MODELING OF UNSTEADY REFRIGERANT FLOW ALONG ADIABATIC CAPILLARY TUBES USING RUNGE-KUTTA METHOD

Vinícius Brandani Labigalini, vini_labigalini@hotmail.com
André Luiz Seixlack, andre@dem.feis.unesp.br
UNESP – Ilha Solteira, Av. Brasil Centro, 56 - CEP 15385-000 - Ilha Solteira, SP

Abstract. *Capillary tubes are commonly used as expansion devices in small refrigeration and air conditioning systems. Due to its great influence on the performance of the refrigeration cycle, the capillary tubes have been the source of many investigations, with experimental techniques and numerical simulations, improving the design and sizing of these devices. The refrigerant fluid flow inside the capillary tube is complex and during the refrigeration system operation, long unsteady periods can happen as a consequence, for example, of starting or stopping the system and variation of operating conditions of the system. During such periods the flow regions can appear or disappear, which difficulties the flow modeling. This work presents a numerical model to simulate unsteady state refrigerant flow along adiabatic capillary tubes. The fundamental equations governing the flow are derived from the mass conservation, momentum and energy conservation laws. The flow along the capillary tube is divided in a single-phase region and a two-phase flow region. The capillary tube is considered straight and horizontal. The refrigerant flow is taken as one-dimensional, the homogeneous flow model is employed for the two-phase flow region and the metastable flow phenomenon is neglected. The system of governing equations is solved using a fourth order Runge-Kutta method. The solution of this system of equations is performed along the tube until the flow is choked, or until the evaporation pressure is reached, if the flow is not choked. From the model the length of capillary tube can be obtained, for a given mass flow rate and operating conditions or, given the tube geometry and operating conditions, the solution yields the mass flow rate. In the first case the numerical solution is performed only once from the inlet to the outlet of the tube and the tube length is obtained. When the tube geometry is known an iteration procedure is required until the mass flow rate obtained is such that choked flow coincides with the outlet of the capillary tube. Comparisons are made with experimental measurements of the mass flow rate and pressure distribution along capillary tubes working with refrigerant R-134a in different operating conditions. The results indicate that the present model provides a good estimation of the refrigerant mass flow rate.*

Keywords*: Adiabatic Capillary Tubes, Two-Phase Flow, Refrigeration.*

1. INTRODUCTION

During approximately seventy years, halogenated refrigerants fluids, chlorofluorocarbons, also known as CFCs, were used in refrigeration systems. From the work of Molina and Rowland (1974), who proposed a theoretical model establishing a link between CFCs released in the atmosphere and the reduction of the ozone layer, this problem has received special attention in the scientific study atmospheric and a series of evidence has accumulated that relationship.

The scientific community has done in recent decades a great effort, aiming at replacement of CFCs and more recently also of HCFCs - hydrochlorofluorocarbons, particularly HCFC-22 or R-22. Several alternatives have been found, mostly of the family of halogenated hydrocarbons, as well as pure substances or mixtures binary or ternary. In the group of halogen, which does not contain chlorine, can be cited the HFC-134a or R-134a, R-404A, R-402B, R-407C and R-417A. Most of the domestic and small commercial applications adopted the HFC-134a as refrigerant and the HCFC-141b/R-141b as an agent in expansion foam. The small commercial applications, which employed HCFC-22 or R-502, began to use R-404A. The HFC-134a has a minimal effect on the ozone layer but has a high potential for global warming, the greenhouse effect. In addition to these alternatives, it is also considered the use of natural refrigerants, such as isobutane, HC-600a, in the domestic refrigeration and the propane, HC-290, and carbon dioxide, CO2/R-744 in the commercial refrigeration.

In the fiel of refrigeration the replacement of CFCs and HCFCs has motivated several researches and big investments in studies of the behavior of components of the refrigeration system operating with new refrigerants. Among these components compressors, heat exchangers, evaporators and condensers and expansion devices, particularly capillary tubes have been analyzed extensively, aiming at improving the thermodynamic efficiency and reducing its manufacturing cost, when operating with refrigerant fluids which are less aggressive to the atmospheric ozone layer.

Capillary tubes are expansion devices with constant inner cross-sectional area widely used in small refrigeration vapor compression systems with a maximum capacity of around 10 kW (34,130 Btu / h), such as those employed in domestic refrigerators and freezers, dehumidifiers and air conditioners. Usually the capillary tubes are made by copper, obtained by drawing and have lengths between 1.0 and 6.0 m and their diameters range from 0.5 and 2.0 mm.

During the operation of vapor compression refrigeration system the refrigerant is maintained at low pressure in the evaporator and at high pressure in the condenser due to the continuous action of the compressor and the expansion device. The capillary has essential role in the cycle, since it reduces the refrigerant pressure, from that condensation to the evaporation, and controls the refrigerant mass flow rate entering in the evaporator.

The main advantages of the capillary tubes are: lower cost, a pressure equalization feature to reduce the torque start-up, allows the operation in reverse cycle and absence of moving parts. However, such devices are not adjustable to attend different conditions of heat load and are designed for a range of operating conditions. The capillary tubes are thus subject to a reduction in operating efficiency, if occur variations in the heat load or temperature of condensation relative to project conditions. Other disadvantages are: the possibility of obstruction by particles and the requirement of a refrigerant charge within narrow limits. This latter condition justifies its use in hermetic units which the possibility of leaks is minimal.

