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NUMERICAL SIMULATION OF THE REFRIGERANT TRANSIENT FLOW IN NON-ADIABATIC CAPILLARY TUBES

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Abstract. *This work presents a numerical model to simulate the unsteady refrigerant flow through capillary tube-suction line heat exchangers. Capillary tubes are commonly used as expansion devices in small refrigeration and air conditioning systems. Capillary tube and suction line are considered straight and horizontal with constant inner diameter, and it is assumed one-dimensional flow. The flow through capillary tube is divided in two distinct regions: a region of subcooled liquid flow and a region of saturated two-phase flow. The homogeneous model is employed for the two-phase flow region. The fundamental equations governing the flow through capillary tube-suction line heat exchanger are derived from the mass conservation, momentum and energy conservation laws. Due to compressibility of the flow in the two-phase region, critical or choked flow condition is generally found in capillary tubes. The model allows prediction, in steady and unsteady states, refrigerant mass flow rates, pressure, quality, refrigerant and wall temperatures distributions along the tubes, as a function of the heat exchanger geometry and operating conditions. Experimental data from the literature for steady flow are compared and discussed with numerical results. The discrepancies between measured and calculated mass flow rate have been found to be about 8.6 %, for concentric heat exchangers, and 5.7 % for lateral heat exchangers. Additionally comparisons between the transient and quasi-steady models are presented and the values for the numerically evaluated mass flow rates have differed by approximately 2 %..*

Keywords *.non-adiabatic capillary tube, transient flow, refrigeration system, heat exchanger, two-phase flow*

1. Introduction

Large systems based on vapour compression refrigerant cycle frequently use a thermostatic valve. Otherwise small hermetically sealed systems, like household refrigerators and room air-conditioners, present a long length and small diameter tube, called capillary tube, as an expansion device. The main advantages of this fixed-geometry expansion device compared to thermostatic valve are: lower cost, a pressure equalization feature to reduce starting torque, and absence of moving parts. The typical length of these tubes lies between 1.0 and 6.0 m and their diameters range from 0.5 to 2.0 mm.

The capillary tube is a simple device, though refrigerant flow through it is quite complex. Regarding to the refrigerant phase change, two phases can be identified along the flow: a single phase-liquid flow region and a two-phase liquid-vapour flow region. The flow is compressible with Reynolds number varying from $4x10^3$ to $20x10^3$, and the choked flow condition can be reached under some circumstances. Furthermore, during the refrigeration system operation, long unsteady 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. Such unsteadiness introduces great complexity in modeling the flow.

The refrigeration capacity of a refrigeration system can be increased lowering the quality at the evaporator inlet by cooling the refrigerant in the capillary tube using the superheated vapour in the compressor suction line. A counter flow heat exchanger, made by soldering the capillary tube on the outside of the suction line (see Fig. 1a) or placing the capillary tube inside the suction line (see Fig. 1b), is used to perform this task. Such device is called capillary tube-suction line heat exchanger or non-adiabatic capillary tube and is widely used in refrigerators.

During the operation of the refrigeration system, the non-adiabatic capillary tube subcool the liquid exiting the condenser, decreasing the amount of vapour at the evaporator inlet. Simultaneously, it superheats the vapour before reaching the compressor, which prevents the entrance of condensate into the compressor and avoids condensation and ice formation outside the suction line.

The refrigerant can enter in the capillary tube as subcooled liquid or as a mixture of saturated liquid and vapour. In the steady state condition, the most common is the subcooled liquid state at the inlet of the tube and a two-phase fluid at the outlet of the tube. Thus, the flow through a capillary tube can be divided in two regions. A liquid region at the entrance of the tube where the pressure decreases linearly until the flash point is reached, and a two-phase region, which presents increasing pressure drop per unit length and refrigerant velocity as the end of the capillary tube is approached.

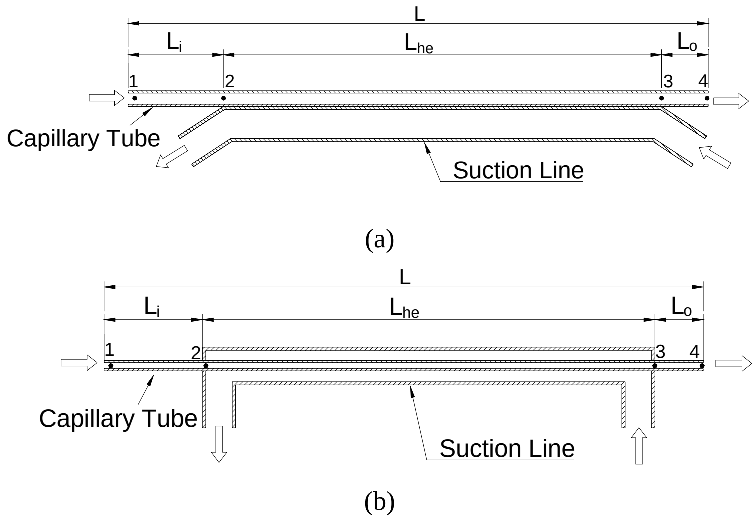


Figure 1. Layout of capillary tube-suction line heat exchangers: (a) lateral and (b) concentric.

Theoretical and experimental investigations of the flow in capillary tube-suction line heat exchanger have been the subject of extensive research, such as reported by: Dirik *et al.* (1994), Mendonça (1996), Zangari (1998), Barbazelli (2000), Melo *et al.* (2002), and Bansal and Xu (2003). These analyses were particularly devoted to the characterization of the refrigerant flow and to determine important geometric parameters of the capillary tube, such as the adiabatic length and the heat exchanger length.

The majority of the models available in the literature consider the steady state operating condition and although the unsteady flow condition has just been account by some researchers, many important questions for a better understanding of the transient flow through non-adiabatic capillary tube needs to be investigate. The transient distributions of the important flow parameters as mass flow rate, pressure, temperature and quality, either experimental as theoretical, through non-adiabatic capillary tubes are almost always inexistent.

The present work intends to contribute this subject and presents a numerical model to simulate the unsteady refrigerant flow through capillary tube-suction line heat exchangers. The fundamental equations governing the flow are derived from the mass conservation, momentum and energy conservation laws and the homogeneous model is employed for the two-phase flow region. Closure of the governing equations is performed with the friction factor correlations and constitutive equations for the convective heat transfer coefficients for the single-phase and two-phase regions. The system of governing equations is solved using a modified finite volume method. The solution of the resulting system of discretized equations is obtained marching along the tube until either choked flow or the established evaporation pressure is reached, which occurs first.

