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Study on Heat Transfer Enhancement of a Coil Zone for Closed Wet Cooling Towers Equipped with Longitudinal Fin Tubes

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In this paper, we numerically investigate the structural parameters and operating parameters of a closed wet cooling tower (CWCT) installed with longitudinal fin tubes using a CFD species transport model. A two-dimensional model of the CWCT is established, the Navier-Stokes equation and κ-ε model are employed to calculate the vapour-phase flow and Reynolds number of the coil zone. In particular, we explore the approaches of heat transfer enhancement and the resulting CWCTs installed with different types of tubes. The results show that the fin tube case-3 (8 fins on one tube) is the optimal one. The heat and mass transfer coefficients of fin tube case-3 are 1.25 ~ 1.7 times that of a plain tube case with an inlet air flow rate of 3.7 $kg \cdot m^{-2} \cdot s^{-1}$. The heat and mass transfer show a decreasing trend with the increase of the inlet air wet-bulb temperature. Remarkably, the heat transfer coefficient of the plain tube case decreases by about 20 %.

1. Introduction

Nowadays, the scarcity of water resources in industrial processing has become an essential issue in global problems (Gamba et al., 2014). Industrial water consumption accounts for more than 60 % of the total water consumption, and the proportion of recirculating cooling water in industrial water consumption has exceeded 80 % (Klemeš et al., 2012). As an important part in a circulating water system, cooling towers are common heat rejection devices used in the process industries as a means of evaporative cooling technology. Cooling towers can be basically classified into open wet cooling towers and closed wet cooling towers (CWCTs). In open wet cooling towers the direct contact between cooling water and inlet air inevitably leads to severe loss of evaporation. In CWCTs, there is no direct contact between the process water and air flows, and the mass and heat transfer primarily occurs only on the tube surface through convection between the spray water and the air. This indirect contacting mode ensures that the process water will keep balanced and clean. CWCTs have received great attention in both experimental study and industrial application under the current situation of the energy saving (Navid Rohani et al., 2016) and emission reduction (Tarighaleslami et al., 2015).

Tremendous efforts have been made to investigate the heat and mass transfer process in CWCTs, and the numerical simulation implemented by CFD is an effective method which can provide a realistic mechanism for the investigation of thermodynamic performance of CWCTs. Research relating to the performance of CWCTs by the CFD simulation and by experiment are in good agreement (Hasan et al., 2004). Riffat et al. (2000) explored the influences of inlet air velocity and spray water combined in the numerical simulation and experiment. Zhu et al. (2013) showed that the standard k-ε model gave the best estimation of CWCTs equipped with plain and oval tubes in terms of the Colburn factor and pressure drop via CFD. The heat transfer enhancement in CWCTs mainly focuses on optimizing operating conditions and some novel designed CWCTs with special tubes, e.g. oval tubes. The heat transfer enhancement of the CWCTs installed with longitudinal fin tubes is rarely reported.

The aim of this study is to numerically investigate the thermo-hydraulic characteristics of CWCTs equipped with plain and longitudinal fin tubes. The structural/operating parameters of CWCTs, including the fin height, fin number, inlet air flow rate, and inlet air wet-bulb temperature, are analysed in detail. The performance of

CWCTs, such as heat and mass transfer coefficients, temperature drop of process water, cooling efficiency, pressure drop, and friction factor are investigated systematically.

2. Analysis of heat and mass transfer process

2.1 Mathematical model

In this model, the structural parameters of plain and longitudinal fin tubes are shown in Table 1. The basic components of a typical CWCT are heat transfer tubes, spray water, pump, fan, and drift eliminators, as shown in Figure 1(a). The process water goes inside the coil, the spray water circulates in a closed loop driven by a pump and the two inlet air streams are injected from the tower bottom through the fan. This counter current contact of spray water and inlet air can indirectly cool the process water through exchanging of sensible and latent heat on the external side of the tubes. The corresponding elementary control volume is given in Figure 1(b). In this paper, the Fin a-b mm is used to define the fin tube parameters, a stands for the fin numbers, b mm is the height of fins.

