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Abstract

A computational model for air-conditioning condensers and evaporators with in-tube enhancement has been developed. The purpose of this model is to provide a tool for the design of plate finned-tube heat exchanger coils in both heating and cooling modes. This model assumes steady state conditions with non-homogeneous, two-phase flow inside the heat exchanger tubes and air in cross-flow on the outside of the tubes. It allows users to specify evaporator and condenser operating conditions, fin-and-tube heat exchanger parameters, refrigerant flow, air flow, and the type of enhanced tubes.

The heat exchanger model uses the Gauss-Seidel iteration scheme to solve the momentum equation, the energy equation, and equations of state, simultaneously. The oveall computer model utilizes a number of correlations from the literature for heat transfer and pressure drop.

Sample solutions were obtained for three different arrangements of heat exchanger tubes. Predicted heat transfer rates, outlet air temperature, local temperatures of the refrigerant, tube wall and air, and local pressure, quality, specific volume, and enthalpy of refrigerant are presented.

UN MODELE DE CONDENSEUR ET D'EVAPORATEUR POUR LE CONDITIONNEMENT D'AIR INSISTANT SUR LE RENFORCEMENT A L'INTERIEUR DES TUBES.

RESUME : L'auteur a mis au point un modèle de calcul des condenseurs et des évaporateurs pour le conditionnement d'air avec renforcement interne des tubes. Le but de ce modèle est d'apporter un outil pour la conception des serpentins de l'échangeur de chaleur à plaques et tubes ailetés en régime de chauffage et en régime de refroidissement. Ce modèle suppose un régime permanent avec écoulement diphasique non homogène dans les tubes de l'échangeur de chaleur et circulation de l'air en courant croisé à l'extérieur des tubes. Il permet aux utilisateurs de préciser les conditions de fonctionnement des évaporateurs et des condenseurs, les paramètres des échangeurs de chaleur à tubes à ailettes, de l'écoulement du frigorigène, de l'écoulement d'air et du type de renforcement des tubes.

Le modèle d'échangeur de chaleur utilise le système d'itération de Gauss-Seidel pour résoudre simultanément les équations des moments de l'énergie et les équations d'état. Le modèle informatique général utilise un certain nombre de corrélations provenant de la littérature pour le transfert de chaleur et la chute de pression.

On a obtenu des solutions pour trois types d'aménagement de tubes d'échangeur de chaleur. On présente les prévisions relatives au taux de transfert de chaleur, à la température de l'air sortant, aux températures locales du frigorigène, de la paroi des tubes et de l'air et à la pression locale, la qualité, le volume massique et l'enthalpie du frigorigène.

A Model of an Air-Conditioning Condenser and Evaporator with Emphasis on In-tube Enhancement

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1. INTRODUCTION

A computational model was developed for air-conditioning condensers and evaporators with either in-tube enhancement or installed smooth tubes. Use of the model is demonstrated by evaluating the effects of installing micro-fin tubes in a condenser and evaporator.

The model can be used to predict the thermal performance of the condenser or evaporator. It uses a numerical solution scheme to solve the momentum equation, energy equation, and equations of state along incremental lengths of each tube. This approach is potentially more accurate than analyzing the whole heat exchanger as a lumped system. The model considers variations in heat transfer coefficients, friction factors, and refrigerant properties throughout the heat exchanger. These variations are due to differing temperatures, qualities, and heat transfer rates in each element. The heat exchanger model can be used to predict the overall heat transfer rate, the outlet temperature of the air flow, the local temperatures of the refrigerant, tube wall, and air, and the local pressure, enthalpy, specific volume, quality, and heat flux of the refrigerant.

