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Study on the heat transfer characteristics for fin-and-tube condenser

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***Abstract*— This paper investigates the heat transfer characteristics of a fin-and-tube heat exchanger in condenser application. The heat transfer coefficients are measured in terms of Nusselt numbers. The method used here is similar to row-by-row method. Using this method the effects of variations in number of tube rows and air velocity on the heat transfer characteristics are investigated and compared with those obtained from the other researchers. Also, this method allows us to find the maximum row number under given conditions of the inlet airstream and working fluid.**

***Keywords*— Air-conditioning, extended surface, fin analysis, heat exchanger, numerical method**

I. Introduction

In an absorption system, the use of the absorption air conditioner reduces significantly the cost and the complexity of the installation because it has air-coil heat exchangers for the condenser, absorber and evaporator. For condenser applications, condensation occurs inside tubes with cooling being provided by air rather than water. Important researches are now largely presented in the open literature to understand the condensation heat transfer mechanisms inside tubes [1]-[6]. Common flow pattern classifications within horizontal tubes are: stratified flow that corresponds to a region of small vapor velocity, slug, plug and wavy flows although still basically stratified and, the annular flow [7]-[16].

A large number of techniques and correlations for predicting the heat transfer coefficients during condensation inside pipes are proposed by many investigators. The correlations of Shah [17] and [18] may be used to evaluate the heat transfer coefficient of the condensing fluid inside horizontal and vertical tubes.

On the air-side, many investigations were devoted to the heat transfer characteristics for different fin geometries during the past years [19]-[23]. The works by McQuiston and Parker [24] and Wang et al [25] can be used to evaluate the air-side convective heat transfer coefficient for finned tube heat exchanger having plate fins. Their heat transfer coefficients are based on the Colburn j-factor.

In this paper the air coil heat exchanger is a finned tube condenser cooled by air. A mathematical procedure has been performed and a new method has been used to evaluate the temperature distribution over the fin surface. The heat transfer coefficients and the temperature rise of air along the flow length have been calculated using a method similar to row-by-row method. The results here are compared with those from other investigators.

II. Mathematical Procedure

The finned tube heat exchanger with continuous fins on an array of straight tubes shown in figure (1) is the most commonly used in heating, ventilating and air-conditioning systems.

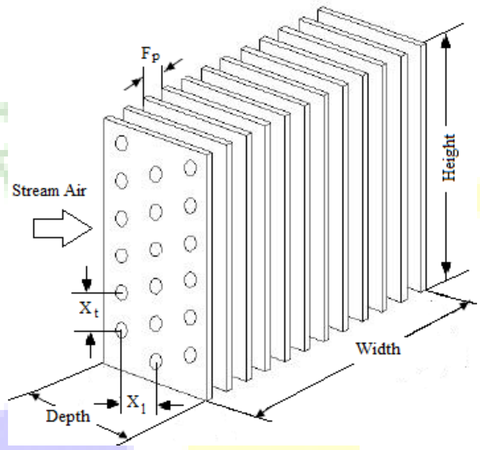

Fig. 1 A sample of tube-fin exchanger with continuous fins

The condenser used here is the cross flow in which the superheated vapour enters the condenser and being cooled by the return air from the indoor space as shown in Fig 2.

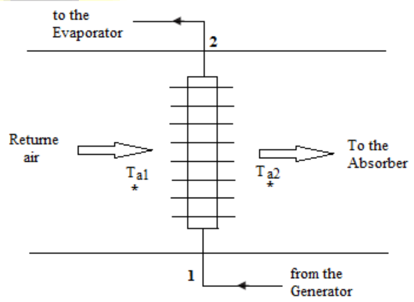

Fig. 2 Air cooled heat exchanger

The classical expression for determining the thermal design problem is defined by the log mean temperature difference method θ_{lm} :

$$Q_t = UA\theta_{lm}F \tag{1}$$

Where, A the heat transfer area, F the correction factor and θ_{lm} the logarithmic mean temperature difference. In condenser application the fluid flowing inside the tube is in two-phase flow. In this case, the mean temperature difference driving force is related to the saturated water, inlet and outlet air temperatures by:

$$\theta_{lm} = \frac{T_{a2} - T_{a1}}{\ln\left(\dfrac{T_2 - T_{a1}}{T_2 - T_{a2}}\right)} \tag{2}$$

