Numerical study on heat and mass transfer characteristics of the counter-flow heat-source tower (CFHST)

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ABSTRACT

The heat-source tower heat pump (HSTHP), as a novel energy-saving unit, extracts low-grade thermal energy from air that can be a promising alternative of boiler in Yangtze River basin, China. A numerical model for analysis of the heat and mass transfer characteristics of a counter-flow heat source tower (CFHST) operating in winter is developed and validated by using experimental results. In this proposed numerical model, the changeable Lewis number is considered, and the effects of various operating, environmental including inlet air dry bulb temperature, inlet air humidity ratio, inlet air flow rate, inlet solution temperature and inlet solution flow rate on the thermal behavior of the heat source tower are studied. Furthermore, the proposed model will also be used to analyze the impact of the porosity and spacing of packing on the heat exchange in the CFHST. Finally, the moisture transfer characteristics inside CFHST under various environmental conditions are also studied. This work can provide a theoretical foundation for performance evaluation and practical design of CFHST.

24 Keywords: counter-flow heat source tower; numerical study; heat transfer; mass transfer

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Nomenclature			
A	area (m^2) / correction coefficient	T	temperature (°C)
B	width of the packing material (m) / correction coefficient	T_a	air temperature (°C)
C	correction coefficient	$T_{a,i}$	inlet air temperature (°C)
c_{da}	specific heat of dry air ($\text{J/kg} \cdot \text{K}$)	$T_{a,o}$	outlet air temperature (°C)
c_a	specific heat of moist air ($\text{J/kg} \cdot \text{K}$)	T_s	solution temperature (°C)
c_s	specific heat of solution ($\text{J/kg} \cdot \text{K}$)	$T_{s,i}$	inlet solution temperature (°C)
c_v	specific heat of water vapor ($\text{J/kg} \cdot \text{K}$)	$T_{s,o}$	outlet solution temperature (°C)
D	spacing of the packing material (m)	V_A	molar volume of water vapor ($\text{cm}^3/\text{g} \cdot \text{mol}$)
D_v	water vapor diffusion coefficient	V_B	molar volume of air ($\text{cm}^3/\text{g} \cdot \text{mol}$)
h_c	heat transfer coefficient ($\text{W/m}^2 \cdot \text{K}$)	X	horizontal direction
h_m	mass transfer coefficient ($\text{W/m}^2 \cdot \text{K}$)	Z	vertical direction
k_a	thermal conductivity of air ($\text{W/m} \cdot \text{°C}$)	Le	Lewis number
L	length of the packing material (m)	<i>Greek symbols</i>	
m	molar concentration of the solution (mol/kg)	β	porosity of the packing material
M_A	molar mass of water vapor (g/mol)	ω_a	air humidity ratio in the bulk air flow (g/kg)
M_B	molar mass of air (g/mol)	ω_s	air humidity ratio at the air-solution interface (g/kg)
m_{da}	mass flow rate of dry air (kg/s)	ρ_a	density of moist air (kg/m^3)
m_s	mass flow rate of water (kg/s)	ξ	ratio of latent heat transfer to the total heat transfer
$m_{s,i}$	mass flow rate of inlet solution (kg/s)	ν_a	kinematic viscosity coefficient of moist air
$m_{s,o}$	mass flow rate of outlet solution (kg/s)	<i>Subscripts</i>	
N	number of the divided sections.	a	moist air
P_0	standard atmospheric pressure (101.325 kPa)	da	dry air
P_s	saturation vapor pressure over solution (kPa)	i	inlet
Re	Reynolds number	o	outlet
r_o	latent heat of condensation of water at 0 °C (J/kg)	s	solution
Pr	Prandtl number	<i>Superscripts</i>	
S_c	Schmidt number	i	the i^{th} control unit

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1. Introduction

The Yangtze River, the longest river in China and the third longest river in the world, has sustained over one third of China's population living in its basin. The climate in Yangtze River basin is characterized by wet and hot summer, and wet and cold winter. The annual average temperature in the southern and northern portions of the mid-lower reaches of Yangtze River is 19 and 15 °C, respectively, and the winter average temperature and relative humidity is around 0°C and 80% [1,2]. Contrary to north China, the central heating system is not available in such areas. Generally, cooling and heating requirements are satisfied by combination of using the water-cooled chiller units and the gas-fired or electric boilers. This is due to that the water-cooled chillers do not have the function of heating in winter, while the boilers have lower energy efficiency for heating compared with the heat pump system [3]. The heat-source tower heat pump (HSTHP), as a novel and promising energy-saving unit that extracts low-grade thermal energy from air, can be a promising alternative compared with boiler in Yangtze River basin. This is because in this region the outdoor temperature is comparatively high all the year around, therefore, the HSTHP can satisfy the heating requirement quite well.

As shown in Fig.2, the main components of water-cooled chiller and HSTHP are the same, and the difference between them is the role of the cooling tower. The cooling tower of water chiller is to dissipate the heat withdrawn from the building to the atmosphere in summer. This process is accomplished by a combination of sensible heat transfer and evaporation of a small proportion of water [4]. For HSTHP, the water is replaced by antifreeze fluid as circulation medium in winter, which can prevent the cooling tower from freezing risk. The tower has hence the function of releasing heat to the atmosphere in summer and absorbing heat from the ambient air in winter without additional investment cost operational expense.