The model allows predicting, in steady and unsteady states, refrigerant mass flow rates, pressure, quality, refrigerant and wall temperatures distributions along the tubes, as a function of the heat exchanger geometry and operating conditions. For validation, results from the model are compared with the experimental data from the literature for steady flow and additionally comparisons between the transient and quasi-steady models which are presented and discussed.

2. Problem Formulation

In the present model, the flow through capillary tube is divided in two distinct regions: a region of subcooled liquid flow and a region of saturated two-phase flow. The capillary tube-suction line heat exchanger is considered insulated from the surrounding air and its length is divided in three parts, as shown in Fig. 1: (i) inlet region (L_i), where the capillary tube does not exchange heat with the suction line; (ii) heat exchanger region (L_{he}), which constitutes the intermediate part of the capillary tube, where heat is exchanged with the suction line; (iii) outlet region (L_o), as in the inlet region, there is not heat exchange with the suction line.

The flow is considered incompressible in the liquid region. The capillary tube and the suction line are taken to be straights, horizontals with inner diameters constants. The gravitational effect is disregard. It is assumed one-dimensional flow of pure refrigerant (oil free). Mechanical equilibrium is also assumed, i.e., the refrigerant pressure is uniform at any given cross-section of the tube and surface tension effects are not considered. The pressure drop along the length of the suction line is neglected. Metastability effects are not considered and the vapour is always saturated with respect to the local pressure.

The two-phase flow along the capillary tube is considered homogeneous, i.e., the flow is mathematically treated as a pseudo single-phase flow which properties are obtained considering the quality and the properties of each individual phase. Consequently, both phases have the same velocity, pressure and temperature at any given cross-section thought the tube. The material properties of the capillary and suction line walls are considered constants and the radiation heat transfer between external surface of the capillary tube and internal surface of the suction line is disregard. Taking the above assumptions into account, the governing equations will be derived next.

a) Mass conservation for the flow in the capillary tube and suction line:

$$\frac{\partial \rho}{\partial t} + \frac{\partial G}{\partial z} = 0 \quad (1)$$

where t is the time [s], z is the distance along the tube [m], ρ is the density [kg/m^3], $G = (\rho u)$ is the refrigerant mass flux [kg/m^{2}s] and u is the refrigerant velocity [m/s]. In the two-phase flow: $\rho = [\rho_l + \alpha(\rho_v - \rho_l)]$ and α is the void fraction (ratio of the portion of the cross-sectional area occupied by the vapour and the total cross-sectional area). The subscripts l and v indicate the liquid and vapor phases, respectively.

b) Momentum equation for the flow in the capillary tube:

$$\frac{\partial(\rho u)}{\partial t} + \frac{\partial(Gu)}{\partial z} = -\frac{\partial p}{\partial z} - \frac{fG^2}{2\rho d_{i,c}} \quad (2)$$

where p is the absolute pressure [Pa], f is the friction factor and $d_{i,c}$ is the inner diameter of the capillary tube [m].

c) Energy equation for the flow in the capillary tube:

$$\frac{\partial(\rho h_o)}{\partial t} + \frac{\partial(G h_o)}{\partial z} = \frac{\partial p}{\partial t} - H_c \frac{P_{i,c}}{A_{i,c}}(T_{rc} - T_{wc}) \quad (3)$$

where $h_o = (h + u^2/2)$ [J/kg] is the specific stagnation enthalpy [J/kg], h is the specific enthalpy [J/kg], H_c is the convective heat transfer coefficient inside the capillary tube [W/m^{2}K], $P_{i,c} = (\pi d_{i,c})$ is the inner perimeter of the capillary tube, $A_{i,c} = (\pi d_{i,c}^2/4)$ is the cross-sectional area of the capillary tube [m^2], T_{rc} is the refrigerant temperature along the capillary tube [K] and T_{wc} is the wall temperature of the capillary tube [K]. In the two-phase flow region: $h = [h_l + x(h_v - h_l)]$ and x is the quality.

d) Energy equation for the capillary tube wall:

$$(\rho c)_{wc} \frac{dT_{wc}}{dt} = H_c \frac{P_{i,c}}{A_{c,c}}(T_{rc} - T_{wc}) - H_s \frac{P_{e,c}}{A_{c,c}}(T_{wc} - T_{rs}) \quad (4)$$

where ρ_{wc} [kg/m^3] and c_{wc} [J/kgK] are the density and specific heat capacity of the wall material of capillary tube, respectively, $A_{c,c}$ is the cross-sectional area of the capillary tube wall [m^2], H_s is the convective heat transfer coefficient inside the suction line [W/m^{2}K], $P_{e,c} = (\pi d_{e,c})$ is the outer perimeter of the capillary tube [m], $d_{e,c}$ is the outer diameter of the capillary tube [m] and T_{rs} is the refrigerant temperature along the suction line [K]. One can observe in equation (4) that the right last term is canceled in the inlet and outlet regions.

e) Energy equations for the flow in the suction line and suction line wall:

$$\frac{\partial(\rho h_0)}{\partial t} + \frac{\partial(G h_0)}{\partial z} = \frac{\partial p}{\partial t} + H_s \frac{P_{e,c}}{A_{i,s}}(T_{wc} - T_{rs}) + H_s \frac{P_{i,s}}{A_{i,s}}(T_{ws} - T_{rs}) \quad (5)$$

$$(\rho c)_{ws} \frac{dT_{ws}}{dt} = -H_s \frac{P_{i,s}}{A_{c,s}}(T_{ws} - T_{rs}) \quad (6)$$

where $A_{i,s}$ is the cross-sectional area of the suction line [m^2], $P_{i,s} = (\pi d_{i,s})$ is the inner perimeter of the suction line [m], $d_{i,s}$ is the inner diameter of the suction line [m], T_{ws} is the wall temperature of the suction line [K]. Notice that: $A_{i,s} = (\pi d_{i,s}^2/4)$ for lateral heat exchanger and $A_{i,s} = [\pi(d_{i,s}^2 - d_{e,c}^2)/4]$ for concentric heat exchanger. In equation

(6) ρ_{ws} [kg/m^3] and c_{ws} [J/kgK] are the density and specific heat capacity of the wall material of suction line, respectively, and $A_{c,s}$ is the cross-sectional area of the suction line wall [m^2], $P_{e,s} = (\pi d_{e,s})$ is the outer perimeter of the suction line [m] and $d_{e,s}$ is the outer diameter of the suction line [m].