Since 1976, several models for condensers and evaporators have been developed. However, computational models of condensers and evaporators with in-tube enhancement have not been reported. Also, most models are not based on the integration of local conditions over the heat exchanger but rather on simplifying assumptions such as the effectiveness-NTU method. Examples of this latter approach are models by Hiller and Glicksman [1] and Ellison et al. [2]. Hiller and Glicksman used the effectiveness-NTU method to model a condenser; later, this same approach was used by Ellison et al. to model an air-cooled, complex tube circuiting condenser. Anand and Tree [3] developed a steady-state simulation of a condenser that consisted of a single tube that was finned. Variations of the heat transfer coefficient and the friction factor due to phase changes along the tube were included in their model. Two-phase pressure drops were based on a homogeneous model. Fischer and Rice [4] modified the model of Ellison et al. to obtain a steady-state computer design model for air-to-air heat pumps . The correlation by Traviss et al. [5] was used to compute local condensation heat transfer coefficients, while the correlation by Chaddock and Noerager [6] was used to compute boiling heat transfer coefficients. The air-side heat transfer coefficients were based on the work of McQuiston [7].

The study reported herein emphasizes in-tube enhancement because past models have not incorporated this feature, because little information has been published regarding total heat exchanger behavior when in-tube enhancement is used, and, finally, because in-tube enhancement is being utilized more frequently by coil manufacturers. Of all the types of in-tube enhancement that are available commercially, micro-fin tubes show the most promise for refrigeration applications because they increase heat transfer significantly while producing only minor increases in pressure drop [8]. Micro-fin tubes are characterized by numerous fins (48-70) that are short in height (0.12-0.25 mm). Several recent reports in the literature on in-tube enhancement of refrigerants with micro-fin tubes by Khanpara et al. [9,10] investigated evaporation and condensation of R-113 with three different mass flow rates through a smooth tube and eight different augmented tubes of 9.52 mm outside diameter. They also compared the effects of several geometrical parameters (peak shape, valley shape, fin height, number of fins, and spiral angle). For evaporation, the average heat transfer was increased by 45% to 238% over the reference smooth tube value. The pressure drop increase was in the range of only 4% to 102%. Enhanced heat transfer for single-phase flow over a range of Reynolds number from 5,000 to 11,000 was also reported. The micro-fin tubes resulted in enhancement factors of 1.3 to 1.8 for single-phase flow.

2. THEORETICAL MODEL

2.1. Description

The model of the air-conditioning condenser and evaporator developed in this study is based on the following assumptions:

- one-dimensional, fully developed, turbulent, steady flow of refrigerant through horizontal tubes with outside plate fins
- dry air flow with constant flow rate and properties on the outside of the tubes
- negligible change in kinetic energy and potential energy
- no energy generation and dissipation in the refrigerant flow and air flow
- constant thermal conductivities for the tube wall and fins

The evaporator and condenser model consist of equations for the refrigerant and air side. For the refrigerant flow, the continuity, momentum, and energy equations along with equations of state are applicable. For the air flow, the energy equation is applicable. The set of governing differential equations, approximated as finite difference equations, were then solved by the Gauss-Seidel iteration scheme. In order to build a more realistic evaporator and condenser model, variations of refrigerant properties, heat transfer coefficients, and friction factors throughout the heat exchanger were considered in this model.

Because of the many similarities between evaporators and condensers, including the governing equations that make up the models, the evaporator and condenser models have been combined into a single computer program. The only special arrangement required is that different heat transfer coefficient correlations, friction factors, enhancement factors, and boundary conditions be used for the evaporator and condenser. Since dehumidification by the evaporator is beyond the scope of this study, the outside air flow for the model derived herein is assumed to be dry. As a result, the air-sides of both models are similar. The model was designed so that the entering refrigerant can be either superheated vapor, two-phase flow, or subcooled liquid for either the heating or cooling process. In addition, the model can be used with any of several refrigerants, including R-12 and R-22. The results to be presented later are R-22 for the evaporator and R-12 for the condenser.

Three kinds of tube circuiting can be considered by the model. The first type is a parallel arrangement as shown in Fig. 1 where the refrigerant enters near the front face and flows in the same general direction as the air flow. A second arrangement is where the main flow stream divides to supply each row of tubes (e.g., three rows of tubes result in three circuits) while a third arrangement is where the main flow divides to supply each tube in the heat exchanger. The three types of possible tube circuits are described in detail in reference 11. Only the results for the first arrangement are presented herein. It is important to note that there are many possible arrangements for tube circuitry in air-to-refrigerant heat exchangers. Because of the approach used herein to solve simultaneous equations on the air and refrigerant side, it was possible to analyze only certain types of tube circuits. For example, the first circuitry is a parallel flow arrangement that is not typically used for condenser coils. However, since the main emphasis herein is a comparison of smooth tubes and enhanced tubes, the conclusions can be applied to other circuits in the absence of other information.