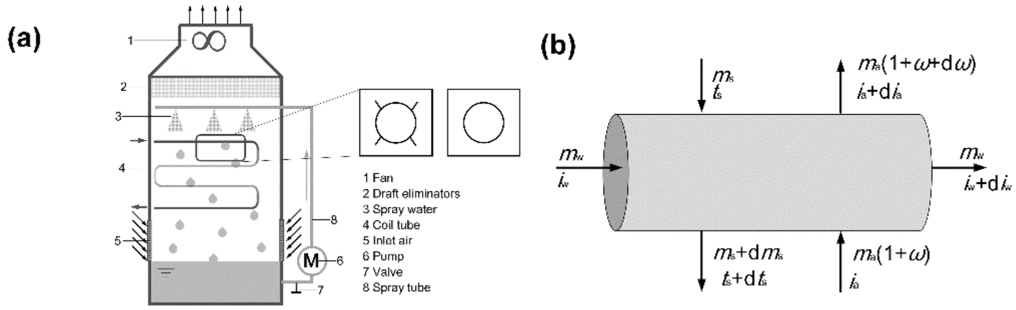

Figure 1: Schematic of CWCTs: (a) main components of CWCTs and (b) an elementary control volume of tube.

Table 1: The structure parameters of different tubes

Case	Tube type	*D*/mm	fins	*C*/°	*H*/mm	*n*
Plain tube case	Plain tubes	10	-	-	-	12
Fin tube case-1	Fin tubes	10	4	90	1	12
Fin tube case-2	Fin tubes	10	4	90	3	12
Fin tube case-3	Fin tubes	10	8	45	1	12

The mass balance principle applied to the control volume is:

$$m_a(1+\omega)+m_s+m_w=m_a(1+\omega+d\omega)+m_s+dm_s+m_w \tag{1}$$

where m is the mass flow rate.

The energy balance over the control volume is given as:

$$m_a i_{ma}+m_s c_{ps} t_s+m_w c_{pw} t_w=(m_s+dm_s)c_{ps}(t_s+dt_s)+m_a(i_{ma}+di_{ma})+m_w c_{pw}(t_w+dt_w) \tag{2}$$

Simplifying Eq. (3) by neglecting its second order terms gives us:

$$dt_s=-\frac{1}{m_s c_{ps}}(m_a di_{ma}+c_{ps} t_s dm_s+m_w dt_w) \tag{3}$$

where i_{ma} refers to the enthalpy of the air-vapour mixture per unit mass of dry air, which can be expressed as:

$$i_{ma}=c_{pa}t_a+\omega(i_{lh}+c_{pv}t_a) \tag{4}$$

The total energy transfer at the air-water interface consists of the sensible heat and latent heat:

$$dQ=dQ_\alpha+dQ_\beta \tag{5}$$

The corresponding energy transfer at the air-water interface results from the concentration difference of the vapour is:

$$dQ_{\beta} = i_{\mathrm{v}} dm_{\mathrm{s}} = i_{\mathrm{v}} \beta(\omega_{\mathrm{saw}} - \omega) dA \tag{6}$$

The total energy transfer at the air-water interface is:

$$dQ = \alpha(t_{\mathrm{s}} - t_{\mathrm{a}}) dA + i_{\mathrm{v}} \beta(\omega_{\mathrm{saw}} - \omega) dA \tag{7}$$

2.2 Computing settings

In order to study the air and water flow process model in two-dimensional CWCTs, the main assumptions are presented as follows: 1) the wettability of the tube surface remains unchanged along the tubes axis. 2) The temperature of tube wall is constant. 3) The air, spray water and vapour can be regarded as in-compressible fluid with the constant property.

The governing equations and the corresponding boundary conditions are solved by the finite volume method using the SIMPLE algorithm, and the convective terms in governing equations are discretized by using the QUICK scheme with second-order precision. The convergence criterions, namely the normalized residuals, are less than 10^{-4} for the flow equations and 10^{-6} for the energy equation. In this simulation, two different kinds of grids are generated to ensure that the results are independent of the grid, namely a fine grid with a quantity around 1,117,800, a coarse one with a quantity around 551,630, as shown in Figure 2. For all the two grid numbers considered, the simulation results show that the differences under various numbers of grids are less than 3 %. The coarse mesh with a grid number of 551,630 is acceptable.

