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The Influence of the Inlet Conditions on the Air Side Heat Transfer Performance of Plain Finned Evaporator

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***Abstract*—**A numerical study has been conducted to investigate the influence of fin pitch and relative humidity on the heat transfer performance of the fin-and-tube heat exchangers having plain fin geometry under dehumidifying conditions. The analysis is done using the ratio between the heat transfer coefficients in totally wet conditions and those in totally dry conditions using the appropriate correlations for both dry and wet conditions. For a constant relative humidity, it is found that the heat transfer coefficient increases with the increase of the air frontal velocity. By contrast, the fin efficiency decreases when the face velocity is increased. Apparently, this phenomenon is attributed to the path of condensate drainage. For the influence of relative humidity, the results showed an increase in heat transfer performance and a decrease in wet fin efficiency when relative humidity increases. This is due to the higher amount of mass transfer encountered at higher relative humidity. However, it is found that the effect of fin pitch on the heat transfer performance depends strongly on the face velocity. At lower frontal velocity the heat transfer increases with fin pitch. Conversely, an increase in fin pitch gives lower heat transfer coefficients when air velocity is increased.

***Keywords*—**Dehumidifying conditions, Fin efficiency, Heat and mass transfer, Heat exchangers.

I. Introduction

FINNED tube heat exchangers are commonly used in a variety of applications in the air-conditioning, refrigeration, and process industry. In practice, the fin surfaces may be fully wet, fully dry or partially wet depending on the difference between dew point temperature of the entering air and surface temperature. However, most investigations of heat exchangers performance under wet conditions consider only fully wet surfaces. A surface is assumed to be fully wet when its temperature is lower than the air dew point temperature throughout the entire heat exchanger. As a result, simultaneous heat and mass transfer occurs along the fin surfaces. The condensate retained on the surface of a heat exchanger has hydrodynamic effects by changing the surface geometry and the air flow pattern. Furthermore, a water layer on the surface increases local surface heat transfer resistance. Therefore, the heat transfer coefficients under wet and dry surface conditions might be significantly different from each other. Elmahdy and Biggs [1] presented a mathematical model to perform row-by-row numerical simulation of finned tube heat exchangers under both dry and wet surface conditions. The results from Mirth and Ramadhyani [2] showed a decrease of Nusselt number with an increase of dew point temperatures. So, it is necessary to have an air side heat transfer correlation that is valid for the specific fin geometry. Because the heat exchanger surface on the air-side consists of prime surface and finned surface, the fin efficiency is an important parameter for evaluating the performance of such fin-and-tube heat exchanger and this becomes more complicated with condensing conditions. Many studies on the fin efficiency of fin-and-tube heat exchangers are now presented in the open literature. For instance, Mirth and Ramadhyani [2] followed the same procedure given in ARI standard to derive fin efficiency equations in both dry and wet surface conditions. Elmahdy and Biggs [3] obtained a numerical solution for wet fin efficiency for circular or longitudinal fins. Their fin efficiency is evaluated using the temperature and specific humidity differences as the driving forces for combined heat and mass transfer. Consequently their results showed dependence between fin efficiency and the air relative humidity. Chen and Wang [4] presented correlations of heat transfer coefficient and fin efficiency in terms of relative humidity. Their results showed a decrease in fin efficiency and an increase in heat transfer coefficient with increasing relative humidity. The present study focuses on the air-side performance of the plain finned tube heat exchangers under wet conditions. The analysis is done using the ratio between heat transfer coefficients under wet conditions and those under dry conditions. We present also the fundamental mathematical formulation applied in a cooling and dehumidifying heat exchangers where a heat transfer coefficient under wet conditions is obtained.

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II. COOLING WITH DEHUMIDIFICATION

In the most of cooling processes, the in-coming dew point temperature of the entering air is higher than the cooling coil surface temperature so that the water vapor in the entering air will be condensed and then the condensate will be drained out. Thus, simultaneous heat and mass transfer occurs during the dehumidification process.