2.2. Governing equations

To simulate the heat exchanger in detail, the tube length is divided into increments and the governing equations are solved along these incremental lengths. Figure 2 shows a typical increment for a finned tube heat exchanger. In the present study, the computation domain starts from the entrance (i.e., refrigerant and air entrance) of each heat exchanger tube and ends at the exit of each tube. The backward difference scheme was used to transform the governing differential equations into finite difference equations. The energy equations for refrigerant flow and air flow were transformed into the finite difference form as follows:

and

$$Q_{per\ element} = m_r (i_{j-1} - i_j) \tag{1}$$

$$Q_{per\ element} = U_o A_o \left(\frac{T_{r,j-1} + T_{r,j}}{2} - T_a \right) \tag{2}$$

$$Q_{per\ element} = m_{a_{per\ element}} \cdot c_{pa} (T_{a,out} - T_{a,in}) \tag{3}$$

where subscripts "r" stands for refrigerant, "a" stands for air, "o" stands for outside, and "j" represents the j-th increment along the tube. The parameters listed in the above equations follow conventions used in most of the literature, namely, m is the mass flow, i is enthalpy, U is overall heat transfer coefficient, A is area, and T is temperature. The "per element" subscript refers to the fact that each tube is divided into increments that are referred to as elements. Since the heat exchanger is divided into increments, the area, A, corresponds to an incremental area. As discussed later, the above equation along with the refrigerant-side momentum equation and the equation of state must be solved simultaneously over each tube increment.

The differential equation for momentum can be approximated by a finite-difference numerical form as follows:

$$P_{j-1} - P_j = G^2 \left[\left(\frac{1}{\rho_j} - \frac{1}{\rho_{j-1}} \right) + \frac{f \cdot (z_j - z_{j-1})}{4D} \left(\frac{1}{\rho_j} + \frac{1}{\rho_{j-1}} \right) \right] \tag{4}$$

where P is pressure, G is mass flux, ρ is density, f is friction factor, z is position and D is inside tube diameter.

2.3. Overall heat transfer coefficient

The equation for the overall heat transfer coefficient, U_o, can be derived by using thermal resistances. The thermal resistances are defined as follows:

$R_1 = 1/h_i A_{pi}$	convection inside the tube
$R_2 = 1/h_{di} A_{pi}$	inside deposit
$R_3 = t_p/A_{pm} k_p$	tube wall
$R_4 = 1/h_c A_{po}$	contact between fin and tube wall
$R_5 = 1/h_{do} A_o$	outside deposit
R_6 = (see discussion)	fin
$R_7 = 1/h_o A_{po}$	convection outside the tube

The values of h_{di} and h_{do} for the inside and outside deposits respectively approach infinity for a clean tube surface. The parameter h_c is the equivalent convection coefficient for fin-tube contact thermal resistance based on the outside tube area. A recent study of 31 coils from 6 manufacturers showed that h_c varies considerably, ranging from 1.045 kW/m^2K to infinity [12]. An average value of 14 kW/m^2K was used in this study. It is convenient to consider the effect of R_6 by combining it with R_7.

As described in detail in reference 13, the final U_o equation is obtained by combining the above resistances.

$$U_o = \frac{1}{\frac{A_o}{A_{pi} h_i} + \frac{A_o t_p}{A_{pm} k_p} + \frac{1 - \phi}{h_o \left(\frac{A_{po}}{A_f} + \phi \right)} + \frac{1}{h_o} + \frac{A_o}{A_{pi} h_{di}} + \frac{1}{h_{do}} + \frac{A_o}{h_c A_{po}}} \tag{5}$$

where ϕ is the fin efficiency.

2.4 Refrigerant-side convection heat transfer coefficients

Correlations for the refrigerant-side heat transfer coefficient were divided into a correlation for single-phase flow and a correlation for two-phase flow.

