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# Seasonal performance research of heat-source tower systems using different work materials

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## Abstract

The heat-source tower system has been evaluated in an enthalpy difference laboratory. This study analyzes the seasonal performance under various operating situations and fluid quality. In the summer and fall seasons, water is used as the circulating fluid in the tower. In the spring and winter seasons, glycol solution is used as the circulating fluid. Some parameters of the heat-source tower (e.g. the temperature of the solution inlet and outlet, the flow rate of the solution inlet and outlet, the temperature of the air inlet and outlet, the inlet air volume and the moisture content of the inlet air) are considered and measured to obtain its heat transfer characteristics. The simulation model of heat-source tower is constructed based on the mechanism of heat and mass transfer. This model is validated by the experimental results. The heat exchange and latent heat exchange of the system are analyzed under different parameters, as well as the variation law of inlet and outlet temperature and moisture content differences. The results show that the system has solution moisture absorption during winter operation. However, for every 9000 $m^3$/h increase in air volume, the rate of solution dilution decreases between 9 and 43%. The rate of solution dilution is reduced by 11–31% for every 1°C in addition to the inlet solution temperature. Meanwhile, the heat dissipation in summer is about 2.8 times of the heat dissipation in winter.

*Keywords:* heat-source tower; heat exchange; simulation model; experiment; heat and mass transfer

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## 1. Introduction

The increase in urban population and living area has also led to the higher increasing trend in building energy consumption. World Energy Outlook released by the World Energy Organization [1] pointed out that the total energy consumption in China in 2030 would reach 5.81 billion tce. Among them, building energy consumption will account for 46.5% of the total energy consumption, of which 21.7% is for building operation energy consumption. Meanwhile, the rate of growth in operational energy consumption continues to increase. Therefore, it is necessary to reduce building operation energy consumption, which is the paramount for the global energy crisis and decarbonization background [2].

At present, buildings commonly use central air conditioning equipment for heating and cooling, e.g. water-cooled chillers and boilers heating in summer [3], air-source heat pump system [4], water-source heat pump system [5] and ground-source heat pump system [6]. Among them, the chiller and boiler solutions have the characteristics of a simple structure, low initial investment and high operational efficiency in summer. However, the chiller is idle in winter and cannot generate heat, resulting in a certain waste of space and energy. Meanwhile, boilers as heat sources have low primary energy utilization and produce a large number of pollutants, which cannot match low-carbon standards [7]. Air-source heat pump system has the characteristics of energy saving and environmental protection [8], and its operating efficiency in winter is better than that of the boiler. However, during winter

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operation, the outdoor units are often troubled by frost problems when they perform heat exchange, resulting in degradation of the system heat exchange and performance [9]. Ground-source heat pump systems and water-source heat pump systems have high cooling and heating efficiency during year-round operation [10, 11], but are more demanding geographically and can have some impact on the building when they are built [12]. Therefore, the different cooling and heat-source options discussed above have their own shortcomings and scope of application.

The existing solutions have great potentials to be improved. Therefore, a new type of heat pump system is currently being developed. The heat-source tower heat pump technology is derived from the theory of heat absorption by cooling tower inversion. Wen *et al.* [13, 14] established the forked-flow heat-source tower device and mathematical model. Meanwhile, the effects of outdoor air parameters and water–air ratio on the heat transfer efficiency in the heat-source tower were investigated. Cheng *et al.* [15] and Hu *et al.* [16] investigated the effects of key parameters on the performance of the system. He *et al.* [17] established a mathematical model of the confined heat-source tower by numerical simulation of the tower under spray conditions and found that the risk of frosting in the heat pump system of the closed heat-source tower was low, which can reduce the energy consumption of the heat pump system. Xu *et al.* [18] combined the effects of heat transfer and antifreezing to provide a reference for design. Huang *et al.* [19] and Lu *et al.* [20] investigated the law of heat and mass exchange in the heat-source tower under different working conditions and optimized the solution parameters and structure of the heat-source tower. Both Jia *et al.* [21] and Li *et al.* [22] analyzed the heat exchange characteristics and energy efficiency ratio of the heat-source tower heat pump system and concluded that the increase in air volume significantly increased the stability and heat exchange efficiency of the system. Meanwhile, for the characteristics of winter solution moisture absorption in heat-source tower system, Xia *et al.* [23] designed a new heat-source tower structure with precondensation function. Experiments showed that the new tower has stronger heat transfer performance and lower solution hygroscopicity. The new heat-source tower solution regeneration system constructed by Wen *et al.* [24] has low-energy consumption and better energy-saving effect. Han *et al.* [25] and Song *et al.* [26] verified the solution regeneration performance, feasibility and energy saving of the heat-source tower. Meanwhile, Liang *et al.* [27] found that the solution regeneration rate could be improved by increasing the auxiliary heat. Xu *et al.* [28] proposed a novel cascade-coupled heat pump system to improve the utilization of waste heat. Meanwhile, a new method of combining extreme learning machine algorithm (OGLCM-ELM) is proposed for the winter frost problem of air-source heat pump system. This method provided significant energy saving potential for defrost control [29].

The introduction and analysis of the heat-source tower heat pump system show that the current research is mainly focused on the heat and mass transfer mechanisms inside the heat-source tower, the optimization of the internal structure and the overall system performance [30, 31]. In the research of antifreeze solution, the main focus is on experimental testing [32] and the study of regeneration mechanisms [33]. However, the effect of different factors on the heat-source tower heat pump system is less studied in practical applications.

In this article, the working conditions of the heat-source tower heat pump system are studied under different inlet and outlet solution temperatures, solution flow rates, air temperatures, inlet air volume and moisture content. The heat exchange and latent heat exchange of the system, as well as the variation characteristics of the inlet and outlet temperature differences and moisture content differences, are analyzed. Also, the operating efficiency and conditions of the unit are analyzed under different conditions. Then, the heat-source tower is studied in the winter heat absorption condition and the summer heat dissipation condition. Moisture absorption in the system under winter conditions was also studied. At the same time, the feasibility and authenticity of the tested data are determined experimentally. This study provides a theoretical basis and data support for heat-source tower design.

## 2. Experimental settings

### 2.1. Heat-source tower heat pump system

This experiment is conducted to determine the heat transfer performance by building a heat-source tower heat pump experimental bench. The system diagram of the experimental bench is shown in Figure 1. A physical diagram of the system is shown in Figure 2.