Neglecting the air-side and the tube-side fouling resistances, the overall heat transfer coefficient U can be defined as [26]:

$$\frac{1}{U} = \frac{1}{h_o\eta_o} + \frac{d_o}{2k_m}\ln\left(\frac{d_o}{d_i}\right) + \frac{1}{h_i}\frac{d_o}{d_i} \tag{3}$$

From the previous equation, the heat must flow from the hot fluid to the cold fluid through three thermal resistances. The first term presents the air-side convective layer resistance, the second term provides the resistance of the exchanger material and the latest term exhibits the tube-side convective layer resistance. The surface efficiency η_o may be written in terms of fin efficiency η_f, fin surface area A_f and total surface area A_o, i.e.

$$\eta_o = 1 - \frac{A_f}{A_o}\left(1 - \eta_f\right) \tag{4}$$

Generally, the condenser can be separated into three regions, superheated, saturated, and sub-cooled regions. Assuming the refrigerant reaches its saturation point at the entrance of the condenser. Referring to Fig. 2 and assuming no extraneous heat losses, the same amount of heat is lost from the coolant:

$$Q_t = Q_w = w_t c_{p,t}\left(T_1 - T_2\right) \tag{5}$$

Or,

$$Q_t = Q_w = w_t\left(i_{v1} - i_{l2}\right) \tag{6}$$

The same amount of heat will be absorbed by the air stream:

$$Q_t = Q_a = w_a c_{p,a}\left(T_{a2} - T_{a1}\right) \tag{7}$$

Here, c_p the isobaric specific heat, $h_{v,1}$ and $h_{l,2}$ the enthalpy of the saturated vapor and the saturated liquid at inlet and outlet and w the mass flow rate. Subscripts a and t refer to air-side and tube-side, respectively.

A. Tube-Side Heat Transfer Coefficient

The heat transfer coefficient in two-phase flow inside tubes is dependent on the flow configuration and fluid properties. In this work, the correlation from Shah [17] can be applicable to all flow regimes. It has been verified by comparison with a wide variety of experimental data. At any given quality, the two phase condensing heat transfer coefficient derived from Shah [17] is:

$$h_i = h_{tp} = h_l\left[\left(1-\chi\right)^{0.8} + \frac{3.8x^{0.76}\left(1-\chi\right)^{0.04}}{\mathrm{P}_r^{0.38}}\right] \tag{8}$$

Where, h_l the heat transfer coefficient assuming all mass to be flowing as liquid, χ thermodynamic vapor quality and P_r the reduced pressure.

The liquid only heat transfer coefficient is calculated from the original work by the Dittus-Boelter [27]. By converting all the variables to basic SI units, their equation can be rewritten as:

$$Nu_t = \frac{h_l\, d_i}{k_l} = 0.0241\,\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4} \tag{9}$$

Where, k thermal conductivity, Nu Nusselt number, Pr Prandtl number and Re the Reynolds number. Subscript l refers to liquid. By integrating Eq. (9) over the length from L_1 to L_2, Shah obtained a sufficiently accurate definition of mean heat transfer coefficient h_{tpm}. For complete condensation, from $\chi = 1$ to $\chi = 0$, Eq. (9) yields :

$$h_{tpm} = h_l\left(0.55 + \frac{2.09}{\mathrm{P}_r^{0.38}}\right) \tag{10}$$

B. Air-Side Heat Transfer Coefficient

The heat transfer coefficient on the air side can be easily found by combining Eqs (1), (3) and (10).

$$\frac{1}{h_o} = \left[\frac{1}{U} - \left(\frac{d_o}{2k_m}\ln\left(\frac{d_o}{di}\right) + \frac{1}{h_{tpm}}\frac{d_o}{d_i}\right)\right]\times\eta_o \tag{11}$$