Single-phase. The Gnielinski correlation [14] was used to compute the single-phase heat transfer coefficients for both superheated vapor and subcooled liquid. This correlation can be expressed as

$$Nu = \frac{(f/2) \cdot (Re - 1000) \cdot Pr}{1 + 12.7 \cdot (f/2)^{1/2} \cdot (Pr^{2/3} - 1)} \tag{6}$$

where $Nu = h_i \cdot D_i/k$ is the Nusselt number and f is the skin friction factor. This correlation is good for a wide range of Reynolds numbers from $2300 \le Re \le 5\ 10^6$.

Since the single-phase enhancement factors used herein are based on the Dittus-Boelter correlation as presented by Khanpara et al. [9,10], the Dittus-Boelter correlation is necessary to compute the heat transfer coefficient for enhanced tubes. The equation is

$$Nu = 0.023 \cdot Re^{4/5} \cdot Pr^n \tag{7}$$

where n is 0.4 for the refrigerant being heated (i.e., evaporator) and 0.3 for the refrigerant being cooled (i.e., condenser).

Condensation. The Traviss et al. [5] correlation was used to predict forced convection condensation. This equation is based on applying the momentum and heat transfer analogy to an annular flow model by using the von Karman universal velocity distribution to describe the liquid film. The radial temperature gradient was neglected and the temperatures in the vapor core and at the liquid-vapor interface were assumed to be equal to the saturation temperature. Axial heat conduction and subcooling of the liquid film were also neglected. Traviss et al. compared the analysis to experimental data for R-12 and R-22, and found that the maximum heat balance error was 7%.

The equation is

$$Nu = \frac{h_t D_t}{k_1} = \left(\frac{Pr_1 \cdot Re_1^{0.9}}{F_2}\right) \cdot F(X_{tt}), \qquad 0.15 < F(X_{tt}) < 15, \tag{8}$$

where F_2 and the Martinelli parameter, X_{tt}, are defined in the original reference. The subscript "l" refers to the liquid phase only.

Evaporation. The Kandlikar correlation [15] was used to predict the two-phase flow boiling heat transfer coefficients. The correlation can be expressed as

$$h_t = D_1 \cdot (Co)^{D_2} \cdot (25 \cdot Fr_1)^{D_5} \cdot h_1 + D_3 \cdot (Bo)^{D_4} \cdot h_1 \cdot F_{fl} \tag{9}$$

where $Co = (1/x - 1)^{0.8}(\rho_v/\rho_l)^{0.5}$ is the convection number, $Bo = q/(G i_{fg})$ is the boiling number, $Fr_l = G^2/(\rho_l^2 g D)$ is the Froude number for an all-liquid flow, and x is quality. All three of the above nondimensional numbers must be greater than one. The parameter F_{fl} is a fluid-dependent multiplication factor. It is 1.5 for R-12 and 2.2 for R-22. The term h_l is the heat transfer coefficient for liquid phase only.

2.5. Refrigerant-side friction factors

Single-phase flow. Shah and Johnson [16] suggested that the Karman-Nikuradse correlation, also referred to as the Prandtl correlation, is the most accurate correlation for isothermal friction factors for turbulent flow in smooth circular tubes. This conclusion was based on the fact that the coefficients were slightly modified to best fit a set of accurate experimental data. The Karman-Nikuradse correlation is implicit in the friction factor, f, and it can be expressed as

$$f = \left(\frac{2}{1.737 \cdot ln(Re\sqrt{f}/2 - 0.4)} \right)^2 \tag{10}$$

Two-phase flow. The two-phase friction multiplier correlations reported by Reddy et al. [17], as given below, are used to compute the pressure drop of the refrigerant flowing through a smooth tube in both condensation and evaporation processes.

$$\Phi_{fo}^2 = 1 + \left(\frac{\rho_l}{\rho_v} - 1 \right) C_1 G^{C_3} \left(\frac{X_{out}^{2+C_2} - X_{into}^{2+C_2}}{2 + C_2} \right) \frac{1}{X_{out} - X_{into}} \tag{11}$$

where Φ_{fo}^2 is the two-phase frictional multiplier based on pressure gradient for the total flow that is assumed to be liquid. The constants C_1 through C_3 are available in the original reference. The two-phase friction multiplier is based on the Blasius correlation for only saturated liquid flowing through a smooth tube.