The solution operating inside the unit is different in winter and summer seasons. In winter conditions, the operating work mass inside the unit of the heat-source tower system is glycol solution. In summer conditions, the operating mass inside the unit is water. The cooling tower receives the necessary fluid at the appropriate temperature through a connection between the fluid loop on one side of the plate heat exchanger and the cooling tower. The other side supplies the necessary heat or cooling for the system by connecting to the external hot and cold-water unit. The solution required for the operation of the heat-source tower is injected through the solution tank, and the solution concentration is controlled by density. From Figure 3, the design ambient wet- and dry-bulb temperatures are achieved and maintained through the company's enthalpy laboratory. Both the fan and the water pump are frequency-regulated. The frequency of the pump and fan are adjusted to produce the desired air volume and solution flow rate. The bypass valve is used to regulate the solution flow rate into the heat-source tower when it needs to be reduced in order to meet the pump frequency's lower limit.

### 2.2. Standard working condition test

For this experiment, a test bench for a heat-source tower heat pump system was constructed. To guarantee the experiment's correctness, the heat-source tower system's functioning is monitored throughout. The heat-source tower was placed in the enthalpy difference laboratory for the standard working condition

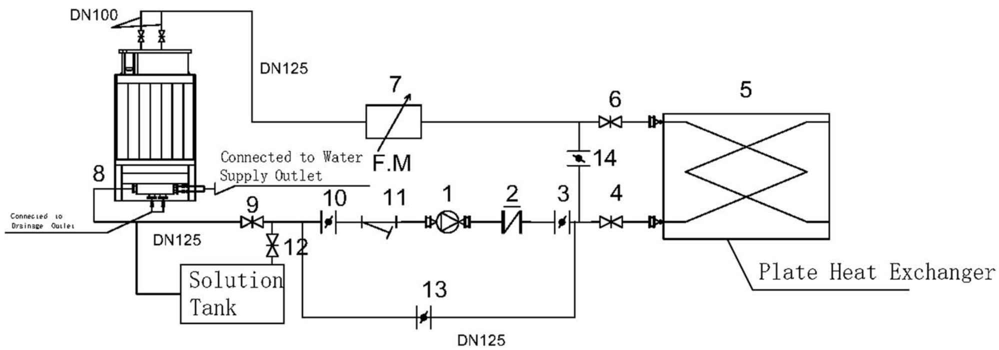



**Figure 1.** *Heat source tower heat pump system diagram. (1. Inverter pump; 2. Check valve; 3. Butterfly valve; 4. Electronic valve; 5. Plate heat exchanger; 6. Electronic valve; 7. Test instrument; 8. Heat source tower; 9. Electronic valve; 10. Butterfly valve; 11. Purifier; 12. Electronic valve; 13. Butterfly valve; 14. Butterfly valve.)*

**Figure 2.** *A physical diagram of the system.*

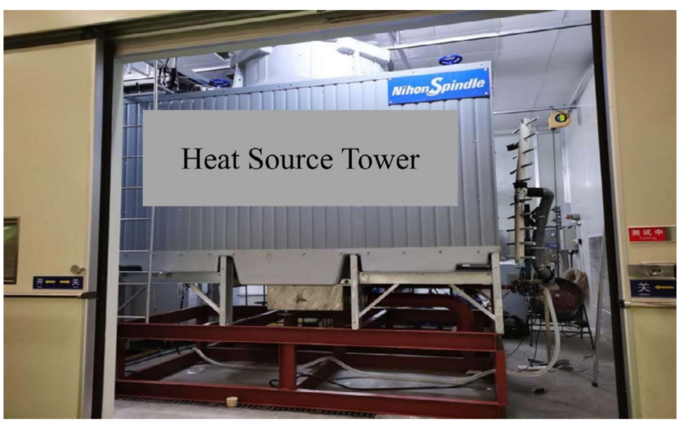



**Figure 3.** *Systems in the enthalpy difference laboratory. (a) Heat source tower solution. (b) Heat source tower internal elements.*

heat-source tower experiment. In the standard condition, the dry- and wet-bulb temperatures of the air inlet and outlet of the heat-source tower are 4.5°C/3.5°C. The glycol solution temperature of heat-source tower inlet and outlet is 3°C/0°C. The flow rate was adjusted at the rated air volume to meet the experimental requirements of the system.

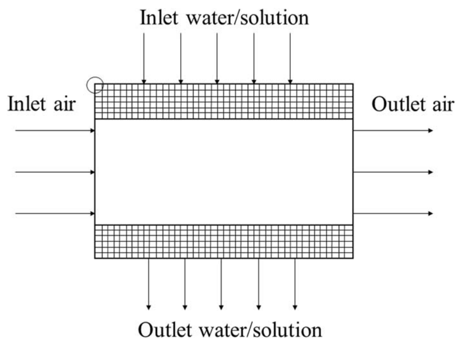
Initially, the pump circulation was activated and the glycol solution was injected via the solution tank. The glycol solution density was measured by densitometer to make its freezing point below 9°C to ensure the normal and stable operation of the whole experimental system. When the concentration of the glycol meets the requirements, the working side of the plate exchange is turned on. The inlet and outlet temperature of the plate exchange was 9°C/6°C to achieve the heat-source tower side to meet the 3°C inlet water temperature requirement.

The heat-source tower pumps and the fans in the tower were turned on. At the same time, the fan frequency was adjusted so that the set airflow was achieved. The water pump frequency was adjusted to achieve the set flow rate of the pump. Since the pump frequency adjustment range was 30 Hz–50 Hz, when the required flow rate still cannot be met at the minimum frequency, the bypass valve connected in parallel with the pump will be open. In order to attain the necessary temperature difference between the inlet and outlet heat-source tower solution, this further lowers the flow rate output to the heat-source tower. Due to the low temperature of the inlet and outlet of the plate exchange, the bypass valve in parallel with the plate exchange can be adjusted for accurate control of the glycol temperature entering the heat-source tower.

## 2.3. Variable working condition experiments

The heat-source tower system's performance in various temperature and humidity conditions can be assessed using a variable duty test. To control the dry- and wet-bulb temperatures around the heat-source tower, the system was placed in an enthalpy difference laboratory. The heating and humidifying equipment in the enthalpy difference laboratory would be PID-controlled to precisely achieve the required temperature and humidity as well as the ambient air speed for the heat-source tower system experiment.

