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Heat transfer Characteristics of Fin-and-Tube heat Exchanger under Condensing Conditions

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Abstract**—**In the present work an investigation of the effects of the air frontal velocity, relative humidity and dry air temperature on the heat transfer characteristics of plain finned tube evaporator has been conducted. Using an appropriate correlation for the air side heat transfer coefficient the temperature distribution along the fin surface was calculated using a dimensionless temperature distribution. For a constant relative humidity and bulb temperature, it is found that the temperature distribution decreases with increasing air frontal velocity. Apparently, it is attributed to the condensate water film flowing over the fin surface. When dry air temperature and face velocity are being kept constant, the temperature distribution decreases with the increase of inlet relative humidity. An increase in the inlet relative humidity is accompanied by a higher amount of moisture on the fin surface. This results in a higher amount of latent heat transfer which involves higher fin surface temperature. For the influence of dry air temperature, the results here show an increase in the dimensionless temperature parameter with a decrease in bulb temperature. Increasing bulb temperature leads to higher amount of sensible and latent heat transfer when other conditions remain constant.

Keywords**—**Fin efficiency, heat and mass transfer, dehumidifying conditions, finned tube heat exchangers.

I. Introduction

FIN performance is very important for accurate design of fin-and-tube heat exchangers. The main objective of fins is to increase the rate of heat transfer between the surface and surrounding air by increasing the surface area. In general, when the fin surface temperature is below the entering dew point temperature, combined heat and mass transfer occurs along the fin surface. A special situation is encountered when the partially wet conditions occurs. In this situation the fin base temperature is lower and the fin tip temperature is higher than the dew point temperature of the ambient air. During condensing conditions, the presence of condensate water and it interact with both fin surface and tube surface makes the study of heat transfer performance very complicated. With dehumidifying coil conditions, however, both moisture and sensible heat are removed from entering air. As a result, the change in fin efficiency becomes an important influential factor to determine coil performance. Elmahdy and Biggs [1] assumed that the difference between the specific humidity of saturated air at the fin surface temperature and that of the adjacent air is the potential for latent heat transfer.

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Their results indicate that the fin efficiency decreases with increasing air relative humidity. An investigation in the Elmahdy and Biggs' work was carried out by Kandlikar [2]. His results indicated that Elmahdy and Biggs' numerical work is probably in error. He preferred, however, the Threlkeld model which is based on the enthalpy potential as driven force. Sharqawy and Zubair [3], [4] proposed a linear relationship between humidity ratio of air and the corresponding temperature over the range from the fin base to dew point temperatures. Kazeminejad [5] proposed theoretical method for analyzing the heat flow within finned surfaces under wet conditions. He obtained a numerical solution of the differential equation for rectangular fins using the concept of sensible to total heat ratio. The effects of bulb temperature, relative humidity and cold fluid temperature variations on the fin temperature distribution have been presented graphically and compared with those under dry conditions. Salah e l-Din [6] assumed, also, that the humidity ratio of the saturated air on the wet fin surface varies linearly with the local fin temperature. His results were in consistent with the data of Kazeminejad [5].

In this work an investigation of the effects of the bulb temperature, relative humidity and face velocity on the temperature distribution over a totally wet fin surface is carried out.

II. Mathematical Formulation

Under dehumidifying conditions, the heat transfer from moist air to a fin surface may be expressed as:

$$dq = h_c dA_f \left(T_a - T_f\right) + h_m dA_f \left(w_a - w_{fs}\right) i_{fg} \tag{1}$$

Where, q "W" is the heat transfer rate, T_a "°C" is the bulk air temperature, T_f "°C" is the fin temperature, A_f "m" is the fin heat transfer area, i_{fg} "Jkg^{-1}" is the enthalpy of vaporization evaluated at the fin surface temperature and, h_c "$Wm^{-2}K^{-1}$" and h_m "$kgm^{-2}s^{-1}$" are the heat and mass transfer coefficients, respectively. Using the Lewis analogy between heat and mass transfer, the precedent equation may be reduced to:

$$dq = h_c \left(1 + \frac{i_{fg}}{c_{pm} Le} \frac{w_a - w_{fs}}{T_a - T_f}\right) \left(T_a - T_f\right) dA_f \tag{2}$$

And,

$$Le = \frac{h_m}{c_{pm} h_c} \tag{3}$$

Many attempts have been made to analyze the air-fin differential equation "(2)". For instance, McQuiston and Parker [7] changed the temperature expressions into enthalpy driven equations. The final expression, with Le=1, was expressed as:

$$dq = h_m dA_f \left(1 + i_{fg} \frac{w_a - w_{fs}}{i_a - i_f}\right)(i_a - i_f) \quad (4)$$

Here, i_a "Jkg^{-1}" is the enthalpy of the moist air in the free stream and i_f "Jkg^{-1}" the enthalpy of the saturated air at the fin surface. To operate in enthalpy terms, Oliet et al [8] introduced a fictitious enthalpy in the aim of changing the temperature in the heat transfer equation into enthalpy expressions.