In summary, the proposed model consists of: Eqs. (1) and (2), which should be solved to give the distributions of *G* and *p*, respectively; Eq. (3) used for calculating the refrigerant stagnation enthalpy through capillary tube; Eq. (4) used for calculating the wall temperature of the capillary tube; Eq. (5) used for calculating the refrigerant stagnation enthalpy through suction line and the Eq. (6) for calculating the wall temperature of the suction line.

In addition the model uses the following closure relationships: (i) the friction factor in the liquid region: Churchill (1977); (ii) the friction factor in the two-phase region: Erth (1970); (iii) the heat transfer coefficient in the liquid region and suction line: Gnielinski (1976); (iv) the heat transfer coefficient in two-phase region: Pate (1982). The refrigerant thermo-physical properties were computed through linear regression of the data provided by McLinden *et al*. (1998).

3. Initial Conditions

In Figure 2, the points 1 to 4 are positions along the capillary tube (see Fig. 1) and together with points 0 and b they represent schematically a common situation of the flow through a non-adiabatic capillary tube. The region located between points 1 and 2, and between points 3 and 4, corresponds to the capillary tube inlet and outlet respectively. The reduction of the refrigerant enthalpy that happens in these regions is very small when compared with the one which happens between points 2 and 3, which is the region of the heat exchanger.

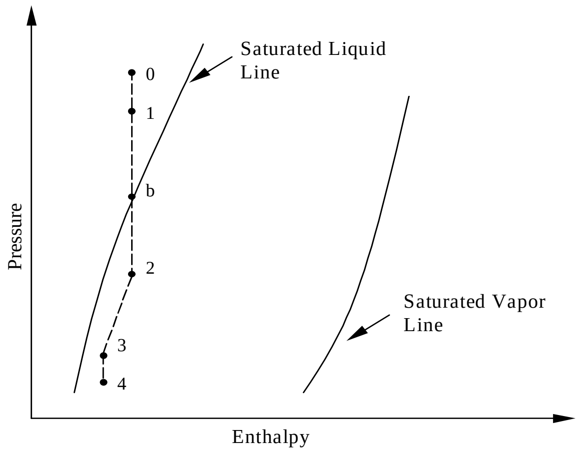


Figure 2. Schematic p-h diagram, representing the thermodynamic states of the refrigerant fluid through capillary tube-suction line heat exchanger.

At the capillary tube inlet, $z=0$ (point 1 in Figs. 1 and 2), the refrigerant pressure is calculated applying the mass conservation and energy conservation requirements between the points 0 and 1, and between the point 1 and the point where the pressure is obtained experimentally at the inlet device, p_e , for that the capillary tube is connected. The refrigerant temperature at the inlet of the capillary tube, $T_{rc,1}$, is determined based on the saturation temperature corresponding to p_1 and on the subcooling, ΔT_{sub}. The wall temperature of the capillary tube at the inlet is considered equal to $T_{rc,1}$. Thus, the initial conditions at the capillary tube inlet are as follows,

$$z = 0 \rightarrow p_1 = p_e - \frac{G^2(1+K)}{2\rho_l} \quad ; \quad T_{rc,1} = T_{sat}(p_e) - \Delta T_{sub} \quad ; \quad T_{wc,1} = T_{rc,1} \tag{7}$$

where $K=0.5$ (Barbazelli, 2000) is the pressure drop coefficient at the capillary tube inlet.

For solving the governing equations in two-phase flow region it is necessary to know initial conditions for dependent variables: *G*, *p* and h_o. As the metastability phenomenon is not been considered, it is assumed that bubble nucleation starts at the saturation pressure $p_{sat}(T_{rc})$, which correspond to a point b in Fig. 2. Thus, the initial condition at two-phase flow inlet is written as,

$$z = z_b \quad \rightarrow \quad p = p_{sat}(T_{rc,b}) \; ; \qquad G = \rho_l(T_{rc,b})u \; ; \qquad h_o = h_l(T_{rc,b}) + \frac{u^2}{2} \tag{8}$$

The refrigerant pressure along the suction line is considered constant and equal to evaporation pressure (p_{evap}) and the refrigerant temperature at the inlet of the suction line is assumed to be known ($T_{rs,i}$). Therefore, the initial conditions at the suction line inlet ($z_{i,s}$) are as followed,

$$z = z_{i,s} \rightarrow \quad p_{i,s} = p_{evap} \quad ; \; T_{rs} = T_{rs,i} \quad ; \; G = \rho_v (T_{rs,i}) u \quad ; \; h_o = h_v (T_{rs,i}) + \frac{u^2}{2} \quad ; \; T_{ws,i} = T_{rs,i} \tag{9}$$

At the capillary tube outlet (point 4 in Fig. 2), none of the dependent variables: G, p and h_o are known. The pressure at this point will correspond to the evaporation pressure, if the flow is not choked. However, if the flow is choked, which is common for capillary tubes encountered in refrigeration systems, the local pressure is higher than the evaporation pressure. Therefore, the determination of the choke pressure is vital to the solution of the equations that govern the flow of refrigerants through capillary tubes.

In the present model, the numerical criterion adopted is the one put forward by Fauske (1962). This criterion is based on theoretical and experimental observations that, under choked flow conditions, the pressure gradient (dp/dz) reaches a maximum value for a given mass flux and quality. This maximum value is generally an arbitrary one. However, it must be such that the position of the choke can be correctly determined; a too big value may lead to a choke pressure which is excessively reduced and therefore unrealistic. In the present work, this condition is numerically implemented evaluating dp/dz at each location along the tube until it becomes positive indicating that the maximum absolute has been reached. Therefore the position of the choke is settled as that one previous to the point where the pressure gradient becomes positive. However, in most of the tests, the refrigerant pressure in the capillary tube outlet becomes lower than evaporation pressure. In these cases, the correct position where the evaporation pressure is reached is obtained by linear interpolation.

