DISTRIBUTED NON HOMOGENEOUS MODEL FOR FLOW SIMULATION IN FINNED TUBE COIL EVAPORATORS

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Abstract. *This work presents a numerical model to simulate the unsteady refrigerant fluid and air flow in dry-expansion finned-tube coil evaporators. The model considers the refrigerant flow inside the tubes divided in a region of two-phase flow and a region of superheated refrigerant flow. The refrigerant pressure drop and the moisture condensation on the air flow crossing the outside of the tubes are also taking into account. The refrigerant two-phase flow is taken as one-dimensional and the slip between the liquid and vapor phases is considered. For the refrigerant flow, mass, momentum and energy conservation equations are solved, and for the air flow, energy and mass (humidity) conservation equations are solved. Also, the solution of energy conservation equation for the tube wall is used to determine the wall temperature distribution. Finite Volume Method is used to discretize the governing equations and a Newton-Raphson scheme is utilized for the solution of the resulting system of equations. From the model the refrigerant mass flow rate can be determined, from a known operating conditions and geometry parameters, using the process of parameter estimation with the method of Levenberg-Marquardt minimization. In order to validate the developed model, the obtained results using R-134a as a refrigerant are compared with experimental and numerical data available in the literature.*

Keywords: *Evaporators, finned-tube coils, two-phase flow, refrigeration, non-linear parameter estimation.*

1. Introduction

The scientific community in the refrigeration field has been accomplishing several researches and big investment since last decades aiming the substitution the refrigerants fluids aggressive to the atmospheric ozone layer. Several alternatives have been appearing mostly derived from the families of hydrocarbons halogenated that do not contain chlorine such as R-134a, R-404A, R-402B, R-407C, and R-417A. The fundamental subject of these researches is analyzing the behavior of main components of the refrigeration system when operating with those alternative refrigerants. In this work the main motivation is the analysis of one of those components: the dry-expansion finned-tube coil evaporators commonly used in refrigeration and air conditioning systems.

The refrigerant fluid flow inside evaporator tubes is complex and generally, due to phase change, it is possible to find two regions: a two-phase flow (liquid-vapor) and a superheated vapor flow. Furthermore, during the refrigeration system operation, long transient periods can happen as a consequence, for example, of starting or stopping the system, starting or stopping the compressor and variation of operating conditions of the system. During such periods, the two-phase flow or superheated vapor flow regions can appear or disappear, become more difficult the flow modeling.

Usually, the heat transfer process between the refrigerant and the air in a finned-tube coil evaporator happens in cross flows, as shown in Fig. 1. For improving the heat transfer efficiency or to seek ideal operating conditions, several tube arrangements are used and various methods to choose the best layout are utilized.

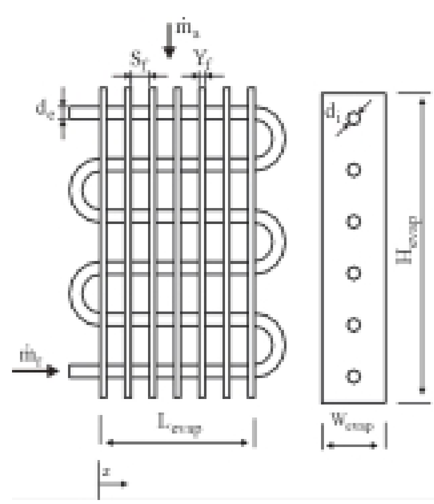

Figure 1. Schematic and geometrical parameters of a finned-tube coil evaporator.

Several studies, dealing with different aspects about the performance of finned-tube coil evaporators can be found in the literature: Wang e Touber (1991), Jia *et al*. (1995, 1999), Liang *et al.* (1999, 2001) and Barbieri (2001). Most researches refer to heat exchangers used in commercial refrigeration and in air-conditioning systems. Due to the large amount of phenomena involved in evaporators operation, different aspects have been analyzed, such as air distribution effect on the evaporator performance; the influence of shape and layout of evaporator tubes; fin type; evaporator dynamical behavior when submitted to different operating conditions; and others.

