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CAPHEAT: AN HOMOGENEOUS MODEL TO SIMULATE REFRIGERANT FLOW THROUGH NON-ADIABATIC CAPILLARY TUBES

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ABSTRACT

This work presents a numerical model to simulate refrigerant flow through capillary tube-suction line heat exchangers. The model can predict performance characteristics of geometrically defined capillary tube-suction line heat exchangers under various operating conditions. Lateral and concentric arrangements are considered. The refrigerant mass flow rate and the pressure, temperature and quality profiles along the flow are computed by the model. Calculated results are compared with experimental data reported in the literature and a good agreement is shown. Some illustrative results are presented to indicate the model potentialities.

NOMENCLATURE

Δt	Temperature difference	A'	Heat exchange area per unit length	g	Vapor
c	Specific heat			in	Inlet
d	Capillary tube diameter	v	Specific volume	int	Inside
D	Suction line diameter	x	Refrigerant quality	out	Outlet
f	Darcy friction factor	Z	Distance from capillary inlet	p	Constant pressure
g	Gravity acceleration	θ	Capillary tube inclination angle	s	Suction line
G	Mass flux			sp	Single-phase flow
h	Convective heat transfer coefficient	*Subscripts*		sub	Subcooling
				hx	Heat exchanger
i	Specific enthalpy	∞	Ambient	tp	Two-phase flow
L	Capillary tube length	c	Capillary tube	v	Constant volume
$\dot{m}$	Mass flow rate	cond	Condensation		
P	Pressure	evap	Evaporation		
t	Temperature	ext	Outside		
U	Overall heat transfer coefficient	f	Liquid		
A	Heat exchange area	fg	Latent heat of evaporation		

INTRODUCTION

Even though capillary tubes appear quite simple they are actually very complex expansion devices. At the tube entrance there is a slight pressure drop, due mainly to the abrupt change in cross sectional area. The pressure then decreases linearly along the tube length due to wall friction. This type of regime holds until the flow reaches saturated conditions. From this point (flash point), where vapor first appears, the pressure drop per unit length increases as the end of the tube is approached. This happens because the pressure drop in the two-phase flow region corresponds to both flow friction and the increase in momentum of the flow media. The critical flow condition or choked flow will occur whenever the fluid velocity reaches the local sonic velocity. After this any further lowering of the evaporating pressure has little effect on the mass flow rate.

Many authors have observed experimentally the existence of a delay in vaporization in adiabatic capillary tubes. This metastability can be characterized by the persistence of the liquid state to pressures less than the saturation pressure

$$\psi = x\cdot\frac{di_g}{dP} + (1-x)\cdot\frac{di_f}{dP} + G^2\cdot v\cdot\left[x\cdot\frac{dv_g}{dP} + (1-x)\cdot\frac{dv_f}{dP}\right] \qquad \lambda = G^2\cdot v_{fg} \qquad \omega = \frac{f_{tp}}{2}\cdot\frac{G^2\cdot v}{d_{int}}$$

The single-phase flow friction factor is calculated by Churchill's equation [7]. This equation is in good agreement with the experimental data obtained by Melo et al. [8]. The convective heat transfer coefficients for the liquid flow in the capillary tube and for the superheated vapor in the suction line are calculated by Gnielinski's correlation [9]. The local two-phase flow friction factor is calculated by Erth's equation [10], using the quality and the Reynolds number at the inlet section of each control volume. The suitability of this approach was confirmed by Mezavila [11]. The convective heat transfer coefficient in the two-phase flow of the capillary tube is assumed to be infinite. This assumption has no effect on the simulation results [11]. The convective heat transfer coefficient between the ambient air and the capillary tube-suction line heat exchanger is evaluated by the equations proposed by Churchill and Chu [12] and by Mc Adams [13] for horizontal and vertical tubes, respectively. The refrigerant thermodynamic and transport properties are evaluated by the REFPROP program version 4.0 [14], which was linked to the present computer code.