4. Solution Methodology

The solution for the system of differential Eqs. (1) to (6), both in single-phase region (liquid) as in two-phase flow region (liquid-vapor) is carried out by using a modified finite volume method, presented by Escanes *et al.* (1995) and also used by Hermes (2000). The domain (capillary tube and suction line) is divided in m control volumes with the discretization nodes located at the inlet and outlet section of these volumes, as showed in Fig. 3.

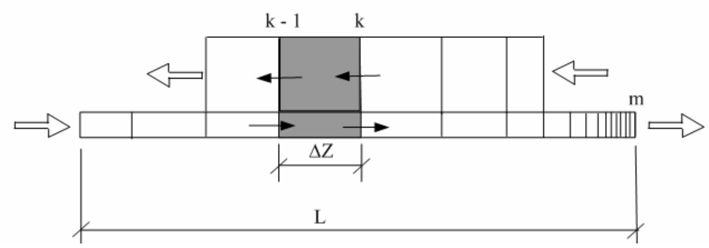


Figura 3. Control volume through capillary tube and suction line for lateral heat exchanger.

The equation used in this work to generate the computational grid is that presented by Escanes *et al.* (1995). In nearly all cases analyzed the grids were generate with 400 volumes: 50 for inlet region, 250 for heat exchanger region and 100 for outlet region.

The governing equations are integrated over time and space along control volumes with length Δz, showed in Fig. 3. A completely implicit scheme is employed to integrate the transient terms, in order to assure numerical stability, independent of the time step employed. The transient terms are discretized using the approximation: $\partial\phi / \partial t = [(\overline{\phi} - \overline{\phi}^o)] / \Delta t$, where ϕ is a specific dependent variable and Δt is the time interval [s]. The subscript ^{o} denotes the previous time step value of the variable ϕ and the superscript – represents the average value of this variable over the control volume, determined in this work from the trapezoidal rule, as follows,

$$\overline{\phi} = \frac{1}{\Delta z} \int_{z_{k-1}}^{z_k} \phi dz = \frac{\phi_k + \phi_{k-1}}{2} \tag{10}$$

The proposed model can be used to determine the refrigerant mass flow rate ($\dot{m}$) along the heat exchanger, known its geometry and operating conditions. The computation procedure is iterative because the equations depends on the mass flow rate, and additionally the refrigerant temperature at the suction line outlet ($T_{s,o}$) is unknown, being thus an inverse problem.

Therefore, at each time step, with a guessed value of $\dot{m}$ with other known operating conditions, the discretized equations are solved by successive substitution along the capillary tube and suction line. The dependent variables G, p, h_o e T_{wc}, for the capillary tube, and G, h_o and T_{ws} for the suction line are calculated for each discrete nodal points in the computational grid until the relative error between two consecutive iterations are less than 10^{-6}. Obtained the solution with the value of $\dot{m}$ guessed, the calculated capillary tube length (L^*) is compared with its respective value measured (L). Then, $\dot{m}$ is adjusted by means of the equation proposal by Melo and Negrão (1988) as follows,

$$\dot{m}_c = C_r\left(\frac{L^*}{L}\right)\dot{m}^* + \left(1 - C_r\right)\dot{m}^* \tag{11}$$

where $\dot{m}^*$ is the guessed value of mass flow rate, the subscript c indicates the correct value and C_r is an under-relaxation coefficient employed to improve the convergence process. Thus, the $\dot{m}$ value is corrected using the Eq. (11) until a given stopping criterion is achieved. Convergence is obtained when the difference between the calculated and actual lengths is lower than 10^{-3} m.

5. Results and Discussion

In this work, it is presented some comparison between computed results and experimental data obtained by Mendonça (1996) and Zangari (1998) for the steady state flow of the refrigerant R134a, through lateral and concentric capillary tube-suction line heat exchanger, respectively. Additionally comparisons for transient and quasi-steady models are presented.

Figures 4(a) and 4(b) present comparison between temperature distributions of the refrigerant flow along the capillary tube and suction line obtained by the present model with those ones available in Zangari (1998) (L=3.0 m; $d_{i,c}$=0.61 mm; $d_{i,s}$=6.3 mm; L_{he}=1.0 m; L_e=0.6 m) and Mendonça (1996) (L=4.0 m; $d_{i,c}$=0.83 mm; $d_{i,s}$=4.8 mm; L_{he}=1.59 m; L_e=0.53 m), respectively. In these figures the numbers I, II and III indicate the entrance, the heat exchanger, and the exit regions along the capillary tube, respectively. The experimental temperature distributions showed are those measured along the external walls of the capillary tube and of the suction line and the temperatures of the refrigerant fluid inside the suction line. The deviations between the measured and calculated mass flow rates are also present in Figs. 4(a) and 4(b).