Although the capillary tube is a simple device, the flow of the refrigerant inside is quite complex, offering many challenges to their phenomenological description, such as phase change, turbulence, multiphase flow, heat transfer, effects of metastability and critical flow. In addition, the capillary tubes can be adiabatic when are completely isolated from the ambient or non-adiabatic, when form a counter flow heat exchanger to the compressor suction line. The former are analyzed in this work.

Theoretical and experimental investigations of the flow in capillary tube, both adiabatic and non-adiabatic, have been the subject of extensive research, such as reported by: Dirik *et al.* (1994), Barbazelli (2000), Melo *et al.* (2002), Bansal and Xu (2003), Yang and Bansal (2005) and Seixlack and Navas (2006). These analyses were particularly applied for refrigerant flow characterization and for determining 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. However, long unsteady periods can happen during the refrigeration system operation as a consequence, for example, of starting or stopping the compressor and variation of the system operating conditions. Such unsteadiness introduces great complexity in the flow modeling. Although the unsteady flow condition has just been account by some researchers, many important questions for a better understanding of the transient flow along capillary tubes needs to be investigated. The transient distributions of the important flow parameters as mass flow rate, pressure, temperature and quality, either experimental as theoretical, along capillary tubes are almost inexistent.

The present work intends to contribute this subject and presents a numerical model to simulate the unsteady refrigerant flow along adiabatic capillary tubes. 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. The system of governing equations is solved using a fourth order Runge-Kutta method. The solution of the system of differential equations is performed along the tube until either choked flow or the established evaporation pressure is reached, which occurs first. The fourth order Runge-Kutta method is employed in order to make the software more agile since the goal is to use it as part of a more general code for the complete simulation of the household refrigerator.

The model allows predicting, in steady and unsteady states, the refrigerant mass flow rate, pressure, quality, refrigerant and wall temperatures distributions along the tubes, as a function of the tube geometry and operating conditions. Moreover, the capillary length can also be obtained for a given mass flow rate and operating conditions. For validation, results from the model are compared with the experimental data from the literature for steady flow and in addition results of the quasi-steady model are presented and discussed. Aspects of the steady-state and transient flow were evaluated and the results were compared with those obtained by Melo *et al*. (1995) and Hermes *et al.* (2000).

2. PROBLEM FORMULATION

In the present model, the flow along the capillary tube is divided in two distinct regions: a subcooled liquid flow region and a saturated two-phase flow region, as shown in Fig. 1. The capillary tube is straight, horizontal and the internal diameter is constant. One-dimensional and adiabatic flow of pure refrigerant (oil free) is assumed. Mechanical equilibrium is also assumed, i.e., the pressure is uniform at any given cross-section of the tube and surface tension effects are not considered. Metastability effects are not considered and the vapor is always saturated with respect to the local pressure.

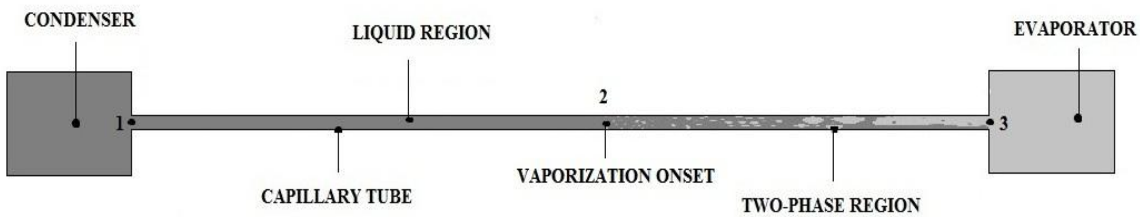


Figure 1. Layout of the refrigerant flow along the capillary tube.

The flow is considered incompressible in the liquid region. 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 tube are considered constants.

The fundamental equations governing the flow along the capillary tube are derived from the mass conservation, momentum and energy conservation laws. Taking the above assumptions into account, the governing equations are,

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

$$\frac{\partial (\rho u)}{\partial t} + \frac{\partial (\rho u u)}{\partial z} = -\frac{\partial p}{\partial z} - \frac{f \rho u^2}{2d} \tag{2}$$

$$\frac{\partial (\rho h_0)}{\partial z} + \frac{\partial (\rho u h_0)}{\partial z} = \frac{\partial p}{\partial t} \tag{3}$$

where t is the time [s], z is the distance along the tube [m], ρ is the density [kg/m^3], u is the refrigerant velocity [m/s], p is the absolute pressure [Pa], f is the friction factor and d is the inner diameter of the capillary tube [m], $h_o = (h + u^2/2)$ is the specific stagnation enthalpy [J/kg], h is the specific enthalpy [J/kg]. In the two-phase flow: $\rho = [\rho_l + \alpha(\rho_v - \rho_l)]$, where α is the void fraction (ratio of the portion of the cross-sectional area occupied by the vapor and the total cross-sectional area), $h = [h_l + x(h_v - h_l)]$ and x is the quality. The subscripts l and v indicate the liquid and vapor phases, respectively.

In summary, the proposed model consists of: Eqs. (1)-(3), which should be solved to give the distributions of u, p and h_0, respectively. To closure the model, constitutive equations are required for the friction factors in the liquid and in the two-phase region. The refrigerant thermo-physical properties were computed through linear regression of the data provided by McLinden *et al*. (1998).