The experiment would test the performance of the heat-source tower system by varying the flow rate, air volume, inlet and outlet temperature and concentration. The variable flow rate experiment can be achieved by adjusting the pump frequency and the opening of the bypass valve connected in parallel with the pump. The



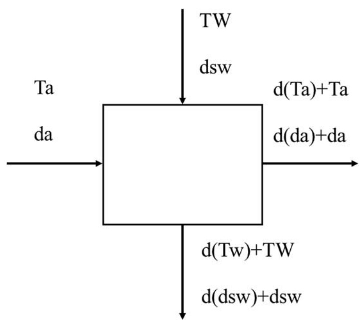




**Figure 4.** *Cross-flow packed heat source tower model.*

variable air volume experiment can be achieved by adjusting the fan frequency to adjust the air volume. The variable inlet solution temperature experiment can be realized by changing the inlet and outlet temperatures on the other side of the plate exchange and the bypass valve connected in parallel with the plate exchange. Variable solution concentration experiment can be achieved by injecting the solution into the solution tank and replenishing water in the system to control and adjust the glycol solution concentration.

# 3. System modeling

## 3.1. Model settings

It was discovered during the experiment that the air in the heat-source tower engages in a thermal mass exchange activity with the water or solution on the packing. The temperature decreased, and the air at the packing was continuously renewed by the flow of air. Thus, the heat mass exchange process was ensured. During the operation, the porous structure of the packing plate made it possible to separate the upper flowing solution or water into a small square grid. The heat mass was exchanged with air in the packing compartment. The liquid flowed into the tray below and was recirculated into the system. The two-dimensional model diagram of the constant-flow packing is shown in Figure 4.

Owing to the ambiguity surrounding the heat transfer process within the system, the model was examined and assumed in the simulation of the heat and mass transfer performance of the heat-source tower system. Within the enthalpy difference laboratory, the stability of the whole heat transfer process was ensured. While the physical quantities of the solution or water entering the packing were stabilized in the range of constant values by proportioning. In addition, the operating parameters were assumed to vary in two dimensions.

In the simulation process, the law of conservation of mass and the law of conservation of energy were used to calculate the temperature distribution on the solution side. In the heat-mass exchange stage, the total heat consisted of latent heat exchange and sensible heat exchange. The calculation of heat exchange can simulate the enthalpy difference law change at the gas–liquid interface. These changes were used to determine whether the gas–liquid sides were in thermal equilibrium. The law of mass conservation is as follows:

$$dG_w = G_{\mathrm{a}} d\left(d_{\mathrm{a}}\right) \tag{1}$$

The law of conservation of energy is

$$G_{\mathrm{a}} dh_{\mathrm{a}} = G_w dh_w + h_w dG_w = c_{\mathrm{pw}} G_w dT_w + c_{\mathrm{pw}} T_w dG_w \tag{2}$$

Substituting (1) into (2) yields the solution-side temperature distribution as

$$dT_w = \frac{G_{\mathrm{a}}}{G_w}\left[\frac{dh_{\mathrm{a}}}{c_{pw}} - T_w d\left(d_{\mathrm{a}}\right)\right] \tag{3}$$

Latent heat exchange:

$$dQ_m = r_{\mathrm{v}} dG_w \tag{4}$$

Apparent heat exchange:

$$dQ_c = h\left(T_{\mathrm{sw}} - T_{\mathrm{a}}\right) dA \tag{5}$$

The total heat exchange is as follows:

$$dQ = dQ_m + dQ_c = h_d\left[\frac{h}{h_d c_{pa}}\left(h_{sw} - h_{\mathrm{a}}\right) + \left(1 - \frac{h}{h_d c_{pa}}\right) r_{\mathrm{v}}\left(d_{sw} - d_{\mathrm{a}}\right)\right] dA \tag{6}$$

$G_w$ is the solution flow rate, kg/s. $g_{\mathrm{a}}$ is the air flow rate, kg/s. $d_{\mathrm{a}}$ is the air moisture content, %. $h_{\mathrm{a}}$ is the air enthalpy, J. $h_{\mathrm{w}}$ is the solution-side enthalpy, J. $C_{\mathrm{pw}}$ is the solution constant pressure-specific heat capacity, J/(kg°C). $T_{\mathrm{w}}$ is the solution temperature, °C. $r_{\mathrm{v}}$ is the latent heat of vaporization, kJ/kg. $h$ is the heat transfer coefficient, W/(m$^2$ K). $T_{\mathrm{sw}}$ is the solution temperature on the liquid film side, °C.

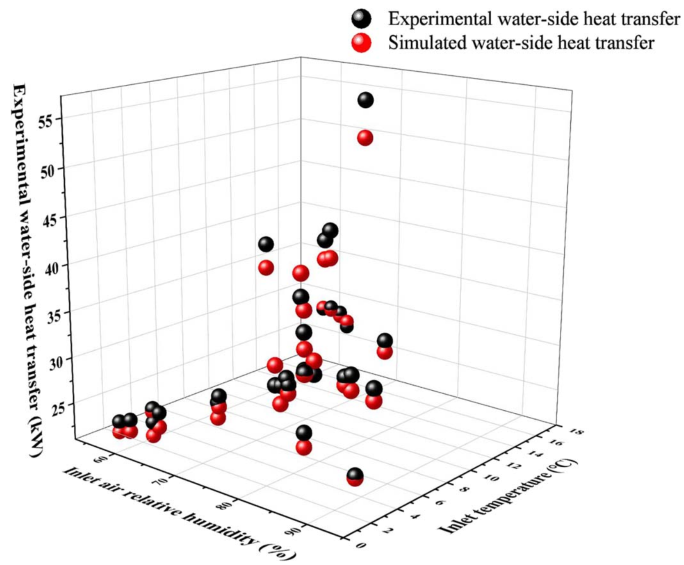



**Figure 5.** *Comparison plot of experimental and simulated data.*

## 3.2. Model validation

During the experimental stage, data from several sets of working conditions were analyzed. Simulation of each set of measured data was also carried out using Visual Basic software. The experimental and simulation results were compared. The comparison results are shown in Figure 5.

From Figure 5, it was derived that both the simulation and the error were within the error range, with the average error ~5% and the maximum error ~8%. The consistency of the simulated and measured results was good, indicating that the experimental model had good accuracy. However, under different working conditions, the experimental error would fluctuate with the test time, system operation and condition of the test instrument.

Based on the test data in standard working conditions, the experiments and simulations can be well fitted. Therefore, in the later study, the simulation was used to derive the operation of the heat-source tower system under different operating parameters.

# 4. Results and discussion

## 4.1. Effects of import water temperature in summer

During summer heat-source tower operation, the heat exchange medium flowing in the piping is water. The dry-bulb temperature of the inlet air is maintained at 31.5°C. The inlet air wet bulb temperature is 28°C. The air flow rate is 27 500 $m^3$/h. The water flow rate in the pipeline is 12 $m^3$/h.