Furthermore, the convective heat transfer coefficient can be calculated as follows, let

$$Q_t = Q_f + Q_{root} \tag{12}$$

Where, Q_f the heat transferred through fin and Q_{root} heat transferred through root metal. Considering a clean outside surface

$$Q_f = h_o^{'} A_f\left(\overline{T_f - Ta}\right) \tag{13}$$

$$Q_{root} = h_o^{'} A_{to}\left(\overline{T_b - T_a}\right) \tag{14}$$

Introducing now the classical definition of the fin efficiency:

$$\eta_f = \frac{Q_{actual}}{Q_{\max}} = \frac{T_f - T_a}{T_b - T_a} \quad (15)$$

Thus:

$$Q_t = h_o' \left(\overline{T_b - T_a}\right)\left(A_{to} + \eta_f A_f\right) \quad (16)$$

As reported by Young and Ward [28], the actual fin side heat transfer coefficient h_o' is related to the equivalent area A_{eq}, the total outside area A_o and the design coefficient h_o as:

$$h_o = h_o' \frac{A_{eq}}{A_o} \quad (17)$$

The equivalent area can be given by:

$$A_{eq} = A_{to} + \eta_f A_f \quad (18)$$

Where, A_{tb} is the exposed area of the tube base and the heat transfer rate Q_t was measured directly from Eq. (6) or Eq. (7). As reported by Sparrow and Samie [29] the quantity $\left(\overline{T_f - T_a}\right)$ is the average of the difference between the surface and the free stream temperatures T_f and T_a, respectively. On the other hand, the surface temperature varies from the fin base to the fin tip, and then a temperature distribution over the fin surface will be required.

It can be noted here that correlations include those of Wang et al [21], McQuiston and Parker [24], Wang et al [25], Katz and Young [30], Elmahdy and Biggs [31], can be used to calculate the air-side heat transfer coefficient for fin-and-tube heat exchanger having plain fin geometry with multiple rows of staggered tubes. Their heat transfer coefficients are based on the Colburn-j factor in which the basic form of the correlation is:

$$j = \frac{Nu}{\mathrm{Re}_m \Pr^{1/3}} = C_1 \mathrm{Re}_m^{C_2} \quad (19)$$

Where, the parameters of C_1 and C_2 depend on the physical dimensions of the heat exchanger. In the works by McQuiston and Parker [24] and Kayansayan [32]-[33], Eq (19) was multiplied by the heat exchanger fining factor that describes the flow geometry effects. This factor can be represented by the following expression:

$$\varsigma = \frac{A_o}{A_{to}} \quad (20)$$

Where, Re_m the Reynolds number based on the outside tube diameter and given by:

$$\mathrm{Re}_m = \frac{G_c d_o}{\mu_a} \quad (21)$$

Here, μ_a the dynamic viscosity of the bulk air and G_c the mass flux of the air based on the free-flow area A_c:

$$G_{\max} = \rho_a u_{\max} \quad (22)$$

Where, ρ_a the air density and $u_{\max}$ the maximum velocity inside the heat exchanger given by:

$$u_{\max} = \frac{u_{fr}}{\sigma} \quad (23)$$

Where, u_{fr} air frontal velocity and σ the ratio of free-flow area to frontal area of the air-side exchanger.

C. Temperature Distribution over the fin surface

The temperature distribution over the fin surface changes from the fin base to the fin tip. This can be obtained by solving a second order differential equation derived from a simple heat balance on a differential element of a fin.