Barbieri (2001) developed a distributed numerical model to simulate the unsteady behavior of finned-tube coil dry expansion evaporators. The results obtained by Barbieri (2001) indicated that the model predicted satisfactory the unsteady behavior of an evaporator subjected to a step change in the inlet refrigerant mass flow rate. In this work, the model proposed by Barbieri (2001) is extended to consider the slip between the liquid and vapor phases aiming to improve the prediction of two-phase flow by reducing the deviation among predicted and experimental parameters, and to gain deeper understanding about the complex phenomena taking place in such flows. It is also included in the model a numerical procedure that allows the computation of refrigerant mass flow rate, once the evaporator geometrical parameters and operating conditions have known, aiming analysis of the evaporator performance. The minimization method of Levenberg-Marquardt is used to do the estimation of mass flow rate. The developed model is validate through a comparison with numerical and experimental data available in the literature using R-134a as a refrigerant. Comparisons are also made with the results obtained by a model that considered the two phase flow as homogeneous.

2. Mathematical model

The refrigerant fluid flow inside the evaporator tube is divided in two regions: a two-phase flow (liquid-vapor) and another one with superheated vapor flow. Pressure drop inside the tubes, due to friction and acceleration effects of the refrigerant, the moisture condensation on air side, and the slip between the liquid and vapor phases are considered. In order to simplify the problem, the following assumptions are done: a) the evaporator has just one circuit (see Fig. 1), since the refrigerant mass flow rate is taken uniform along each circuit and the heat conduction among those circuits is neglected; b) the refrigerant and air flow are taken to be one-dimensional; c) the change in refrigerant potential energy is neglected; d) the heat transfer coefficient is uniform on the air side (may be different on dry and wet coil regions); e) the heat conduction in the tube wall is neglected; f) the air flow is incompressible; g) there isn't ice formation; h) the refrigerant is oil free.

The governing equations for the flow inside the tube are divided according to the kind of flow: single-phase or two-phase. The equations presented bellow are related to the two-phase flow region. Considering the void fraction $\alpha = 1$, in such equations it is obtained the governing equations to the superheated vapor region.

Based on above assumptions, considering the evaporator geometrical parameters, shown in Fig. 1, and applying the principles of mass conservation, momentum and energy conservation for the refrigerant flow, it is obtained, respectively, the following governing equations:

$$\frac{\partial \tilde{\rho}}{\partial t} + \frac{\partial(\tilde{\rho} u)}{\partial z} = 0 \tag{1}$$

$$\frac{\partial(\tilde{\rho} u)}{\partial t} + \frac{\partial}{\partial z}\left\{\tilde{\rho} u^2 x^2\left[1+\frac{\rho_l(1-\alpha)}{\alpha\rho_v}\right] + \tilde{\rho} u^2 (1-x)^2\left[1+\frac{\alpha\rho_v}{(1-\alpha)\rho_l}\right]\right\} = -\frac{\partial p}{\partial z} - F_z \tag{2}$$

$$\frac{\partial(\tilde{\rho} h_r)}{\partial t} + \frac{\partial(\tilde{\rho} u h_r)}{\partial z} = \frac{\partial p}{\partial t} + \frac{A_i'}{A_{tub}} dq_{wr}'' \tag{3}$$

where $\tilde{\rho} = [\alpha\rho_v + (1-\alpha)\rho_l]$ is the density [kg/m^3], $\alpha = \{1 + [S\rho_v(1-x)/\rho_l x]\}^{-1}$ is the void fraction, x is the vapor quality, $S = (u_v / u_l)$ is the slip ratio between the phases, $u = G_r / \tilde{\rho}$ is the mean velocity [m/s], G_r is the refrigerant mass flux [kg/m^{2}s], t is the time [s], z is the coordinate along the evaporator tube [m] (see Fig. 1), and p is the flow pressure inside the tube [Pa], calculated by using the state equation $p = p(\tilde{\rho}, h_r)$ presented by Barbieri (2001). The subscripts l and v indicate the liquid and vapor phases, respectively. In equation (2) $F_z = (dp_F / dz)$ is the friction pressure loss [N/m^3].