From the change curve in Figure 6, it can be seen that as the water temperature increases, the heat exchange and the water temperature difference increase significantly. The system operates in summer when the outdoor temperature and moisture content are higher. In order to ensure that the system has enough heat exchange with outdoor air, it makes the unit operate at a higher performance. Therefore, the inlet water temperature of the heat-source tower is increased in summer conditions. Meanwhile, according to the relevant research, the optimal temperature difference between the inlet and outlet water for summer conditions is controlled at 5°C.

Therefore, the study examined the water temperature difference in the 5°C working condition. In this working condition, the inlet water temperature is 35°C and the heat exchange is 76 kW. At this time, the difference between the inlet air temperature and the inlet water temperature is only 3.5°C. The temperature of the cold-side water circulation system on the heat-source tower is only lower than the outdoor dry-bulb temperature when it is 3.5°C. Therefore, the heat-source tower can meet the heat requirements by the system.

Figure 6 also shows the air import and export temperature differences and moisture content differences with the inlet water temperature change curve. The air temperature and moisture content difference both rise linearly with the increase in inlet water temperature. The reason is that the medium flowing in the pipe is water. Therefore, the increase in moisture content has little impact on the operation of the system.

## 4.2. Effects of wind volume in summer

Inlet air temperature is 31.5°C. Inlet air wet bulb temperature is 28°C. The inlet water temperature is 35°C. The water flow rate is 12 $m^3$/h. The changes in heat exchange capacity and air inlet and outlet states of the heat-source tower are analyzed for different air volumes.

From the variation in Figure 7, it can be concluded that the water-side heat exchange and latent heat exchange of the system increase basically in parabolic form with the change in the air volume. With the growth of air volume, the heat exchange also increases. But the rate of growth gradually becomes slower. Because the number of air–water contacts increases with the addition of air volume. At the same time, the wind speed growth and the heat exchange between air and water are not sufficient compared to the previous heat exchange. Therefore, although the heat exchange increases, the rate of increment becomes slower.

With the addition of air volume, the latent heat exchange of the system also gains, and the heat exchange increases from 39 to 89 kW. The gain in air volume increases the total heat exchange of the system, thus the heat-source tower system. However, as the air volume increases, the rate of heat exchange increases slowly, and the increase in air volume should be kept within a certain range.

Figure 7 also shows the air temperature difference and moisture content difference vary linearly with the change in air volume. As the air volume increases, the temperature difference in the air decreases, and the moisture content difference reduces. When the air volume is 9000 $m^3$/h, the air temperature difference is as high as 5.15°C and the moisture content difference is 0.93 g/kg. When the air volume is 33 500 $m^3$/h, the temperature difference is only 2.99°C and the moisture content difference is only 0.08 g/kg. With the growth of air volume, the system moisture increase becomes slower. In an environment of high humidity, the air flow can be

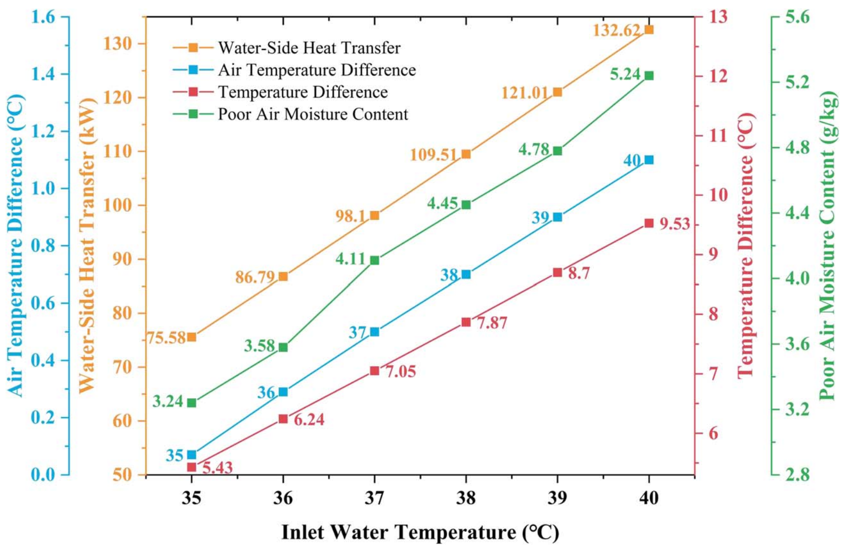



**Figure 6.** *Effect of inlet water temperature on the performance of heat source towers in summer.*

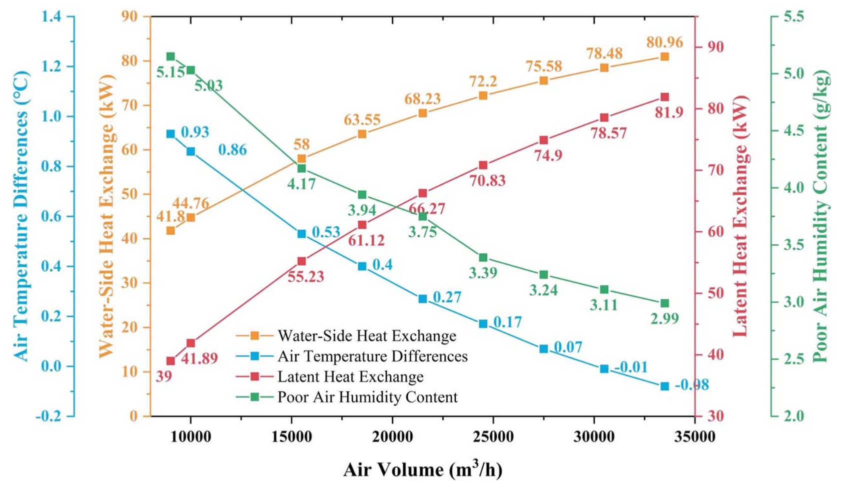



**Figure 7.** *Effect of airflow on the performance of heat source towers in summer.*

increased to reduce the moisture entering the system, which can impact the operation of the system.

## 4.3. Effects of water flow in summer

Inlet air temperature is 31.5°C. Inlet air wet bulb temperature is 28°C. Inlet water temperature is 35°C. The air volume is 15 000 $m^3$/h. The changes in heat exchange capacity and air inlet and outlet states of the heat-source tower are analyzed under different water flow rates.

From the curve in Figure 8, it can be concluded that with the growth of the water flow rate, the heat exchange capacity of the heat-source tower increases, but the rate becomes gradually slower. At the same time, the heat exchange temperature difference gradually reduces. At the water flow rate of 8 $m^3$/h, the heat exchange is 46.07 kW, and the heat exchange temperature difference is 4.97°C. When the water flow rate is 19 $m^3$/h, the heat exchange is 69.25 kW, and the heat exchange temperature difference is 3.14°C. The heat exchange increased by 50%, and the rate of increase became slower. The increase in water flow rate can reduce the heat exchange temperature difference. The heat exchange performance of the heat-source tower can be improved by the reduction in temperature difference.