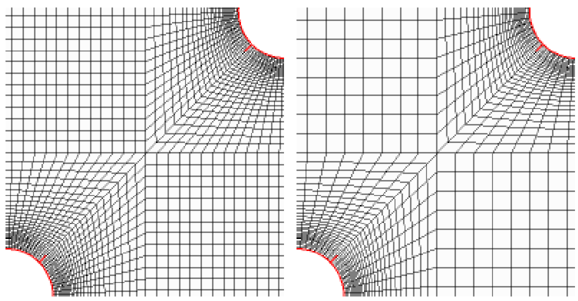

Figure 2: Mesh generation (a) fine and (b) coarse

3. Results and discussion

3.1 Model validation

The aim here was to accurately study the heat and mass transfer between the water film and the tube surface through numerical simulation. Figure 3 shows the results of CFD simulation by combining the species transport model under different air flow rates compared with the experimental results obtained from Heyns et al. (2010). It is observed that the difference between the simulation and experimental results is smaller than 10 %. The species transport model is employed to describe the heat and mass transfer of spray water and air in this work.

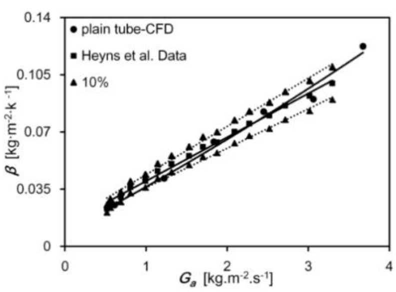

Figure 3: Model validation

3.2 Influences of inlet air flow rate

The influence of the inlet air flow rate (G_a) on the heat and mass transfer coefficients (α, β) and cooling efficiency (η) have been investigated. As depicted in Figure 4 (a - c), the thermal performances for the fin tube case is higher than that of the plain tube case which agrees with the experimental findings (Niitsu et al., 1969). This phenomenon can be further explained by the velocity distribution shown in Figure 5 in which the fins on the outer surface of the fin tubes can effectively intensify the turbulence of the airflow. As for 4 fin number cases, the α of the 1 mm fin height case is almost 8 % higher than that of the 3 mm fin height case at a G_a of 3.7 kg·m^{-2}·s^{-1}. Increasing the fin height would greatly disturb the movement of the falling film, which inevitably affects the heat transfer between the wall surface and bulk airflow. The process water temperature drop (Δt) of fin tube cases is greater than the plain tube case, as shown in Figure 6. The Δt of fin tube case-3 is 1.2 ºC greater than that of the plain tube case. This can be further demonstrated by the temperature field provided in Figure 7, where the outlet air temperature of the fin tube case is higher than the plain tube case. We can conclude that the increase in fin number on tube surface and the G_a has a positive effect on the heat and mass transfer in CWCTs. Note that, increasing the G_a would unavoidably increase the power consumption of the fan, which further increases the operating cost. This is related to the hydraulic characteristics of CWCTs which we will discuss as follows.

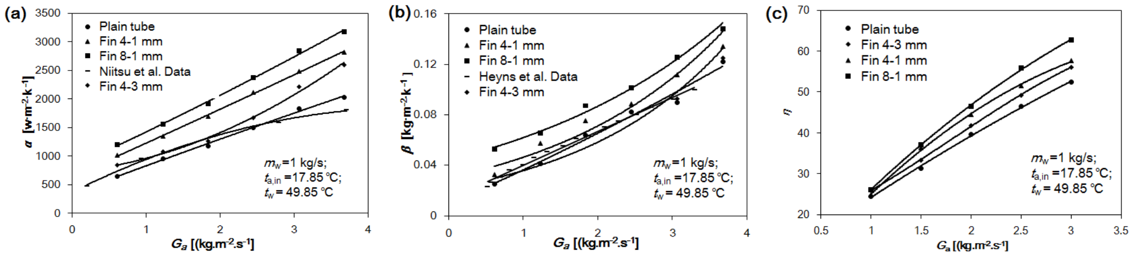

Figure 4: Influence of G_a on (a) heat transfer coefficient, (b) mass transfer coefficient, (c) cooling capacity