The heat and mass transfer mechanism and thermal performance of a cooling tower have been well studied when it is normally operated in summer mode [5-8]. However, it is believed that the thermal behavior of heat source tower would be different in three aspects of water-side heat transfer resistance, latent heat exchange and chilled water flow, as it is operated to extract heat from the ambient air for hot water supply [9, 10]. Therefore, it should be well understood. Liang et al. [11] discussed the performance of a cross-flow HSTHP unit by changing the circulating solution temperature differences. They found that the COP of heat pump declined from 3.02 to 2.72, and the difference between air temperature and evaporating temperature increased from 7.64°C to 11.96°C, when the environment

temperature was -1.2°C , and the temperature difference of solution between the heat-source tower inlet and outlet changed from 1.5°C to 3°C . Wen et al. [12] performed an experimental study on the performance of a cross-flow heat-source tower, and examined the impact of air flow rate, inlet solution temperature and solution flow rate on the tower performance. The experimental results showed that when the inlet solution temperature is increased from -2°C to 4°C , the latent heat percentage reduced from 27% to zero. With the increase of the air flow rate and inlet air temperature, the latent heat percentage decreased and the total heat transfer rate increased. As the flow rate of solution increased from $2.9\text{ L}\cdot\text{min}^{-1}$ to $6.4\text{ L}\cdot\text{min}^{-1}$, the latent heat percentage was raised from 15% to 28% and the total heat transfer rate was also increased nearly 50%. Meanwhile, Wen et al. [13] experimentally determined the optimized liquid to gas ratio for the HSTHP under different outdoor air parameters by using water and solution (glycol solution) as the circulation medium. They claimed that the optimized ratio always remains at 0.25 to 0.4 when the outlet air temperature is regulated from 5.5°C to 18.5°C with the medium of water and glycol solution. Then, Wen's team [14] analyzed the effect of various parameters on the tower performance. It was concluded that both air flow rate and liquid flow rate have significant effect on heat transfer coefficient, but the influences of inlet air temperature and inlet liquid temperature on heat transfer coefficient are small.

Tan and Deng [15] identified the differences in heat and mass transfer process taking place in a reversibly used water cooling tower (RUWCT), a spray room and a standard water cooling tower, and firstly proposed a mathematical model of a counter-flow RUWCT where it is heated by warmer ambient air. This research work provided a fundamental basis for studying heat and mass transfer characteristics in a counter-flow RUWCT. However, considering that the work was based on field experiments, further experimental work in a controlled environmental laboratory is highly required.

Tan and Deng [16] presented a numerical analysis by which the air and water states at any horizontal sections along the tower height may be determined. The numerical analysis was based on the heat and mass exchange within the tower, which was also partially validated by the on-site experimental data. However, they did not consider the effect of Lewis number, and the results are only applicable when

Lewis number is equal to one. Zhang et al. [17] proposed a heat and mass transfer model taking account of the effect of Lewis number to study the thermal behavior of the counter flow RUWCT under various operating and environmental conditions. Liang et al. [18] pointed out that when the solution is used as the circulating medium in heat source tower, the solution would be diluted due to moisture entering, and the freezing phenomenon may occur in solution that deteriorates the performance of the heat source tower. However, the moisture transfer characteristics of the heat source tower in winter are little considered in those published studies.

From the present research status, the studies are mainly concentrated on the heat and mass transfer characteristics of cross-flow heat source tower, while few efforts are paid on investigation of the thermal behavior of counter-flow heat source tower (CFHST) operating in winter. In fact, the air flow direction in counter-flow heat source tower is different from that in cross-flow heat source tower, and the operating and climatic parameters may have different effects on the heat and mass transfer characteristic of CFHST. What's more, the influence is non-linear, and some of the parameters are interactional and intercoupling. Moreover, to reduce the risk of solution freezing due to the solution dilution, the moisture transfer characteristics of the heat source tower needs to be well understood. Therefore, it is highly required a comprehensive study on the heat and mass transfer characteristic of CFHST.

The aim of this paper is to develop a numerical model for analysis of the heat and mass transfer characteristics of a CFHST operating in winter. In this model, the changeable Lewis number is considered, and the effects of various operating and environmental parameters including inlet air dry bulb temperature, inlet air humidity ratio, inlet air flow rate, inlet solution temperature and inlet solution flow rate on the thermal behavior of the heat source tower are studied. Furthermore, the

proposed model will also be used to analyze the impact of the porosity and spacing of packing on the heat and mass exchange in the CFHST. Finally, the moisture transfer characteristics inside HSTHP under various environmental conditions are also studied. This work can provide a theoretical foundation for performance evaluation and practical design of CFHST.

2. Mathematical modeling of CFHST

2.1 Description of CFHST

As shown in Fig.1, in a CFHST, the hot solution enters at the top of the tower as a spray and flows downward through the tower when it is operated in summer, while the ambient air enters at the bottom of tower and flows upward. The packing material inside tower provides a large surface area for heat and mass transfer from air to solution droplets. The chilled solution is sprayed into the upward flowing air stream by using a number of nozzles. The nozzles are arranged in such a way that the water is uniformly distributed over the packing material. Due to heat and mass transfer, the temperature of solution decreases while the air enthalpy increases because the air is heated and humidified by the solution as it flows up. However, when the cooling tower is reversibly used as part of a heat pump system in winter, the cold solution enters the tower and extracts the free heat from the upward flowing air. The air is cooled and dehumidified when the solution temperature is increased.

Fig.1. Schematic diagram of a typical CFHST.

As shown in Fig.2, in summer, the HSTHP system is operated in the cooling mode. The cooled water in the cooling tower is pumped into the condenser to absorb heat, and then flows back to the cooling tower in which the absorbed heat is taken away by air by evaporative cooling. In the evaporator, heat is released to meet the cooling needs for end users. In winter, the system is operated in the heating mode, and the water in the cooling tower could be replaced by antifreeze fluid. The heated liquid in the

heat-source tower is pumped into the evaporator to release heat, and then the cooled liquid flows back into the heat-source tower. In the condenser, heat is extracted to meet the heating needs for users.

Fig. 2. Schematic diagram of heat-source tower heat pump system.

2.2 Governing equations

The control volume of a counter-flow cooling tower is shown in Fig. 3. The major assumptions and simplifications used to derive the governing equations are given as below:

1. The heat and mass transfer occurs along the vertical direction;
2. Both inlet air and solution flow rates are steady and uniform;
3. The density and the specific heat of air and solution are constant.
4. The temperature throughout the solution stream at each cross section is uniform;
5. Cross-sectional area of the tower is constant.
6. The heat and mass transfer through the tower walls to the environment are negligible;
7. Thermal dispersion effects in both air and solution domains are negligible;
8. The packing is uniformly wet and in thermal equilibrium with the aqueous phase.
9. The interface temperature between solution film and air is assumed to be equal to the solution film temperature.

According to Fig. 2, the driving force for mass transfer is the pressure difference between vapor partial pressure of air and vapor partial pressure at the solution surface, thus dm_w can be written as:

$$dm_w = h_m (\omega_a - \omega_s) dA \quad (1)$$

where ω_a is the air humidity ratio in the bulk air flow, g/ kg. ω_s is the air humidity ratio at the air-solution interface, g/ kg, h_m is the mass transfer coefficient.

