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Design calculation of porous ceramics tube type dew point indirect evaporative cooler

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Abstract: To improve the wall surface hydrophilicity of a tube type indirect evaporative cooler, a new method adopting porous ceramics is proposed. This method realizes the combination of porous ceramics and the evaporative cooling technique. The design calculation of the porous ceramics tube type dew point indirect evaporative cooler are carried out from such aspects as the volumes and status parameters of the primary and secondary air, the cooler structure, the heat transfer of the solid porous ceramic tubes and the resistance of the cooler. The calculation results show that the design is reasonable. Finally, based on the design calculation, the porous ceramics tube type dew point indirect evaporative cooler is successfully manufactured.

Key words: tube type indirect evaporative cooler; dew point; porous ceramics; design calculation

The research progress of the evaporative cooling technology at home and abroad is presented detailedly in literature[1–3], and it can be seen that evaporative cooling is an environmental-protected, efficient and economical cooling mode. However, an evaporative cooling air conditioning system is not perfect because there are still some problems. The most important issue is the poor hydrophilicity of the wall surface, which causes low transfer efficiency. In order to improve the hydrophilicity of the second channel wall surface of the indirect evaporative cooler, a hydrophilic aluminum foil coating is adopted; unfortunately, it is not satisfactory. Fiber paper is also used, but the poor efficiency and easy deterioration in a moist environment limit its application[4]. Wang[5] made a research in theory and experiments on wrapping functional water-absorbing materials outside the tubes of the elliptical tube type indirect evaporative cooler from the perspective of structures and materials. However, after a long operating time, a gravity action can result in the material detachment from the bottom of the heat exchange tubes, which seriously affects the normal operation of the system. Moreover, water cannot fully wet the surface of the heat exchange tubes, and, eventually, the heat transfer efficiency of the cooler is reduced. Therefore, it is necessary to find a more effective way to improve the heat transfer efficiency of the tube type indirect evaporative cooler.

Porous ceramics are potential materials used for evaporative cooling due to their advanced properties, such as high porosity, low volume density, large specific surface area, high capacity force, high thermal conductivity, waterproof, durability and capillarity, which can provide such good performances as good moisture retain, high heat and moisture exchange efficiency. When used in indirect evaporation systems, porous ceramics with a thin impermeable ceramic film can avoid the penetration of moisture from the tube outside to the tube inside[6–7]. Herein, we adopt porous ceramics to the tube type dew point indirect evaporative cooler. And the detailed design calculation of the porous ceramics tube type dew point indirect evaporative cooler is carried out.

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1 Basic Parameter Determination

1.1 Air volume

Considering practical factors, the primary air volume is set to 500 m^3/h. And the secondary air volume is set at 70% of the primary air volume experimentally, namely, 350 m^3/h.

1.2 Air status parameter

The summer air-conditioning outdoor dry-bulb temperature of Xi'an is 35.1 ℃ and the wet-bulb temperature is 25.8 ℃[8], which are the same as the primary air and secondary air parameters of the entrance. The cooler can cool the primary air to a dry-bulb temperature of 31.8 ℃, with a temperature drop of 3.3 ℃.

1.3 Heat quantity released by primary air

The average temperature t_{pm} of the primary air in the tube is selected as the qualitative temperature. And it can be calculated by[9]

$$t_{pm} = \frac{t_{p1} + t_{p2}}{2} \tag{1}$$

where t_{p1} and t_{p2} are the inlet and outlet primary air dry bulb temperatures, respectively.

The parameters of the primary air include the primary air density ρ_{pm}, the specific isobaric heat capacity of the primary air $C_{p,pm}$, the Prandtl number of the primary air Pr_{pm}, the thermal conductivity of the primary air λ_{pm}, and the kinematic viscosity of the primary air γ_{pm}. Then, the heat quantity released by the primary air can be calculated by

$$Q = m_p C_{p,pm}(t_{p1} - t_{p2}) \tag{2}$$

where m_p is the mass flow rate of the primary air.

1.4 Average temperature of secondary air

The secondary air and water film under the combined effect of temperature difference and water vapor concentration difference, namely the enthalpy difference, result in the heat and mass transfer. In the micro-process line, the heat and mass exchange process is the composite process involving the secondary air isoenthalpic adiabatic humidification process and the sensible heating process[10]. The heat exchange process between the primary air and the secondary air can achieve a heat balance; that is,

$$m_s(h_{s2} - h_{s1}) = m_p C_p(t_{p1} - t_{p2}) \tag{3}$$

where m_s is the mass flow rate of the secondary air; h_{s1} and h_{s2} are the enthalpy values of the inlet secondary air and the outlet secondary air, respectively.