For a rectangular fin profile:

$$\frac{d^2\theta}{dX^2} = m^2 L^2 \theta \quad (24)$$

Where $X = (x / L)$ the dimensionless distance from the fin base, x distance from the fin base and L the fin length and θ represents the temperature difference between the air temperature and the fin surface temperature. The parameter m defined by:

$$m = \sqrt{\left(\frac{2 h_o'}{k_f t_b}\right)} \quad (25)$$

Where, k_f the fin thermal conductivity and t_b the fin thickness at fin base. The following boundary conditions are needed to solve Eq. (24):

$$\theta = \theta_b = T_a - T_b \quad \text{at} \quad X = 1 \quad (26)$$

$$\frac{d\theta}{dX} = 0 \quad \text{at} \quad X = 0 \quad (27)$$

For a constant thickness circular fin:

$$r^2 \frac{d^2\theta}{dr^2} + r \frac{d\theta}{dr} = r^2 m^2 \theta \quad (28)$$

Here, r the fin radius measured from the tube centerline. The following boundary conditions are needed to solve Eq. (28):

$$\theta = \theta_b = T_a - T_b \quad \text{at} \quad r = r_b \quad (29)$$

$$\frac{d\theta}{dr} = 0 \quad \text{at} \quad r = r_e \quad (30)$$

The analytical solution of Eqs (24) and (28) subjected to boundary conditions Eqs (26), (27), (29) and (30) gives us the temperature distribution along the fin surface. These solutions are:

For rectangular fin profile

$$\theta = \theta_b \frac{\cosh(mLX)}{\cosh(mL)} \quad (31)$$

For circular fin, Bourabaa et al [34] used the modified Bessel functions to solve their second order differential equation under wet fin surface. Just neglecting the condensation effects, their analytical solution for a dry circular fin can be modified as:

$$\theta = \theta_b \frac{I_o(mr)K_1(mr_e)+I_1(mr_e)K_o(mr)}{I_o(mr_o)K(mr_e)+I_1(mr_e)K_o(mr_o)} \quad (32)$$

Where, I_n and K_n are the modified Bessel of first and second kinds, r_e and r_o are the outer and the inner fin radius, respectively. Eqs (24) and (28) can be solved numerically using the same methods presented by Bourabaa et al [34]-[35]. Another solution for the temperature distribution over the dry fin surface will be presented here. This solution proceeds by assuming a temperature distribution that can be used to evaluate the fin efficiency, Eq. (15), and thus the average heat transfer coefficient on the fin side, Eq. (16).

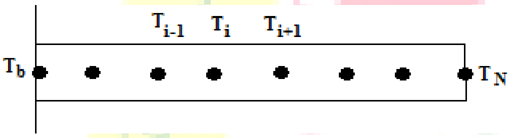

Fig. 3 Control volume for an internal node

The first step is to position the nodes throughout the computational domain. For our case, the nodes will be distributed uniformly as shown in Fig. 3 so:

$$x_i = \frac{i-1}{N-1}L \quad \text{for} \quad i = \overline{1:N} \quad (33)$$

Where, N the number of nodal points where the fin surface temperature is to be determined. The distance between adjacent nodes is:

$$\Delta x = \frac{L}{N-1} \quad (34)$$

Based on Fig. 3, a linear temperature distribution provides:

$$T_i = T_1 - \frac{i-1}{N-1}(T_N - T_1) \quad (35)$$

Since we used this approach, the temperature at first node and at the end node must be known as special cases, lets:

$$T_1 = T_b \quad \text{and} \quad T_N = T_a \quad (36)$$

As the fin temperature changes from the fin base to the fin tip, we assume that the maximum temperature at the fin tip is the dry bulb temperature of the air stream.

D. Fin efficiency

The theoretical fin efficiency is defined as the ratio of the actual heat transfer rate to the maximum heat transfer rate when the fin is at the same base temperature. In the present work, the fin efficiency can be obtained using the model of McQuiston and Parker [24]. Their method is summarized as follows:

For a rectangular fin

$$\eta_f = \frac{\tanh(mL)}{mL} \quad (37)$$

For a circular fin

$$\eta_f = \frac{\tanh(mr\phi)}{mr\phi} \quad (38)$$

$$\phi = \left(\frac{R_{eq}}{r_i} - 1\right)\left[1 + 0.35\ln\left(\frac{R_{eq}}{r_i}\right)\right] \quad (39)$$

$$\frac{R_{eq}}{r_i} = 1.27\frac{X_M}{r_i}\left(\frac{X_L}{X_M} - 0.3\right)^{0.5} \quad (40)$$

Where, R_{eq} is the equivalent radius for circular fin and r_i is the tube inside diameter. The geometric parameters X_L and X_M for staggered tube layout are:

$$X_L = \left[\left(\frac{X_t}{2}\right)^2 + \frac{X_l^2}{2}\right]^{0.5} \quad (41)$$

$$X_M = \frac{X_t}{2} \quad (42)$$

Where, X_l the longitudinal tube pitch and X_t the transverse tube pitch.