$$f_{fo} = 0.3164 \cdot Re^{-1/4} \tag{12}$$

Therefore, the frictional pressure gradient of two-phase flow is

$$-\left(\frac{dP}{dx} \right) = -\left(\frac{dP}{dx} \right)_{fo} \Phi_{fo}^2 \tag{13}$$

2.6. Air-side convection heat transfer coefficients

The air-side heat transfer coefficients used in this study are similar to those used by Fischer and Rice [4] in that they are based on the work of McQuiston [18], Yoshii [19], and Senshu and Yoshii [20]. This correlation can be expressed as

$$h_o = C_o G_a C_{pa} Pr_a^{-2/3} \cdot j \cdot \left(\frac{1 - 1280 N_T Re_w^{-1.2}}{1 - 5120 Re_w^{-1.2}} \right) \tag{14}$$

and

$$j = 0.0014 + 0.2618 \cdot \left(\frac{1}{1 - AR} \right)^{-0.15} \left(\frac{G_a D_o}{\mu_a} \right)^{-0.4} \tag{15}$$

where N_T is the number of tubes in air flow direction, $Re_w = G \cdot c/\mu_a$ is the Reynolds number based on the tube spacings, c, in the air flow direction, AR is the ratio of fin surface area to total outside surface area, and $C_o = 1.0, 1.45$, or 1.75 depending respectively on whether the fins are smooth, wavy, or louvered. The above correlation was obtained from test data for smooth fins over the Reynolds number range $3000 \le Re_w \le 15000$. The multiplicative constant C_o is used to correct the heat transfer coefficients, which are based on the smooth fin equation, for wavy and louvered fins. This approach was suggested by Fischer and Rice [4].

2.7. Enhancement factors for micro-fin tubes

Heat transfer coefficients and pressure drops for a cooling or heating process (i.e., condensation or evaporation) in a micro-fin tube can be expressed in terms of enhancement factors, E_h and E_p, as follows:

$$h_{i,a} = E_h(x, m_r) \cdot h_{i,s} \tag{16}$$

$$\Delta P_{i,a} = E_p(x, m_r) \cdot \Delta P_{i,s} \tag{17}$$

where E_h is the ratio of the heat transfer coefficient based on a smooth area of the root diameter for an enhanced tube to a smooth tube with a similar inside diameter. Similarly, E_p is the ratio for pressure drop. The subscript "a" refers to the augmented tube (i.e., micro-fin tube) and "s" refers to the smooth tube. The enhancement factors depend on the geometry of the enhanced tubes, flow rate (m_r), and quality (x), and on whether a condensation or evaporation process is occurring.

For micro-fin tubes, E_h and E_p have been obtained only experimentally to date. A procedure is presented herein for obtaining enhancement factors as a function of mass velocity and quality from experimental data. This procedure was used to calculate enhancement factors from experimental data reported by Khanpara et al. [9,10]. Even though these data were obtained for R-113 as the test fluid, they were used in this study because of the lack of detailed information on micro-fin tube performance with other refrigerants, at least at the time the study reported herein was performed. At three different qualities and mass velocities, Newton's interpolation scheme for three points was used to find the enhancement factors as a function of quality, and then, also by using further interpolation, as a function of mass velocity for each tube. Selecting qualities of 0.25, 0.5, and 0.75 as reference points for every mass velocity, we can express the enhancement factor as a function of quality thus:

$$E(x) = C_o + C_1 \cdot x + C_2 \cdot x^2 \tag{18}$$

where C_0, C_1, and C_2 are constants for either the heat transfer coefficient or pressure drop at three different mass velocities. Mass velocities of 224, 378, and 571 kg/m^{2}s were chosen as reference points. By interpolating the mass velocities, the final equation for enhancement factor can be expressed as

$$E(m_r, x) = (A/53611.5 - B/29741.25 + C/66797.5)(m_r^2 - 601.5m_r + 8448.3) \tag{19}$$

$$+ (B - A)(m_r - 223.5)/154.5 + A$$

where A(x), B(x), and C(x) represent enhancement factors as functions of quality at low, medium, and high mass velocities, respectively. The coefficients, C_0 through C_2, for these equations are available for each mass velocity in reference 11.