Figure 8 also shows the air temperature difference and moisture content difference with the water flow rate change curve. Increasing the water flow can parabolically increase the air temperature difference. In this case, the moisture content difference increases linearly but slowly. Because, as the water flow rate increases, the rate of growth for latent heat exchange in the circulation line inside the heat-source tower is increased, and the heat exchange rate increases. At the same time, it causes water from outside to enter the system, but since circulating water is water in summer. Therefore, during summer operations, the efficiency of the heat-source tower can be increased by providing water flow.

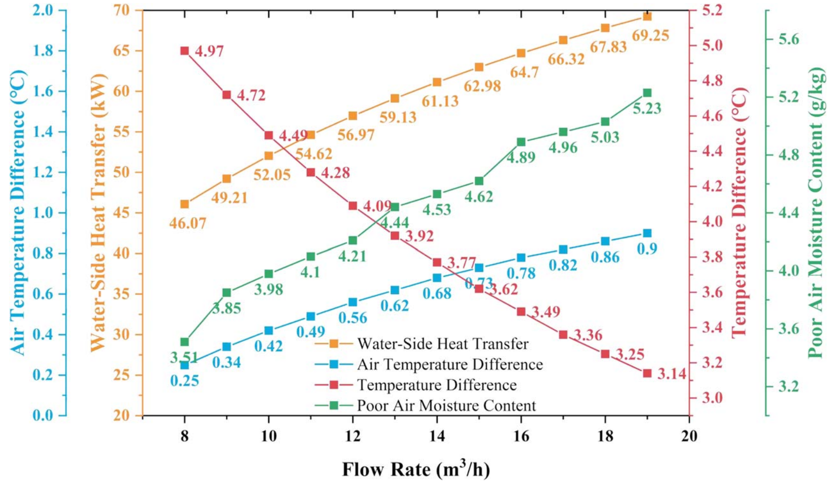



**Figure 8.** *Effect of water flow rate on the performance of heat source towers in summer.*

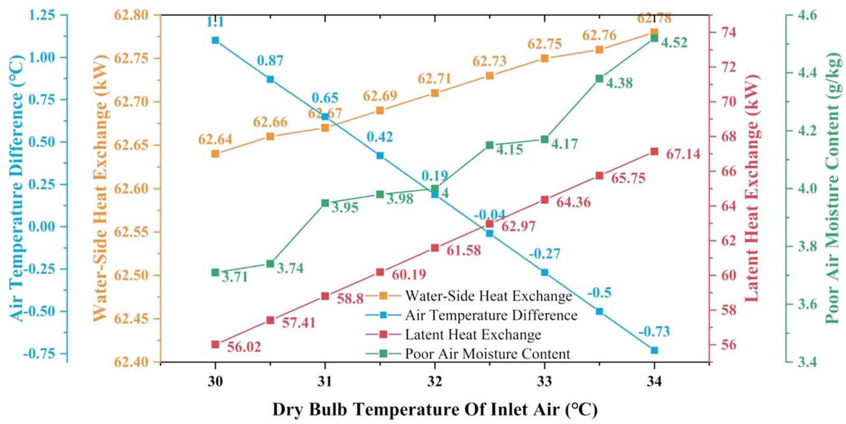



**Figure 9.** *Effect of inlet air dry-bulb temperature on the performance of heat source towers in summer.*

## 4.4. Effects of dry-bulb temperature on imported air in summer

The water flow rate is 12 $m^3$/h. The inlet air wet bulb temperature is 28°C. The inlet water temperature is 35°C. The air flow rate is 18 000 $m^3$/h. The heat transfer capacity of the heat-source tower and the change of air inlet and outlet states are studied by varying the dry-bulb temperature of the inlet air.

As shown in Figure 9, the total heat exchange and water-side heat exchange of the system grow linearly with the increase in inlet air temperature, and the rate of increase is fast. The latent heat exchange of the system increases by 11.02 kW when the inlet dry-bulb temperature increases from 30°C to 34°C. Therefore, the operation of the heat-source tower system benefits from increased air inlet temperature.

As shown in Figure 9, with the increase in the dry-bulb temperature of the inlet air, the difference in temperature of the inlet and outlet air decreases, while the difference in moisture content of the inlet and outlet air increases linearly. The increase in inlet air temperature affects the change in inlet air temperature. It improves the sensible heat exchange and latent heat exchange of the system, and the slow growth of heat exchange capacity does not have much effect on the interface water vapor partial pressure, so the difference between the water vapor partial pressure on the air side and the interface side water vapor partial pressure does not have much effect, and the difference between the moisture content of the import and export air becomes larger.

## 4.5. Effects of moisture content of imported air in the summer

The inlet air dry-bulb temperature is 31.5°C. The water flow rate is 12 $m^3$/h. The inlet solution temperature is 35°C. The air volume is 18 000 $m^3$/h. The air volume is 18 000 $m^3$/h. The changes of heat exchange capacity and air inlet and outlet states of the heat-source tower are analyzed for different inlet air moisture content.

From the curve in Figure 10, it can be seen that the system heat exchange and the system latent heat exchange decrease linearly with the increase in inlet air moisture content. When the moisture content of inlet air increases from 21.74 to 26.56 g/kg, the total heat exchange of the system decreases by 31% and the latent heat exchange decreases by 35%. The increase in moisture content of

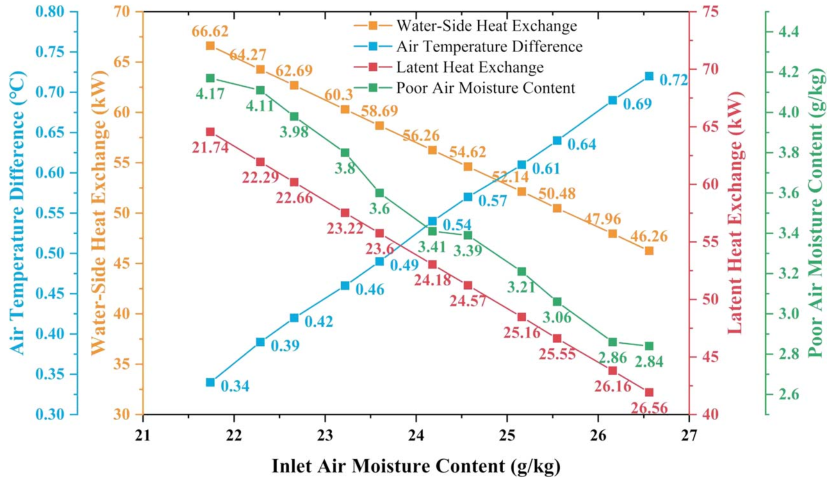



**Figure 10.** *Effect of humidity content of inlet air on the performance of heat source towers in summer.*

inlet air is equivalent to the increase in inlet air temperature. This reduces both the heat exchange and latent heat exchange in the system. Therefore, as the moisture content of the air increases, it affects the operation of the heat-source tower system. However, the effect on operating efficiency is very low.