By definition, the fin efficiency of fin-and-tube heat exchanger is the ratio between the actual heat transfer rate to the maximum heat transfer rate. However, the temperature distribution over the fin surface is required. From an energy balance on an element of straight fin with rectangular profile shown in fig. 1 the following differential equation of the temperature distribution can be derived:

$$\frac{d}{dx}\left(kA \frac{dT_f}{dx}\right) + ph_c (T_a - T_f) + ph_m (w_a - w_{fs}) = 0 \quad (5)$$

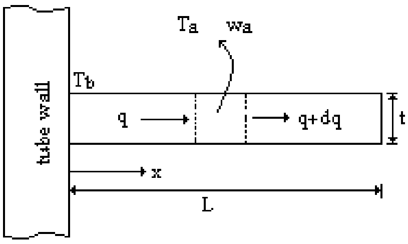

Fig. 1 Schematic diagram of a rectangular fin

In the above equation, the air specific humidity difference is a function of the local fin temperature. Elmahdy and Biggs [1], assumed a linear relation between w_{fs} and T_f over a small temperature range. Therefore

$$w_f = a + b_f T_f \quad (6)$$

The constants a and b_f are to be determined from the psychrometric data for the range of water temperatures considered. Sharqawy and Zubair [4], proposed a linear relationship between w_{fs} and T_f over the temperature range (T_b<T_f<T_{dp}) as follows:

$$w_{fs} = a_2 + b_2 T_f \quad (7)$$

Where;

$$a_2 = w_{fb} - \frac{w_{f,dp} - w_{fb}}{T_{dp} - T_b} T_b \quad (8)$$

$$b_2 = \frac{w_{f,dp} - w_{fb}}{T_{dp} - T_b} \quad (9)$$

Subscripts, b and dp referred to base and dew point, respectively.

Putting expression (7) into (5), the following equation may be obtained:

$$\frac{d^2\theta}{dX^2} - m_0^2 L^2 (1 + Bb_2)\theta - m_0^2 BL^2 (w_a - a_2 - b_2 T_a) = 0 \quad (10)$$

Where, t "m" is the fin thickness, k_f "$Wm^{-1}K^{-1}$" is the fin thermal conductivity and

$$\theta = T_a - T_f \quad (11)$$

$$B = \frac{i_{fg}}{c_{pm} Le} \quad (12)$$

$$m_0 = \left(\frac{2h_c}{k_f t}\right)^{0.5} \quad (13)$$

To obtain the temperature distribution over the fin surface, equation (10) can be solved by using these boundary conditions:

Fin base: at $X = 0$

$$\theta_b = T_a - T_b \quad (14)$$

And at fin tip: $X = 1$

$$\frac{d\theta}{dX} = -\frac{h_c L}{k_f}(1 + Bb_2)\theta - \frac{h_c LB}{k_f}(a_2 - w_a + b_2 T_a) \quad (15)$$

Kazeminejad [5] developed a solution to the problem described by the non homogeneous second order differential equation (10) when only sensible heat transfer occurs. When dehumidification occurs he solved numerically this equation by combining the classical Runge-Kutta method and the Newton-Raphson iteration.

Following to the work by Elmahdy and Biggs [1], the dimensionless fin surface temperature, θ_f, is plotted against the dimensionless distance from the fin base, thus

$$\theta_f = \frac{T_a - T_f}{T_a - T_b} \quad (16)$$

$$X = \frac{x}{L} \quad (17)$$

In the above equations, the air side heat transfer coefficient under fully wet conditions, h_c, can be calculated in terms of the non-dimensional Chilton-Colburn j-factor, given by

$$h_c = j G_{a,\max} c_{pm} \Pr^{-2/3} \quad (18)$$

Where; $G_{a,max}$ "$kg\ m^{-2}s^{-1}$" is the mass flux of the air based on the minimum flow area and Pr is the Prandtl number.

The proposed correlation for coils operating under wet conditions is that from Wang et al [9].

$$j = 19.36 \operatorname{Re}_{D_o}^{j_1} \left(\frac{F_p}{D_o}\right)^{1.352} \left(\frac{X_l}{X_t}\right)^{0.6795} N^{-1.291}$$
$$j_1 = 0.3745 - 1.554 \left(\frac{F_p}{D_o}\right)^{0.24} \left(\frac{X_l}{X_t}\right)^{0.12} N^{-0.19} \quad (19)$$

Here; D_o "m" is the tube outside diameter, F_p "m" is the fin pitch, N is the number of tube rows, Re is the Reynolds number, X_l "m" is the longitudinal tube pitch and X_t "m" the transverse tube pitch.

