is verified by simulation, and the economy and environmental impact are compared with the traditional GSHP system.

5.1. Analysis of System Cascade Operation and Soil Temperature Variations

Figure 6 shows the heating capacity, COP, and outdoor ambient temperature of the ETHP unit during the series operation of the system (from 1 November to 24 November). Due to the relatively high ambient temperature, the heating capacity of the ETHP is strong. It can be seen from the figure that the change in outdoor environment temperature has a great influence on the ETHP unit. During this period, the maximum outdoor temperature per day is 17.4 °C, the heating capacity of the unit can reach 488.09 kW, and the COP can reach 3.56. The minimum outdoor temperature per day is −12.75 °C, the heating capacity of the unit is only 385.97 kW, and the COP can only reach 2.73. It can be seen that the system has a large fluctuation. The COP difference during the single-day operation period can reach 0.45, and the average COP during the whole operation period is 3.03.

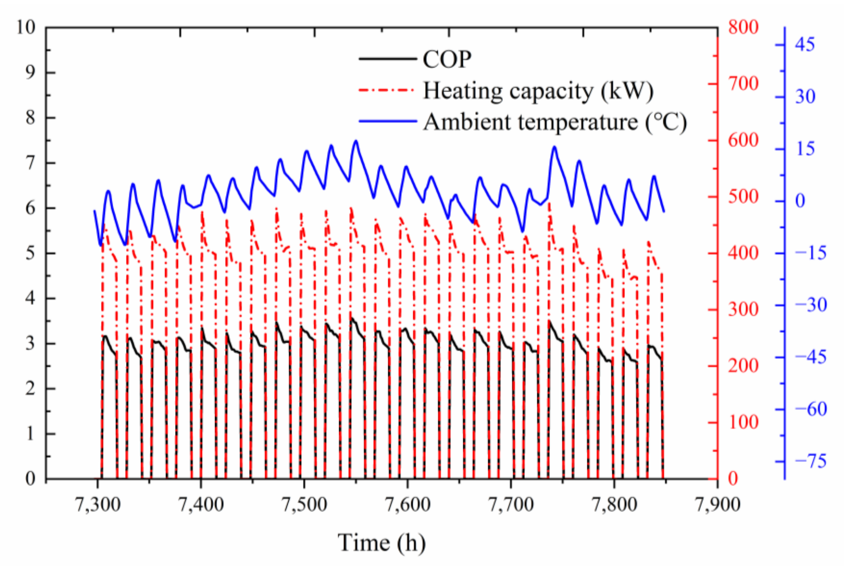

Figure 6. COP and heat capacity of ETHP system in series operation.

The series operation stage stores heat in the soil while meeting the heating demand. It is assumed that the average temperature of the soil during the operation of the system is 20 °C. The change in average soil temperature during the series operation is shown in Figure 7. The average soil temperature increased from 20 °C to 28.6 °C, an increase of 8.6 °C. Figure 8 shows the inlet and outlet temperature of the ground heat exchanger in series operation, and the average temperature difference between the inlet and outlet is 4.02 °C. It can be seen from the figure that the temperature difference between the inlet and outlet is proportional to the heating capacity of the ETHP unit. When the COP of the unit is high, the temperature difference between the inlet and outlet becomes higher. When the COP of the unit decreases, the temperature difference between the inlet and outlet also decreases. The outlet water temperature of the buried pipe continues to increase with the heat storage. The reason is that the average soil temperature is increasing, the temperature difference between the inlet and outlet is decreasing, and the heat storage rate in the soil is also decreasing.