Because air property changes with the secondary air temperature and the relative humidity variation is relatively small, the secondary air quality is set to 70% of the primary air quality; that is, $m_s/m_p = 0.7$. Based on the psychrometric chart, $h_{s1} = 80.00$ kJ/kg. So according to Eq. (3), $h_{s2} = 84.74$ kJ/kg. According to experience, the relative humidity of the outlet secondary air is assumed to be 80%, and then the dry bulb temperature of the outlet secondary air t_{s2} is 29.7 ℃. The average enthalpy value of the secondary air is given by

$$h_{sm} = \frac{h_{s1} + h_{s2}}{2} = 82.37 \text{ kJ/kg}$$

The average temperature t_{sm} of the secondary air outside the tube is given by[9]

$$t_{sm} = \frac{t_{s1} + t_{s2}}{2} \tag{4}$$

where t_{s1} is the dry bulb temperature of the inlet secondary air. Similarly, the parameters of the secondary air ρ_{sm}, $C_{p,sm}$, Pr_{sm}, λ_{sm}, γ_{sm} can be calculated.

2 Preliminary Design of Cooler Structure

2.1 Heat transfer area

The preliminary estimate of the heat flux per unit area is $q_o = 750$ kJ/(m^2·h). Thus, the required outside tube heat transfer area is given by

$$F_o = \frac{Q}{q_o} \tag{5}$$

2.2 Structure layout

Select a porous ceramic tube with an external diameter of 30 mm and a wall thickness of 5 mm. The inner surface of the tube is an impermeable ceramic thin film. At the cross-section of the cooler, there are 7 rows along the vertical direction, with 7 tubes averagely set at the odd row and 6 tubes averagely set at the even row. There are 46 tubes totally. Both the space between two horizontal tubes s_1 and the space between two vertical tubes s_2 are 40 mm, and the rows are staggered. The primary air head-on size is 290 mm × 290 mm; the secondary air head-on size is 290 mm × 600 mm. The tube length is 0.6 m. The actual heat transfer area for each tube is 2.60 m^2.

3 Calculation of Heat Transfer

3.1 Convective heat transfer coefficient

For the turbulent fluid in the slippery tube, the following relationship is adopted[9]:

$$Nu = 0.023 Re_{pm}^{0.8} Pr_{pm}^{0.3} \tag{6}$$

where Nu and Re_{pm} are the Nuselt number and the Reynolds number of the primary air, respectively.

The average temperature t_{pm} of the primary air is selected as the qualitative temperature, and the tube inner diameter d_i is selected as the type size. Then,

$$Re_{pm} = \frac{u_p d_i}{\gamma_{pm}} \tag{7}$$

where $u_p = 9.6$ m/s is the primary air velocity in the tube.

The convective heat transfer coefficient α_i can be calculated by

$$\alpha_i = Nu \frac{\lambda}{d_i} \tag{8}$$

3.2 Heat transfer coefficient

In engineering, in order to facilitate the calculation, the heat transfer coefficient of the water spray α_w under ambient temperature can be calculated by[11]

$$\alpha_w = 217 \frac{G_w}{2Lnd_o} \tag{9}$$

where $G_w = 580$ kg/h is the volume of the water spray; L is the tube length; n is the tube number of the first row; d_o is the outer diameter of the heat transfer tube.

3.3 Heat exchange between water film and secondary air

First, the convective heat transfer coefficient of the outside tube α_o is calculated. The calculation formulae are as follows[12]:

$$\alpha_o = \frac{0.88 C \lambda_{sm} Re_{sm}^m Pr_{sm}^{0.36}}{d_o} \tag{10}$$

$$Re_{sm} = \frac{u_{max} d_o}{\gamma_{sm}} \tag{11}$$

$$u_{max} = \frac{s_1 u_s}{s_1 - d_o} \tag{12}$$

where $C = 0.35$ and $m = 0.6$ are constant; u_{max} is the secondary air velocity at the narrowest cross-section; u_s is the face velocity of the secondary air.

Then, the convective mass transfer coefficient of the outside tube σ is calculated. According to the Lewis relationship, for a water-air system, $Le^{-2/3} \approx 1$, so $\sigma = \alpha_o / C_{p,sm}$. If the heat and mass transfer between the flowing air and the water film outside the tube is considered as a comparative sensible heat exchange process, then the heat exchange capacity can be calculated by[13]

$$Q = \alpha_j(t_w - t_{sm}) F_o = \frac{A\alpha_o}{C_{p,sm}}(h_w - h_{sm}) F_w \tag{13}$$

where α_j is the equivalent heat transfer coefficient; t_w is the average water film temperature; $A = 0.96$ is a coefficient of the water film temperature; h_w is the enthalpy of the saturated air boundary layer; $F_w = f_w F_0$ is the contact area between the water film and air; the coefficient f_w is 1.3 to 1.5, and for the porous ceramic tube, $f_w = 2.3$.