In Equation (3), $h_r = [x h_v + (1-x) h_l]$ is the refrigerant enthalpy [J/kg], h_l e h_v are the liquid and vapor enthalpy [J/kg], respectively, A_i' is the tube inner wall area by unity length [m], A_{tub} is the inside cross-section area of tube [m^2], $dq_{wr}'' = [H_r(T_w - T_r)]$ is the heat flux from the tube wall to the refrigerant [W/m^2], H_r is the heat transfer coefficient inside the tube [W/m^{2}K], T_w is the tube wall temperature [°C] and T_r is the refrigerant temperature [°C]. In superheated vapor flow region the Eqs. (1) to (3) have the same form with parameters α and x equals to the unit.

To obtain the governing equations for the air flow, besides the previous assumptions, the thermal inertia of the air is neglected. The finned-tube coils are largely used in applications for atmospheric air cooling, being common the occurrence of the moisture condensation on the air side. Because of dehumidification, a liquid water film that eventually could freeze covers the coil surface on the air side. Thus, the cooling and dehumidifying process involves both sensible and latent heat transfer.

Applying the principles of mass conservation (humidity) and energy conservation for the air, it is obtained the following equations, respectively,

$$\dot{m}_a \frac{d\omega_a}{dz} dz = H_m A_t' dz\,(\omega_a - \omega_{a,sat}) \tag{4}$$

$$\dot{m}_a \frac{dh_a}{dz} dz = H_a A_t' dz (T_a - T_w) + H_m A_t' dz\,\lambda_{water} (\omega_a - \omega_{a,sat}) + H_m A_t' dz (\omega_a - \omega_{a,sat}) h_{l,water} \tag{5}$$

where $\dot{m}_a = [\rho_a u_a (W_{evap} L_{evap})]$ is the air mass flow rate [kg/s], ρ_a is the air density [kg/m^3], u_a is the air velocity [m/s], L_{evap} is the evaporator straight length [m], W_{evap} is the evaporator width [m], $A_t' = (A_r' + A_f' \eta_f)$ is the heat transfer total area by unity length [m], A_r' is the outer area not covered by fins by unity length [m], A_f' is the fins surface area by unity length [m], η_f is the fin efficiency presented by McQuiston and Parker (1994), H_a is the air heat transfer coefficient [W/m^{2}K], T_a is the air temperature [°C], ω_a is the air absolute humidity, $\omega_{a,sat}$ is the saturated air humidity at temperature T_w, λ_{water} is the latent heat of water condensation at temperature T_w [J/kg], $h_{l,water}$ is the liquid water enthalpy at temperature T_w [J/kg] and H_m is the mass transfer coefficient [kg/m^{2}s], calculated by the Lewis correlation, $[H_m = H_a / Le\, c_{p,a}]$, where Le is the Lewis number, $Le = [k_a / (\rho_a c_{p,a} D_{ab})]$, D_{ab} is the water-air mass diffusivity [m^2/s], k_a is the air thermal conductivity [W/mK] and $c_{p,a}$ is the specific heat at constant pressure for the air [J/kgK].

Appling the principle of energy conservation at tube wall, it is obtained,

$$M'_{wf}\, c_{wf}\, \frac{dT_w}{dt} = H_a A_t' dz (T_a - T_w) + H_m A_t' dz\, \lambda_{water} (\omega_a - \omega_{a,sat}) - H_r A_i' dz (T_w - T_r) \tag{6}$$

where M'_{wf} is the mass of the fins and the tube wall by unity length and $c_{p,wf}$ is the mean specific heat at constant pressure, considering the tube and fins materials (see Barbieri, 2001).

In summary, the Eqs. (1) to (6) should be solved to give the distributions of $\tilde{\rho}$, u, h_r, T_a, T_a and ω_a. To perform such task, closure relationships are thus necessary to compute the slip ratio, the refrigerant, air and water thermo-physical properties, the friction coefficient, the heat transfer coefficients to the refrigerant and to the air and the mass transfer coefficient to the air.