III. Heat Exchanger Design

In the application of the basic heat transfer data to the geometry of the heat exchanger, certain geometrical relations are necessary. Expressions provide the geometrical description for different fin profiles and tube layouts are presented in details by Kays and London [36]. For the plate-fin, cross flow heat exchanger with staggered tube layout, the following relations are derived from the works by Kayansayan [32]-[33] and Kays and London [36], referring to Fig 1:

The relation between the number of tubes per row n_t, the transverse pitch X_t and the exchanger height H is:

$$H = n_t X_t \quad (43)$$

The number of tube rows N_r, the longitudinal tube spacing X_l and the flow length L_f are related as:

$$L_f = N_r X_l \quad (44)$$

The free-flow area per unit length A_c and the exchanger frontal area A_{fr} per unit length are:

$$A_c = n_t(X_t - d_o)(1 - t_b S_f) \quad (45)$$

$$A_{fr} = n_t X_t \qquad (46)$$

The fin density S_f has units of fins per unit length. The ratio of free-flow area to frontal area of exchanger σ is then:

$$\sigma = \frac{A_c}{A_{fr}} = (1 - d_o)(1 - t_b S_f) \qquad (47)$$

The finned area per unit length A_f and the tube outside area per unit length A_{to} are given by:

$$A_f = N_r n_t \frac{\pi d_o^2}{2}\left(\frac{4}{\pi}\frac{X_l X_t}{d_o^2} - 1\right) S_f \qquad (48)$$

$$A_{to} = N_r n_t \pi d_o (1 - t_b S_f) \qquad (49)$$

The total outside transfer area of exchanger per unit length A_o is the sum of A_f and A_{to}.

The total exchanger volume per unit length V is:

$$V = L_f A_{fr} = L_f n_t X_t \qquad (50)$$

The flow-passage hydraulic diameter is:

$$d_h = \frac{4 L_f A_{min}}{A_o} = \frac{4\sigma A_{fr} L_f}{A_o} \qquad (51)$$

The ratio of total transfer area of exchanger to total volume α is:

$$\alpha = \frac{A_o}{V} = \frac{A_o}{L_f A_{fr}} = \frac{4\sigma}{d_h} \qquad (52)$$

Furthermore, the total inside area of exchanger tubes per unit length is defined as:

$$A_i = N_r n_t \pi d_i \qquad (53)$$

IV. RESULTS AND DISCUSSION

The variations of the Nusselt number with Reynolds number for the first and the fourth rows of the present fin-and-tube condenser are presented in Fig. 4. We assumed that the outlet temperature of the stream air for such row is assumed to be the inlet temperature of the air for the next row. The condensing temperature of the working fluid is assumed to be the outlet temperature of the water. The outlet temperature of such row is usually not greater than the condensing temperature of the working fluid. As can be seen from Fig. 4, the Nusselt number of the present work is calculated using equations (11) and (17) and compared with those of McAdams [1], Wang et al [21] and Ward [37]. The Nusselt number measured from Eqs. (11) and (17) gives us a mean deviation of 5.3 % for the first row. We can see that the deviation increases with increasing tube rows and reaches a mean value of 22 % for the fourth row. Apparently, the divergence can mainly be attributed to the methods used for the temperature distribution along the fin surfaces. The divergence between the results of the present work and those from McAdams, Wang et al and Ward may be attributed to the data bank and methods used in their correlations. It should be recalled, here, that the present method assumes that each row is considered as a heat exchanger and thus the outlet conditions of such row will be used as inlet conditions for the next row.