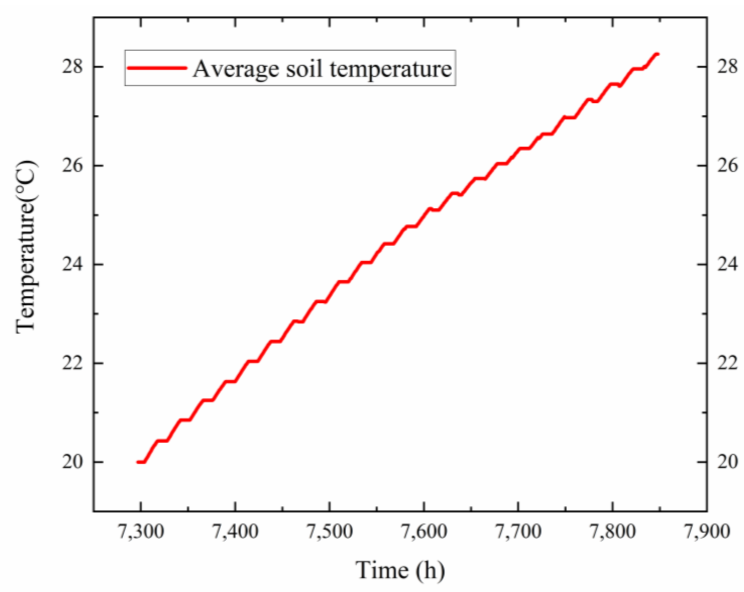

Figure 7. The change in average soil temperature in series operation.

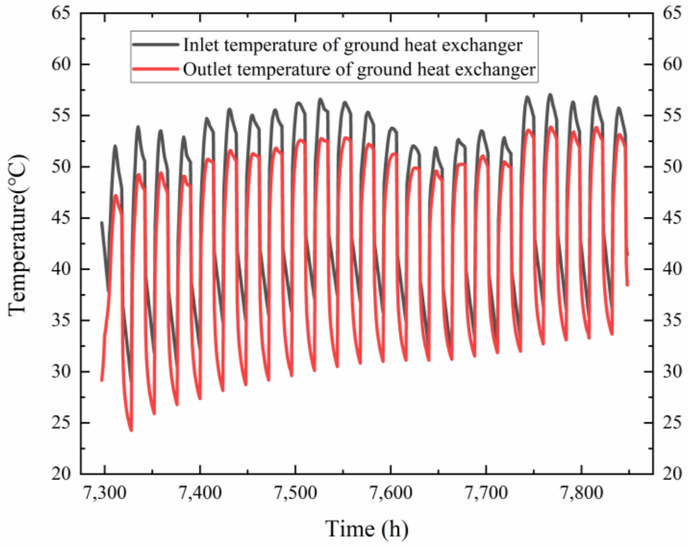

Figure 8. The inlet and outlet temperature of the ground heat exchanger in series operation.

5.2. Analysis of System Parallel Operation and Extreme Cold Condition Operation

The system parallel operation is divided into two periods: the first stage is from 24 November to 12 January and the second stage is from 17 February to 31 March. In the parallel operation of the system during heating, the ETHP system is the main system. When the outdoor ambient temperature is low, ETHP energy efficiency is reduced, the water supply temperature does not meet the requirements, and the soil source heat pump is turned on to regulate the heating peak in the water supply temperature to meet the requirements of the ground source heat pump unit until it is turned off in order to save energy. Figure 9 shows the change in COP of the ETHP unit and the soil source heat pump unit with the outdoor ambient temperature in the first stage of the parallel operation of the system. From the figure, it can be seen that the COP of the soil source heat pump unit is relatively stable and is always maintained around 4. The COP of the ETHP is about 23.3% lower than that of the series period, the average COP is only 2.32, and the COP of the ETHP unit is even lower than 1.75 when the ambient temperature drops to −13.25 °C. Considering all these factors, the average COP of the whole system is 3.19, which shows that it is necessary to turn on the GSHP to make up for the heat.

