Effects of the operation parameters of a closed-type heating tower on the performance of heating tower heat pump system

Feng Rong¹,², Li Xiuzhen³, Zhao Xingcheng¹ and Fang Junfei¹,²

Abstract
The ventilator of the heating tower and the circulating pump of the anti-freeze solution are the main electrical equipment of a heating tower heat pump system, besides the compressor. By controlling the working frequencies of the ventilator of the heating tower and circulating pump of the anti-freeze solution, the effects of the operation parameters of a closed-type heating tower on its heat absorption and the performance of heating tower heat pump system were investigated under winter heat conditions. The results indicated that reducing the frequency of the circulating pump for the anti-freeze solution leads to a decrease in the temperature of the outlet evaporator of the anti-freezing solution and an increased temperature difference between the anti-freeze solution flowing into and out of the heating tower; meanwhile, excessively high and low anti-freeze flow rates lead to reduced heat absorption of the closed-type heating tower. The coefficient of performance fluctuates slightly if the frequency of circulating pump is above 20 Hz, but a slight drop in coefficient of performance is observed when the frequency is less than 15 Hz. The system energy efficiency ratio tends to increase as the frequency of circulating pump is reduced, although a substantial reduction occurs at 10 Hz. Furthermore, a reduced ventilator frequency decreases the temperatures of the anti-freeze solution at the inlet and outlet of the heating tower and the temperature difference, hindering the heat absorption of the heating tower. With reductions in the ventilator frequency, the coefficient of performance exhibits an initial increase followed by subsequent decreases, while the system energy efficiency ratio showed continual increases until the ventilator frequency dropped to 10 Hz. When the ventilator frequency or

¹School of Mechanical Engineering, Shaanxi University of Technology, Hanzhong, China
²Shaanxi Key Laboratory of Industrial Automation, Hanzhong, China
³Institute of Refrigeration and Air Conditioning, Henan University of Science and Technology, Luoyang, China

Corresponding author:
Feng Rong, Shaanxi University of Technology, Dongyihuan Road 1#, Hanzhong 723001, China.
Email: fengrong19871221@126.com

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circulating pump frequency drops to 15 Hz and 10 Hz, the evaporation temperature of the heat pump unit decreases, resulting in an excessively exhaust temperature, which is not favorable for the safe operation of the system.

Keywords
Closed-type heating tower, operation parameters, heat pump system, energy efficiency, coefficient of performance

Introduction
The heating tower heat pump system (HTHP) is a novel heat pump which was developed based on conventional air-source heat pump systems and water-source/geothermal heat pump systems. Using air as the heat source, the HTHP transfers low-temperature heat from the air to heating towers that contain working fluids with low freezing points under heating conditions. The heat is then transported to the evaporator of the water-source heat pump unit and subsequently released to generate high temperature heat for users such as buildings. The HTHP mitigates issues such as frost, which is prevalent in evaporators of conventional air-source heat pump systems, and the constraints of geological conditions present in water-source/geothermal heat pump systems (Kim et al., 2017; Liu et al., 2019a, 2019b; Wang et al., 2019; Xu et al., 2020). Therefore, HTHPs have attracted significant research attention in recent years.

Many studies have focused on the heat and mass transfer in heating tower at present. Tan and Deng (2002) established the heat and mass transfer model based on the standard Merkel equation. Referring to the problem that the thermal resistance of the liquid film cannot be ignored during heat absorption, the average temperature of air and water is used to replace the temperature at the air–liquid interface. Later, they improved the model to calculate the state parameters of air and water at any position inside the Heating Tower (Tan and Deng, 2003). Lu et al. (2017) set up a model in which a variable Lewis number is considered, the results show that an increase in the air dry bulb temperature can significantly boost the sensible heat transfer and slightly reduce the transfer of latent heat. Wen et al. (2012) conducted studies on the heat transfer coefficient between liquid and air. They observed that the flow rate of the liquid and air considerably affect the heat transfer coefficient, but these have a small effect on the inlet temperature. Then, the correlation formula of the heat transfer coefficient is acquired by regression. Huang et al. (2017) constructed an experimental system with glycol solution as the working medium. The heat transfer coefficient is studied without consideration of the Lewis law. They also obtained the correlation equation of the heat and mass transfer coefficient and found that Lewis number in these experiments is between 0.91 and 1.12. He et al. (2015) established a heat and mass transfer model of a counter-current closed-type heating tower under frost-proof conditions. It is found that the distribution of the air enthalpy in heating tower is nearly linear. The heat transfer rate in the upper half part of the finned coil tube heat exchanger is better than that in the lower half part. Song et al. (2019) conducted an experimental investigation to figure out the heat and mass transfer characteristics of a cross-flow closed-type heating tower, and the correlations of heat and mass transfer coefficients for heat tower were developed. Aiming at the problem
of moisture absorption in heating tower in winter, Wen et al. (2018, 2019) proposed and studied a new type of anti-freezing solution regeneration system based on vacuum boiling.