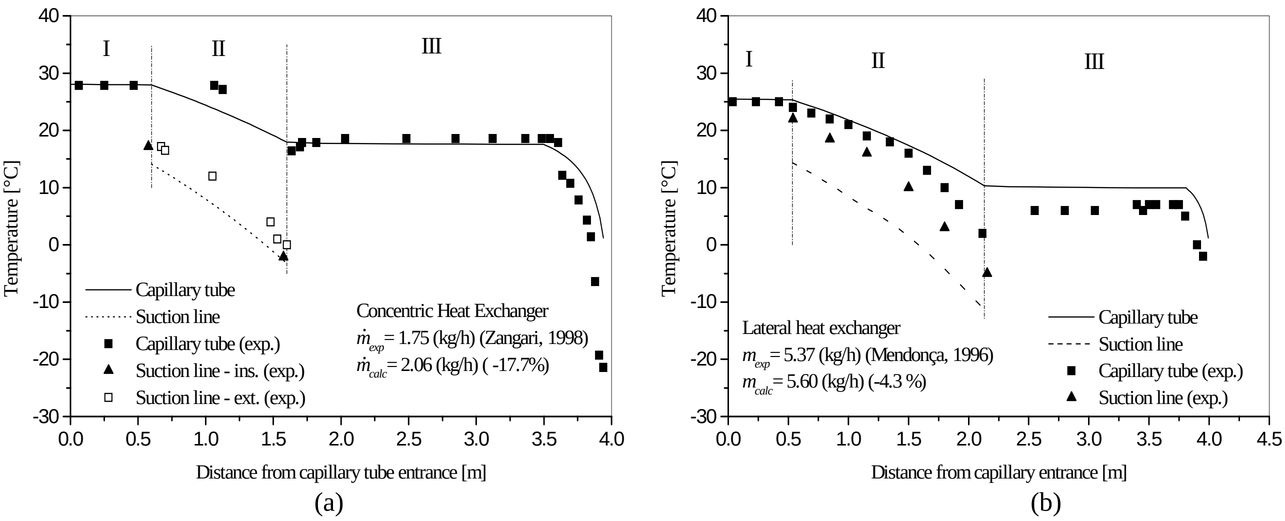
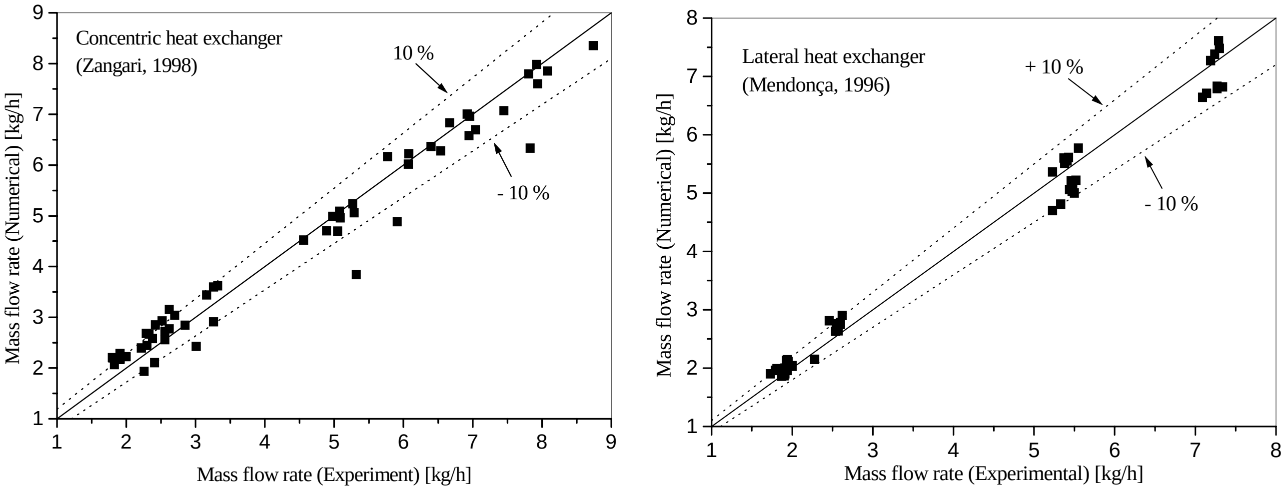


Figura 4. Comparison between temperature profiles measured and calculated: (a) Zangari, (1998): p_e=902 kPa, ΔT_{sub}=7.7 °C, $T_{i,s}$=-3.2 °C; (b) Mendonça, (1996): p_e=906 kPa, ΔT_{sub}=10.4 °C, $T_{i,s}$= – 10.9 °C).

It can be seen in Figs. 4(a) and 4(b) a good agreement between computed and experimental results for temperature profile regardless the deviation between the measured and calculated mass flow rate. It can also be notice the small extension of the two-phase flow region. In almost all experiments described by Mendonça (1996) and Zangari (1998) the refrigerant flash point is reached within an average distance of 0.20 m from the capillary tube outlet, and the modeling of the two-phase flow region becomes of small influence over the global result.

Comparisons between mass flow rates obtained in this work and the ones measured by Zangari (1998) and Mendonça (1996) are presented in Figs. 5(a) and 5(b), respectively. As can be seen in Figs. 5(a), for the concentric heat exchangers, 81 % of the mass flow rates calculated, using the present methodology, are within ±10 % of the experimental values, and the maximum difference found was of 25 %. For the lateral heat exchangers (see Fig. 5b), the present model give results which are within ±10 % of the experimental ones. Thus, considering all tests, the mean absolute deviations were of 8.6 % and 5.7 %, for the concentric and for the lateral heat exchangers, respectively.



(a) (b)

Figura 5. Comparison between measured and predicted mass flow rate: (a) concentric heat exchanger (Zangari, 1998) and (b) lateral heat exchanger (Mendonça, 1996).

In order to analyze the transient flow through capillary tube-suction line heat exchangers, the model is firstly tested considering the pseudo-transitory or quasi-steady regime flow. In the quasi-steady analysis the solution at any instant is obtained using a very large time step (steady state), but with the boundary conditions corresponding to the given instant. Following the procedure of Escanes *et al.* (1995), the solution of the steady state is obtained initially, for a given operating condition, then a linear temporal variation is imposed to the capillary tube inlet refrigerant temperature, $T_{rc,i}$. For a constant pressure at the inlet of the capillary tube, the temperature $T_{rc,i.}$ is decreased 12 °C linearly from its steady state value ($T_{rc,e}\big|_{\infty}$), in 240 s, i.e., $T_{rc,e} = T_{rc,e}\big|_{\infty} - 0{,}05t$ **.**

The linear temporal reduction of the temperature $T_{rc,i}$ increase the refrigerant mass flow rate through the heat exchanger. This fact happens because the reduction of $T_{rc,i}$, maintaining constant the pressure in the entrance, increases the refrigerant subcooling degree in the tube entrance and also increases the region along the tube which the refrigerant flow as subcooled liquid.

This fact can be observed in Fig. 6 which presents, for the steady refrigerant flow and for different instants of time: (a) the characteristics of the flow in the p-h diagram and the distributions of: (b) temperature; (c) pressure and (d) quality along the capillary tube. The geometric characteristics of the capillary tube-suction line lateral heat exchanger of the Fig. 6 are: L=4.0 m; $d_{i,c}$=0.83 mm; $d_{i,s}$=4.8 mm; L_{he}=1.59 m; L_e=0.53 m and the operating conditions are: p_e=906 kPa, ΔT_{sub}=10.4 °C, $T_{i,s}$= – 10.9 °C.