Several two-phase critical flow models, representing a broad range of theories are used to predict choked flow in capillary tubes. In the present work the critical flow conditions are established at the point of maximum pressure gradient ($dP/dZ \to \infty$) [15]. In doing so no assumptions about the flow path along capillary tube-suction line heat exchangers have been made.

NUMERICAL SOLUTION

The present model is essentially an initial value marching problem. The governing differential equations are integrated by the 4th order Runge-Kutta method. For each control volume, and from the flow conditions at its inlet section, solution of the equations gives the flow variables at its outlet section. This process is moved forward in the flow direction until either critical flow or evaporator pressure is reached. The solution of the flow through the capillary tube-suction line heat exchanger is carried out in an iterative process. The mass flow rate and the suction line exit temperature are iteratively evaluated, by means of a Newton-Raphson algorithm.

COMPARISON WITH EXISTING EXPERIMENTAL DATA

Figure 3 shows the computations of the **CAPHEAT** program against the experimental data for HFC-134a, reported by Dirik et al. [5]. It can be observed that for this data set the mass flow rate measurements tended to be under-estimated by the model. Dirik et al. [5] achieved a better agreement between calculations and measurements because their model took into account the metastable flow effects. This was done by using the correlation proposed by Chen et al. [16], which was later found to be inappropriate for the calculation of the underpressure of vaporization in capillary tubes [8].

Comparisons were also made with recent experimental data for CFC-12, reported by Bittle et al. [17]. Figure 4 shows that a good agreement between calculations and measurements was achieved for this data set, with all the data points falling inside a 10% band.

Unfortunately very little experimental data have been published for capillary tube-suction line heat exchangers. More experimental data is needed to validate the model. Additional experiments are currently being performed by the authors and the results will be reported in a near future.

NUMERICAL RESULTS

Calculated mass flow rates for a representative case are presented in Figure 5. The position of the heat exchanger is taken as the independent variable. With subcooled refrigerant entering the capillary and with the heat exchanger close to the capillary inlet, the mass flow rate is unaffected by variations in heat exchanger position. The reason for this is that, in this condition, the refrigerant remains liquid for at least the entire length of the heat exchanger. When the heat exchanger is moved towards the end of the capillary tube a reduction in the mass flow rate is observed. This is due to the increase in vapor quality

Capillary tube single-phase flow regions:

$$\frac{dP_c}{dZ} = -\frac{f_{sp}}{2} \cdot \frac{G^2 \cdot v}{d_{int}} - \frac{g \cdot \sin\theta}{v} \tag{1}$$

Heat exchanger region (concentric hx):

$$\frac{dt_c}{dZ} = -\frac{v}{c_v} \cdot \frac{dP_c}{dZ} - \frac{1}{\dot{m} \cdot c_v} \cdot UA'_{c,s} \cdot (t_c - t_s) - \frac{g \cdot \sin\theta}{c_v} \qquad UA'_{c,s} = \frac{1}{\frac{1}{h_c \cdot \pi \cdot d_{int}} + \frac{1}{h_s \cdot \pi \cdot d_{ext}}} \tag{2}$$

Heat exchanger region (lateral hx):

$$\frac{dt_c}{dZ} = -\frac{v}{c_v} \cdot \frac{dP_c}{dZ} - \frac{1}{\dot{m} \cdot c_v} \cdot UA'_{c,s} \cdot (t_c - t_s) - \frac{g \cdot \sin\theta}{c_v} \qquad UA'_{c,s} = \frac{1}{\frac{1}{h_c \cdot \pi \cdot d_{int}} + \frac{1}{h_s \cdot \pi \cdot D_{int}}} \tag{3}$$

Inlet and outlet regions:

$$\frac{dt_c}{dZ} = -\frac{v}{c_v} \cdot \frac{dP_c}{dZ} - \frac{1}{\dot{m} \cdot c_v} \cdot UA'_{c,\infty} \cdot (t_c - t_\infty) - \frac{g \cdot \sin\theta}{c_v} \qquad UA'_{c,\infty} = \frac{1}{\frac{1}{h_c \cdot \pi \cdot d_{int}} + \frac{1}{h_\infty \cdot \pi \cdot d_{ext}}} \tag{4}$$