Some works also paid their attention to the heating performance of HTHPs. Meng et al. (2011) tested the heating performance of an open-type HTHPs in winter in Nanjing; the results indicate that the heating capacity of the heat pump attenuates with a decrease in the air wet bulb temperature. Jia et al. (2015) put forward the idea of a heat pump system in which the heating tower is coupled with an auto-cascade refrigeration system. Huang et al. (2019a) carried out a large-scale comprehensive performance evaluation of the HTHPs in 869 typical locations in the warm, mixed and cool climate zones over the world; the results show that the HTHPs have excellent performance in the warm and mixed climate zones and also applicable in the cool climate zone. To investigate the performance improvements of the HTHP over the air-source heat pump, a comprehensive comparison between the two systems was carried out based on a simulation study by Huang et al. (2019b); it was found that the HTHP achieves an increase of 7.4% in efficiency that air-source heat pump in winter.

In the winter season, the enthalpy difference of air at the inlet and outlet of the heating tower tends to be small, resulting in large volume of air needed. Meanwhile, the working fluids have low freezing points, high viscosity and low specific heat capacities compared with water, resulting in variations of the required flow. In an HTHP, the ventilator of the heating tower and circulating pump for the anti-freezing solution are key power-consuming equipment apart from the compressor (Xiong et al., 2018). The frequency of the ventilator and circulating pump has a significant effect on the flow rate of air and anti-freezing solution, which not only related to the heat transfer between the air and anti-freezing but also to the energy consumption by the system. However, to date, the effects of varying the operation parameters of the heating tower ventilator and the circulating pump for the anti-freezing solution on the performance of the HTHPs have not been thoroughly investigated. Hence, the purpose of this study is to investigate the influence of the operation parameters of a closed-type heating tower on its heat absorption characteristics, and then the performance of the heat pump system was further studied under heating conditions in winter condition by tuning the operation frequencies of the ventilator and circulating pump in the proposed small-scale closed-type HTHPs.

Closed-type HTHPs

A schematic diagram of the small-scale closed-type HTHP proposed is shown in Figure 1, consisting of a closed-type heating tower, a heat pump unit and several valves. The components of the closed-type heating tower from top down include an axial ventilator, sprinkler, finned tube heat exchanger, air outlet, sprinkling pool and a sprinkling pump. As heat absorption from air by the finned tube heat exchanger leads to a reduced temperature and increased density of air in winter, the direction of ventilation of the axial ventilator is oriented from top to bottom. Meanwhile, the anti-freeze solution in the closed-type heating tower flows vertically upwards from the bottom to generate a counter-current during the heat exchange between the anti-freeze solution and air. The heat pump unit consists of a compressor, a plate-type heat exchanger used as an evaporator, a shell-tube heat exchanger used as a condenser, a four-way reversing valve, a throttle valve, a reservoir and a gas/liquid separator. R22 was used as the refrigerant in the HTHPs, while glycol solution (volume fraction = 30%) was used as the anti-freeze working media in the heating tower. To ensure smooth circulation, an expansion water tank was installed in front of the inlet of the
circulating pump for the anti-freeze solution, and the heating tower ventilator and circulating pump for the anti-freeze solution were also equipped with a frequency inverter to allow for tuning of the operation parameters. Table 1 summarizes the model and parameters of the main equipment of the closed-type HTHPs.

**Mechanism and method**

Two types of tests were conducted, in which variables and parameters were varied or fixed to investigate the influences of each. In Test A, the working frequency of the heating tower ventilator was fixed, and the operation frequency of the circulating pump for the anti-freeze solution was tuned, while in Test B, the operation frequency of the circulating pump for the anti-freeze solution was fixed, and the working frequency of the heating tower ventilator was varied. The effects of air volume and flow rate of the anti-freeze solution on the system performance were reflected by the heat absorption of the heating tower, coefficient of
performance (COP) of the heat pump unit and the system energy efficiency ratio (SEER) of the HTHPs.