3. INITIAL AND BOUNDARY CONDITIONS

In Figure 2, the line between the points 1 to 3 represent schematically a common situation of the flow along the adiabatic capillary tube (see Fig. 1). The regions located between points 1-2 and 2-3 correspond to the subcooled flow and two-phase flow, respectively. The governing equations, Eqs. (1) to (3), are first order partial differential equations and, therefore, require one initial and one boundary condition to be solved.

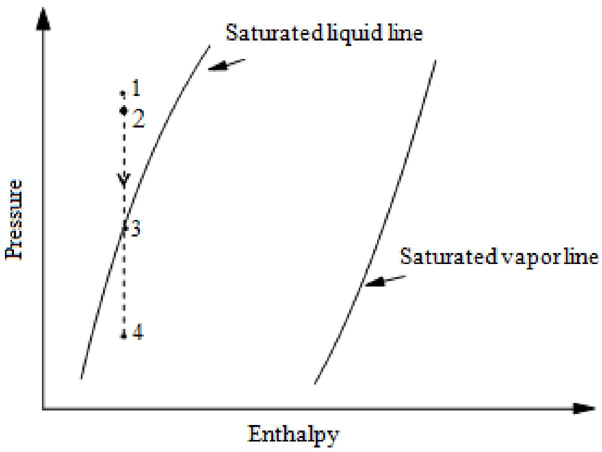


Figure 2. Schematic *p-h* diagram, representing the thermodynamic states of the refrigerant fluid along capillary tube.

At the capillary tube inlet, $z = 0$ (state 1 in Figs. 1 and 2), the refrigerant can be in the subcooled liquid state or in the saturation state, with quality equal or greater than zero. If the proposed model is used to determine the total length, L, of the capillary tube as a function of the tube geometry and of a prescribed mass flux, for a set of operating conditions, the refrigerant velocity at the tube inlet, u_{ent}, is determined based on the mass flux. The pressure and the temperature at the tube inlet, p_{ent} e T_{ent} are known. If the model is used to determine the total mass flux, G [kg/m^{2}s], as a function of the capillary tube geometry, for a set of operating conditions, the pressure and the temperature at the tube inlet, p_{ent} e T_{ent}

are also known. However in this case an iterative process is required, since the governing equations are dependent on the mass flux. Thus, the boundary conditions at the capillary tube inlet are as follow,

$$z = 0 \;\rightarrow\; p_1 = p_{ent} \;;\; T_1 = T_{ent} \;;\; u_1 = u_{ent} = G/\rho_1 \tag{4}$$

Since metastability effects are disregarded, it is hypothesized that the two-phase region starts when the local pressure equals the saturation pressure corresponding to the temperature at state 2 of Fig. 1. This point is characterized through a comparison between the refrigerant temperature in the liquid region and the saturation temperature, $T_{sat}(p)$, obtained using the calculated pressure as an input to a thermodynamic correlation (McLinden *et al*., 1998). Thus, the boundary conditions at the inlet of the two-phase region are given by,

$$z = z_3 \;\rightarrow\; p = p_{sat}(T_2) \;;\; u = u(z_2) \;;\; h_0 = h_{01}(z_2) \tag{5}$$

At the capillary tube outlet (state 4 in Fig. 1), none of the parameters u, p and h_o are known. The pressure at this point will correspond to the evaporator 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, determining the choke pressure is vital for solving the equations that govern the flow of refrigerants through the capillary tubes. In the present model, the Fauske (1962) criterion is used, considering the position of the choke as determined through the change of the pressure gradient (dp/dz) sign. In a marching algorithm using length steps of the order of millimeters, the numerical solution of the system of equations is obtained up to the step at which the sign of (dp/dz) is inverted. This is an indication that, between this step and the previous one, (dp/dz) reached a maximum. This procedure was adopted due to the great variability of the maximum (dp/dz) values.

The transient behavior of the capillary tube is analyzed according to Hermes and Melo (2008) by imposing time dependent boundary conditions, generated from the steady-state results performed by Klein (1998), and the experimental results of pull-down testes performed by Hermes and Melo (2008). Such tests consist of monitoring the evolution of the transient pressure, temperature and power consumed by the refrigeration system from the start-up to the steady–state condition. According to Hermes and Melo (2008), in general, the boundary conditions imposed to simulate the system start-up can be represented as: $\beta = \beta_\infty + (\beta_o - \beta_\infty)e^{-t/a}$, where β is generic dependent variable, β_o is the initial value of the variable calculated according to the pressure equalization system and the ambient temperature, β_∞ is the value of β under steady–state conditions and *a* is a time constant. Notice that the function given by β is increasing for $\beta_o < \beta_\infty$ and decreasing for $\beta_o > \beta_\infty$, with an asymptotic behavior with respect to β.

4. SOLUTION METHODOLOGY

The flow of refrigerants along the capillary tubes is an initial value problem, since parameters such as pressure and temperature or subcooling are known at the tube inlet. Besides, the flow has strong unidirectional characteristics and thus, information are transmitted only in one direction, witch means that the flow is parabolic to purpose of solution.

The fourth-order Runge-Kutta technique has been extensively used in numerical integration of parabolic systems. Here, this method is employed in the solution of Eqs. (1) to (3) both in single-phase region (liquid) as in two-phase flow region (liquid-vapor). In this method, the value of the derivatives must be known at the tube entrance (z_0), and the recently updated derivatives are used in the calculation of the following step.