Capillary tube two-phase flow regions:

$$\frac{dP_c}{dZ} = -\frac{\frac{f_{tp}}{2} \cdot \frac{G^2}{d_{int}} \cdot \left[x \cdot v_g + (1-x) \cdot v_f\right] + G^2 \cdot v_{fg} \cdot \frac{dx}{dZ} + \frac{g \cdot \sin\theta}{v}}{1 + G^2 \cdot \left[x \cdot \frac{dv_g}{dP} + (1-x) \cdot \frac{dv_f}{dP}\right]} \tag{5}$$

Heat exchanger region:

$$\frac{dx}{dZ} = \frac{-\frac{\delta}{\dot{m}} \cdot UA'_{c,s} \cdot (t_c - t_s) + \psi \cdot \left(\omega + \frac{g \cdot \sin\theta}{v}\right) - \delta \cdot g \cdot \sin\theta}{\phi \cdot \delta - \psi \cdot \lambda} \tag{6}$$

Inlet and outlet regions:

$$\frac{dx}{dZ} = \frac{-\frac{\delta}{\dot{m}} \cdot UA'_{c,\infty} \cdot (t_c - t_\infty) + \psi \cdot \left(\omega + \frac{g \cdot \sin\theta}{v}\right) - \delta \cdot g \cdot \sin\theta}{\phi \cdot \delta - \psi \cdot \lambda} \tag{7}$$

Suction line:

$$\frac{dt_s}{dZ} = -\frac{UA'_{c,s} \cdot (t_c - t_s) + U_{s,\infty} \cdot \pi \cdot D_s \cdot (t_\infty - t_s)}{\dot{m} \cdot c_{p,s}} \qquad U_{s,\infty} = \frac{1}{\frac{1}{h_s} + \frac{1}{h_\infty}} \tag{8}$$

Where:

$$\phi = i_{fg} + G^2 \cdot v_{fg} \cdot \left[x \cdot v_g + (1-x) \cdot v_f\right] \qquad \delta = 1 + G^2 \cdot \left[x \cdot \frac{dv_g}{dP} + (1-x) \cdot \frac{dv_f}{dP}\right]$$

corresponding to its temperature. There is no conclusive evidence about the existence of this phenomena for situations where the flash point takes place in the suction line heat exchanger region of the capillary tube [1].

During the refrigerant passage through the capillary tube some of the liquid turns into vapor. The refrigerant leaving the capillary tube is therefore a two-phase mixture. The liquid fraction of this mixture boils away in the evaporator, taking heat from the bodies or fluids being refrigerated. The vapor fraction of the mixture, however, plays no further part until it is compressed again. In order to reduce the refrigerant quality at the evaporator inlet the capillary tube is often in thermal contact with the suction line. This configuration provides heat exchange to the low-pressure vapor passing through the suction line from the high-pressure refrigerant passing through the capillary tube. The benefit of such heat exchanger, in terms of coefficient of performance and volumetric capacity, depends on the combination of refrigerant properties and operating conditions [2].

There are basically, two different types of capillary tube-suction line heat exchangers; lateral and concentric. The capillary tube is soldered to the suction line in the lateral arrangement, while it passes inside the suction line in the concentric arrangement. In both cases the capillary tube and the suction line form a counterflow heat exchanger.

The study conducted by Pate and Tree [1] indicated that the flow pattern in capillary tube-suction line heat exchangers is considerably different from the one encountered in adiabatic capillary tubes. It was observed, for example, that the frictional effects that encourage flashing are suppressed by the heat transfer to the suction line. Using their experimental data base, Pate and Tree [3] developed a simplified model for lateral capillary tube-suction line heat exchangers, assuming a linear quality profile along the heat exchanger region. This assumption reduced the sensitivity of their model to the assumed flash point location. However, as pointed out by Peixoto and Bullard [4], such quality profile is unlikely to exist in most cases.

Dirik et al. [5] developed a model for concentric capillary tube-suction line heat exchangers with no restrictive assumption on the quality distribution. The model considers only those cases where flashing begins at a location upstream of the heat exchanger. It was argued that this is the common situation with refrigerators, although the experimental data of Pate and Tree [1] demonstrated that the flash point can definitely exist within the heat exchanger region.