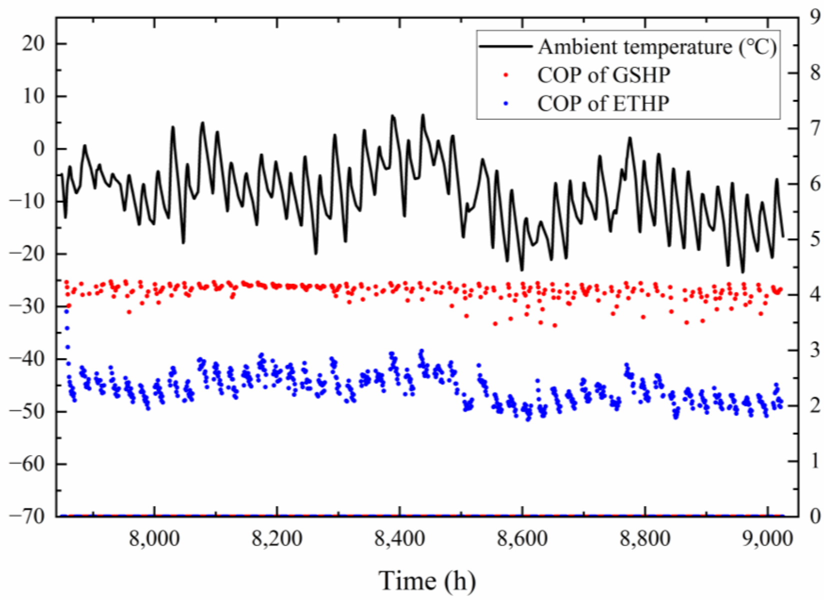

Figure 9. COP of the GSHP and ETHP in the first stage of parallel operation.

Figure 10 shows the comparison of the average soil temperature change between this operation mode and the traditional GSHP system. It can be seen from the diagram that under the same heat storage conditions, the average temperature of the soil during the heating of the traditional GSHP decreases rapidly, from 28.26 °C to 25.94 °C, a decrease of 2.32 °C; meanwhile, the average soil temperature decreased from 28.26 °C to 27.43 °C, only a decrease of 0.83 °C. It can be seen that the parallel operation strategy can slow down the decline rate of the average soil temperature so that more heat in the soil can be preserved, which is conducive to the subsequent separate heating of the GSHP in extremely cold weather. It can be seen from the diagram that the average soil temperature decreases faster when the traditional ground source heat pump is heating alone, while the average soil temperature decreases slower when the parallel operation is running. This scheme can save more heat in the soil, which is conducive to the separate heating of the ground source heat pump in the subsequent extremely cold weather.

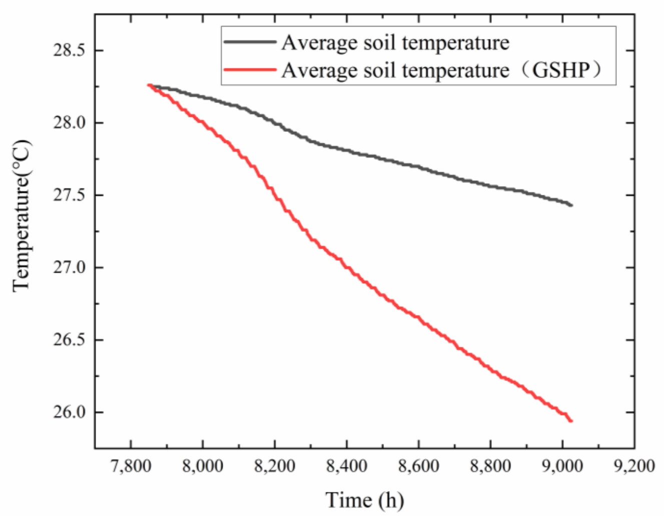

Figure 10. Comparison of soil average temperature changes.

In extremely cold weather, the system is switched to GSHP heating alone. The heating period is from 12 January to 17 February. Because of its stable heating capacity, its COP can still be stabilized at about 4, so its COP value and heating capacity are not discussed here. The change in average soil temperature during this operation is shown in Figure 11. The soil temperature decreased from 27.43 °C to 25.62 °C, a decrease of 1.81 °C.