The heat absorbed by the closed-type heating tower from air (denoted as $Q_e$) is equivalent to that absorbed by the refrigerant in the evaporator and can be calculated by

$$Q_e = c_1 L_1 \rho_1 (T_2 - T_1)$$  \hspace{1cm} (1)

where $c_1$, $L_1$ and $\rho_1$ are the specific heat (kJ/(kg°C)), volume flow rate (m$^3$/s) and density (kg/m$^3$) of the anti-freeze solution, respectively; $T_2$ is the temperature of the anti-freeze solution flowing out of heating tower and into the evaporator (°C) and $T_1$ is the temperature of the anti-freeze solution flowing out of evaporator and into the heating tower (°C).

Denoted as $Q_c$, the heat generated by the system can be calculated by

$$Q_c = c_2 L_2 \rho_2 (T_4 - T_3)$$  \hspace{1cm} (2)

where $c_2$ is the specific heat of the working fluid, which is 4.187 kJ/(kg°C) in this case; $L_2$ is the volume flow rate of the working fluid (m$^3$/s); $\rho_2$ is the water density, which is 1000 kg/m$^3$ in this case; $T_4$ is the temperature of working fluid flowing out of condenser (°C); $T_3$ is the temperature of working fluid flowing into condenser (°C).

Subsequently, the COP and SEER of the heat pump unit can be calculated by

$$\text{COP} = \frac{Q_c}{E_c}$$  \hspace{1cm} (3)

$$\text{SEER} = \frac{Q_c}{E_c + E_v + E_p}$$  \hspace{1cm} (4)

where $E_c$, $E_v$, and $E_p$ are energy consumption by compressor, axial ventilator and circulating pump, respectively, kW.

According to the law of energy conservation, the sum of the heat absorption of the evaporator in heat pump unit and power consumption of the compressor shall be equivalent to the heat release of the condenser. Hence, the errors at energy equilibrium were obtained by

$$\eta = \frac{E_c + Q_e - Q_c}{Q_c} \times 100\%$$  \hspace{1cm} (5)

Figure 2 shows the structure of the system and the arrangement of measuring points. As illustrated in the diagram, the temperature sensors were installed in the pipes between the evaporator outlet of the anti-freeze solution and the inlet of the heating tower; between the
outlet of the heating tower and the evaporator inlet near the anti-freeze solution, and between the condenser outlet and the inlet near the anti-freeze solution of the working media. Flowmeters were installed in the pipes between the outlet of heating tower and the evaporator inlet near the anti-freeze solution and the pipes between the outlet of the heating tower and the condenser outlet near the working media. In addition, temperature and humidity sensors were installed at the upper inlet of the closed-type heating tower and the compressor power was monitored in real time using a three-way power transmitter.

Furthermore, external insulation patch-type temperature sensors were installed near the inlets and outlets of the compressor, condenser and evaporator to monitor the temperature of the refrigerant, which can facilitate the analysis of the heat pump unit during operation.

Table 2 summarizes the model and parameters of the equipment used. All data were collected automatically by a data collector (Agilent-349702) at time intervals of 10 s.

Several assumptions were involved to simplify calculation and analysis. First, the heat losses of pipes to heat were considered to be negligible for the pipes which were covered with insulation. Second, the power consumptions of the axial ventilator and circulating pump for the anti-freeze solution were assumed to remain constant after frequency tuning and are the products of their rate powers with ratios of actual frequencies with 50 Hz. Third, the temperatures measured by the patch-type sensors were taken to be equivalent to the refrigerant temperatures due to the small thickness and large thermal conductivity of the copper tubes. Finally, the specific heat and density of water were deemed to be constant. Based on the specific heat and density of the glycol solution in literature (Liu, 1995) and the temperatures measured, correlations were obtained by linear regression fitting and the specific heat and

Figure 2. System structure and arrangement of measuring points. 1 – axial ventilator; 2 – sprinkler; 3 – finned tube heat exchanger; 4 – air outlet; 5 – sprinkling pool; 6 – sprinkling pump; 7 – expansion water tank; 8 – circulating pump for anti-freezing solution; 9 – evaporator; 10 – compressor; 11 – four-way reversing valve; 12 – condenser; 13 – throttle valve; T: temperature sensor; L: flowmeter; R: humidity sensor.
density of the anti-freeze solution were calculated by substituting in the average temperature of the anti-freeze solution. Table 3, equations (6) and (7) show the relevant values used in the calculations

\[ c_1 = 0.0029 \left( \frac{T_1 + T_2}{2} + 273.15 \right) + 2.802 \quad (R^2 = 0.9998) \] (6)
\[
\rho_1 = -0.2654 \left(\frac{T_1 + T_2}{2} + 273.15\right) + 1124.2 \quad (R^2 = 0.9982) \tag{7}
\]