Air and water thermo-physical properties are obtained, respectively, from data presented by ASHRAE (1993) and Incropera and DeWitt (2002). The properties of the refrigerant fluid and tube wall material are computed, respectively, by using the data given by McLinden *et al.* (1998) and Incropera and DeWitt (2002). Moreover, the model uses the following correlations to compute other parameters:

1. friction factor in the superheated vapor region: Churchill (1977);
2. pressure drop due to shear in the two-phase flow region: Paliwoda (1989);
3. heat transfer coefficient in the single-phase flow region: Dittus-Boelter (1930);
4. heat transfer coefficient in the two-phase flow region: Jung and Radermacher (1991) and Chen (1966);
5. heat transfer coefficient on the air side: McQuiston (1981) and Turaga *et al.* (1988).
6. slip ratio: Zivi (1964).

3. Initial conditions and solution methodology

For a given coil the model could be used to: a) determine evaporator performance parameters, such as refrigeration load; outlet refrigerant and air temperatures; among others, since the evaporator operating conditions and dimensions are known. In this case a direct problem is solved, departing from a set of inlet conditions for the refrigerant and also for the air; b) determine the refrigerant mass flow rate along evaporator tubes, once known its dimensions and other operating conditions. In this case, an inverse problem must be solved, since the governing equations are mass flow rate dependent. The solution procedure chosen in each case is shown following.

3.1. Direct Problem

In this case, the solution for the system of Eqs. (1) to (6) is obtained from known refrigerant conditions $\dot{m}_r$, x and T_r, at the tube inlet and from known air conditions T_a, ω_a and $\dot{m}_a$ at evaporator inlet. The numerical solution for this system of equations is obtained by using the finite volume method with staggered grid for a relative placement of variables in the computational mesh. Since the problem is strongly convective, an upwind scheme is used to discretize the governing equations.

To obtain the discretized equations by using the finite volume method, the governing equations are integrated over time and space, along the control volume. Unsteady terms in governing equations are discretized using the approximation formula: $\partial\phi / \partial t = [(\phi - \phi^o) / \Delta t]$, in which superscript "o" means the time step immediately before. To do the integrals over time, it was used a completely implicit scheme, aiming to reach numerical stability for the algorithm.

To improve the efficiency of the solution process, it is utilized an iterative method in two levels. First, it is computed refrigerant and tube wall variables. Second, variables for the air are calculated. The Newton-Raphson method cell by cell is used to solve the system of equations.

In the air side, the correction of temperature and humidity values is done for the whole mesh and not cell by cell. Convergence is achieved when the summation of all corrections of the air conditions, T_a and ω_a, will be less than 10^{-4}. It is considered that steady state regime is reached when a change of $\tilde{\rho}$, u, h_r and T_w, from a time step to another is less than 10^{-3}. Once the steady state regime is achieved, the refrigerant mass flow rate is increased or decreased to simulate the evaporator unsteady behavior. As the evaporator was in steady state regime, this perturbation will promote an unsteady state period followed by a steady state. Such change in refrigerant mass flow rate simulates varying operation system conditions.

3.2. Inverse Problem

The inverse problem corresponds to that one which is desired to calculate the refrigerant mass rate along the evaporator once known its geometry and operating conditions.

Since the problem is non-linear due to the governing equations be mass dependent, an additional calculation procedure is necessary, beyond that presented in the item 3.1. Therefore, the initial value of $\dot{m}_r$ is guessed and the refrigerant temperature at the coil exit, $T_{s,c}$, is calculated and compared to the respective measured (or computed using another model) value, $T_{s,m}$. After, the value of $\dot{m}_r$ is corrected by using the non-linear parameter estimation method of Levenberg-Marquardt. The process is repeated until some convergence criteria are matched.

4. Results and discussion

In this work, it is presented some comparison among results obtained by the model for the steady and unsteady state regime computed using the present model and results obtained by Liang *et al.* (1999) and Jia *et al.* (1995), respectively. The results are presented for the direct problem as well as for the inverse problem. Geometrical parameters for the evaporator analyzed by Liang *et al.* (1999) are presented in Table 1.

Table 1. Evaporator geometrical parameters simulated by Liang *et al.* (1999).