The heat transfer enhancement factor for single-phase refrigerant flow is only a function of Reynolds number and can be expressed as

$$E_h(Re) = C_o + C_1 \cdot Re + C_2 \cdot Re^2 \tag{20}$$

Because only a small region of Reynolds numbers was tested by Khanpara et al., the author chose Reynolds numbers 4,000, 7,000, and 10,000 as the reference points and assumed that the enhancement factor did not change for Reynolds numbers over 16,000.

3.0. METHOD OF SOLUTION

Thermodynamic and flow properties of both the refrigerant and air must be specified at the entrance of the heat exchanger. These entrance properties are the initial values to be used in solving for the exit properties of the first element. The exit properties of the first element are then the initial values of the second element and the calculation is repeated. Therefore, the solution scheme is essentially an initial value problem. Using entrance data for an element to compute the exit data for the same element constitutes one calculation cycle. The average of the properties at the beginning and the end of an element is used to represent the average thermodynamic and flow properties of an element.

For each element in each calculation cycle, five nonlinear governing equations must be solved for five unknowns. These five equations are the energy and momentum equations for the refrigerant flow, the energy equation for the air flow, and the two equations of state. The five unknowns to be solved by these equations are the temperature, enthalpy, pressure, specific volume of the refrigerant, and the temperature of air. The Gauss-Seidel iteration scheme was selected to solve the system of equations as the solution is marched from node to node. The solution procedure is a cyclic space-marching procedure from the refrigerant side entrance to the exit of the condenser or evaporator.

Since a correlation for two-phase friction factors in micro-fin tubes cannot be found, the calculation procedure for two-phase flow pressure drop is slightly complicated. First, the pressure drop of two-phase flow in a smooth tube is obtained, and then the result is modified by the pressure drop ratios for the enhanced tube in order to get the actual pressure for an enhanced tube. Once the actual pressure is known, the other properties are easily obtained.

For the inside heat transfer coefficient calculations, the entrance properties of each element are first identified to be either single phase or two phase and then the distribution between a refrigerant heating or cooling process is made. Next the appropriate heat transfer coefficient correlation and enhancement factor equation are used to calculate the inside heat transfer coefficient of a smooth tube or an enhanced tube.

4.0. RESULTS

The condenser and evaporator models developed herein were used to analyze the effects of adding in-tube enhancement (i.e., micro-fin tubes) to a typical condenser and evaporator. Plots of refrigerant temperature and qualities along with air temperatures are presented for heat exchangers with enhanced and smooth tubes. The geometrical and flow details of a typical heat pump evaporator and condenser are listed in Table 1.

4.1. Condenser

The refrigerant enters the condenser in a superheated state and exits in a subcooled state. The constants of interpolation equations for enhancement factors for heat transfer coefficients and pressure drops of enhanced tubes are listed in reference 11. For the example analyzed herein, 28 grids are uniformly distributed along each tube (the tube circuitry shown in Fig. 1 was used). It is important to note that each tube has several passes across the air flow stream. Since the conditions of each horizontally parallel circuit are the same, only the local properties for one parallel circuit are presented.

Figure 3 shows the refrigerant temperatures inside a micro-fin tube and inside a similar sized smooth tube. It is obvious that the refrigerant flow inside the micro-fin tube enters the two-phase region much earlier than that in the smooth tube. Also, its two-phase regime is shorter, meaning that a subcooled condition is reached sooner, than that of the smooth tube. The above behavior is evidence of higher in-tube heat transfer coefficients for the enhanced tube. It should be pointed out that the refrigerant temperature of the two-phase regime is not constant but rather decreases very slowly because of the refrigerant pressure drop.