According to the law of conservation of mass, the air side water-vapor mass balance can be written as

[17].

$$-m_{da}d\omega_a = h_m(\omega_a - \omega_s)dA \quad (2)$$

According to the law of conservation of energy, the energy balance between air and solution is given by the following equation:

$$\begin{aligned} [m_{da}c_{da}T_{a,i} + m_{da}\omega_{a,i}(c_vT_{a,i} + r_0)] - [m_{da}c_{da}T_{a,o} + m_{da}\omega_{a,o}(c_vT_{a,o} + r_0)] = m_{da}d\omega_a(c_vT_{a,i} + \\ r_0) + h_c(T_a - T_s)dA \end{aligned} \quad (3)$$

where c_{da} is the specific heat of dry air, J/kg·°C, c_v is specific heat of water vapor, J/kg·°C, r_0 is the latent heat of condensation of water at 0°C, J/kg, h_c is heat transfer coefficient, W/ m² · °C.

The Eq. (3) can be rewritten as below:

$$(m_{da}c_{da}T_{a,i} + m_{da}\omega_{a,i}c_vT_{a,i}) - (m_{da}c_{da}T_{a,o} + m_{da}\omega_{a,o}c_vT_{a,o}) = m_{da}c_vT_{a,i}d\omega_a + h_c(T_a - T_s)dA \quad (4)$$

Similarly, the energy balance for the solution side is given in terms of the increase of the enthalpy of the solution.

$$m_{s,o}c_sT_{s,o} - m_{s,i}c_sT_{s,i} = dm_w(c_vT_{s,i} + r_0) + h_c(T_a - T_s)dA \quad (5)$$

$$m_{s,o}c_sT_{s,o} - m_{s,i}c_sT_{s,i} = (m_{s,o} - m_{s,i})(c_vT_{s,i} + r_0) + h_c(T_a - T_s)dA \quad (6)$$

The Eq. (1), (2), (4) and (6) describe the basic heat and mass transfer mechanism between solution and air in a CFHST.

Fig. 3 Schematic diagram of heat and mass transfer in a CFHST.

2.3 Numerical method

As it is reasonable to assume that no heat and mass transfer occurs in horizontal direction and the operating parameters of the CFHST vary only in the vertical direction. As shown in Fig. 4, the whole heat and mass transfer process is divided into finite sections, and the area of each section is dA. Section

i is filled with solution which enters this section at the temperature of T_s^i and exits at temperature of T_s^{i+1} . Similarly, in section i+1, an iterative computation is developed to calculate the output parameters according to the input parameters. Eventually, outlet solution temperature, outlet solution mass flow rate, outlet air temperature and humidity ratio of air within the CFHST can be obtained. The governing equations could be written in the form of differential equations and are solved by a finite difference method [19, 20]. If the step length is short enough along the vertical direction, the method is reliable. The CFHST divided into 10 to 20 sections would suffice for the required accuracy of the calculation results [21-23]. In this study, the packing material of CFHST is equally divided into 10 sections along vertical direction (N=10).

Fig. 4 The divided segments of padding along vertical direction.

The area of the air-solution interface dA can be calculated by the following equation.

$$dA = A^i = \frac{\beta LBH}{DN} \quad (7)$$

where β is the porosity of the packing material, L is the length of the packing material, m , B is the width of the packing material, m , H is the height of the packing material, m , D is the spacing of the packing material, m , N is the number of the divided sections.

The Eq. (1), (2), (4) and (6) can be rewritten in form of differential equations.

$$m_s^{i+1} - m_s^i = h_m^i (\omega_a^i - \omega_s^i) A^i \quad (8)$$

$$m_{da} (\omega_a^{i+1} - \omega_a^i) = h_m^i (\omega_a^i - \omega_s^i) A^i \quad (9)$$

$$(m_{da} c_{da} T_a^{i+1} + m_{da} c_v \omega_a^{i+1} T_a^{i+1}) - (m_{da} c_{da} T_a^i + m_{da} c_v \omega_a^i T_a^i) = m_{da} c_v T_a^i (\omega_a^{i+1} - \omega_a^i) + h_c^i (T_a^i - T_s^i) A^i \quad (10)$$

$$c_s m_s^{i+1} T_s^{i+1} - c_s m_s^i T_s^i = (m_s^{i+1} - m_s^i) (c_v m_s^i + r_0) + h_c^i (T_a^i - T_s^i) A^i \quad (11)$$

The above Eq. (8) - (11) can be reformulated in form of explicit differential equations.

$$m_s^{i+1} = m_s^i + h_m^i (\omega_a^i - \omega_s^i) A^i \quad (12)$$

$$\omega_a^{i+1} = \omega_a^i + \frac{h_m^i}{m_{da}} (\omega_a^i - \omega_s^i) A^i \quad (13)$$

$$T_a^{i+1} = \frac{(m_{da} c_{da} + m_{da} c_v \omega_a^i) T_a^i + m_{da} c_v T_a^i (\omega_a^{i+1} - \omega_a^i) + h_c^i (T_a^i - T_w^i) A^i}{m_{da} (c_{da} + c_v \omega_a^{i+1})} \quad (14)$$

$$T_s^{i+1} = \frac{c_s m_s^i T_s^i + (m_s^{i+1} - m_s^i) (c_v m_s^i + r_0) + h_c^i (T_a^i - T_s^i) A^i}{c_s m_s^{i+1}} \quad (15)$$

2.4 Condition parameters of air and solution

The humidity ratio of saturated moist air at solution surface

Calcium chloride (CaCl_2) aqueous solution is selected as the circulating medium due to its stability, high performance, and relative lower freezing point [24]. The reference [20] pointed out that the saturated vapor pressure at CaCl_2 aqueous solution surface is lower than that for pure water, and the pressure is a function of the temperature and concentration of solution. The saturated vapor pressure at solution surface can be obtained by the following equation.

$$\log P_s = A(m) + \frac{B(m)}{T_s + 273.15 + C} + 3 \quad (16)$$

$$A(m) = A_0 + A_1 m + A_2 m^2 + A_3 m^3 \quad (17)$$

$$B(m) = B_0 + B_1 m + B_2 m^2 + B_3 m^3 \quad (18)$$

where P_s is the saturated vapor pressure at CaCl_2 aqueous solution surface, kPa, m is the molar concentration of the solution, mol/kg, T_s is the solution temperature, $^\circ\text{C}$.