Geometrical parameters			
Straight tube length (m)	1,0	Tube inner diameter (mm)	8,83
Transversal tube spacing (m)	0,025	Fins thickness (mm)	0,12
Longitudinal tube spacing (m)	0,0216	Fins spacing (mm)	2,41
Tube outer diameter (mm)	9,53	Quantity of fins	394

In Figures 2 and 3, it is presented, respectively, for relative humidity of air at evaporator inlet, R.H., of 60% and 90%, some comparison of the air temperature profile, the refrigerant temperature and the tube wall temperature calculated using the present model with the results computed by Liang *et al.* (1999). The relative length refers to the ratio of refrigerant flowing length measured from the inlet to the total path. These figures also show predictions obtained by homogeneous model (S=1). For these cases, the direct problem is solved considering known refrigerant mass flow rate of 0,0055 kg/s and 0,0086 kg/s, respectively, for relative humidity of air at evaporator inlet of 60% and 90%. The other operating conditions used by Liang *et al.*(1999), for such cases, are presented in Table 2. The enthalpy at evaporator inlet is determined through operating conditions at condenser exit (condensation temperature = 45°C, sub-cooling degree = 5°C).

Table 2. Operating conditions (Liang *et al*., 1999).

Operation conditions at inlet	Case 01	Case 02
Refrigerant temperature (°C)	10,5	11,0
Refrig. mass flow rate 10^3 (kg/s)	5,5	8,6
Vapor quality	0,22	0,22
Air temperature (°C)	28,0	28,0
Relative humidity (%)	60	90
Air velocity (m/s)	2,0	2,0

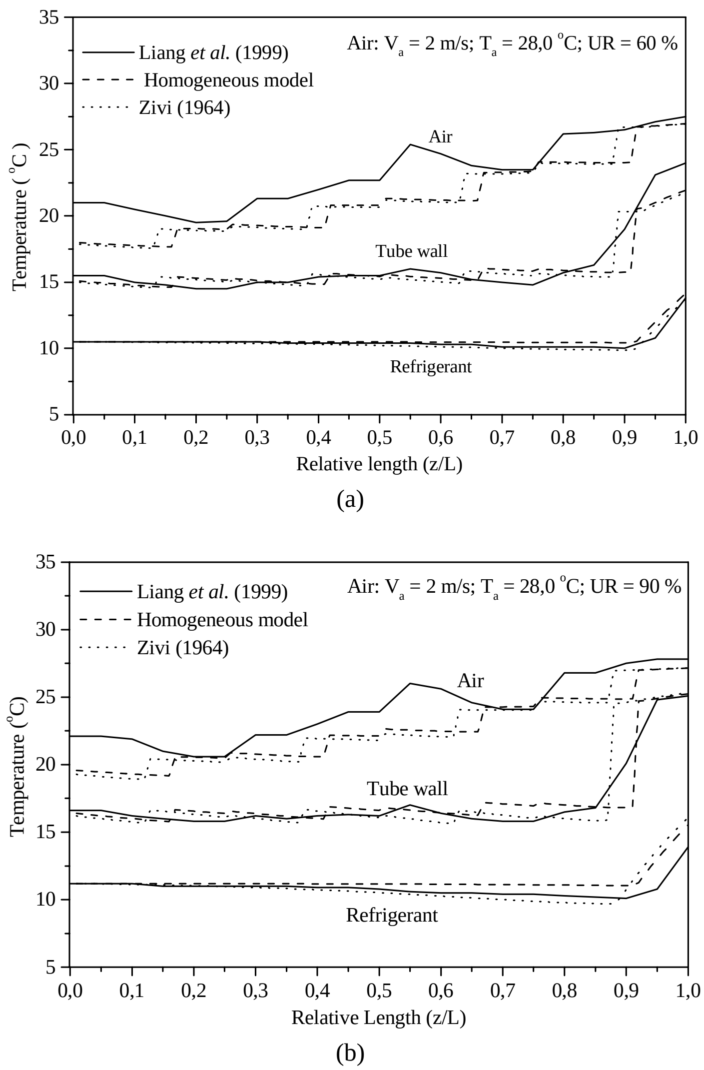

Figure 2. Comparison between temperature profiles for air, tube wall and refrigerant for: (a) R.H. = 60% and (b) 90%.