Peixoto and Bullard [4] presented a model for lateral and concentric capillary tube-suction line heat exchangers. The finite difference method was used to solve the governing equations, in conjunction with constitutive equations taken from the literature. Of special note are the equations used to evaluate the two-phase flow friction factor. For tubes of unknown roughness, for example, an experimental correlation applicable only to a specific capillary tube [3] was indiscriminately used.

Escanes et al. [6] presented a model to simulate the evaporating flow through capillary tubes. The heat exchange with the ambient air or with the suction line was prescribed as a boundary condition in terms of a local distribution along the capillary tube wall of the temperature or the heat flux exchanged. This approach limits the application of this model as a general tool for analyzing the performance of capillary tube-suction line heat exchangers.

The present work is intended to establish a model to simulate the effects of lateral and concentric heat exchangers on the flow characteristics of capillary tubes, without most of the limitations encountered in the previous models.

GOVERNING EQUATIONS

The governing equations of the flow are the mass, momentum and energy equations. To derive these equations the following simplifying assumptions are made: i) straight and constant inner diameter capillary tube, ii) steady-state, pure refrigerant one-dimensional flow, iii) negligible axial conduction in the fluid and tube walls, iv) homogeneous two-phase flow, v) radial and axisymmetric isothermal tube walls and vi) thermodynamic equilibrium.

For modeling purposes the flow in the capillary tube is divided into three different regions. In the heat exchanger region the capillary tube is in thermal contact with the suction line. In the other two regions (inlet and outlet) heat exchange occurs only with the surrounding air.

along the heat exchanger length (see Figures 6 and 7). The vapor quality profile presented in Figure 6, shows two flash points, one upstream and another dowstream of the heat exchanger. The recondensation of the refrigerant within the heat exchanger region of the capillary tube was also observed by Peixoto et al. [4].

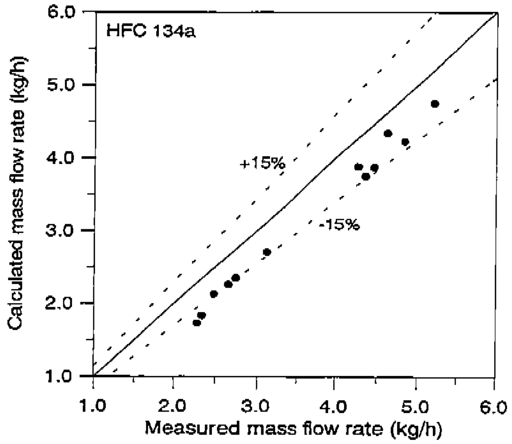

Fig. 3 - Comparison with experimental data of Dirik et al. [5]

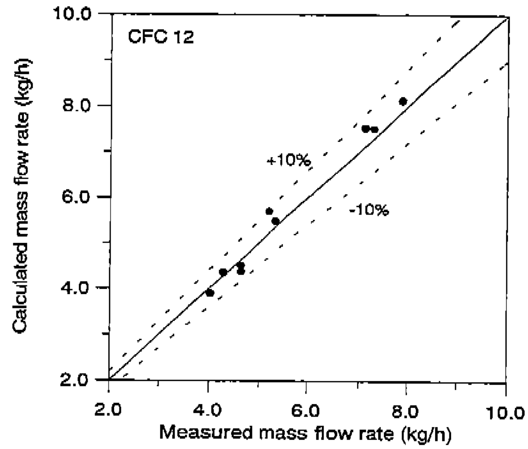

Fig. 4 - Comparison with experimental data of Bittle et al. [17]