The values for correction coefficient A and B are given in Table 1.

The molar concentration of the solution is 1.3243 mol/kg when the density of the solution is 1130 kg/m³, and the temperature varies within -10°C to 10°C . Therefore, the saturated vapor pressure is changed to:

$$\log P_s = 10.055 - \frac{1668.374}{T_s^i + 228.043} \quad (19)$$

The humidity ratio of saturated moist air at the solution surface is given as below.

$$\omega_s = 0.622 \frac{P_s}{P_0 - P_s} \quad (20)$$

where P_0 is the standard atmospheric pressure, 101.325 kPa.

Combining Eq. (19) and (20), the humidity ratio of saturated moist air at the solution surface for each section is obtained.

$$\omega_s^i = \frac{0.622 \times 10^{\frac{10.055 - \frac{1668.374}{T_s^i + 228.043}}}{101325 - 10^{\frac{10.055 - \frac{1668.374}{T_s^i + 228.043}}}} \quad (21)$$

Heat transfer coefficient

According to literature [25], the heat transfer coefficient h_c can be calculated as follows.

$$h_c = k_a (2.0 + 0.6 Re^{0.5} Pr^{0.33}) / D \quad (22)$$

where k_a is the thermal conductivity of air, the value of 0.0244 W/(m • °C) is used in this study.

Re and Pr can be obtained by the following equations.

$$Re = \frac{D \times 10^5}{L \times B} \left(\frac{m_{da} + m_{da} \omega_s^i}{1.2} + \frac{m_s^i}{1130} \right) \quad (23)$$

$$Pr = \frac{\nu_a c_{pa} \rho_a}{k_a} \quad (24)$$

Combined with Eq. (22) and (23), Eq. (22) becomes:

$$h_c^i = \frac{0.0488}{D} + \frac{3.676}{(L \times B \times D)^{0.5}} \left(\frac{m_{da} + m_{da} \omega_s^i}{1.2} + \frac{m_s^i}{1130} \right)^{0.5} \quad (25)$$

Mass transfer coefficient

The mass transfer coefficient h_m can be calculated as follows.

$$h_m = \frac{h_c}{\rho_a c_a} \left(\frac{Pr}{Sc} \right)^{2/3} \quad (26)$$

where Sc is Schmidt number and is used to characterize fluid flows in which there are simultaneous momentum and mass diffusion convection processes.

$$Sc = \nu_a / D_v \quad (27)$$

D_v is water vapor diffusion coefficient, m²/s. According to literature [26], it can be obtained as follows

$$D_v = \frac{435.7 (T_a + 273.15)^{1.5} \left(\frac{1}{M_A} + \frac{1}{M_B} \right)^{1/2}}{P_0 (V_A^{1/3} + V_B^{1/3})^2} \times 10^{-4} \quad (28)$$

where M_A is the molar mass of water vapor, g/mol, V_A is the molar volume of water vapor, cm³/g·mol,

M_B is the molar mass of air, g/mol, V_B is the molar volume of air, $\text{cm}^3/\text{g} \cdot \text{mol}$. So Eq. (28) can be rewritten as below.

$$D_v = 3.881 \times 10^{-9} (T_a^i + 273.15)^{3/2} \quad (29)$$

Taking Eq. (29) into Eq. (27), Eq. (27) is then modified to

$$S_c = \frac{2576.656}{(T_a^i + 273.15)^{3/2}} \quad (30)$$

Combining Eq. (25), (26) and (30), Eq. (26) can be rewritten as follows

$$h_m^i = 2.754 \times 10^{-6} h_c^i (T_a^i + 273.15) \quad (31)$$

Based on the above-detailed analysis of the heat source tower, a simulation program is written in “MATLAB” software to predict the performance of the heat source tower. The computational flow diagram is shown in Fig. 5.

Fig.5. Flowchart of the simulation program.

3. Validation of the developed model

To validate the developed model, field experimental work has been carried out in a chiller plant for a hotel building in Guizhou province, a hot summer and cold winter region in China. Six heat source towers heat pumps were installed to supply hot water in winter to the hotel building (see Fig.6). The tested heat source towers were filled with packing material, and each piece of packing is made with dimensions of 5000 mm long $\times$ 3300 mm wide $\times$ 2900 mm high and 98 % porosity. The spacing between the packing is set to 30 mm. Calcium chloride (CaCl_2) aqueous solution with density of 1130 kg/m^3 was used as the circulation medium due to its stability, high performance, and relative lower freezing point.

The schematic diagram of the experimental site is shown in Fig. 6. The field experiments were carried out from 11:50 to 18:30 in a period of 5 days in winter. During the period of field measurements, the

CaCl₂ aqueous solution mass flow rate $m_{w,i}$, inlet CaCl₂ aqueous solution temperature $T_{w,i}$, outlet CaCl₂ aqueous solution temperature $T_{w,o}$, inlet air dry-bulb temperature $T_{a,i}$, outlet air dry-bulb temperature $T_{a,o}$, humidity ratio of inlet moist air $\omega_{a,i}$, humidity ratio of outlet moist air $\omega_{a,o}$ and air mass flow rate $m_{a,i}$ were measured at an interval of 10 min by the devices that listed in Table 2. To measure the air mass flow rate $m_{a,i}$, the air velocities at the inlet of CFHST were measured by three hot-wire anemometers 9535-A with the measurement range of 0~30m/s and accuracy of ± 3 % of reading. In order to avoid the influence of disturbances from the grilles installed on inlet which may induce the overestimating or underestimating the real air velocity, a duct with internal dimensions of 1500 mm deep $\times$ 5000 mm long $\times$ 1000 mm wide was made and the cross-sectional dimensions of the duct are equal to those of the inlet. As the inlet on each side of the CFHST has the same size, the air velocities were measured from one inlet. The duct was connected to the inlet and the velocities measured at three points are therefore considered to represent the average air velocity of the inlet. An average velocity is measured with 20 measurements per 20 s to represent the airflow velocity for each point. The positions of three points are shown in Fig.7.