It is verified on Figs. 2(a) and 2(b) a good agreement among results presented by Liang *et al.* (1999) and the results calculated using the present model, mainly for refrigerant temperature and tube wall temperature profiles. However, for the air relative humidity at inlet of 30%, the present model does not predict complete refrigerant vaporization. In this case, the refrigerant vapor quality in the coil exit is 0.81 differing from Liang *et al.* (1999) results. In spite of this, the temperature profiles for the air, the refrigerant and the tube wall are well approximated, with a major difference just in the region of superheat steam. Notice in Figs 2(a) and 2(b) that temperatures profiles predicted by the present model and homogeneous model are almost coincident along the entire refrigerant circuit.

In Figure 3(a) it is shown the refrigerant mass flow rate change as a function of the air relative humidity at evaporator inlet. In this case, the inverse problem is solved; thus, the mass flow rate is computed, known the evaporator geometrical parameters and the operating conditions. The geometric parameters for the tube and fins are the same presented in Table 1, with the operating conditions corresponding to the case 1 presented in the Table 2. Computed values for the mass flow rate are compared to those ones obtained by Liang *et al.* (1999) for a relative humidity range from 20% up to 90%.

As it can be observed in Fig. 3(a), the major deviations among results happen for relative humidity bellow 50%. For relative humidity above 50% there is a good agreement among those results, and considering whole air relative humidity range, from 20% up to 90 %, the mean absolute deviation from Liang *et al.* (1999) results is 12,1 %. Yet, it is also observable a significant increase in refrigerant mass flow rate for relative humidity above 40%. That behavior, as shown by Liang *et al.* (1999) and Barbieri (2001), is due to the increase of the cooling load needed, that approximately double its value for humidities in the range from 50 up to 90%. The increase on cooling load is mainly due to the quick increase in the portion of latent heat as consequence of the increase of air relative humidity.

In Figure 3(b) it is presented the refrigerant mass flow rate change as a function of evaporation temperature and the air temperature at evaporator inlet. Once more, the tube and fins geometrical parameters are those available on Table 1. The operating conditions are refrigerant superheating degree of 5 ^{o}C, air velocity at inlet of 2 m/s, air relative humidity of 60% and evaporation temperature of 24 ^{o}C and 28 ^{o}C. For such cases, the mean absolute deviation between the mass flow rate values calculated by using the present model and those ones obtained by Liang *et al.* (1999) is 5 %.

Keeping the air inlet temperature constant, one can observe on Fig. 3(b) that the refrigerant mass flow rate, necessary to keep the same refrigerant superheating degree, decreases with the increase in the evaporation temperature. In the same way, for the same evaporation temperature, the refrigerant mass flow rate decreases with the decreasing in air inlet temperature.