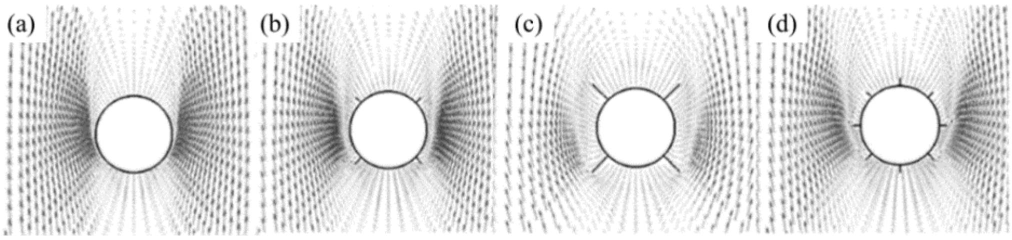

Figure 5: Local velocity field (a) plain tube case (b) fin tube case-1, (c) fin tube case-2, (d) fin tube case-3

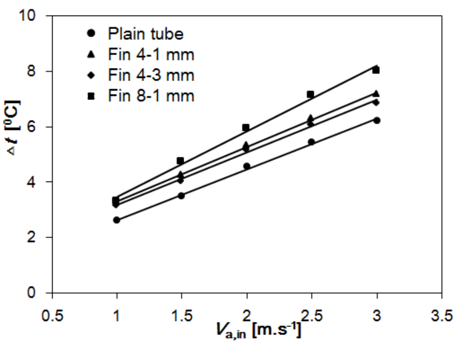

Figure 6: Temperature drop of process water under different inlet air velocity

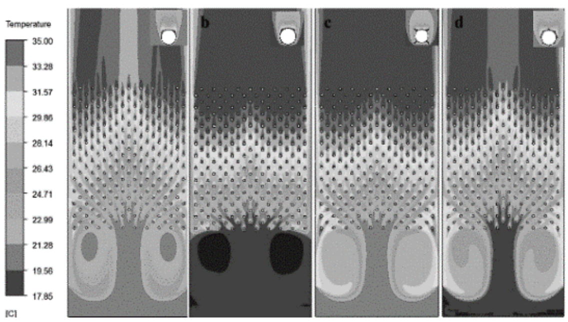

Figure 7: Temperature field of (a) Plain tube case, (b) Fin tube case-1, (c) Fin tube case-2, (d) Fin tube case-3

As shown in Figure 8(a) and 8(b), the pressure drop (ΔP) and related friction factor (f) under different G_a are investigated. Increasing G_a would greatly affect the speed and direction of air velocity, which further inevitably causes a multitude of kinetic energy losses as air flows around the tube bundle, namely, increasing the pressure drop. The fins on the tube surface not only hinder the movement of airflow but also increase the friction between the airflow and the wall surface. These fin geometries further cause greater inertial and viscous resistances in CWCTs. It can be found that when the G_a is 3.7 kg·m^{-2}·s^{-1}, the ΔP of the fin tube case-2 and fin tube case-3 are 1.37 times and 1.47 times higher than fin tube case-1. This is validated by Figure 9, which depicts the pressure field of the plain and fin tube cases as the inlet air velocity is 3 m·s^{-1}. Figure 8(b) shows the friction factors of the four cases with respect to the air flow rate. It can be seen that f decreases with the increase of G_a.

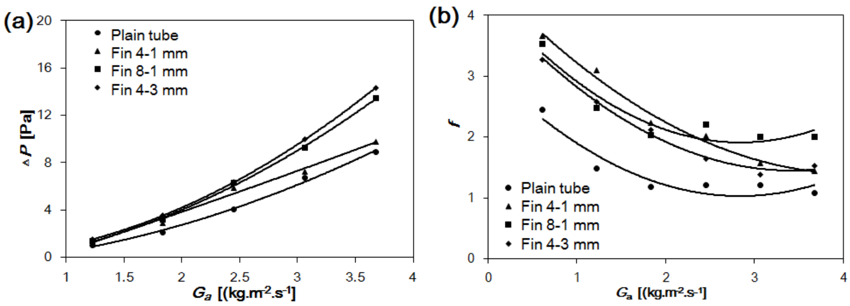

Figure 8: Influences of inlet air flow rate on (a) pressure drop and (b) friction factor