With the obtained testing results, the outlet air dry-bulb temperature $T_{a,o}$, humidity ratio of outlet moist air $\omega_{a,o}$ and outlet CaCl₂ aqueous solution temperature $T_{w,o}$ can be calculated by using the proposed simulation program. The average values of the five input parameters from 5 days were applied to the simulation program, and the simulated output parameters were compared with the results obtained from experiments. Appendix A lists the calculated and testing values of outlet solution temperature, outlet air temperature and humidity ratio of outlet air.

Fig.6 Photos of the tested counter flow heat source tower.

Fig.7. Dimensions of constructed duct and positions of the measuring points: (a) lateral section; (b)

cross section (all units are in mm).

Figs. 8, 9 and 10 present the error of outlet solution temperature, outlet air dry-bulb temperature and humidity ratio of outlet air between those values obtained from simulation program, respectively. It is seen that the errors for three parameters are all within 10%. The results illustrate that the developed numerical program has the acceptable accuracy to predict the heat and mass transfer characteristics of the CFHST.

Fig.8. The simulated outlet solution temperature vs. the experimental values.

Fig.9. The simulated outlet air dry-bulb temperature vs. the experimental values.

Fig.10. The simulated humidity ratio of outlet air vs. the experimental values.

4. Results and discussion

4.1 Heat and mass transfer process within heat source tower

The proposed mathematical model can be used to calculate the solution and air states at any horizontal sections along the tower height within the CFHST.

In order to investigate the heat and mass transfer process within heat source tower, the following boundary conditions are applied: 1) The boundary conditions at the bottom of the tower ($Z = H$) are the inlet air temperature ($T_{a,i} = 4.5\text{ }^{\circ}\text{C}$), inlet humidity ratio ($\omega_{a,i} = 4.5\text{ g/kg}$) and the inlet air mass flow rate ($m_{a,i} = 40\text{kg/s}$). 2) The boundary conditions at the top of the tower ($Z = 0$) are the inlet solution temperature ($T_{w,i} = -3\text{ }^{\circ}\text{C}$), and the inlet solution mass flow rate ($m_{w,i} = 37.7\text{kg/s}$). The structural parameters of the packing material are the same to those mentioned previously. The heat and mass transfer characteristics within the tower are plotted in Fig. 11-13.

It is seen from Fig.11, the humidity ratio of the upward flowing air decreases until the air exits the top of packing. This is because when the low-temperature and low-freezing-point solution is uniformly

sprayed into the heat source tower, it extracts sensible heat and latent heat from the air at the same time due to the temperature and water vapor partial pressure difference between air-solution interface and air. The humidity ratio of the upward flowing air decreases hence with height of packing increasing. Contrarily, as shown in Fig.12, the mass flow rate of solution increases continuously as it downward flows to the bottom of packing because of moisture entering. It should be noticed that the diluted circulating solution by the absorbed water vapor may reduce the solution concentration and cause the problem of solution freezing.

Fig.11. Variation of air humidity ratio in the packing material.

Fig.12. Variation of solution mass flow rate in the packing material.

Fig.13 shows the air and solution temperature distributions in the packing material. Similarly, it is seen that the temperature of upward flowing air decreases with the height of packing until the top of packing, and the temperature of solution increases continuously as it flows down to the bottom of packing so that the sensible heat transfer process continues.

Fig.13. Air and solution temperature variations in the packing material.

4.2 Parametric study

Design and operating parameters may affect the heating performance of the heat source tower. In order to understand how various parameters influence the tower's performance for given design and operating conditions, a parametric study is performed by using the experimentally validated numerical model. The simulation conditions for the following parametric cases are given in Table 3.

4.2.1 Effect of inlet air dry-bulb temperature on heat transfer

Fig. 14 shows the effect of the inlet air dry-bulb temperature on heat transfer. It is seen that the sensible heat transfer increases quickly with the inlet air dry-bulb temperature. With an increase of 9 °C in the

inlet air dry-bulb temperature (from 1 °C to 10 °C), the sensible heat transfer increases from 90.3 kW to 291.3 kW. However, the latent heat transfer is slightly reduced. It is obvious that the effect of inlet air dry-bulb temperature on sensible heat transfer is more significant than that on latent heat transfer. ξ represents the ratio of latent heat transfer to the total heat transfer, which can be used to quantify the degree of variation of latent heat transfer in a heat source tower. As shown in Fig.14, ξ decreases from 0.28 to 0.095 with the inlet air dry-bulb temperature increasing, while the total heat transfer is increased. That is, a higher air temperature increases the heat transfer performance of the heat source tower but reduces the latent heat transfer and therefore reduces the absorbed water vapor by solution.

Fig.14. Effect of inlet air dry-bulb temperature on sensible and latent heat transfers.

4.2.2 Effect of air humidity ratio on heat transfer

The effect of air humidity ratio on the heat transfer is given in Fig.15. Converse to the effect of air dry-bulb temperature, the latent heat transfer increases from 9.1kW to 108.7 kW quickly when the air humidity ratio is increased from 3.0 g/kg to 5.0 g/kg. However, the sensible heat transfer is slightly reduced. That is, the impact of the air humidity ratio on sensible heat transfer is not obvious. The reason for this phenomenon is that with the increase of humidity ratio, the water vapour partial pressure of air side becomes higher, and therefore increases the transfer of water vapor into the solution. The ratio of latent heat transfer to the total heat transfer ξ increases from 0.04 to 0.35 with the increase of air humidity ratio, while the total heat transfer is increased from 215 kW to 306 kW. Though the total heat transfer is greatly increased, the increment of heat transfer is totally from the latent heat transfer. The analysis indicates that the air humidity is an important factor in diluting the solution concentration that air with high humidity ratio may result in serious freezing problem during winter. For this case, solution-regenerating measures are required for heat source tower.

Fig.15. Effect of air humidity ratio on sensible and latent heat transfers.