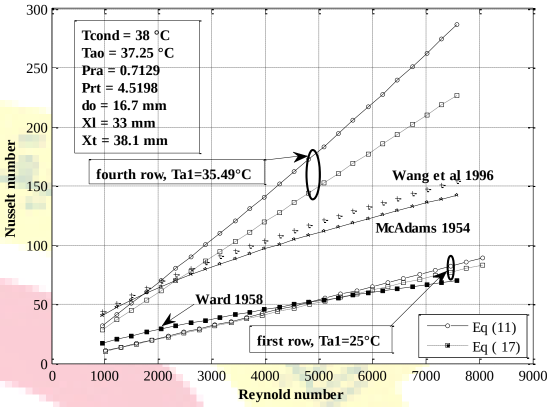

Fig. 4 Nusselt number vs Reynolds number

The second attention of the present work is turned to the effect of the tube rows on the Nusselt number. Fig.5 indicates that the Nusselt number increases with increasing Reynolds number for all rows. As the Reynolds number increases, the downstream turbulence increases and this causing air flow mixing to increase. As a result, higher heat transfer coefficient can be seen at higher Reynolds number.

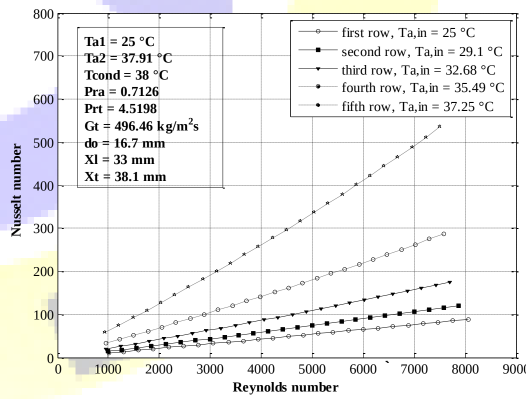

Fig. 5 Effect of the number of tube rows on Nusselt number

For the effect of tube rows on the air-side heat transfer characteristics, Fig. 4 shows that the Nusselt number increases with the increase in tube rows and this effect becomes more important at higher Reynolds number. It is found from these results that the mean value of the heat transfer coefficient for the first row is about 33 % smaller than the second row. These results are in consistent with those by Young and Ward [28] and Sparrow and Samie [29]. Their heat transfer coefficient for the first row of staggered arrangement was found 30 % smaller than the second row. In addition, this method allows us to find the maximum number of rows. Under the conditions of the present work the maximum number of rows was five.

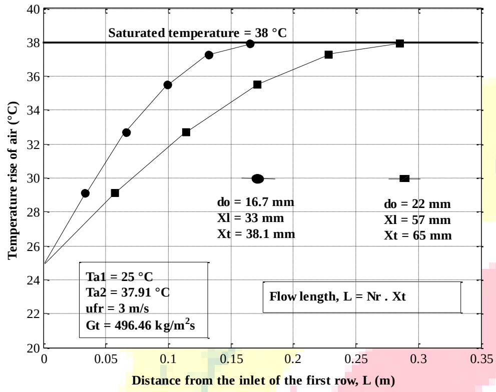

Fig. 6 Temperature profile of exit air across flow length

Fig. 6 depicts the variation of the exit air temperature along the heat exchanger depth. It can be seen that the curves for higher geometrical parameters lie below that of smaller ones. The possible explanation is as follow: increasing tube diameter and transverse tube spacing can be lead to incomplete mixing of the air across the depth of the exchanger and this can be offset by increasing the longitudinal tube pitch.

V. CONCLUSIONS

A method similar to row-by-row method has been performed to investigate the heat transfer characteristics of an air-cooled heat exchanger. The condenser here is a fin-and-tube heat exchanger. The results showed an increase of Nusselt numbers with both Reynolds number and tube rows. The heat transfer coefficient for the first row was found 33 % smaller than the second row. The air temperature curves diminish with increasing tube diameter and transverse tube pitch. This is due to incomplete mixing of the air across the flow length.

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