Results and discussion
In order to make it easier to distinguish, the data for Test A and Test B are marked with “a” and “b”, respectively, later. In Test A, the frequency of the circulating pump was decreased by 5 Hz at 15 min intervals from 50 Hz to 10 Hz, while the ventilator frequency was fixed at 40 Hz. In Test B, the frequency of circulating pump was fixed at 20 Hz and the ventilator frequency was decreased by 5 Hz every 15 min from 50 Hz to 10 Hz. The relationship between the frequency of the ventilator and circulating pump and the average air speed and flow rate of anti-freeze solution are shown in Figure 3. Figure 4 shows the variations in temperature and humidity of air during the two tests, where environmental temperature and relative humidity are seen to fluctuate between 3.9°C–5.4°C and 64.4%–71.6%, and 6.1°C–7.1°C and 58.9%–64.3% in Tests A and B, respectively.

Effects on heat adsorption and heat production
The changes in temperature and flow rate of the anti-freeze solution in Tests A and B are shown in Figure 5(a) and (b), respectively. In Test A, the flow rate of the anti-freeze solution decreased with the frequency of the circulating pump and became constant as the frequency of circulating pump was stabilized. The temperature of the anti-freeze solution flowing out of the evaporator decreases with a reduction in its flow rate, while the temperature of the anti-freeze solution flowing out of heating tower was independent from its flow rate if the frequency of the circulating pump was >20 Hz. As the frequency of the circulating pump dropped continuously, the temperature of anti-freeze solution flowing out of the heating tower increased slightly. Overall, a reduction in frequency of the circulating pump resulted in a lower anti-freeze flow rate and the temperature difference of the anti-freeze solutions flowing into and out of heating tower increased. Meanwhile, in Test B, the frequency of the ventilator was varied, while the circulating flow rate of the anti-freeze solution remained constant. The temperatures of the anti-freeze solutions at the inlet and outlet of the heating tower seem to decrease. At ventilator frequencies of <20 Hz, significant reductions in the temperatures of the anti-freeze solutions were observed, while the temperature difference of the anti-freeze solution flowing into and out of the heating tower decreased with the ventilator frequency. Figure 6(a) and (b) presents the trends of the cooling water flow rate at the end and temperature in Tests A and B, respectively. The results indicate that the flow rate of cooling water fluctuated at around 2.90 m³/h, and that the cooling water temperature failed to remain constant due to the influences of end heating and system operation.

The variations in the heat produced by the HTHPs, heating tower heat absorption and compressor power consumption in Tests A and B are shown in Figure 7(a) and (b), respectively. The heat absorption of the heating tower exhibits an initial increase as the circulating pump frequency was reduced. Once the circulating pump frequency dropped to 15 Hz and 10 Hz, the heat absorbed by the heating tower decreased significantly, demonstrating that the excessive flow rate of the anti-freeze solution leads to reduced heat exchange due to the shortened time for heat exchange between the anti-freeze solution and air. Conversely, an
Figure 3. Relationship between the frequency and air speed and flow rate of solution.

Figure 4. Environmental temperature and humidity of air. In order to make it easier to distinguish, the data for Test A and Test B are marked with “a” and “b”, respectively.

Figure 5. Flow and temperature of anti-freezing solution. In order to make it easier to distinguish, the data for Test A and Test B are marked with “a” and “b”, respectively.
insufficient anti-freeze flow rate leads to reductions in the heat transfer coefficient and heat exchange within the finned tubes. The heating absorbed by the heating tower was observed to decrease with the ventilator frequency at a rate which was inversely proportional to the ventilator frequency, indicating that an increase in the ventilator frequency is favorable for enhancing the heat absorption of the heating tower.

In this study, the maximum heat generated by the system exceeded 13 kW. At circulating pump frequencies of >20 Hz, the heat generation of the system was observed to decrease gradually with the frequency. If the circulating pump frequency dropped to 15 Hz and 10 Hz, greater reductions in heat generation were found as the frequency decreased. At maximum ventilator frequency, the heat generation of the system was lower than that at a ventilator frequency of 45 Hz, due to the high temperature of the cooling water, although the heat absorption of the heating tower was maximized. Meanwhile, at ventilator frequencies of <40 Hz, the heat generation of the system dropped with ventilator frequency. Likewise, the decreasing rate of heat generated by the system was inversely proportional

![Figure 6](a) Flow and temperature of cooling water. In order to make it easier to distinguish, the data for Test A and Test B are marked with “a” and “b”, respectively.