4.2.3 Effect of air flow rate on heat transfer

As shown in the Fig.16, sensible heat transfer is much higher than latent heat transfer, and they both increase with the increase of air flow rate. However, the increment rates of sensible and latent heat transfers are both slowing down. When air flow rate varies from 20 kg /s to 70 kg /s, sensible heat transfer increases from 126.4 kW to 277.8 kW, while latent heat transfer increases only from 22.2 kW to 38.6 kW. The reason is that the heat and mass transfer coefficients are increased when the air flow rate increases, but the increased air velocity shortens the heat and mass transfer time that reduces the increment rates of heat and mass transfer coefficients. ξ is decreased with the increase of air flow rate, varying from 0.135 to 0.122. This demonstrates that increasing air flow rate can increase the sensible heat transfer and reduce the ratio of latent heat transfer to the total heat transfer. However, the effect of air flow rate on the ratio of latent heat transfer to the total heat transfer is not remarkable and on the sensible heat transfer weakens when the air flow rate increases. Therefore, it is hard to reduce the absorbed water vapor greatly while increase the total heat transfer by continuously increasing air flow rate.

Fig.16. Effect of air flow rate on sensible and latent heat transfers.

4.2.4 Effect of inlet solution temperature on heat transfer

As presented in Fig. 17, the inlet solution temperature has a great effect on the sensible heat transfer and latent heat transfer. The higher the inlet solution temperature, the lower sensible heat transfer and latent heat transfer. The sensible heat transfer decreases from 325 kW to 100 kW, while the latent heat transfer varies from 75 kW to -25 kW. The variation of the latent heat transfer indicates that a transition occurs as the inlet solution temperature increases. It is seen that the solution extracts latent heat from

air until the solution temperature rises up to 0 °C, and there is no latent heat transferred from air to solution at temperature of 0 °C. Then, partial moisture in the solution evaporates into the air when the solution temperature increases above 0 °C, and the heat-source tower solution has the regeneration effect, while the total heat transfer and the value of ξ remain reducing with the increase of inlet solution temperature. It implies that though increasing the inlet solution temperature can reduce the absorbed moisture of the solution, the total heat transfer amount would be reduced at same time. Therefore, the inlet solution temperature should be as high as possible under the precondition of meeting the heating demand.

Fig.17. Effect of inlet solution temperature on sensible and latent heat transfers.

4.2.5 Effect of inlet solution mass flow rate on heat transfer

Fig.18 describes the effect of the inlet solution mass flow rate on sensible heat transfer and latent heat transfer. As shown, the sensible heat transfer and latent heat transfer increase with the increase of solution mass flow rate. However, the increment rates of sensible and latent heat transfers are both slowing down. When mass flow rate varies from 20 kg/s to 70 kg/s, the sensible heat transfer increases from 185.5 kW to 237.9 kW, while the latent heat transfer increases from 24.6 kW to 33.9 kW. It is also seen that the value of ξ keeps increasing with the increase of inlet solution temperature. Therefore, a higher solution mass flow rate leads to a higher sensible heat transfer and latent heat transfer, and leads to more moisture extracted by the solution. In order to maintain the solution concentration, the solution mass flow rate should be set at the lowest value under the precondition of meeting heating demand.

Fig.18. Effect of inlet solution mass flow rate on sensible and latent heat transfers.

4.2.6 Effects of porosity and packing spacing on heat transfer

The porosity and spacing of the packing material play an important role on performance of the heat

source tower. This is because the porosity and packing spacing directly affects the total heat and mass transfer area between the air and solution and therefore affects the heat transfer within the tower.

The effects of porosity and packing spacing on the sensible and latent heat transfer are described in Fig.19. It is clearly seen that for all studied cases, the sensible heat transfer and latent heat transfer are decreased with the increase of packing spacing. The sensible heat transfer is reduced by 44.6%, 46.1 % and 51%, for the porosity of 0.98, 0.78 and 0.58, respectively, while the latent heat transfer are reduced by 48.9 %, 49.1% and 49.5%, accordingly when packing spacing increases from 20 mm to 40 mm.

However, the decreases in the sensible and latent heat transfers vary from 10.7 % to 12.9 % and from 16.7 % to 32 %, respectively for the packing spacing varying from 20 mm to 40 mm, when the porosity is reduced from 0.98 to 0.58. The results indicate that the effect of the packing spacing on heat transfer is more significant than that for the porosity. This is due to the packing spacing not only affect the heat and mass transfer areas but also affect the heat and mass transfer coefficients.

Fig.19. Effects of porosity and packing spacing on heat transfer.

4.3 Study on characteristics of solution dilution

As described previously, when a heat-source tower runs in winter, the heat and mass transfer occurs between air and solution, where the solution extracts sensible heat and latent heat from the air, and the solution is diluted during the heat and mass transfer process due to moisture entering. The dilution of the solution is a hazard to the heat source tower as the freezing point of the diluted solution would be increased and hence the solution would be easier to be freezing. Therefore, in order to prevent the solution from freezing, it is necessary to understand the characteristics of solution dilution in heat source tower.

Fig.20 presents the variation of moisture transfer between air and solution with relative humidity of the

inlet air. These plots are generated based on the following data: $T_{a,i} = 6\text{ }^{\circ}\text{C}$, $m_{a,i} = 40\text{ kg/s}$, $T_{s,i} = -3\text{ }^{\circ}\text{C}$, $m_{s,i} = 37.7\text{ kg/s}$. It is seen that the moisture transfer amount varies with the relative humidity of air in a linear trend. When the relative humidity varies from 30% to 49 %, it is seen that the moisture transfers from solution to air, conversely, when the relative humidity continue to increase from 49 % to 100 %, the moisture is transferred from air to solution. Therefore, reducing the relative humidity is beneficial to the solution regeneration.

Fig.20. Effect of relative humidity of air on moisture transfer.

The variations of w_a and w_s along with packing height when relative humidity of the inlet air varying from 40 % to 60% are given in Fig.21-23, respectively. In the counter-flow heat-source tower, the air flows upwards to the top, and the solution flows downwards to the bottom. As can be seen from Fig. 21, for the relative humidity of 60 %, the humidity ratio of air, w_a steadily decreases to the top, while the humidity ratio at the interface between solution and air, w_s increases continuously to the bottom. Moreover, it is always smaller than that of air along the height of packing, meaning that the moisture is transferred from air to solution, and the moisture transfer amount from air to solution is 12.1g/s. As seen from Fig.22, for relative humidity of 50%, the humidity ratio of air, w_a increases from bottom to height of packing of 1.22 m, then w_a decreases from 1.22 m to the top of packing. It indicates that the moisture is released from air to solution below the height of 1.22 m from bottom, while air absorbs moisture from solution above the height of 1.22 m. The moisture transfer amount is 0.68 g/s, that is, the overall moisture is transferred from air to solution. For relative humidity of 40%, the humidity ratio at the interface between solution and air w_s is always higher than the humidity ratio of air, w_a along the height of packing. It means that the moisture is transferred from solution to air, and the moisture transfer amount is 11.1g/s.