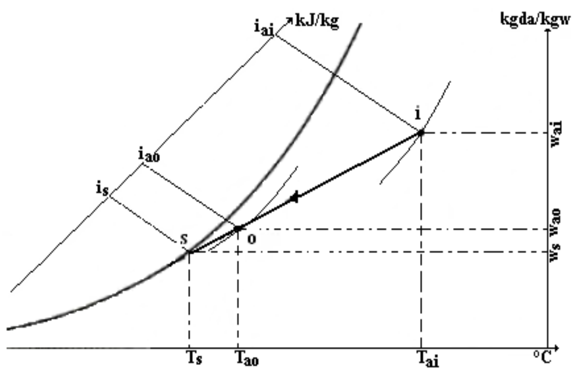

Fig. 1 Psychrometric chart of cooling and dehumidifying process

As a result, both the specific humidity and the local dew point temperature of the leaving air will be lowered. Fig. 1 shows psychrometric chart of cooling and dehumidifying process

A. Formulas and equations

By approximating the saturation curve on the psychrometric chart by a straight line between the inlet and the saturation temperatures, the heat transfer equation on the air side is the sum of the convective heat transfer and the mass transfer equations.

$$\delta \dot{Q} = h_{hta} dA_a (T_a - T_s) + h_{mta} dA_a (w_a - w_s) i_{fg} \quad (1)$$

Where, A_a "m" is the total air-side area, h_{hta} "W m-^{2}K^{-1}" and h_{mta} "kg m^{-2} s^{-1}" are the air-side heat and mass transfer coefficients, respectively; and w_s "kg kg^{-1}" is the humidity ratio of saturated air at the water film temperature T_s "K". the enthalpy of vaporization i_{fg} "J kg^{-1}" is evaluated at the wall temperature neglecting the thermal resistance of the film. When establishing equation (1), the Lewis analogy between heat transfer and mass transfer were used, where c_{pa} "J kg^{-1}K^{-1}" is the moist air specific heat at constant pressure.

$$Le = \frac{h_{hta}}{h_{mta}\, c_{pa}} \quad (2)$$

The validity of equation (2) relies heavily on the mass transfer rate and the assumption of unit Lewis factor is actually incorrect. However, with the need to allow for *Le* different from unity, the use of an appropriate correlation is required. In the present study, the correlation proposed by Pirompugd et al [5] was selected for calculating the Lewis Factor of a wet cooling coil.

$$Le = 2.28 N^{0.2393} \left(\frac{F_s}{D_o}\right)^{(0.0239 N + 0.4332)} \left(\frac{A_a}{A_t}\right)^{(0.0321 N + 0.0747} \times \mathrm{Re}_{D_o}^{(-0.01833 N + 0.194 F_s / D_o - 0.0026 X_L / D_o - 0.03012 X_t / D_o + 0.0418)} \quad (3)$$

Where, A_t "m" is the tube surface, Do "m" is the outside diameter, F_s "m" is the fin spacing, N is the tube row number, X_l "m" is the longitudinal tube pitch, and X_t "m" is the transverse tube pitch. The Reynolds number Re_{Do} is based on the tube outside diameter D_o.

Combining equations (1) and (2) yield

$$\delta \dot{Q} = h_{wet} dA_a (T_a - T_s) \quad (4)$$

Here, h_{wet} is the total heat transfer coefficient for wet external surface, given by:

$$h_{wet} = h_{hta} \left(1 + \frac{i_{fg}}{Le\, c_{pa}} \frac{w_a - w_s}{T_a - T_s}\right) \quad (5)$$

Assuming that equation (4) is for air-fin heat transfer and considering the corresponding air-tube heat transfer equation; a differential heat transfer from the air to an element of the evaporator is obtained:

$$\delta \dot{Q} = h_{wet} \eta_{s,wet} dA_a (T_a - T_s) \quad (6)$$

The overall fin efficiency is: where, A_F "m" is the fin area

$$\eta_{s,wet} = 1 - \frac{A_F}{A_a}(1 - \eta_{F,wet}) \quad (7)$$

The fin efficiency under wet condition is calculated as follows:

$$\eta_{F,wet} = \frac{tgh(m r_o \phi)}{(m r_o \phi)} \quad (8)$$

Here;

$$\phi = \left(\frac{r_{eq}}{r_o} - 1\right)\left(1 + 0.35 \ln\left(\frac{r_{eq}}{r_o}\right)\right) \quad (9)$$

And

$$m^2 = \frac{2 h_{wet}}{k_F \delta_F} \quad (10)$$

With, r_{eq} "m" is the radius of equivalent area circular fin, k_F "W m^{-1}K^{-1}" thermal conductivity of fin and δ_F "m" is the fin thickness.