![Figure 7](a) Heat generation of the system, heat absorption of the heating tower and power consumption of the compressor. In order to make it easier to distinguish, the data for Test A and Test B are marked with “a” and “b”, respectively.
to the ventilator frequency. Additionally, the power consumption of compressor increased with the temperature of cooling water, and the differences of maximum and minimum power consumption of the compressor were 0.21 kW and 0.25 kW in Tests A and B, respectively.

Figure 8 illustrates the relative errors of the energy equilibrium in the two tests. As the experimental data were collected under practical conditions, the energy consumption at the loaded side was difficult to control and the errors and uncertainties of the measuring devices were inevitable, resulting in large relative errors of <10% in most cases. Large relative errors were observed after each adjustment of circulating pump frequency.

Effects on COP and SEER

Figure 9(a) and (b) shows the changes in the COP of the heat pump unit and the SEER of the HTHPs against the frequencies of the circulating pump and ventilator, respectively. If the frequency of the circulating pump is above 20 Hz, the COP fluctuates at around 2.9, and a slight drop in COP was observed at a circulating frequency of 15 Hz. The COP exhibited a significant reduction at a circulating pump frequency of 10 Hz. Meanwhile, the SEER tends to increase as the frequency of circulating pump is reduced, although a substantial reduction in the SEER occurs at a circulating pump frequency of 10 Hz. In addition, it was revealed that with reductions in the ventilator frequency, the COP exhibited an initial increase followed by subsequent decreases, while the SEER showed continual increases until the ventilator frequency dropped to 10 Hz, at which a slight decrease was observed.

In order to quantify the effects of the frequencies of the circulating pump and ventilator on the COP and SEER, and to minimize any interferences on the system’s responses due to frequency changes, the average values of COP and SEER over a 5–15 min interval after frequency tuning were calculated (Table 4). As detailed in the results, the average COP of
the heat pump unit reached 2.94, while the average SEER of the heat pump unit had a maximum value of 2.15, which were achieved under separate conditions.

Effects on the operation of the heat pump unit

The variations in refrigerant temperatures at different locations against the frequencies of the circulating pump and ventilator are shown in Figure 10(a) and (b), respectively. The temperature of the refrigerant at the compressor outlet was found to increase with reductions in both frequencies, while the temperature of the refrigerant at the compressor inlet and the evaporator inlet and outlet decreased. If the frequency of either the circulating pump or ventilator dropped to 15 Hz and 10 Hz, the temperatures of the refrigerant at the evaporator inlet and outlet exhibited a reduction due to the relatively low temperature of the anti-freeze solution, resulting in a reduced evaporation pressure. Furthermore, the condensing pressure can be considered to be constant as the cooling water temperature remained constant. As a result, both the compression ratio of the heat pump unit and refrigerant
temperature at its outlet increased at a growing rate and approached the critical temperature of heat pump unit. This indicates that despite the high SEER achieved, excessively low operation frequencies of the circulating pump and ventilator are not favorable for heat pump unit operation.

Conclusions

The findings of this investigation demonstrated that reducing the frequency of the circulating pump for the anti-freeze solution leads to a reduction in the temperature of the outlet evaporator of the anti-freeze solution and a greater temperature difference between anti-freeze solution at the inlet and outlet of the heating tower; meanwhile, excessively high or low anti-freeze flow rates result in reduced heat absorption of the closed-type heating tower. Furthermore, decreasing the ventilator frequency causes the temperatures of the anti-freeze solution at the inlet and outlet of the heating tower to be reduced and minimizes the temperature difference, hindering the heat absorption of the heating tower. If the air ventilator or circulating pump frequency drops to 15 Hz and 10 Hz, the evaporation temperature of the heat pump unit decreases, resulting in an excessive exhaust temperature, which is not favorable for the safe operation of the heat pump unit.

Additionally, the maximum heat generated by the system exceeded 13 kW, and the maximum average values of the COP and SEER reached 2.94 and 2.15, respectively, albeit under different operating conditions. In the view of achieving higher SEER, the optimal frequency of circulating pump and ventilator frequency was all 20 Hz. If the heating quantity needs to be further increased, the frequency of the ventilator should be increased to strengthen the heating capacity of the system.

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ORCID iD
Feng Rong https://orcid.org/0000-0002-5233-7663

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