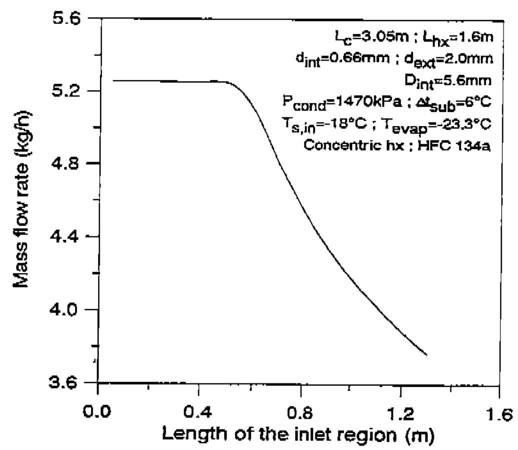

Fig. 5 - Mass flow rate versus hx position

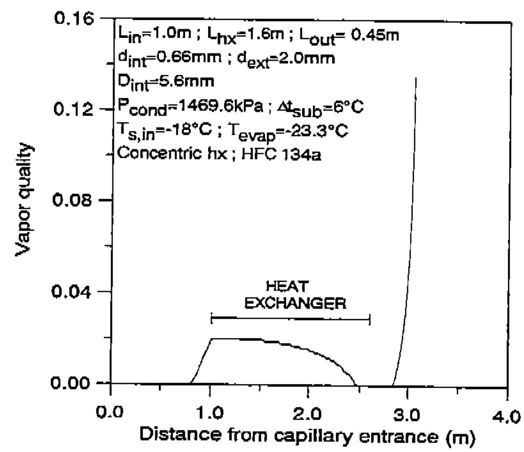

Fig. 6 - Refrigerant quality along the capillary tube

Figure 8 illustrates how the mass flow rate through a capillary tube-suction line heat exchanger is higher for the case of horizontal tubes than for vertical tubes. This can be easily explained by the gravity effects on the conservation laws. It can also be seen in this figure how the heat transfer between the ambient air at 25°C and the capillary tube-suction line heat exchanger affects the mass flow rate.

CONCLUSIONS

An homogeneous model to simulate the refrigerant flow through lateral and concentric capillary tube-suction line heat exchangers has been presented. The model, based on a control volume formulation of the governing equations, can predict performance characteristics of geometrically defined capillary tube-suction line heat exchangers under various operating conditions and with several refrigerants. The flow rates predicted by the model show good agreement with experimental data available in the literature. An user-friendly computer interface was developed for the **CAPHEAT** program. It is believed that this software can be used to assist the selection of capillary tubes in household refrigeration systems, where traditionally graphical methods are still widely used.

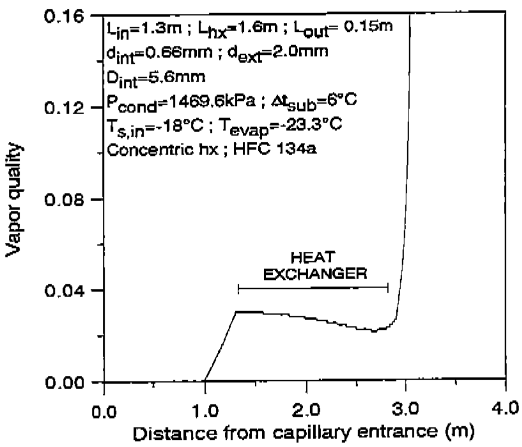

Fig. 7 - Refrigerant quality along the capillary tube

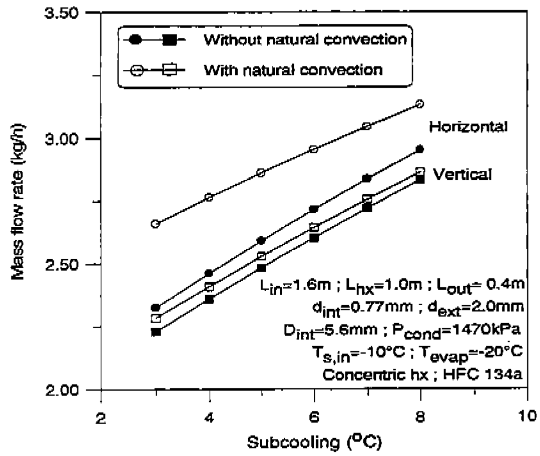

Fig. 8 - Heat transfer and inclination effects

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