3.4 Heat load per unit area

Because the following calculation has a relationship with the average temperature of the water film t_w, the heat transfer process can be divided into two processes: the first is that the heat of the primary air transfers through the fouling layer, the tube wall, the fouling layer outside the tube and the cooling water film; the second is the heat exchange process between the cooling water film and the secondary air[13].

The heat flux of the first process can be calculated by

$$q'_o = K_o(t_{pm} - t_w) \tag{14}$$

The heat flux of the second process can be calculated by

$$q''_o = Af_w\sigma(h_w - h_{sm}) \tag{15}$$

where K_o is the heat transfer coefficient between the primary air and the water film. For a new heat exchanger, the fouling resistance on both sides of the tube can be ignored. And the thermal resistance of the water film can also be ignored. Because the ceramic membrane separating the primary air and the water film is very thin, the thermal resistance of the porous ceramic tube wall can be ignored. Then, K_o can be calculated by

$$K_o = \frac{1}{\frac{1}{a_i}\frac{d'_o}{d_i} + \frac{1}{\alpha_w}} \tag{16}$$

where d'_o is the equivalent diameter of the tube. Because the wall is composed of porous ceramics, $d'_o = (d_i + d_o)/2$. It is more difficult to directly solve h_w because it is a function of the water film temperature. So we adopt the trial method to calculate it in this paper. We can obtain that $t_w = 26.8$ ℃ and $q'_o = 752$ kJ/(m^2·h). q'_o has a 3% discrepancy with the assumed value 750 kJ/(m^2·h), indicating that the previous assumption is desirable. The heat transfer area outside the tube $F_o = Q/q'_o = 2.54$ m^2. Thus, the eventual heat transfer area is similar to the assumption. So we adopt the assumed cooler structure.

4 Resistance Calculation

4.1 Resistance calculation for primary air side

The pressure at the primary air side can be calculated by[14]

$$\Delta P_p = \Delta P_f + \Delta P_j \tag{17}$$

where ΔP_f is the loss of the frictional resistance and ΔP_j is the loss of the local resistance.

4.2 Resistance calculation for secondary air side

The pressure at the secondary air side can be calculated by[15]

$$\Delta P_s = 1.3 \times 0.334 C_f n' \frac{G_{max}^2}{2\rho_{sm}} \tag{18}$$

where $C_f \approx 2.11$ is the coefficient; $n' = 7$ is the number of the tube rows along the flow direction; G_{max} is the mass velocity of wet air at the narrowest cross-section. Fig. 1 presents the cooler designed by the above calculations.

Fig. 1 Photo of the porous ceramics tube type dew point indirect evaporative cooler

5 Conclusion

To solve the hydrophilic problem of the wall surface of a tube type indirect evaporative cooler, we adopt porous ceramics to a tube type indirect evaporative cooler. The design calculation of the porous ceramics tube type dew point indirect evaporative cooler is carried out from such aspects as the air volumes and the status parameters of the primary and secondary air, the cooler structure, the heat transfer and the resistance of the cooler. Porous ceramics are new class materials for the evaporative cooling technology, and the porous ceramic indirect evaporative cooling exchanger is a new type of heat exchanger, so there is no experimental research on the porous ceramic indirect evaporative cooling exchanger and some design parameters selected may be not reasonable. The next work is focused on experimental studies on the designed porous ceramics tube type dew point indirect evaporative cooler to justify the reasonability of the selected parameters and make corrections.

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多孔陶瓷管式露点间接蒸发冷却器的设计计算

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摘要:为了提高管式间接蒸发冷却器壁面的亲水性,提出了一种采用多孔陶瓷材料的新方法. 该方法将多孔陶瓷材料与蒸发冷却技术有机结合,从一次空气风量和二次空气风量、一次空气状态参数和二次空气状态参数、冷却器结构、固体多孔陶瓷管的传热以及冷却器阻力等方面对多孔陶瓷管式露点间接蒸发冷却器进行了设计计算. 计算结果表明该设计合理可行. 最后,依据设计制作出多孔陶瓷管式露点间接蒸发冷却器.

关键词:管式间接蒸发冷却器;露点;多孔陶瓷;设计计算

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