Fig.21. The variations of w_a and w_s along with packing height (RH = 60 %).

Fig.22. The variations of w_a and w_s along with packing height (RH = 50 %).

Fig.23. The variations of w_a and w_s along with packing height (RH = 40 %).

5. Conclusions

The heat and mass transfer characteristic of a CFHST was studied by using an experimentally validated numerical model in this paper. The developed model is able to analyze and predict the air and solution temperature profiles, humidity ratio of air and the solution mass flow rate change along the height of the CFHST. The following conclusions can be drawn from the analysis:

(1) The higher the air dry-bulb temperature, the higher the sensible heat transfer, but the lower the latent heat transfer. The ratio of latent heat transfer to the total heat transfer decreases with the increase of inlet air dry-bulb temperature. That is, a higher air temperature increases the heat transfer performance of the heat source tower but reduces the latent heat transfer and therefore reduces the absorbed water vapor by solution.

(2) Higher air humidity ratio increases the latent heat transfer amount but slightly reduces the sensible heat transfer. Therefore, the ratio of latent heat transfer to the total heat transfer is increased. The analysis indicates that the air humidity is an important factor in diluting the solution concentration that air with high humidity ratio may result in serious freezing problem during winter.

(3) Both sensible heat transfer and latent heat transfer increase with the increase of air flow rate. However, the effect of air flow rate on the ratio of latent heat transfer to the total heat transfer is not remarkable. Therefore, it is hard to reduce the absorbed moisture greatly by continuously increasing air flow rate.

(4) The inlet solution temperature has a great effect on the sensible heat transfer and latent heat

transfer. The higher the inlet solution temperature, the lower sensible heat transfer and latent heat transfer. The value of ξ remain reducing with the increase of inlet solution temperature. Therefore, the inlet solution temperature should be as high as possible under the precondition of meeting the heating demand.

(5) The sensible heat transfer, latent heat transfer and the value of ξ increase with the solution mass flow rate increases. Therefore, a higher solution mass flow rate leads to a higher sensible heat transfer and latent heat transfer, and leads to more moisture extracted by the solution. In order to maintain the solution concentration, the solution mass flow rate should be set at the lowest value under the precondition of meeting heating demand.

(6) The sensible heat transfer and latent heat transfer are decreased with the increase of packing spacing and with the decrease of porosity. Moreover, the effect of the packing spacing on heat transfer is more significant than that for the porosity. Because the packing spacing not only affects the heat and mass transfer areas but also affects the heat and mass transfer coefficients.

(7) The moisture transfer amount varies with the relative humidity of air. For this study, when the relative humidity varies from 30% to 49 %, the moisture transfers from solution to air, conversely, when the relative humidity continue to increase from 49 % to 100 %, the moisture is transferred from air to solution. Typically, for relative humidity of 50%, though the overall moisture is transferred from air to solution. There is a transition occurs to the moisture transfer at the height of 1.22m.

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Appendix I The calculated and testing values of outlet solution temperature, outlet air temperature and humidity ratio of outlet air

Time	$T_{s,i}$ (°C)	$m_{s,i}$ (kg/s)	$T_{s,o}$ (°C)	$T_{a,i}$ (°C)	$W_{i,o}$ (kg/kg)	$T_{a,o}$ (°C)	$W_{a,o}$ (kg/kg)	$T_{s,o}'$ (°C)	$T_{a,o}'$ (°C)	$W_{a,o}'$ (kg/kg)
11:50	-3.762	57.654	-1.675	4	0.00349	-1.9	0.00267	-1.555	-1.88	0.00281
12:00	-4.344	57.306	-1.996	3.5	0.00341	-2.25	0.00261	-2.06	-2.42	0.00271
12:10	-5.13	57.574	-2.553	3.5	0.00336	-3	0.00255	-2.623	-3.02	0.00259
12:20	-5.277	57.281	-2.711	3.6	0.00333	-3.1	0.00246	-2.713	-3.11	0.00256
12:30	-5.001	57.917	-2.233	3.9	0.00353	-3	0.00258	-2.372	-2.82	0.00265
12:40	-4.665	57.448	-2.124	3.7	0.00344	-2.5	0.00254	-2.229	-2.62	0.00267
12:50	-4.344	57.274	-1.955	4.2	0.00346	-2.1	0.00267	-1.902	-2.27	0.00272
13:00	-4.012	57.345	-1.789	3.5	0.00338	-2.25	0.00276	-1.861	-2.18	0.00275
13:10	-3.555	56.861	-1.583	3.7	0.00339	-1.7	0.00278	-1.505	-1.8	0.00282
13:20	-3.321	57.664	-1.312	4.1	0.00343	-1.6	0.00275	-1.282	-1.54	0.00286
13:30	-3.287	57.432	-1.311	3.7	0.00335	-1.6	0.00276	-1.368	-1.61	0.00285
13:40	-3.215	57.34	-1.212	3.8	0.0034	-1.5	0.00273	-1.277	-1.53	0.00287
13:50	-3.189	57.362	-1.117	4.4	0.00348	-1.4	0.00278	-1.077	-1.36	0.0029
14:00	-3.742	57.386	-1.506	4.1	0.00348	-1.7	0.00264	-1.518	-1.84	0.00281
14:10	-3.612	57.409	-1.573	3.8	0.00339	-1.7	0.00263	-1.538	-1.82	0.00281
14:20	-4.525	57.432	-2.053	3.7	0.00335	-2.7	0.0026	-2.185	-2.53	0.00267
14:30	-5.501	57.455	-3.147	3.1	0.00324	-3.6	0.00254	-3.021	-3.4	0.00251
14:40	-5.121	57.478	-2.764	2.9	0.0033	-3.4	0.00268	-2.79	-3.16	0.00257
14:50	-4.814	57.501	-2.353	3.8	0.00338	-2.8	0.00289	-2.33	-2.71	0.00264
15:00	-4.397	57.524	-1.901	4	0.00344	-2.3	0.00289	-1.993	-2.36	0.00271
15:10	-3.905	57.547	-1.875	3.4	0.0033	-2.2	0.00278	-1.876	-2.14	0.00274
15:20	-3.907	57.443	-1.763	3.7	0.00345	-2.1	0.00282	-1.726	-2.06	0.00278
15:30	-3.734	57.34	-1.593	3.6	0.00344	-1.9	0.00266	-1.639	-1.95	0.0028
15:40	-3.703	57.362	-1.412	4.1	0.00347	-1.8	0.00289	-1.483	-1.81	0.00282
15:50	-3.97	57.386	-1.661	4	0.00349	-1.9	0.00271	-1.68	-2.03	0.00278
16:00	-4.069	57.409	-1.767	3.7	0.00344	-2.1	0.00272	-1.845	-2.18	0.00275
16:10	-3.988	57.432	-1.731	3.7	0.00341	-2.1	0.00262	-1.795	-2.12	0.00276
16:20	-3.948	57.455	-1.749	3.8	0.00343	-2.1	0.00266	-1.743	-2.07	0.00277
16:30	-3.979	57.478	-1.887	3.2	0.00342	-2.1	0.00269	-1.9	-2.23	0.00276
16:40	-3.893	57.501	-1.669	3.5	0.00344	-2	0.00264	-1.761	-2.09	0.00278
16:50	-3.908	57.524	-1.749	3.3	0.00339	-2.2	0.00261	-1.857	-2.16	0.00276
17:00	-3.942	57.547	-1.866	3.3	0.0034	-2.2	0.00288	-1.867	-2.18	0.00276
17:10	-3.872	57.443	-1.629	3.5	0.00342	-2	0.00292	-1.756	-2.08	0.00278
17:20	-3.928	57.57	-1.867	3.2	0.00346	-2.2	0.00266	-1.861	-2.19	0.00277
17:30	-3.911	57.593	-1.859	3.2	0.00342	-2.2	0.00261	-1.849	-2.17	0.00277