III. Results and Discussion

The special features of the present work is to study the effect of the variations of dry air temperature, relative humidity and the face velocity on the fin surface temperature distribution under dehumidification. Via finite difference method the differential equation that describes the temperature distribution was solved.

The effect of the relative humidity variations on the fin temperature distribution for two values of air frontal velocity (u_{fr}=0.5 m/s and u_{fr}=4 m/s) is shown in fig. 2. Firstly, the curves with dehumidification lie below that of a dry condition for a constant air frontal velocity. As can be seen, the curves for u_{fr}=4 m/s lie below that for u_{fr}=0.5 m/s. This phenomenon is attributed to the presence of condensate water. With increasing air velocity, a larger amount of moisture flowing across the fin surface under the same value of relative humidity can be observed. As a result, the latent heat transfer enhancement owing to the flow of condensate water film becomes more prfound. On the other hand, higher amount of latent energy lead to a higher fin surface temperature. In addition, when only sensible heat transfer occurs the results shown in fi. 2 indicate that the temperature distribution with u_{fr}=4 m/s is lower. The results here are associated with the higher heat transfer coefficients encountered in higher velocities.

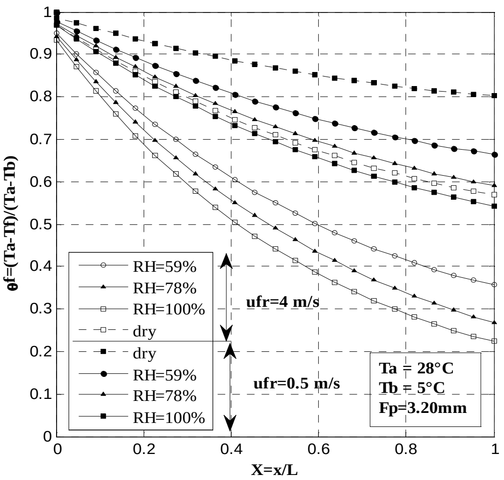

Fig. 2 Effect of relative humidity on the fin temperature distribution

Therefore, a larger heat transfer coefficient translates to a higher fin surface temperature. Thus, changing air velocity affects the temperature distribution even when the surface is dry.

For a constant air frontal velocity and dry air temperature, fig. 2 show that as relative humidity is increased, the dimensionless temperature distribution is decreased. As relative humidity increased the number of droplets condensing on the fin surface becomes more profound. This results in higher amount of mass transfer and higher latent energy and by consequence higher fin surface temperature. As a result, lower curve is that for higher relative humidity value.

Fig. 3 shows the effect of dry air temperature on the temperature distribution parameter. For a constant relative humidity, air velocity and fin base temperature, the amount of moisture increases with the increase in air temperature. In the other word, a higher air temperature translates to a larger amount of moisture content. These mean that the latent heat transfer contribution increases.

Fig. 4 shows the distribution of the curve-fitted fin temperature for various relative humidity values. The results here are in consistent with those by Sharqawy and Zubair [4] and Kazeminejad [5]. Their results indicate that the fin temperature increases with increasing relative humidity when other conditions are being kept constant.

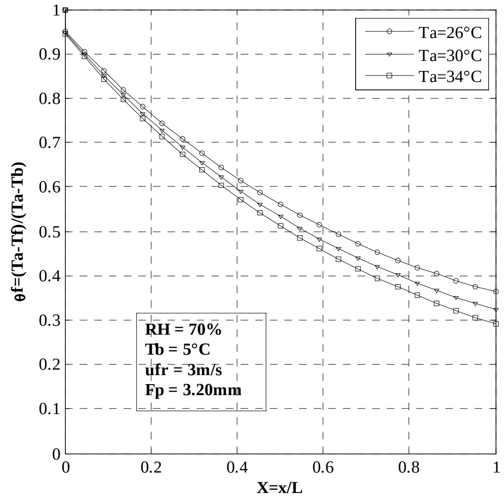

Fig. 3 Effect of dry bulb temperature on the temperature distribution

This is due to the presence of condensate water along the fin surface which involves a larger amount of mass transfer. On the other hand, there is more droplets accumulation for higher relative humidity under the same conditions. However, the latent heat transfer contribution becomes a significant portion when dehumidification occurs.