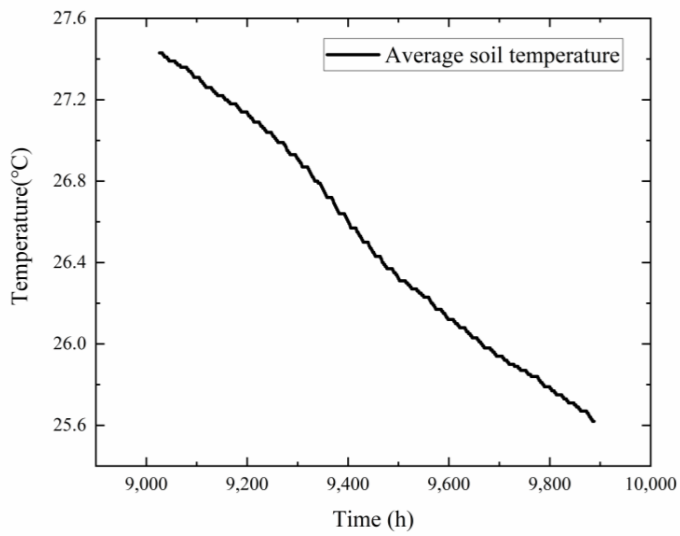

Figure 11. The average soil temperature of GSHP when heating alone.

Figure 12 shows that the COP of the ETHP unit and the GSHP unit in the second stage of the parallel operation of the system changes with the outdoor environment temperature. As the outdoor temperature gradually warms up, the heating performance of the ETHP unit also increases. The maximum COP can reach 3.01, and the average COP is 2.09. Considering the GSHP, the average COP of the whole system is 3.04. Figure 13 shows the average temperature of the soil in the second stage of parallel operation. The final temperature at the end of heating is 25.07 °C, which is conducive to summer heat recovery and ensures the soil heat balance throughout the year. Figure 14 shows the average temperature change in the soil during the whole heating period of the GSHP. The temperature decreases from 20 °C to 16.11 °C, a decrease of 19.45%, which is a difference of 8.96 °C from the final temperature of the coupling system. If there is no effective non-heating season heating system, the heat storage capacity of the perennial running soil will decrease, resulting in the heat imbalance of the soil.

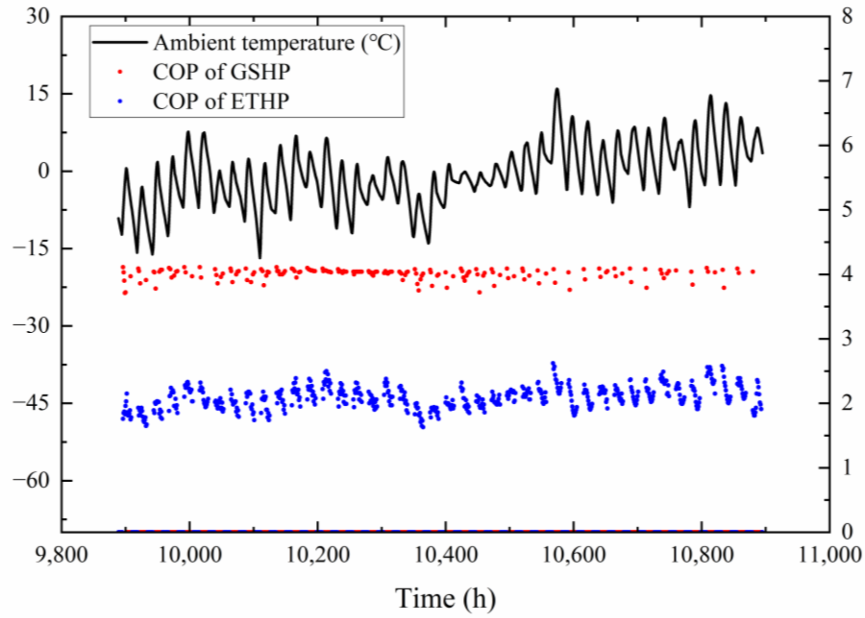

Figure 12. COP of the GSHP and ETHP in the second stage of parallel operation.