B. Correlations

In general, the sensible heat transfer coefficient may be calculated in terms of the non-dimensional Chilton-Colburn j-factor as follow:

$$h_{hta} = j\, G_{a,\max}\, c_{pa} / \Pr^{2/3} \quad (11)$$

In this paper, the proposed correlation for coils operating under dehumidifying conditions is that derived from Wang et al [6].

$$j = 19.36 \,\mathrm{Re}_{D_o}^{j_1} \left(\frac{F_p}{D_o}\right)^{1.352} \left(\frac{X_l}{X_t}\right)^{0.6795} N^{-1.291}$$
$$j_1 = 0.3745 - 1.554 \left(\frac{F_p}{D_o}\right)^{0.24} \left(\frac{X_l}{X_t}\right)^{0.12} N^{-0.19} \quad (12)$$

The correlation proposed by Wang et al [7] can be used to calculate the air-side heat transfer coefficient under dry surface conditions:

$$j = 0.394 \,\mathrm{Re}_{D_o}^{-0.392} \left(\frac{\delta_F}{D_o}\right)^{-0.0449} N^{-0.0897} \left(\frac{F_p}{D_o}\right)^{-0.212} \quad (13)$$

In the above equations, F_p "m" is the fin pitch, $G_{a,max}$ "kg m^{-2}s^{-1}" is the mass flux of the air based on the minimum flow area and Pr is the Prandtl number.

III. Results And Discussion

The variations of the fin efficiency as a function of the in-coming air relative humidity for two different values of air frontal velocity (u_{fr}=0.5 and u_{fr}=4m/s) are shown in fig. 2. Firstly, the fin efficiency decreases with increasing relative humidity. This is due to the higher amount of latent energy encountered at higher relative humidity. In the other word, the accumulation of droplets becomes more efficient for higher relative humidity. Furthermore, the presence of condensate water on the fin surface leads to a higher latent heat transfer. Consequently, higher amount of mass transfer gives lower fin efficiency. As seen in Fig. 2 the influence of fin pitch on the fin efficiency is rather small for u_{fr}=0.5m/s than for u_{fr}=4m/s. Apparently it is attributed to the drop size and airflow pattern. As reported by Chen et al [4] the whole fin is in partially wet condition for lower relative humidity and is in totally wet condition for higher relative humidity when u_{fr}=0.5m/s. Possible explanation about this phenomenon is that the small droplet may grow up and join with the neighbouring fine size droplet to become larger. This larger droplet falls off to the fin base under the force due to surface tension. This phenomenon is more profound for a higher relative humidity. By increasing the air velocity, the path of condensate drainage is inclined to the airflow direction.

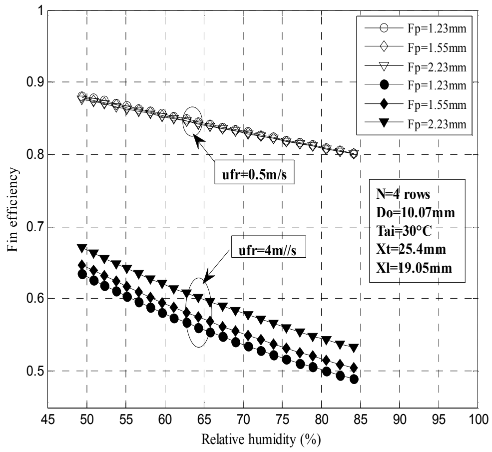

Fig. 2 Variation of fin efficiency with the relative humidity

One can be seen from Fig. 2 that the fin efficiency increased with the fin pitch at u_{fr}=4m/s. This is associated with the presence of condensate under dehumidification. A large amount of condensate may be blowing off the fin surface and this becomes much more rapidly when the fin pitch is increased.