The proposed model can be used to simulate the flow in the following situations: (i) to determine the total length, L, of the capillary tube as a function of the tube diameter and of a prescribed mass flux, for a set of operating conditions; (ii) to determine the total mass flux, G, as a function of the capillary tube geometry, for a set of operating conditions. In the first case, the system should be solved only once so that the total length of the tube can be determined. In case (ii), an iterative process is required, since the conservation equations are dependent on the mass flux. Thus, an initial mass flux is guessed, $\dot{m}^*$, the total length, L^*, is calculated and compared with the actual length, L. The mass flux is then corrected accordingly until convergence is obtained. This correction can be done in different ways. Here is used the same approach of Mezavila (1995) that consists in correcting the mass flow rate using the calculated capillary tube length, according to the following equation,

$$\dot{m}_c = C_r\left(\frac{L^*}{L}\right)\dot{m}^* + (1 - C_r)\dot{m}^* \tag{6}$$

where the use of the relaxation coefficient, C_r, is also necessary to improve convergence. In the above equation, if the calculated length is larger then the actual length, the mass flow rate has to be increased for the next iteration. Otherwise, if the calculated length is smaller than the actual one, the mass flow rate must be decreased. Convergence is obtained when the difference between the calculated and actual lengths is lower than 10^{-3} m .

5. STEADY STATE RESULTS

The steady-state model presented was validated using the experimental data of Melo *et al*. (1995) for R-134a. This database comprises measurements of mass flux and local temperature and pressure along two different capillary tubes (here identified as capillary 1 and capillary 2), whose geometric characteristics are presented in Tab. 1.

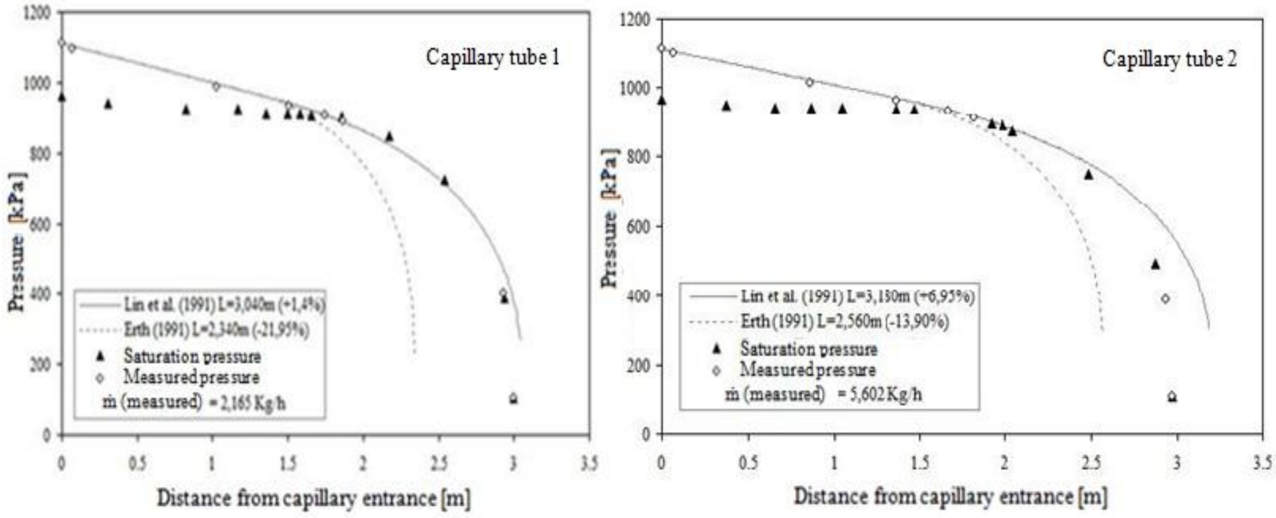
Table 1. Geometric characteristics of capillary tubes 1 and 2 (Melo *et al.*, 1995).

Capillary	Length [m]	Inner Diameter [mm]	Absolute Roughness (μm)
1	2.998	0.606	1.08
2	2.973	0.871	0.78
Measurement Uncertainty	0.002	0.005	0.2

Operating conditions were such that condensing pressures ranged from 900 to 1600 kPa (9 to 16 bar) and subcooling from 2 to 16 ^{o}C. All tests were carried out under choked flow conditions, which were verified through the insensitivity of the mass flux with respect to a variation of the vapor pressure. Tests were carried out using tubes without pressure taps, in order to identify their effect. It was found that these do not alter the mass flux significantly. The experimental results presented in this paper are from tests which metastability effects are less pronounced or inexistent. This is because such effects are not dealt with in the proposed model.

The Churchill (1977) correlation was used to evaluate the friction factor, *f*, in the single phase region. Two procedures were analyzed in the two-phase region to evaluate the frictional component of the momentum equation, Eq. (2): (i) using the Erth (1970) correlation to calculate the two-phase friction factor; (ii) using the Lin *et al.* (1991) correlation to calculate the two-phase frictional multiplier ϕ_l^2 .