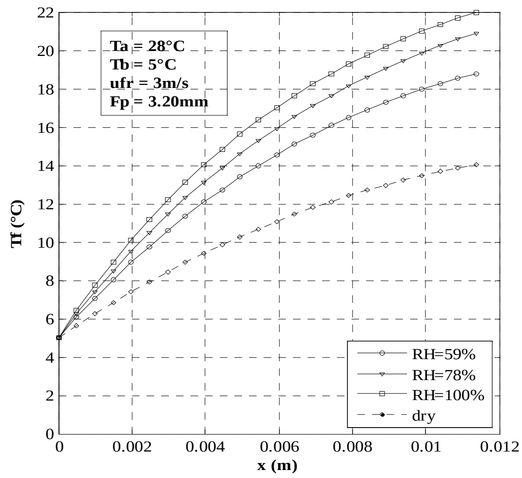

Fig. 4 distribution of fin temperature

Fig. 5 shows the effect of fin pitch on the wet fin temperature distribution. As can be seen, for a given air and fin base temperatures and inlet relative humidity, the dimensionless temperature distribution increases as the fin pitch increases. Apparently it is attributed to the presence of condensate water on the fin surface. The condensate water is easier to adhere between fin surfaces when the fin pitch is small enough. Further increase of fin pitch would result in a lower heat transfer coefficient. Therefore, the temperature distribution curves increases as the fin pitch increases.

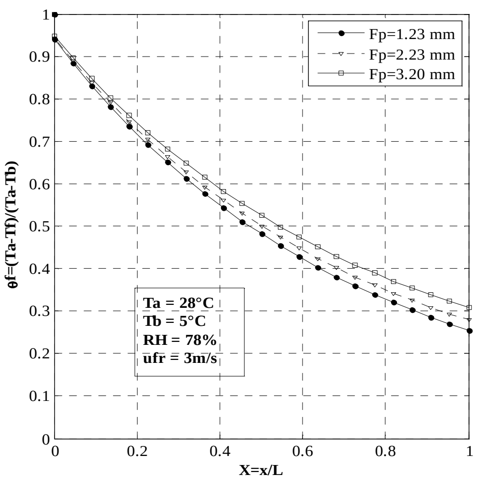

Fig. 5 Effect of fin pitch on the temperature distribution

IV. Conclusion

In the present paper an investigation of the influence of dry bulb temperature, relative humidity and air face velocity on the temperature distribution of a fully wet fin of rectangular profile has been conducted. With increasing air face velocity lower curves were obtained. This is attributed to the condensate water film flowing along the fin surface and to the heat transfer coefficients encountered at higher velocity. Thus, the temperature distribution can be affected by the air velocity even when the surface is dry. For the influence of relative humidity, the results here show a decrease in fin temperature parameter with the rise of relative humidity indicating that the latent heat transfer is a very significant portion under condensing conditions. The fin temperature distribution decreases with increasing dry temperature. The latent and sensible heat transfer increase as the air temperature increases. In addition, it is found that the temperature distribution curves increase with increasing fin pitch. This is because the condensate adhered phenomena.

References

[1] A.H. Elmahdy, and R.C. Biggs, “Efficiency of extended surfaces with simultaneous heat and mass transfer,” ASHRAE *Transaction.* vol. 89, Part 1A, pp. 135–143, 1983.

[2] Satish G. Kandlikar , “Thermal design theory for compact evaporators” *Visiting Scientist, 1988-1990, Mechanical Engineering Department,, Massachusetts Institute of Technology, Cambridge, Mass. 02139.*

[3] Mostafa H. Sharqawy, and Seyed M. Zubair, “Efficiency and optimization of an annular fin with combined heat and mass transfer-An analytical solution,” *International Journal of Refrigeration 30 (2007) 751-757*.

[4] Mostafa H. Sharqawy, and Seyed M. Zubair, “Efficiency and optimization of straight fins with combined heat and mass transfer-An analytical solution,” *Applied Thermal Engineering28 (2008) 2279-2288*.

[5] H. Kazeminejad, “Analysis of one-dimensional fin assembly heat transfer with dehumidification,” *Int. J. Heat and Mass Transfer. Vol. 38, No. 3, pp. 455-462, 1995*.

[6] M. M. Salah El Din, “Performance analysis of partially wet fin assembly,” Applied Thermal Engineering Vol. 18, No. 5, pp. 337-349, 1998.

[7] Faye C. McQuiston and Jerald D. Parker, “ Heating, Ventilating, and Air Conditioning: Analysis and Design” Second Edition, New York, 1982.

[8] C. Oliet, C.D. Pérez-Segarra, S. Danov and A. Oliva, “Numerical simulation of dehumidifying fin-and-tube heat exchangers: Semi-analytical modelling and experimental comparison,” *International Journal of Refrigeration 30 (2007) 1266-1277*.

[9] C. C. Wang, Y. T. Lin and C. J. Lee, “An airside correlation for plain fin-and-tube heat exchangers in wet conditions”, International Journal of Heat and Mass Transfer 43, 2000, pp. 1869–1872.