As shown in Figure 10, with the rise in inlet air moisture content, the inlet air temperature difference increases in a straight line. The air moisture content difference decreases linearly. Since the circulating solution medium of the summer system is water, the influence of moisture content is not a major factor for the system. In a high temperature and high humidity environment, the system can still ensure normal operation.

## 4.6. Effects of imported solution temperature in winter

Dry-bulb temperature of inlet air is 4.5°C, while the inlet air wet bulb temperature is 3.5°C. The air flow rate is 27 500 $m^3$/h, and the glycol solution flow rate is 12 $m^3$/h. The air flow rate is 27 500 $m^3$/h. The glycol solution flow rate is 12 $m^3$/h. The effect of different inlet solution temperatures on the performance of the heat-source tower is analyzed.

As shown in Figure 11, the heat exchange and glycol solution temperature difference decrease almost linearly with the increase in solution temperature. The heat-source tower is operated in winter. Since the outdoor temperature and moisture content are much smaller compared to summer, which is necessary to ensure that enough heat is absorbed from the air. At the same time, to ensure that the system operates at a higher level of performance, it is necessary to increase the inlet solution temperature of the heat-source tower as much as possible. For the same heat exchange, the glycol solution flow rate is increased, and the temperature difference between the inlet and outlet solutions is reduced. Therefore, related scholars believe that the temperature difference between the inlet and outlet solutions of the heat source tower should be controlled at 3°C during winter operation, instead of its 5°C in summer.

For the heat-source tower solution operating at a low-temperature difference of 3°C, the inlet solution temperature is 3°C and the heat exchange is 35.8 kW. The difference between the inlet air temperature and solution temperature is only 7.5°C at this time. This indicates that the heat-source tower can provide enough low-temperature heat when the lowest point temperature of the cold side solution circulation system is only 7.5°C below the outdoor dry-bulb temperature.

Figure 11 also shows the graphs of air inlet and outlet temperature differences and moisture content differences with the temperature of the inlet glycol solution. The air temperature difference and moisture content difference decrease linearly with the growth of the inlet solution temperature. At the same time, during the operation, the moisture content gradually decreases with the increase in operation time, and the glycol solution gradually dilutes. Therefore, increasing the concentration of the glycol solution has an important impact on the operation of the system.

## 4.7. Effects of airflow in winter

Dry-bulb temperature of inlet air is 4.5°C. Inlet air wet bulb temperature is 3.5 v. The inlet glycol solution temperature is 3°C. The inlet glycol solution flow rate is 12 $m^3$/h. The glycol solution flow rate is 12 $m^3$/h. The effect of winter air volume on the performance of the heat-source tower is analyzed.

As shown in Figure 12, the total heat exchange of the system basically increases parabolically with the change in air volume, while the latent heat exchange of the system shows a weak growth trend with the change in air volume. With the increase in air volume, the heat exchange increases. However, the increase in air volume becomes slower. The main reason is that as the air volume increases, the amount of air-solution contact increases. At the same time, the air speed increases, and the heat exchange between air and glycol solution is not sufficient compared with before. Therefore, the heat exchange increases, but the rate of increase becomes slower.

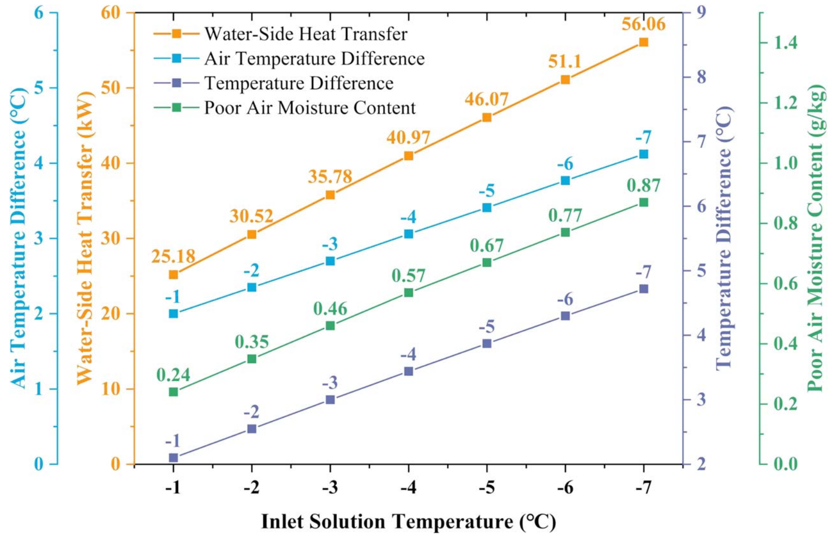



**Figure 11.** *Effect of inlet glycol solution temperature on the performance of heat source towers in winter.*

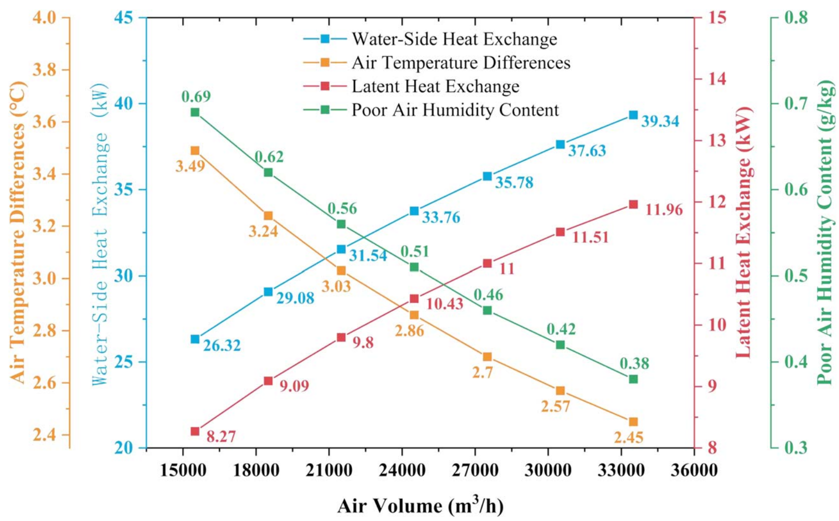



**Figure 12.** *Effect of airflow on the performance of heat source towers in winter.*

With the increase in air volume, the latent heat exchange of the system shows a weak increase, and the heat exchange increases from 8.3 to 12 kW. The increase in air volume increases the total heat exchange of the system, and the latent heat exchange shows a weak increase. Thus, for the heat-source tower system, increasing the air volume is beneficial to improve the system's performance. However, as the air volume increases, the rate of heat exchange increases slowly, and the increase in air volume should be kept within a certain range.