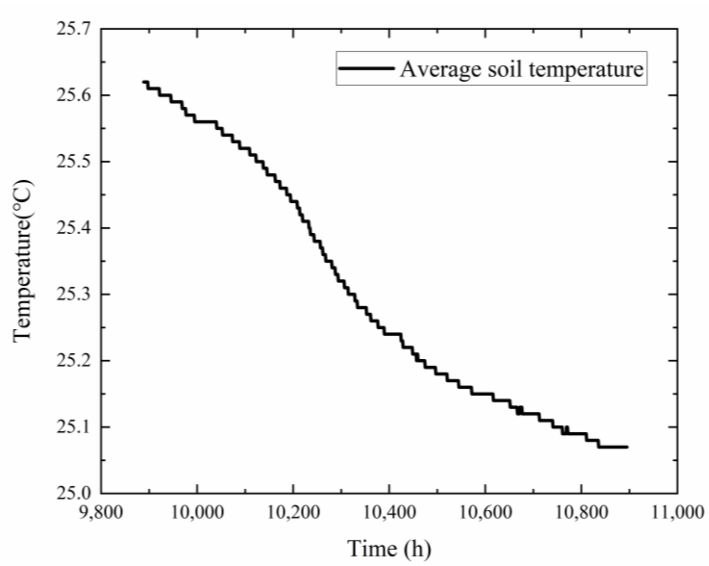

Figure 13. Average soil temperature in the second stage of parallel operation.

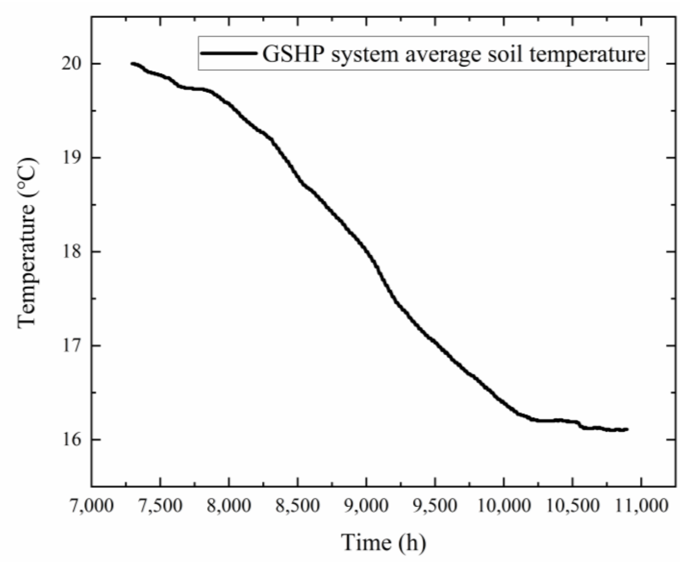

Figure 14. Soil temperature of traditional GSHP system heating.

5.3. Analysis of System Energy Consumption

Because the heating capacity of the ETHP system drops sharply after the outdoor ambient temperature reaches −15 °C, it cannot bear the building heating demand in extreme cold conditions. Therefore, the energy consumption of the system is compared with that of the GSHP heating system during the whole heating period. The comparison of energy consumption in different periods of the system is shown in Table 2. As can be seen from Table 2, the total system energy consumption during the operation of the first period of the ET–GSHP system is higher than the total energy consumption of the fourth period, but the operation time of the first period is only half of that of the fourth period. The reason for this is that the system needs to store heat in the soil in addition to meeting the load-side demand in the first period of the ETHP, so the outlet temperature of the heat pump unit is increased by 5 °C compared to the outlet temperature of the unit for the rest of the time period, and the energy consumption is increased as well. In addition, in the fourth period of parallel operation, the system uses the soil source heat pump as the main heat source while the energy tower only plays an auxiliary role. The soil source heat pump itself undergoes less energy consumption than the energy tower, and due to the first period of heat storage, the soil temperature at this time is higher, so the fourth period of energy consumption is relatively low. The combined effect of the above two reasons has resulted in the phenomenon of a short running time but a higher energy consumption of the system. The comparison is shown in Figure 15. In the early stage of the series operation of heating, the total energy consumption of the coupling system is higher than that of the GSHP system. The reason is that the ETHP unit increases the outlet water temperature in order to store heat in the soil. At this time, the energy tower unit has a better heating effect, which can meet the heat demand of the load side while storing heat. For the traditional GSHP system, the average temperature of the soil at this time is higher, the heating capacity of the GSHP unit is strong, and the power consumption of the compressor is lower, so the initial energy consumption is less. When the series operation is completed, the energy consumption of the coupling system is 7896.08 kW·h more than that of the GSHP system. When the parallel mode is turned on, the daily energy consumption of the system begins to decrease through the adoption of a reasonable operation mode until the total energy consumption of the system is 15,577.59 kW·h. At this point, the soil source separate heating stage begins, which shows the advantages of the parallel coupling operation mode. The total energy consumption of the traditional GSHP unit in the whole heating season is 292,977.34 kW·h, while the total energy consumption of the coupling system is 280,266.92 kW·h, meaning 12,710.42 kW·h is saved.