The air-side performance of four-row configurations analysis is done using the ratio between the heat transfer coefficients in totally wet conditions and those in totally dry conditions. The effects of both inlet relative humidity and fin pitch on the air side heat transfer performance are shown in Fig. 3. The ordinates are h_{wet}/h_{dry} and the abscissa is the inlet relative humidity. It can be seen that the heat transfer coefficients increased as the relative humidity is increased. Apparently, it is attributed to the presence of condensate that provides an augmentation of latent heat transfer. In addition, the ratio h_{wet}/h_{dry} increased with frontal velocity. A larger mass of moisture flows across the fin surface when the air velocity is increased. Furthermore, the presence of condensate water acts as a roughening mechanism and thus improves the heat transfer performance. However, the effect of roughness on the heat transfer performance is more pronounced when the air velocity is increased.

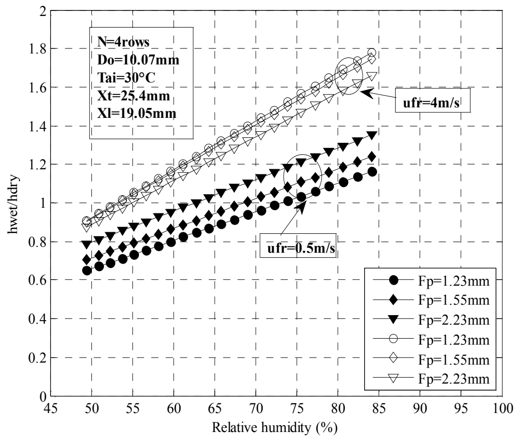

Fig. 3 Effect of inlet relative humidity on h_{wet}/h_{dry}

For the influence of fin pitch on heat transfer characteristics, Fig. 3 showed that the effect of fin pitch for u_{fr}=4m/s is slightly lower than for u_{fr}=0.5m/s. As can be seen, the heat transfer augmentation increases with a rise in fin pitch when u_{fr}=0.5m/s. This is because a large amount of condensate may rolls alongside the fin under the effect of gravity when the fin pitch is increased and the heat transfer enhancement owing to the flow of condensate becomes more pronounced for larger fin pitch. At u_{fr}=4m/s, the effect of fin pitch on the heat transfer performance increases with the increase of relative humidity. For lower relative humidity, the effect of fin pitch is comparatively small. A possible explanation for the effect of fin pitch in this region may be related to the presence of the condensate water and air velocity. First, a further increase of fin pitch would result in an increase of cross-stream width of vortex region behind the tube. As a result, the heat transfer performance decreases with the increase of fin pitch. Second, the flow of condensate water on the fin surface provides a good air flow mixing when the fin pitch is increased. This better mixing leads to an increase of the heat transfer performance. Combining both phenomena yields no effect of fin pitch on the heat transfer coefficient. At higher values of relative humidity, lower heat transfer coefficient is seen for larger fin pitch. When the fin pitch is increased, the condensate will exist on the fin surface in the form of water film, and can be drained vertically due to the force of gravity. This condensate will not block the air flow pattern when the fin pitch is increased. However, the recirculation zone is consistently extended. As a result, lower heat transfer coefficient is seen for larger fin pitch.

IV. Conclusion

This study presents a numerical investigation of the effects of relative humidity and fin pitch on the air side heat transfer performance of plain finned tube heat exchangers having four rows of tubes. The analysis is done using the ratio between the heat transfer coefficients under wet conditions and those under dry conditions. It is found that the influence of fin pitch on this ratio depends strongly on air face velocity. For the influence of relative humidity, it is found that the heat transfer coefficients increase with the rise of relative humidity indicating that the latent heat transfer is a very significant portion under dehumidifying conditions. The effect of fin pitch on fin efficiency is rather small at low velocity indicating that the fin surface is in fully wet condition. By increasing air velocity the fin efficiency diminishes with the rise of fin pitch. This is associated with the presence of condensate under dehumidification.

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