As shown in Figure 12, as the air volume increases, the temperature difference of the air decreases and the moisture content difference decreases. Therefore, the increase in air flow will greatly reduce moisture transfer from the system. Thus, in the case of higher outdoor air moisture content, the air flow rate can be increased appropriately to reduce the moisture in the solution.

## 4.8. Effects of glycol solution flow in winter

Dry-bulb temperature of inlet air is 4.5°C. Inlet air wet bulb temperature is 3.5°C. The inlet glycol solution temperature is 3°C. The air volume is 12 $m^3$/h. The air volume is 12 $m^3$/h. The effect of the variation of the glycol solution flow rate on the performance of the heat-source tower under winter conditions is analyzed.

As shown in Figure 13, with the increase in the glycol solution flow rate, the heat exchange of the heat-source tower increases, but the increase rate becomes slower. The heat exchange temperature difference gradually decreases. When the glycol solution flow rate increased from 8 to 20 $m^3$/h, the heat exchange increased by 40%. However, as can be inferred from the trend of heat exchange in the graph, the rate of increase becomes gradually slower. Meanwhile, the increase in the glycol solution flow rate can reduce the heat transfer temperature difference, which is beneficial to improve the heat transfer performance of the heat-source tower.

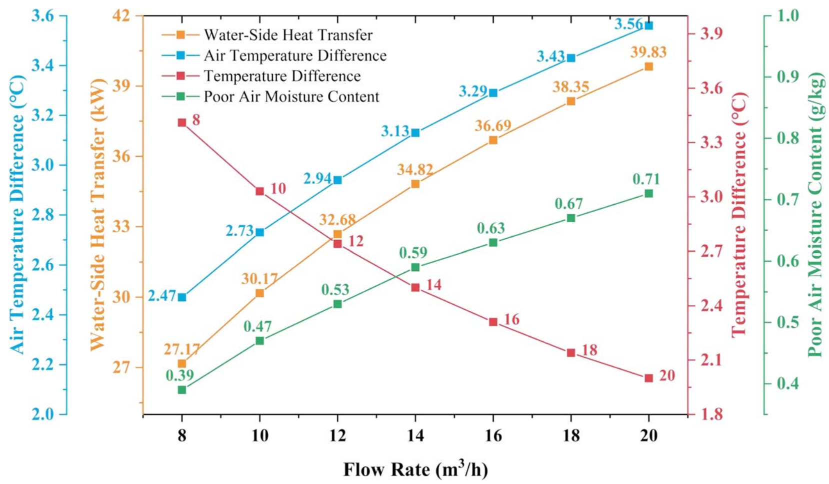



**Figure 13.** *Effect of glycol solution flow rate on the performance of a heat source tower in winter.*

As shown in Figure 13, the air temperature difference and moisture content difference increased parabolically with the increase in glycol solution flow rate. When the glycol solution flow rate increased from 8 to 20 $m^3$/h, the air temperature difference between import and export increased by 1.09°C, while the moisture content difference increased by 0.32 g/kg. The increase in glycol solution flow rate increased the rate of latent heat exchange, which caused an increase in water absorption. Therefore, it is not conducive to the normal operation of the heat-source tower solution.

Therefore, increasing the solution flow rate is conducive to enhancing the system heat exchange and reducing the temperature difference between the inlet and outlet solution. However, the latent heat exchange in the system increases with the increase in solution flow rate, and more moisture in the air enters the solution. It is not conducive to the stable operation of the system. Therefore, it is necessary to consider the performance of the heat-source tower to determine the appropriate solution flow rate.

## 4.9. Effects of inlet air temperature in winter

The solution flow rate is 12 $m^3$/h. The inlet air wet bulb temperature is 3.5°C. The inlet glycol solution temperature is 3°C. The air volume is 18 000 $m^3$/h. The air volume is 18 000 $m^3$/h. The effect of the variation in inlet air temperature on the performance of the heat-source tower under winter conditions is analyzed.

As shown in Figure 14, with the increase in inlet air temperature, the total heat exchange of the system increases linearly, while the latent heat exchange of the system decreases almost linearly. When the outdoor air temperature is 3.5°C, the heat exchange of the system is 27.6 kW, of which the latent heat exchange due to the condensation of moisture in the air into the glycol solution is up to 12.7 kW. When the outdoor air temperature is up to 7.5°C, the total heat exchange of the system is up to 31.1 kW, while the latent heat exchange is close to 0 kW, accounting for <20% of the total heat exchange. Therefore, with other parameters unchanged, increasing the inlet air temperature is beneficial to the heat-source tower system.

As shown in Figure 14, with the increase in inlet air temperature, the difference between import and export air temperatures increases linearly, while the difference of moisture content of import and export air decreases almost exponentially. The increase of inlet air temperature greatly increases the variation of inlet and outlet air temperatures, which improves the sensible heat exchange capacity of the system and thus the overall heat exchange capacity of the system. Due to the increase in heat transfer capacity, the average temperature of the import and export glycol solution increases, and thus its interface water vapor partial pressure increases accordingly. The difference between the water vapor partial pressure on the air side and the water vapor partial pressure on the interface side decreases, and thus the moisture content difference between the import and export air becomes smaller.

## 4.10. Effects of moisture content of imported air in winter

Inlet air dry-bulb temperature is 4.5°C. The inlet glycol solution temperature is 3°C. The glycol solution flow rate is 12 $m^3$/h. The air volume is 18 000 $m^3$/h. The glycol solution flow rate is 12 $m^3$/h. The air volume is 18 000 $m^3$/h. The effect of the change in inlet air temperature on the performance of the heat-source tower under winter conditions is analyzed.

As shown in Figure 15, with the increase in inlet air moisture content, the system heat exchange and the system latent heat exchange basically increase linearly. When the moisture content of inlet air is 3.35 rises to 4.83 g/kg, the total heat exchange of the system increases by 8.1 kW. While the latent heat exchange of the system increased by 15.5 kW, the increase in latent heat exchange was greater than the growth of total heat exchange when the moisture content of the air increased. Therefore, with the increase in air moisture content, the latent heat exchange generated due to the moisture in the air entering the solution increases, and the total heat exchange is greatly improved.