Table 2. Comparison of energy consumption in different periods of the system.

Operation Period	Energy Consumption (kW·h)	
	GSHP	ET–GSHP
11.1–11.24	58,553.16	66,449.24
11.24–1.12	99,939.84	76,466.16
1.12–2.17	87,919.3	78,131.8
2.17–3.31	46,565	59,219.7
The whole operation period	292,977.34	280,266.92

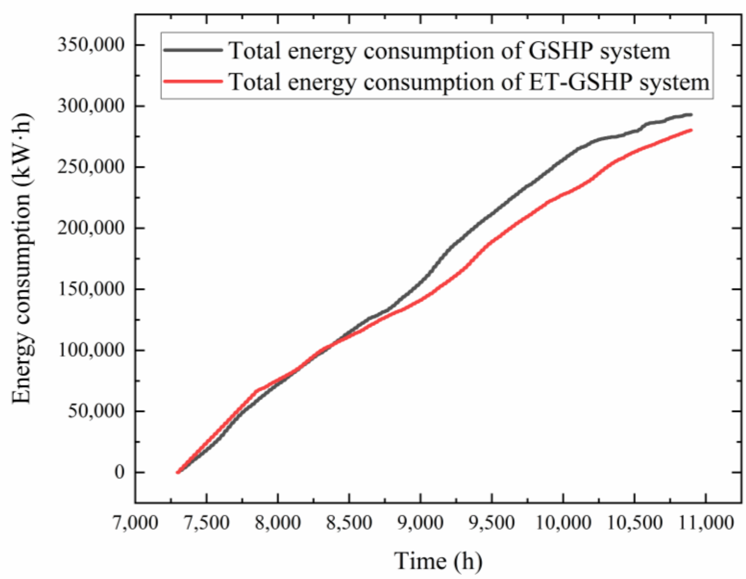

Figure 15. Comparison of total system energy consumption.

6. Conclusions

In order to solve the problems of performance degradation of ETHP systems under low temperature conditions and soil heat imbalances of GSHP systems in cold regions, a new coupled system of ETHP and GSHP systems (ET–GSHP system) and its operating mode were proposed. The system divides the heating season into four working conditions and adopts three modes of operation to achieve the efficient coupling of heat sources. With the help of TRNSYS software, the author took a shopping mall in a cold area as an example to simulate the performance and energy consumption of the system during the whole heating period, and reached the following conclusions:

(1) The heating season is divided into four operation stages according to the change in outdoor ambient temperature: the first stage is from 1 November to 24 November and uses the series heat storage operation mode; the second stage is from 24 November to 12 January and uses parallel operation and the soil source heat pump to make up for the heat; the third stage is from 12 January to 17 February, during which time the energy tower unit is shut down and the soil source heat pump takes on the heat supply task alone; the fourth stage is from 17 February to 31 March and uses the parallel operation mode. The simulation results showed that the distribution method has high economy and is appropriate for year-round operation. The series period accounts for approximately 15.36% of the entire heating period, the parallel operation period accounts for approximately 60% of the entire heating period, and the ground source alone operation period accounts for approximately 24.64% of the entire heating period.