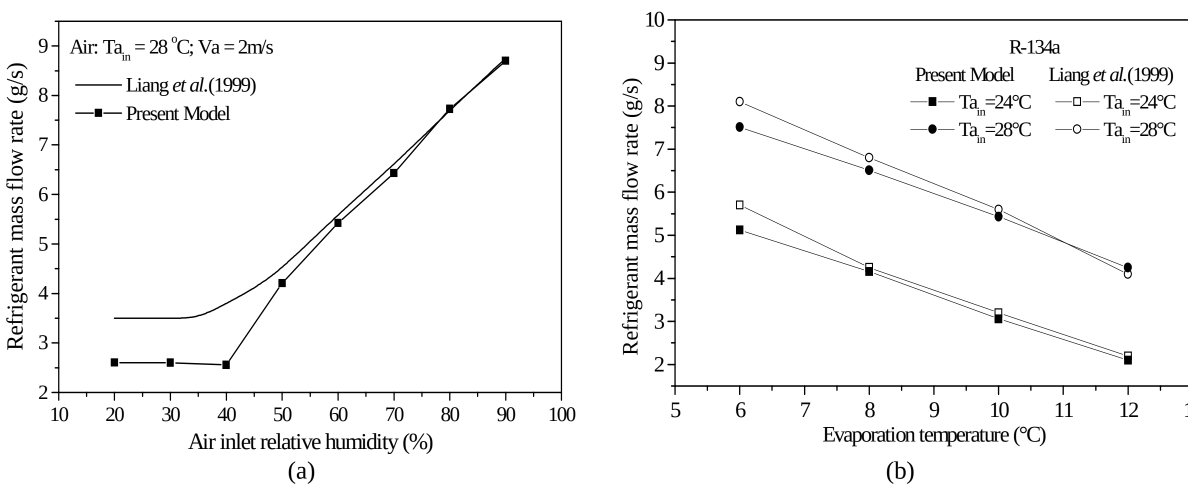

Figure 3. Refrigerant mass flow rate as function of: (a) air relative humidity at evaporator inlet; (b) evaporation temperature and air temperature at evaporator inlet.

Some results were also obtained to demonstrate the transient response of the evaporator to a step change of the refrigerant mass flow rate. These results are compared with experimental data of Jia *et al.* (1995) that measured the degree of superheated of the refrigerant along the evaporator and the outlet air temperature at the left side of the coil. To obtain the transient response the steady state condition of the system was established first, and then a change in the refrigerant flow rate was initiated by opening or closing a step of an electronic expansion valve. Geometrical parameters and operating conditions for the evaporator analyzed by Jia *et al.* (1995) are presented in Tabs. 3 and 4, respectively.

Table 3. Evaporator geometrical parameters analyzed by Jia *et al.* (1995).

Geometrical parameters			
Straight tube length (m)	1,632	Tube inner diameter (mm)	11,84
Transversal tube spacing (m)	0,0275	Fins thickness (mm)	0,12
Longitudinal tube spacing (m)	0,03178	Quantity of fins	514
Tube outer diameter (mm)	12,70		

Figures 4(a) and 4(b) show, respectively, comparisons of the degree of superheated and outlet air temperature at the left side of the coil between the experimental data by Jia *et al* (1995) and the predictions of the present model using different correlations to compute the friction pressure loss, F_z. The simulations were performed dividing the evaporator in 180 cells and using a time step of 10 s. The friction pressure loss correlations analyzed are: Bandarra Filho (2002), Jung e Radermacher (1991) and Paliwoda (1989).

Table 4. Operating conditions (Jia *et al.*,1995).

Operation conditions at inlet			
Refrigerant temperature (°C)	-2,0	Vapor quality	0,359
Refrigerant pressure (kPa)	273	Air temperature (°C)	14,5
Refrig. mass flow rate 10^3 (kg/s)	8,11	Air velocity (m/s)	2,0

As it can be seen in Fig. 4(a), the degree of superheated distributions predicted by the model presents the same behavior that was obtained by Jia *et al.* (1995), although the calculated values are on average 8,8% higher than the experimental data. Therefore, the superheated vapor flow region is overestimated by the present model when compared with experimental data. Thus, the "dry-out" point, at which the liquid vaporizes completely, predicted by the model is closer to the evaporator inlet than one measured, resulting in a larger degree of superheated. Also can be seen in Fig. 4(a), the degree of superheated distributions calculated are nearly coincident, regardless of the correlations used to compute the friction pressure loss.