From the point of view of the normal operation of the heat-source tower, a large amount of moisture entering the glycol

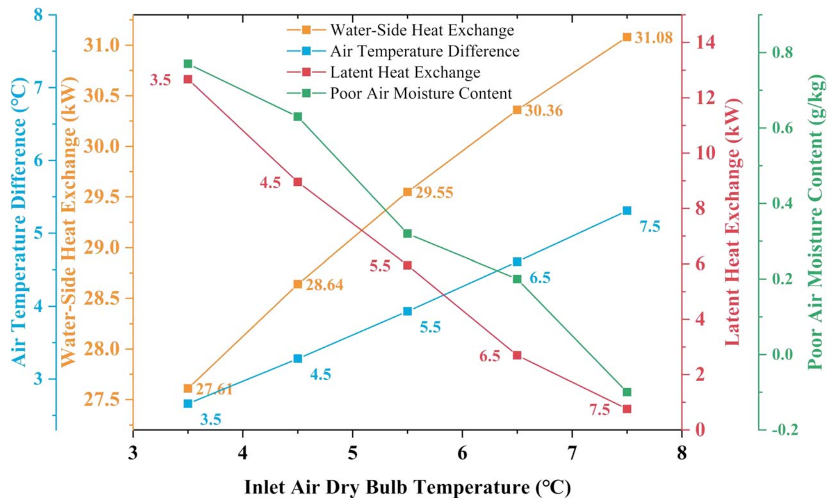



**Figure 14.** *Effect of inlet air dry-bulb temperature on the performance of heat source towers in winter.*

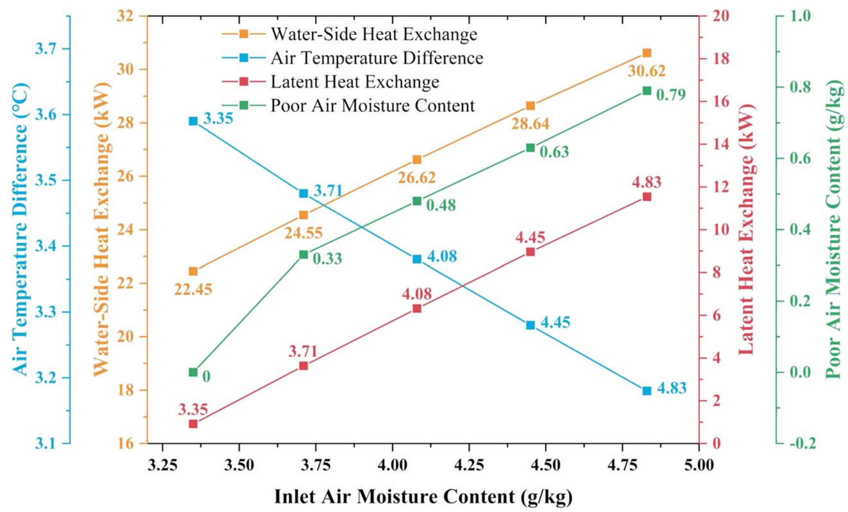



**Figure 15.** *Effect of humidity content of inlet air on the performance of heat source towers in winter.*

solution will cause the solution to become diluted faster. It is not conducive to the normal operation of the system. However, from the perspective of heat exchange, increasing the moisture content in the air is conducive to increasing the total heat exchange of the system. This allows the unit to operate at higher efficiency.

As shown in Figure 15, as the moisture content of imported air increases, the temperature difference between imported and exported air decreases in a straight line, while the moisture content difference increases. The increase in outdoor air moisture content makes the apparent heat exchange of the system due to the temperature difference decrease to a certain extent. The latent heat exchange due to the moisture content difference is greatly increased. The solution absorbs a large amount of moisture in the air. Therefore, under high outdoor moisture content conditions, the concentration of the solution becomes diluted at a dramatically faster rate. A special solution regeneration system or continuous solute addition is required to make the system operate properly.

## 5. Conclusion

In this article, a cross-flow heat-source tower experimental bench is built, and the experimental study is conducted to test and investigate the performance of the existing constant-flow packed heat-source tower in terms of heat transfer and water absorption performance. Furthermore, the corresponding simulation model is established. The variation of heat transfer and water absorption in the system with the inlet solution temperature, solution flow rate, air volume, inlet air temperature and moisture content of the inlet air is pointed out. Theoretical support and data support are provided for the optimization of the heat-source tower system. The main conclusions drawn are as follows:

1) The heat exchange process of the heat-source tower needs to be considered from both performance and safety aspects. From the perspective of performance, it is necessary to

improve the system's heat exchange. For safety, it is necessary to make the latent heat exchange of the heat exchange process as zero as possible. The two aspects need to be considered together.

2) The heat exchange rate of the heat-source tower is 88.68 kW in summer and 31.56 kW in winter, so the heat exchange rate in summer is about 2.8 times the heat exchange rate in winter. Therefore, the heat exchange during summer operation is stronger than that in winter.

3) For the operation parameters, the heat exchange can be improved by increasing the air volume and solution flow rate within a reasonable range. For the safety of the system, the increase in air volume will reduce the latent heat exchange in the system. It is conducive to the safe and stable operation of the system. The increase in solution flow rate will lead to an increase in latent heat exchange in the system. The amount of water entering the solution is accelerated, which is not conducive to the safe operation of the system.

4) For the inlet air parameters, increasing the air temperature and moisture content can increase the heat exchange in the system. As for the safety of the system, the increase in air temperature can reduce the amount of water vapor in the air that enters the solution. While increasing the air moisture content will increase the amount of water vapor in the air, which is not conducive to the safe operation of the system.

5) For the inlet solution parameters, increasing the water temperature helps to improve the heat exchange in the system. However, increasing the solution temperature will reduce the heat exchange in the system. At the same time, the latent heat exchange and latent heat exchange percentages are reduced to some extent. In order to achieve system heat exchange, the inlet solution temperature should be increased as much as possible to make the system operate safely.

## Author contributions

Yifan Mao (Conceptualization [lead], Resources [lead], Writing – original draft [lead]), Yongcun Li (Funding acquisition [lead], Resources [lead]), Xiantai Wen (Methodology [lead], Resources [supporting], Validation [lead]), Xiaolei Yuan (Formal analysis [supporting], Validation [supporting], Writing – review & editing [lead]) and Zhaofan Wu (formal analysis [Supporting], Software [supporting]).

## Data availability

No data was used for the research described in the article.

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