Figure 4 shows quality along the tube. The flow in the micro-fin tube enters the two-phase region earlier than it does in the smooth tube. In addition, the quality in the micro-fin tube decreases much faster compared to the smooth tube, indicating a higher rate of condensation. Especially important in Fig. 4 is the fact that the refrigerant is completely condensed at the 2.25 m position when enhanced tubes are installed as compared to 3.0 m when smooth tubes are installed. Therefore, the conclusion from Fig. 4 is that the condenser size can be reduced by about 25%, thus eliminating one of the four rows of tubes, when micro-fin tubes are installed in an air-conditioning condenser.

Figures 5 compares the air temperatures of the micro-fin and smooth tubes. The uneven distribution of air temperatures along the tube length is due to the fact that there are four rows of tubes; thus, the 3.0 m length of tubing makes four passes across the air-flow stream. It can be observed that the air temperatures for the micro-fin tube are higher than those for the smooth tube. In fact, the average air temperature leaving the third row of tubes, when in-tube enhancement is installed, is comparable to the average air temperature leaving the four-row smooth tube heat exchanger.

4.2. Evaporator

The refrigerant enters the evaporator in a two-phase state and exits in a superheated state. In this example, 28 grids are uniformly distributed along the tube. The evaporator uses the same tube circuitry used in the above condenser analyses.

Figures 6 and 7 show the refrigerant temperatures and quality respectively inside evaporators with micro-fin tubes and smooth tubes installed. The heat exchanger with micro-fin tubes takes a shorter length of tubing to evaporate the refrigerant flowing through it. For example, the refrigerant is saturated vapor when it reaches the 5.2 m position for the enhanced tube, compared to the refrigerant, which is still a saturated mixture as it exits the heat exchanger with smooth tubes. In fact, as evidenced by air temperatures plotted in Fig. 8, the last 0.7m length of the heat exchanger with micro-fin does not provide any significant cooling of the air. As a result, it would be possible to reduce the length of the heat exchanger with micro-fin tubes by about 12%.

5.0. CONCLUSIONS AND RECOMMENDATIONS

A computational model of an air-conditioning condenser and evaporator has been develped to provide a tool for comparing the effects of installing smooth tubes and enhanced tubes. Parameters that can be predicted throughout the heat exchanger are the local temperature, pressure, heat flux and quality of the refrigerant flow, the local wall temperature, and the local air-flow temperature. Variations of flow properties, heat transfer coefficients, and friction factors with location are considered in the model. By observing local temperatures and qualities of the refrigerant and local air-temperatures, it was determined that the size of a condenser can be reduced by about 25% and the size of an evaporator by 12% through the use of micro-fin tube. These size reductions are approximate, and they are for a specific tube circuiting arrangement. However, in the absence of additional information, they provide useful information regarding possible heat exchanger size reductions when micro-fin tubes are installed.

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Table 1. Geometry and flow characteristics for heat exchangers

	Condenser	Evaporator
Outside diameter of tubes	10 16 mm	same
Inside diameter of smooth tubes (root diameter of enhanced tubes)	8.5344 mm	same
Width of the exchanger	0.725 m	1.5 m
Fin thickness	1.6154×10^{-4} m	same
Fin pitch	551 1811/m	same
Fin surface area/total outside surface area	0.948	same
Horizontal tube spacing	22 225 mm	same
Vertical tube spacing	25.4 mm	same
Inside tube area ratio of enhanced tubes to smooth tubes	1 47	same
No. of tubes along air flow direction	4	same
No. of tubes vertical to air flow direction	12	same
Free air flow area/frontal area	0.5466	same
Inlet refrigerant temperature	357.5 °K	274.8 °K
Inlet refrigerant pressure	2.1×10^6 kPa	-
Inlet refrigerant enthalpy	–	78 kJ/kg
Inlet refrigerant quality	>1.0	between 0–0 1
Refrigerant mass flow rate	0 209 kg/s	0.157 kg/s
Refrigerant	R 12	R–22
Inlet air temperature	294 5 °K	same
Air mass flow rate	0 6747 kg/s	1.85 kg/s
Equivalent convection coefficient of fin-tube contact	14 kW/m²·K	same

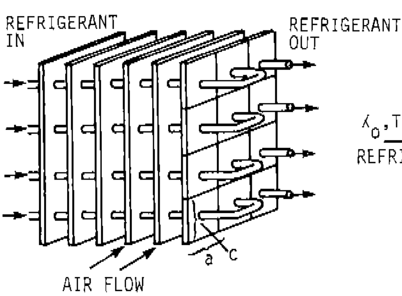

Fig. 1. Schematic diagram of refrigerant circuit.