17:40	-3.845	57.617	-1.762	3.2	0.00341	-2	0.00256	-1.83	-2.13	0.00277
17:50	-3.63	57.64	-1.692	3.3	0.00343	-2	0.00269	-1.647	-1.94	0.00281
18:00	-3.569	57.663	-1.611	3.5	0.00347	-1.9	0.00271	-1.546	-1.85	0.00283
18:10	-3.692	57.686	-1.705	3.1	0.00343	-2	0.0029	-1.731	-2.03	0.0028
18:20	-3.786	57.709	-1.905	2.7	0.00342	-2.1	0.00298	-1.895	-2.2	0.00278
18:30	-3.156	57.732	-1.711	3.3	0.0035	-2	0.0031	-1.559	-1.88	0.00284

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Fig.1. Schematic diagram of a typical CFHST.

Fig. 2. Schematic diagram of heat-source tower heat pump system.

Fig. 3 Schematic diagram of heat and mass transfer in a CFHST.

Fig. 4 The divided segments of padding along vertical direction.

Fig.5. Flowchart of the simulation program.

Fig.6 Photos of the tested counter flow heat source tower.

Fig.9. The simulated outlet air dry-bulb temperature vs. the experimental values.

Fig.10. The simulated humidity ratio of outlet air vs. the experimental values.

Fig.11. Variation of air humidity ratio in the packing material.

Fig.12. Variation of solution mass flow rate in the packing material.

Fig.13. Air and solution temperature variations in the packing material.

Fig.14. Effect of inlet air dry-bulb temperature on sensible and latent heat transfers.

Fig.15. Effect of air humidity ratio on sensible and latent heat transfers.

Fig.16. Effect of air flow rate on sensible and latent heat transfers.

Fig.17. Effect of inlet solution temperature on sensible and latent heat transfers.

Fig.18. Effect of inlet solution mass flow rate on sensible and latent heat transfers.

Fig.19. Effects of porosity and packing spacing on heat transfer.

Fig.20. Effect of relative humidity of air on moisture transfer.

Fig.21. The variations of w_a and w_s along with packing height (RH = 60 %).

Fig.22. The variations of w_a and w_s along with packing height (RH = 50 %).

Fig.23. The variations of w_a and w_s along with packing height (RH = 40 %).

Table 1

The values for correction coefficient in Eq. (17) and (18)

Correction coefficient	A_0	$10^2 A_1$	$10^3 A_2$	B_0	B_1	B_2	$10B_3$	C
Value	7.0945	-2.8495	-2.7670	-1669.84	3.7632	-1.4530	-4.1724	-45.107

Table 2

Specifications of the different measuring devices

Device	Type	Measuring range	Accuracy
Humidity and temperature meter	HIOKI 3641-20	Temperature: -20.0~70.0 °C Relative humidity: 0.0%~100.0%	Temperature: ± 0.5 °C Relative humidity: $\pm 5\%$
Flowmeter	DCT1288	-12~12m/s	± 0.015 m/s
Hot-wire anemometer	9535-A	0~30m/s	$\pm 3\%$
Data acquisition instrument	Agilent 34970A	-100~760 °C	± 0.5 °C

Table 3

Simulation conditions for different parametric cases

Parameter	Inlet air dry-bulb temperature (°C)	Humidity ratio of inlet air (g/kg)	Inlet air mass flow rate (kg/s)	Inlet solution temperature (°C)	Inlet solution mass flow rate (kg/s)	Packing spacing (m)	Porosity of packing
Inlet air dry-bulb temperature	variable	3.5	40	-3	37.7	0.03	0.98
Humidity ratio of inlet air	6	variable	40	-3	37.7	0.03	0.98
Inlet air mass flow rate	6	3.5	variable	-3	37.7	0.03	0.98
Inlet solution temperature	6	3.5	40	variable	37.7	0.03	0.98
Packing spacing	6	3.5	40	-3	37.7	variable	0.98
Porosity of packing	6	3.5	40	-3	37.7	0.03	variable