(2) During the series operation, the soil's average temperature rose from 20 degrees Celsius to 28 degrees Celsius, considerably enhancing the soil's ability to store heat. The system's COP was kept constant between 2.73 and 3.56, with an average COP of

3.03. Due to the drop in outdoor temperature, the COP of the ETHP unit dramatically reduced during the parallel operating time, dropping by roughly 23.3% compared to the series operation period, with an average COP of 3.19 during the first period of the parallel operation. The average soil temperature decreased only 0.83 °C, and the soil still maintained a strong heat transfer capacity. In the second stage, the energy tower heat pump unit's performance started to improve. Its highest COP was 3.01, its average COP was 2.09, and the system's average COP was 3.04. The average soil temperature after heating is 25.07 °C, 8.96 °C higher than it would be with a conventional soil source heat pump. This is helpful for preserving the soil heat balance during continuous operation.

(3) The average COP of the system for the whole heating period is 3.315, which shows that it can maintain a high heating capacity throughout. The total energy consumption of the traditional soil source heat pump system for the whole heating season is 292,977.34 kW·h, and the total energy consumption of the coupled system for the whole heating season is 280,266.92 kW·h, which is 4.34% less than the total energy consumption of the traditional soil source heat pump for heat supply, indicating high economy and environmental protection.

In conclusion, the proposed new energy tower-coupled ground source heat pump system operation deployment method can achieve efficient energy utilization, reducing the total energy consumption of the system, resolving the soil's thermal imbalance, and increasing the power of the heat pump unit, which is especially suitable for the cold regions in the north of China.

However, because just one region was simulated in this work, there is a lack of generalizability in the findings. Additionally, no simulated verification is provided; only the results are presented. Future research should take into account buildings in various cold climates to offer the best generalized control strategies and system design methodologies based on building type, climate, and load demand. To validate the simulation results, a testbed should also be constructed.

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

Nomenclature

Symbols	
c_h	specific heat capacity kJ/(kg·°C)
c_{pa}	constant pressure-specific heat capacity of air kJ/(kg·°C)
c_{pg}	constant pressure-specific heat capacity of air, kJ/(kg·°C)
m_a	volume flow rate of air, m^3/s
m_g	volume flow rate of antifreeze, m^3/s
A_i	total area of the inner surface of the tube, m^2
A_o	total area of the outer surface of the tube, m^2
h_i	heat transfer coefficient of the inner surface of the tube, $W/(m^2 \cdot K)$
h_o	heat transfer coefficient of the outer surface of the tube, $W/(m^2 \cdot K)$
T_{aout}	air outlet temperature, °C
T_{ain}	air inlet temperature, °C
T_{gout}	antifreeze outlet temperature, °C
T_{gin}	antifreeze inlet temperature, °C
Q_g	antifreeze absorbs heat, W

$T_{\text{GHE, out}}$	outlet temperature of GHE, °C
$T_{\text{GHE, in}}$	inlet temperature of GHE, °C
T_{soil}	soil temperature, °C
k_{soil}	thermal conductivity of soil, W/(m^3·K)
k_{GHE}	thermal conductivity of GHE, W/(m·K)
v_{soil}	volume of soil, m^3
L_{GHE}	length of GHE, m
c_{f}	specific heat capacity of fluid, J/(kg·°C)
m_{f}	mass flow rate of the fluid, kg/s
Abbreviations	
NTU	number of transfer units
COP	coefficient of performance
ETHP	energy tower heat pump
GHE	ground heat exchanger
ET–GSHP	energy tower-coupled ground source heat pump
Greek symbols	
ρ_{a}	density of air, kg/m^3
ρ_{g}	density of antifreeze, kg/m^3
η	efficiency
ξ	heat transfer efficiency
β	damping coefficient

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