Figures 4(a) and 4(b) present for capillary tubes 1 and 2, respectively, comparison between pressure distributions of the refrigerant flow along the capillary tube obtained by the present model with those ones available in Melo *et al*. (1995). In these figures the solid line and the dashed line represent pressure distributions obtained when frictional component of the momentum equation in the two-phase region is evaluated by Lin *et al.* (1991) correlation and by Erth (1970) correlation, respectively. As can be observed in Figs. 4(a) and 4(b), the pressure drop is linear in the single-phase liquid flow region, since the flow is considered incompressible in this region. It is also observed that the length of liquid region ranging approximately from 1.5 to 2.0 m from the capillary entrance. Since metastability is not considered in the present model, the flow reaches the saturation point, at 1.5 to 2.0 m from the capillary entrance, which corresponds to the onset of refrigerant vaporization. Thereafter the local pressure coincides with the saturation pressure and the pressure gradient is considerably larger due to acceleration effects and to increased friction associated with the two-phase flow.

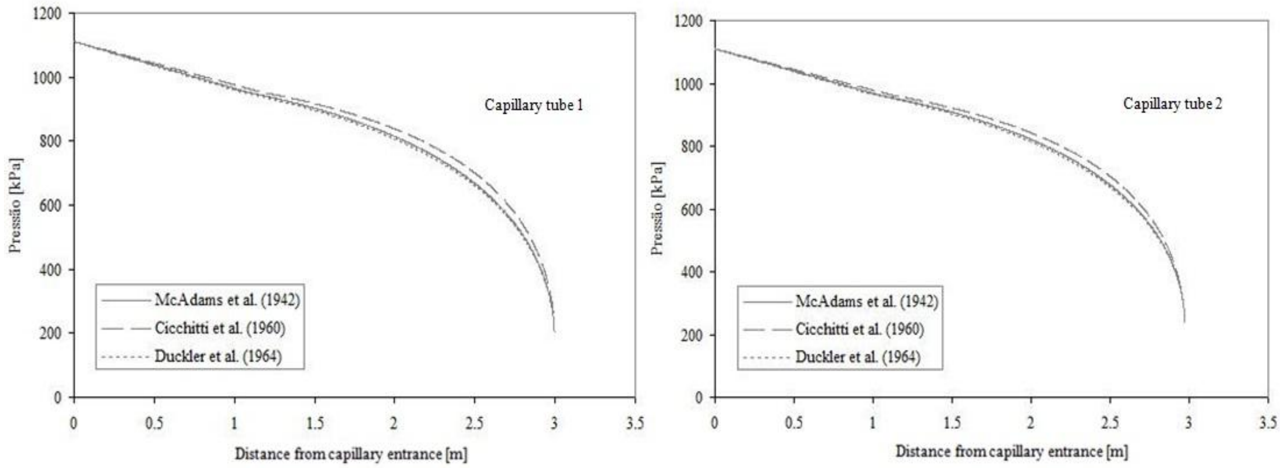


(a) (b)

Figure 4. Influence of the correlation to evaluated the frictional component of the momentum equation in two-phase region on the refrigerant pressure distribution. (a) capillary tube 1: p_{ent}=1111 kPa, T_{ent}=37,6°C; (b) capillary tube 2: p_{ent}=1110 kPa, T_{ent}=38°C.

In all cases tested, the correlation of Lin *et al*. (1991) provided results closer to the real values of capillary tubes length. The case which this difference becomes quite evident is that shown in Fig. 4(a), which shows the deviation of -21.95% obtained when Erth (1970) correlation is used and +1.4% when Lin *et al*. (1991) correlation is used.

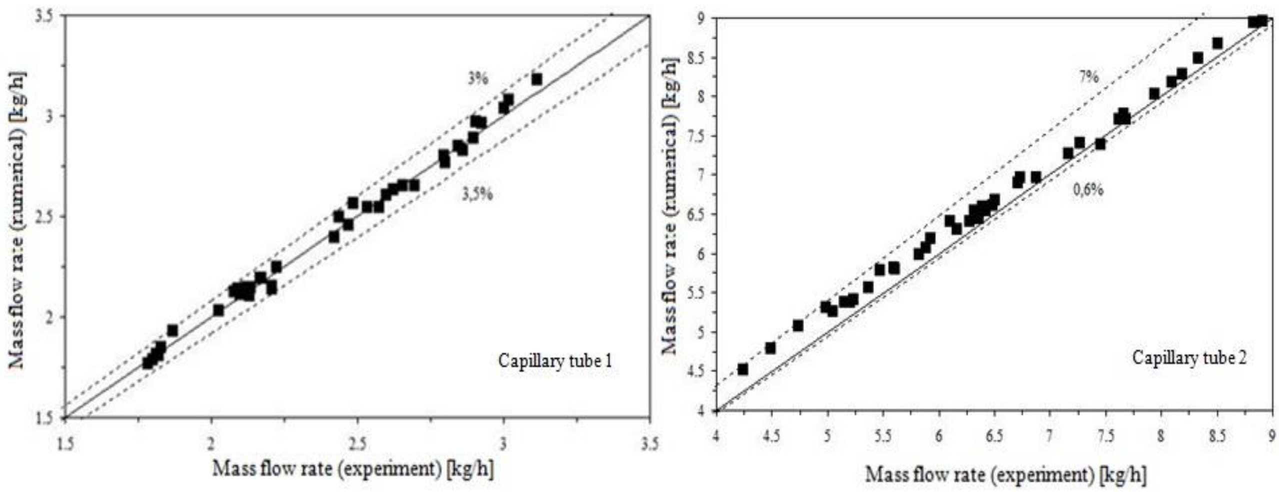
The best results obtained by Lin *et al.* (1991) correlation may be related to the fact that this correlation regards the variables related to the vapor phase, since the mean two-phase viscosity is calculated by the equations of McAdams *et al.* (1942), or Cicchitti *et al.* (1960) or Duckler *et al.* (1964). The correlation of Erth (1970), however, is used an approximate form and consider only the viscosity of the liquid phase. Thus, the correlation of Lin *et al.* (1991) was adopted in this model. The three above equations to calculate the mean two-phase viscosity were tested with the Lin *et al.* (1991) correlation [see Figs. 5(a) and 5(b)] and the difference between the pressure profiles was negligible. McAdams *et al.* (1942) equation was then adopted because it is more commonly used in the definition of mean two-phase viscosity.