As can be observed in Figs. 6 the temporal reduction of the refrigerant temperature at the inlet of the capillary tube, increases the distance from the tube entrance where the refrigerant flash point is reached. Thus, the lines that represent the flow along the tube in the diagram p-h intercept the saturated liquid line in smaller pressures, or enthalpies, in each instant of time. The delay of the refrigerant flash point along the tube as a function of time can also be observed in Figs. 6(b) and 6(c), in which the largest gradients of temperature and pressure, respectively, take place towards the end of the tube as the time goes by.

As it can also be noticed in Figs. 6(d) the two-phase flow region is reduced with the time. The quality change with distance is not linear, with the rate of change progressively increasing towards the end of the tube. This is in accordance with the physical interpretation of the phenomenon, as acceleration and friction effects become gradually more important giving rise to higher vaporization rates.

Figure 7 shows the temporal mass flow rate distributions computed with the transient and pseudo-transitory or quasi-steady models through a concentric non-adiabatic capillary tube. In the quasi-steady analysis the solution at any instant is obtained using a very large time step (steady state), but with the boundary conditions corresponding to the given instant. The geometric characteristics of the capillary tube of the Fig. 7 are: $d_{i,c}$=0.61 mm e L=3.0 m. The others geometric characteristics of the heat exchangers numbers 1, 2 and 3, and operating conditions showed in Fig. 7 are presented in Tab. 1.

Notice in Fig. 7 that the transient variation of the mass flow rate along the tubes is linear in almost all time interval analyzed. As it can also be observed, the mass flow rate values computed by quasi-steady model are higher than that ones computed by transient model. The medium difference between the mass flow rates computed by transient and quasi-steady models for the heat exchangers 1, 2 e 3 was, 1.0%, 1.7% e 1.5%, respectively. This fact shows a small influence of the transient terms on the dynamic behavior of the flow along the non-adiabatic capillary tubes, although it is more significant than in the adiabatic case mentioned by Escanes *et al*. (1995) e Hermes (2000).

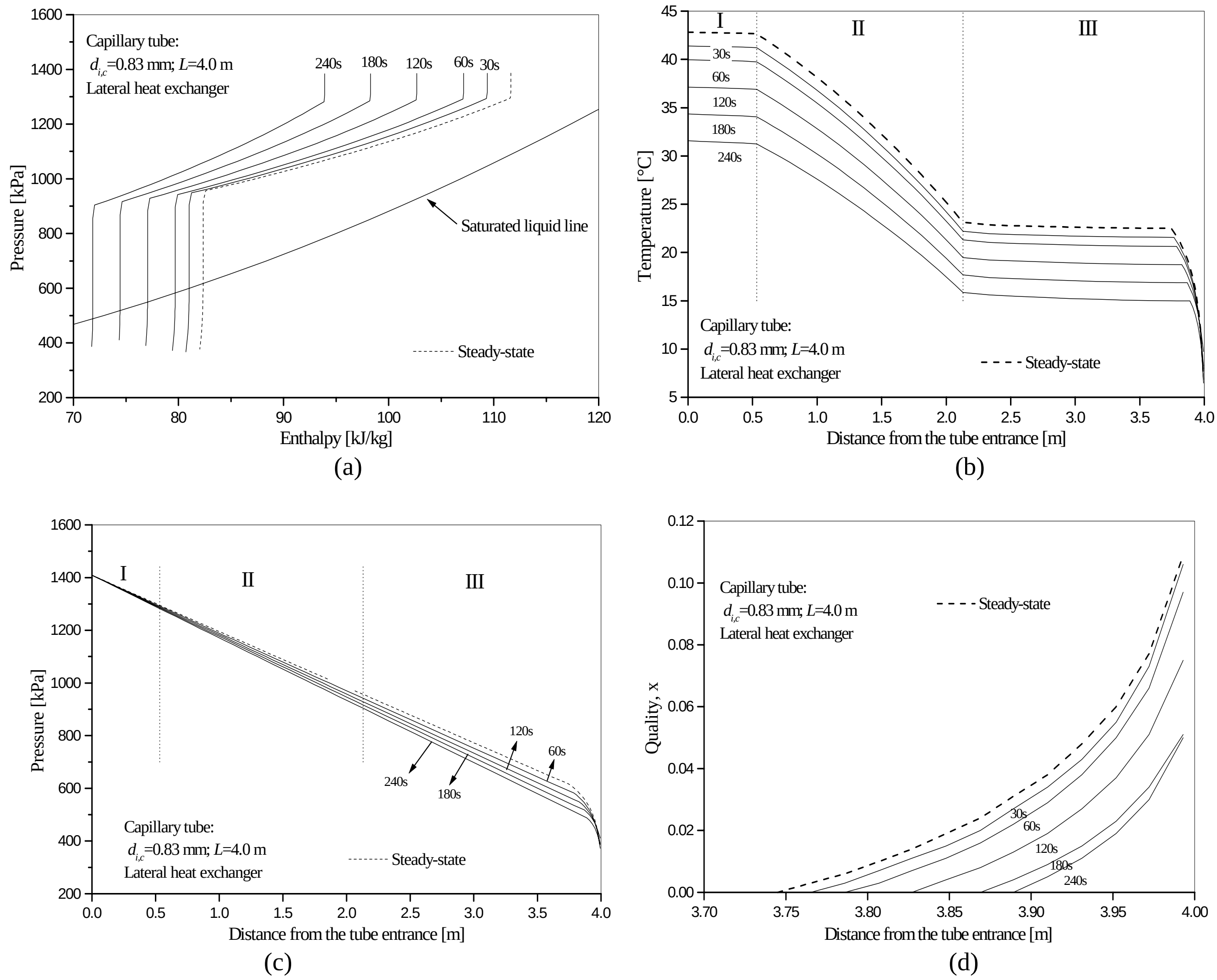


Figure 6 – (a) *p-h* diagram and distributions of: (b) temperature, (c) pressure and (d) quality along the capillary tube – lateral heat exchanger.