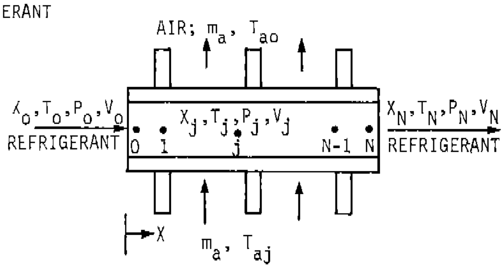

Fig. 2. Increments for a finned-tube heat exchanger.

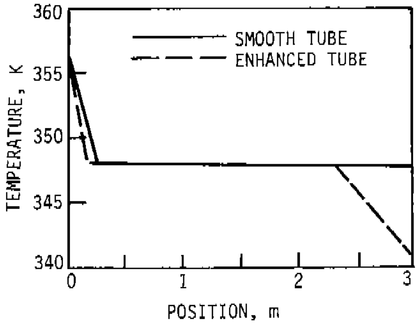

Fig. 3. Condenser refrigerant temperatures for smooth and enhanced tubes.

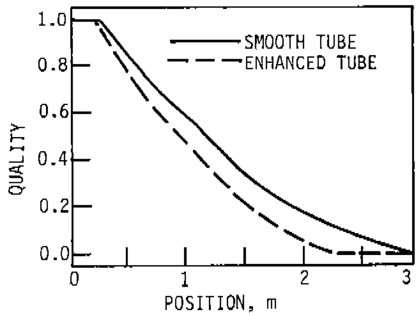

Fig. 4. Condenser-refrigerant quality for smooth and enhanced tubes.

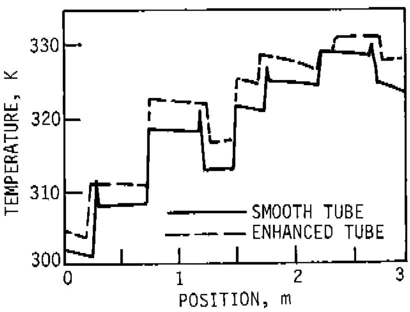

Fig. 5. Condenser-air temperatures for smooth and enhanced tubes.

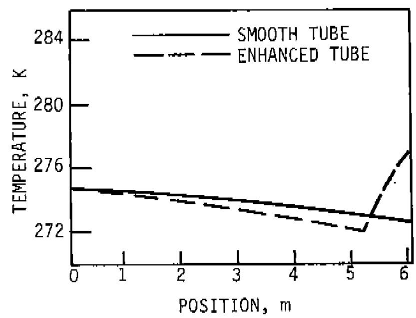

Fig. 6. Evaporator-refrigerant temperatures for smooth and enhanced tubes.

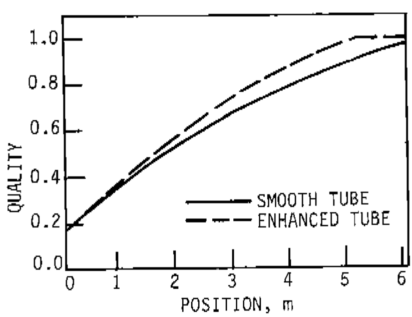

Fig. 7. Evaporator-refrigerant quality for smooth and enhanced tubes.

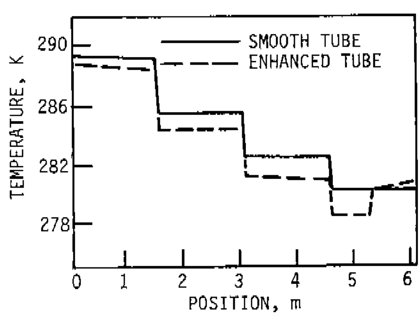

Fig. 8. Evaporator-air temperatures for smooth and enhanced tubes.