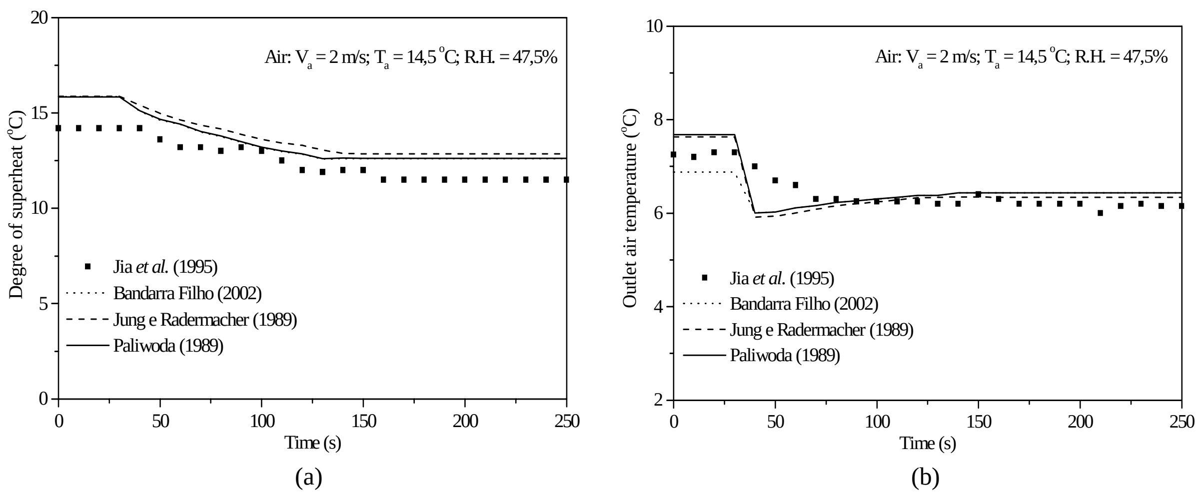

Figure 4. Comparison between: (a) degree of superheat and (b) outlet air temperature measured (Jia *et al.*,1995) and predicted: influence of the friction pressure loss correlations.

In Figure 4(b) it is observed the reasonably good agreement between the outlet air temperature distributions calculated and measured by Jia *et al.* (1995) after 75 s, although the predicted temperature exhibits a shorter transient period. As it was pointed out by Jia *et al.* (1995), in the simulation the air temperature at the evaporator inlet was specified as a constant during the transient period, but in practice this temperature decreases slightly when a step increases in refrigerant mass flow rate was applied, due to constructive aspects of the experimental setup. It was also shown in Figure 4(b), the outlet air temperature distributions calculated using different correlations to compute F_z are nearly coincident after 35 s.

5. Conclusion

In this work, it is presented a numerical model to simulate the unsteady refrigerant fluid flow and air flow in dry-expansion finned-tube coil evaporators, widely used in air conditioning and refrigeration systems. The model takes into account the refrigerant R-134a flow inside just one tube and the crossed air flow outside that tube. The refrigerant pressure drop, the moisture condensation on the air side, and the effects of the ambient external and internal conditions are also considered. The transient response of the evaporator under a step change in the inlet refrigerant flow rate is investigated. The transient behavior happens to be one the main concerns of the manufacturers of refrigeration systems with respect to the technological applications, because under real conditions the steady state is rarely achieved. In order to simplify the model, the flow inside the tube is divided in two distinct regions: one, two-phase evaporating flow and another, single-phase superheated vapor flow. Additionally, the flow in two-phase region is modeled as one-dimensional and the slip between the liquid and vapor phases is considered.

The results obtained are in satisfactory agreement with prediction and experimental results from the literature. Comparisons were carried out in terms of the temperature distributions of air, refrigerant and tube wall, refrigerant mass flow rate, the degree of superheated of the refrigerant along the evaporator and the outlet air temperature at the left side of the coil. The transient response of the evaporator is reasonably well predicted by the model, although it overestimated the degree of superheated and the outlet air temperature exhibits a shorter transient period than experimental data.

The minimization method of Levenberg-Marquardt used to compute the mass flow rate for refrigerant fluids along the evaporator presented high efficacy. Due to its robustness the Levenberg-Marquardt method does not present divergence or oscillation problems during iteration process. Hence, this method overcomes the Newton-Raphson that is conditionally convergent, or it is frequently unstable depending strongly on the initial guess.

Such instability and non-convergence problems were found in some tested cases, since the Newton-Raphson method is still used in part of the computational code to solve the non-linear governing equations, both for the refrigerant as well as the air flows. Some of such cases presented convergence difficulties and other ones presented oscillatory behavior around some points, resulting in the non-convergence of the iteration process. This fact implies the necessity to implement in the computational code more efficient methods to solve the non-linear system of governing equations.

6. Acknowledgment

Authors acknowledge ***FAPESP–Fundação de Amparo à Pesquisa do Estado de São Paulo*** by its financial support that is allowing the development of this project.

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