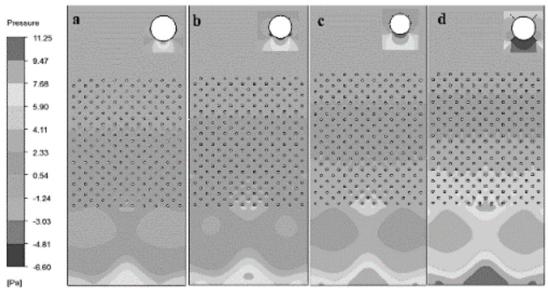

Figure 9: The pressure field of (a) plain tube case, (b) fin tube case-1, (c) fin tube case-2, (d) fin tube case-3

3.3 Influence of the inlet air wet-bulb temperature

The inlet air wet-bulb temperature ($t_{a,in}$) that can vary in different seasons and operating conditions is also a key parameter that influences the mass and heat transfer coefficients of CWCTs, as shown in Figure 10 (a and b). Increasing the $t_{a,in}$ can reduce the temperature difference between the falling film and airflow, further lowering the heat transfer driving force. The increase in $t_{a,in}$ would decrease the heat and mass transfer coefficients. The heat transfer coefficients of the plain tube case decreased by about 20 %, and the α of the fin tube case is about 1.2 times - 1.4 times that of the plain tube case. It is observed that the β of fin tube cases are 1 - 2 times that of the plain tube case, as plotted in Figure 10(b). In this regard, the high $t_{a,in}$ is relatively adverse to the heat exchange in CWCTs.

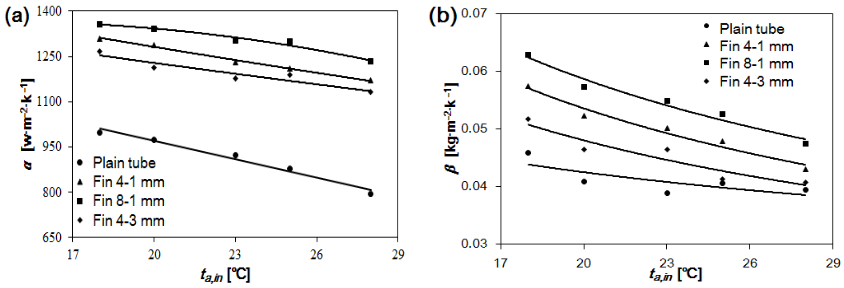

Figure 10: Influences of $t_{a,in}$ on (a) heat transfer coefficient and (b) mass transfer coefficient

4. Conclusions

In this study, the influences of operational and structural parameters on the thermo-hydraulic performances of CWCTs equipped with plain, longitudinal fin tubes have been numerically investigated, and the following conclusions are drawn: The CWCTs installed with longitudinal fin tubes have better thermal performance compared with the plain tube case. The heat and mass transfer coefficients of fin tube cases are more than 1.25 times than the plain tube case, at the air flow rate of 3.7 kg·m^{-2}·s^{-1}, the pressure drop of fin tube cases is about 1.5 times higher than that of the plain tube case. The increase in inlet air wet-bulb temperature would decrease the heat and mass transfer coefficients, and the heat transfer coefficients of the fin tube case is about 1.2 times - 1.4 times that of the plain tube case. It is observed that the mass transfer coefficients of fin tube cases are 1~2 times of plain tube case.

Acknowledgments

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Nomenclature

A	heat or mass transfer area (m^2)
C	the angle between the two adjacent fins (°)
c_p	specific heat capacity (J·kg^{-1}·K^{-1})
D	tube diameter (m)
f	friction factor
G	air flow rate (kg·m^{-2} s^{-1})
H	fin height (m)
i	enthalpy (J·kg^{-1})
m	mass flow rate (kg·s^{-1})
n	number of tube rows
P	pressure (pa)
Q	heat exchange capacity (W)
t	temperature (°C)
V	velocity (m·s^{-1})
Greek symbols	
α	heat transfer coefficient (W·m^{-2}·K^{-1})
β	mass transfer coefficient (kg·m^{-2}· s^{-1})
η	cooling efficiency (%)
ω	air humidity (kg·kg^{-1})
Subscripts	
a	dry air
in	inlet
ma	the air-vapour mixture per unit mass of dry air
out	outlet
s	spray water
saw	the saturated wet air under spray water temperature
v	vapour
w	process water

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