(a) (b)

Figure 5. Influence of the mean two-phase viscosity equations on the pressure distribution along the capillary tube (a) 1 and (b) 2.

Comparisons between mass flow rates obtained in this work and the ones measured by Melo *et al.* (1995) are presented in Figs. 6(a) and 6(b) for the capillary tubes 1 and 2, respectively. As can be seen in Fig. 6(a), the mass flow rates calculated, using the present methodology, are within -3,5% and +3% of the experimental values. For the capillary tube 2 (see Fig. 6b), the present model give results which are within -0,6% and +7% of the experimental ones. Thus, considering all tests, the mean relative deviations were of 1.3% and 3.4%, for the capillary tubes 1 and 2, respectively.



(a) (b)

Figure 6 - Comparison between measured (Melo *et al.*, 1995) and predicted flow rate for capillary tubes: (a) 1 and (b) 2.

6. TRANSIENT RESULTS

In order to analyze the transient behavior of the capillary tube, has been considered the particular case in which, starting from the steady–state solution for a specific operating condition, the time dependent boundary conditions are imposed in accordance to Hermes and Melo (2008) until a new steady-state has been reached. This procedure is known as quasi-steady model since the dynamic behavior is defined only by the temporal variation of tube inlet conditions. The geometrical parameters used in this analysis were those of the capillary 1 shown in Tab. 1.

Two situations were considered. The first represents a decrease of the heat load of the refrigerator, varying the temperature at the capillary tube inlet and maintaining constant the other operating conditions. The solution of the steady-state is then obtained initially, next the temperature at the capillary inlet is varied and the mass flow is calculated for each time step until it reaches a new steady- state. The equation used to vary the temperature at the inlet, T_e , is given by Hermes and Melo (2008) as follow,

$$T_e = T_\infty + \left(T_e^o - T_\infty\right)\exp\left(-t/100\right) \tag{7}$$

where T_e^o and T_∞ are the initial (37,56 ^{o}C) and final (17,56 ^{o}C) steady-state temperature at the capillary tube inlet, respectively.

The second analysis represents the compressor start-up and the solution procedure is analogous to that used for the condition given by Eq. (7). The refrigerant pressure at the capillary tube inlet, p_e , is varying with time according to equation (Hermes and Melo, 2008) given by,

$$p_e = p_{op} + \left(p_{eq} - p_{op}\right) exp\left(-t/5\right) \tag{8}$$

where p_{eq} is the equalization pressure of the refrigeration system (4.9 bar), p_{op} is the pressure at the capillary tube inlet of the final steady-state (15,53 bar).

Figures 7(a)-(d) shows the distributions of: (a) temperature; (b) pressure; (c) quality of the refrigerant along the capillary tube 1, at certain instants and (d) refrigerant mass flow rate distribution with time at a time dependent condition of temperature at the inlet of the capillary tube 1.