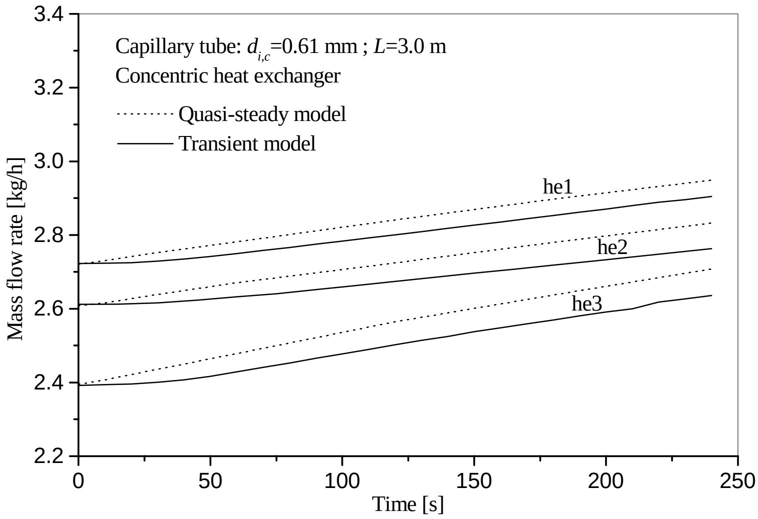
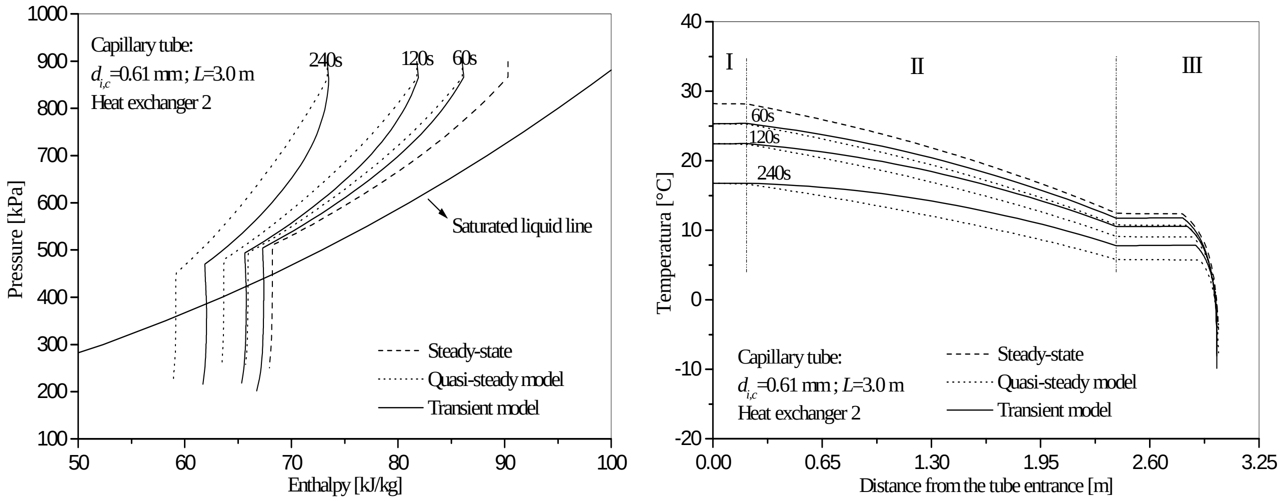


Figura 7. Mass flow rate variations with time computed by transient and quasi-steady models.

Table 1. Geometrical parameters and operating conditions - concentric heat exchangers of Fig. 7 (Zangari, 1998).

Heat exchanger	**$d_{i,s}$(mm)**	**L_{he}(m)**	**L_i(m)**	**p_e (kPa)**	**ΔT_{sub} (°C)**	**$T_{rc,e}$(°C)**	**$T_{rs,e}$ (°C)**
1	7,86	2,2	0,6	900	10,2	25,5	-17,0
2	6,3	2,2	0,2	903	7,5	28,3	-3,1
3	6,3	1,0	0,2	899	7,4	28,2	-6,0

For highlighting that influence, Figs. 8(a) and 8(b) illustrate for the heat exchanger 2 presented in Fig. 7, spatial and temporal variations of the refrigerant enthalpy and the refrigerant temperature along the capillary tube, respectively. These variations are computed with the transient and quasi-steady models, for the instants 60 s, 120 s and 240 s and also for the steady state. It can be seen in Fig. 8(a), the enthalpy profiles obtained by the transient model are moved for right in relationship those obtained by the quasi-steady model. Consequently, the temperatures of the refrigerant through capillary tube in each instant obtained by the transient model are larger than those obtained by the quasi-steady model (see Fig. 8b).



(a) (b)

Figure 8. Comparison between the transient and quasi-steady models: (a) p-h diagram; (b) temperature profiles along the capillary tube.

Another form of analyzing the transient behavior of the flow through non-adiabatic capillary tubes is that employed by Hermes (2000). In this case, boundary conditions based in experimental results of pull-down tests is imposed. Such tests consist of the accompaniment of the transient evolution of the pressure, temperature and power consumed by the refrigeration system, from the start-up to the steady–state condition. In the present model, it is analyzed the case in which starting from the steady–state solution, for a specific operating condition, the refrigerant temperature at the inlet of the capillary tube, $T_{rc,i}$, is reduced in accordance with the equation proposed by Hermes (2000) as,

$$T_{rc,i} = T_{rc,\infty} + \left(T^o_{rc,i} - T_{rc,\infty}\right)e^{-t/a} \tag{12}$$

where $T^o_{rc,i}$ and $T_{rc,\infty}$ are the refrigerant temperature at the inlet of the capillary tube for the initial and final steady–state conditions, respectively, and a is a constant time. After the reduction of $T_{rc,i}$, in accordance with the Eq. (12), a new steady-state condition will achieve, simulating the variation of the system operating conditions, which an equipment of refrigeration or air conditioning is submitted frequently.

Considering the same geometric configuration of the capillary tube of the Figs. 7 and 8 with the concentric heat exchanger 1 (see Table 1), Fig. 9 shows the temporal variation of the mass flow rate through capillary tube obtained by the transient and quasi-steady models. In this case the initial operating conditions are: $T^o_{rc,i} = 42{,}5\ °C$; $\Delta T^o_{sub} = 10{,}1\ °C$; $T^o_{rs,i} = -11{,}2\ °C$; $T_{rc,\infty} = 27{,}5\ °C$ and $a=100$. For the transient model, the inlet and outlet mass flow rates predicted are shown in Fig. 9, and it can be verified that practically there is not difference between its values.