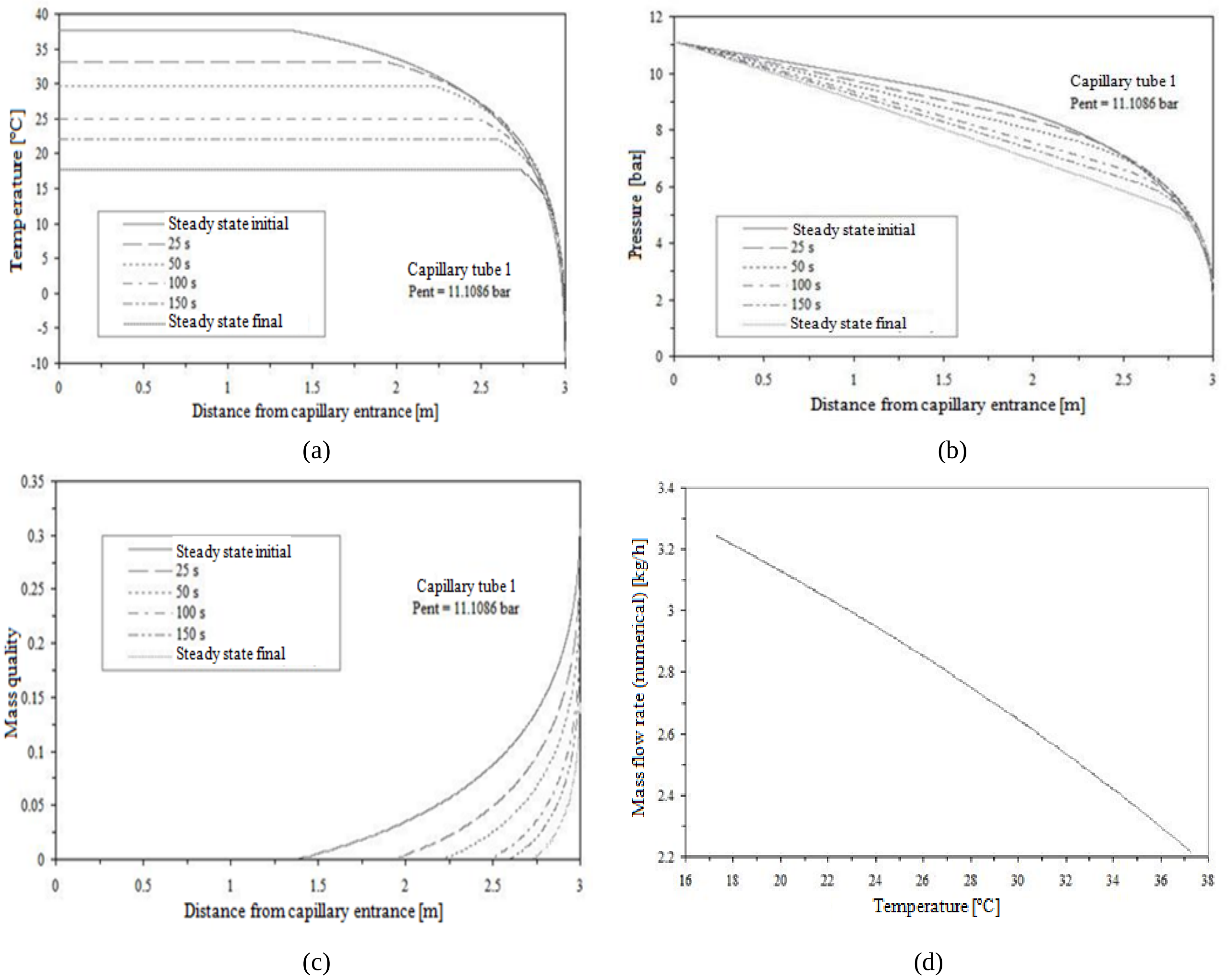


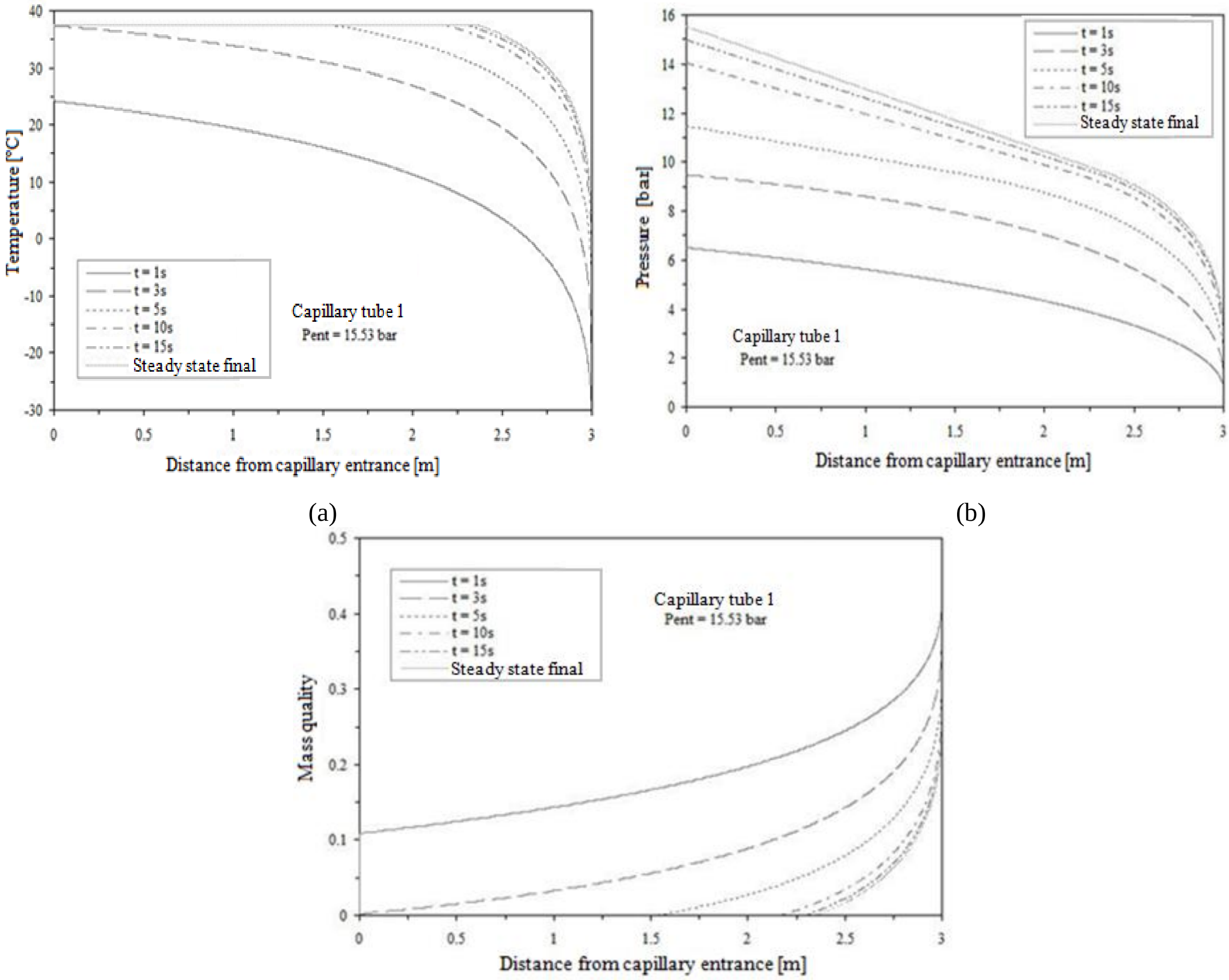
Figure 7. Refrigerant distributions of: (a) temperature, (b) pressure and (c) quality along the capillary 1 at some instants in time and (d) refrigerant mass flow rate distribution with time at a time dependent temperature of the capillary tube 1 inlet.

Notice that the temporal reduction of the refrigerant temperature at the inlet of the capillary tube moves the onset of vaporization toward the capillary tube outlet with time. The delay of the refrigerant flash point along the tube as a function of time can also be observed in Figs. 7(a) and 7(b), in which the largest gradients of temperature and pressure, respectively, take place towards the end of the tube with time.

As it can also be noticed in Fig. 7(c) the two-phase flow region is reduced with the time. The quality change along the tube is not linear and its gradient progressively increases 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.

Notice in Fig. 7(d) that the mass flow rate distribution with time along the tube is nearly linear along the whole temperature range considered. As it can also be observed, the temporal reduction of the temperature T_e increases the refrigerant mass flow rate through the capillary tube. This fact happens because the reduction of T_e , maintaining constant the pressure in the inlet, increases the refrigerant subcooling degree in the tube inlet and also increases the region along the tube which the refrigerant flow as subcooled liquid.

Figures 8(a)-(c) shows the distributions of: (a) temperature; (b) pressure; (c) quality of the refrigerant along the capillary tube 1, at certain instants, to a transient condition of pressure at the inlet of the capillary tube 1. As can be observed at the compressor start-up the pressure at the inlet of the capillary tube is reduced and there is only two-phase flow along the capillary.



(c)

Figure 8. Refrigerant distributions of (a) temperature, (b) pressure and (c) quality along the capillary 1at some instants in time to a transient condition of pressure at the inlet of the capillary tube 1.

As the inlet pressure approaches to operating pressure of the steady-state, the quality of the refrigerant will become zero near the tube inlet, showing the onset of the liquid region. This phenomenon is viewed from 3s for the capillary 1 in Fig. 8(c). It is also notice in Fig. 8(a) when the pressure at the tube outlet reaches the evaporator pressure or the choked criterion, the temperatures at the tube outlet at the initial instants of the compressor start-up reach very low values around -30 º C.

Notice in Figure 8 (b) that the pressure gradients in the tube outlet become more pronounced as the pressure at the inlet of the capillary increases. In the initial instants, although the flow is fully two-phase, the choking flow does not occur and the pressure in the tube outlet is equal to the evaporator. The distributions of the refrigerant quality along the capillary [see Fig. 8(c)] are similar those observed in the transient condition of temperature at the capillary tube inlet [see Fig. 7(c)]. It is notice that the quality gradients are very high near the capillary outlet, showing the high acceleration of the fluid in this region. Notice also, that in the first instants after the compressor start-up the values of the quality at the tube outlet are approximately 40%, which from the standpoint of the refrigerator performance is undesirable, due to a large portion of the vapor flow along the evaporator.

7. CONCLUSIONS

This paper presents a numerical model for the simulation of one-dimensional unsteady state flows in adiabatic capillary tubes, commonly used in small refrigeration systems. The model allows prediction, in steady and unsteady states, of the refrigerant mass flow rate, pressure, quality and refrigerant temperature distributions along the tube as a function of the tube geometry and operating conditions. Additionally the model allows prediction, in steady state, of the total length, L, of the capillary tube as a function of the tube diameter and of a prescribed refrigerant mass flow rate, as a function of operating conditions.

Comparisons between the results obtained by Melo *et al*. (1995) and those obtained in this work show good agreement with respect to mass flow rate as well as temperature profiles. Considering all tests, the mean relative deviations between experimental and numerical values of mass flow rate are of 1.3% and 3.4%, for the capillary tubes 1 and 2, respectively. In all cases tested, the Lin *et al*. (1991) correlation to evaluate the frictional component of the momentum equation in the two-phase region provided results closer to the real value of capillary tube length, than the Erth (1970) correlation to calculate the two-phase friction factor.

The results of the quasi-steady model showed that the temporal reduction of the refrigerant temperature at the inlet of the capillary tube moves the onset of vaporization toward the capillary tube outlet with time, due to a reduction of the saturation pressure. Since the pressure gradients in the liquid region are smaller than two-phase region, the mass flow rate required to reach the evaporating pressure becomes larger. The results of the quasi-steady model, obtained for the temporal reduction of the refrigerant inlet pressure, allowed verifying the flow behavior along the capillary tube in the refrigeration system start-up from the equalization pressure. It was found, in the initial instants of the refrigerator operation, that two-phase flow prevails along of the capillary tube and the pressure gradient along the flow are smaller for inlet pressures lower.

8. ACKNOWLEDGMENTS

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