It can also be seen in Fig. 9, the mass flow rates predicted by the transient and quasi-steady models increase from 3.51 up to 3.8 kg/h in approximately 400 s. However, during the transient period the values of the mass flow rates predicted by quasi-steady model are higher than the ones of the transient model. Such behavior is the same observed previously in the comparisons presented in the Fig. 7. Once, this difference indicates that the transient terms can have influenced on the dynamic behavior of the refrigerant flow through non-adiabatic capillary tubes.

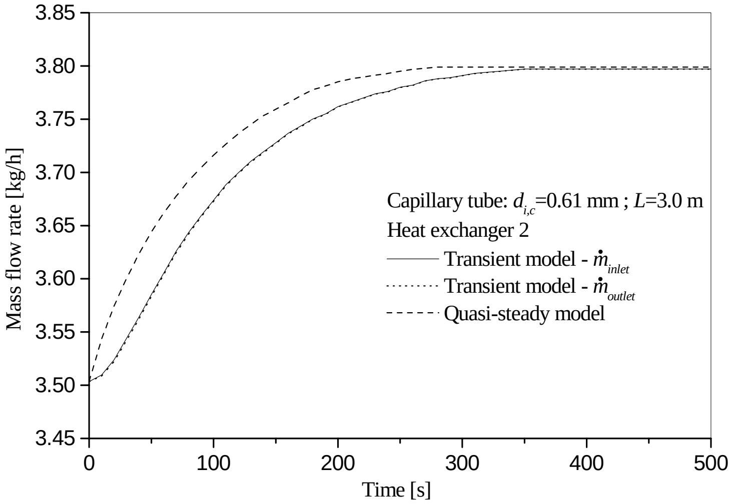


Figure 9. Mass flow rate variations with time computed by transient (a=100) and quasi-steady models.

4. Conclusions

This paper presents a numerical model for the simulation of one-dimensional unsteady state flows in capillary tube-suction line heat exchangers, commonly used in small refrigeration systems. Comparisons between experimental results available in open literature and the ones obtained in this work show good agreement, regarded the mass flow rate as well as temperature profiles. Taking the whole data into account, the relative mean deviation in the prediction of the choked mass flow rate in steady state was 8.6% for concentric heat exchangers and 5.7% for lateral heat exchangers.

The results of the transient and quasi steady models for the refrigerant flow were compared and differences about 2%, between the calculated values of the mass flow rates, were obtained. It was also observed differences among the pressure, temperature and enthalpy profiles of the refrigerant flow along the capillary tube, calculated by the transient and quasi steady models. Such facts, regardless of the small difference between the values of the mass flow rates, indicate that the influence of the transients terms of the governing equations on the dynamic behavior of the refrigerant flow through non-adiabatic capillary tubes can be more significant than in the adiabatic configuration.

The numerical method used in the solution of the governing equations and the iterative procedure to compute the refrigerant mass flow rate showed to be efficient, although they have presented convergence difficulties in some tests. In addition, the procedure of the mass flow rate correction, in function of the calculated length of the capillary tube, was not very sensitive for small variations of the refrigerant temperature at the inlet of the capillary tube.

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6. References

Bansal, P. K. and Xu, B., 2003, A Parametric Study of Refrigerant Flow in Non-Adiabatic Cappilary Tubes, Applied Thermal Engineering, 23, 397-408.

Barbazelli, M. R., 2000, Analysis of the Two-Phase Flow Through Non Adiabatic Capillary Tubes Using a Two-Fluid Model, Master´s Thesis, Department of Mechanical Engineering, São Paulo State University, Ilha Solteira, Brazil, 129p. (in Portuguese).

Churchill, S. W., 1977, Friction Factor Equation Spans all Fluid Flow Regimes, Chemical Engineering, 84, 91-92.

Dirik, E.; Inam, C.; Tanes, M. Y., 1994, Numerical and Experimental Studies on Adiabatic and Non-Adiabatic Capillary Tubes, Proceedings of the 1994 International Refrigeration Conference at Purdue University, West Lafayette, U.S.A., 365-370.

Erth, R. A., 1970, Two–Phase Flow in Refrigeration Capillary-Tube: Analysis and Prediction, Ph.D. Thesis, Purdue University, U.S.A., 120p.

Escanes, F., Pérez-Segarra, C. D. and Oliva, A., 1995, Numerical Simulation of Capillary-Tube Expansion Devices, International Journal of Refrigeration, 18 (2), 113-122.

Fauske, H. K., 1962, Contribution to the Theory of Two-Phase, One-Component Critical Flow, Argonne National

Laboratory, ANL-6633, Argonne, Illinois, U.S.A.

Gnielinski, V., 1976, New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow, International Chemical Engineering, 16, 359-368.

Hermes, C. J. L., 2000, Development of Mathematical Models to Numerical Simulation of Domestic Refrigerators in Transient Regime, Master´s Thesis, Department of Mechanical Engineering, Federal University of Santa Catarina, Florianópolis, Brazil, 255p. (in Portuguese).

McLinden, M. O., Klein, S. A., Lemmon, E. W. and Peskin, A. P., 1998, NIST Thermodynamic and Transport Properties of Refrigerants and Refrigerants Mixtures - REFPROP, Version 6.01, National Institute of Standards and Technology, Gaithersburg Maryland, USA.

Melo, C. and Negrão, C. O. R., 1988, Numerical Simulation of Capillary Tubes Used as Expansion Device in Refrigeration Systems, Proceedings of the II ENCIT, Águas de Lindóia, Brazil, 221-224 (in Portuguese).

Melo, C., Vieira, L. A. T. and Pereira, R. H., 2002, Non-Adiabatic Capillary tube Flow with Isobutane, Applied Thermal Engineering, 22, 166-1672.

Mendonça, K. C., 1996, Experimental Analysis of Lateral Capillary Tube-Suction Line Heat Exchangers, Master´s Thesis, Department of Mechanical Engineering, Federal University of Santa Catarina, Florianópolis, Brazil, 105p. (in Portuguese).

Pate, M. B., 1982, A Theorical and Experimental Analysis of a Capillary Tube – Suction Line Heat Exchanger", Ph. D. Thesis, Purdue University, West Lafayette, U.S.A.

Zangari, J. M., 1998, Experimental Analysis of Concentric Capillary Tube-Suction Line Heat Exchangers, Master´s Thesis, Department of Mechanical Engineering, Federal University of Santa Catarina, Florianópolis, Brazil, 160p. (in Portuguese).