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MODELING THE HFC 134a FLOW THROUGH CAPILLARY TUBES USING A TWO-FLUID MODEL

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ABSTRACT

This work presents a numerical model to simulate the HFC-134a flow through capillary tubes, commonly used as expansion devices in refrigeration systems. The capillary tube is considered straight and horizontal; the flow is taken as one-dimensional, steady and adiabatic. A two-fluid model, involving five conservation equations and considering the hydrodynamic and thermal nonequilibrium between the liquid and vapor phases, is applied to the two-phase flow region. Computed pressure profiles and mass flow rates obtained from the model are compared with experimental data. It is shown that the present model yields better results than the commonly employed homogeneous model.

INTRODUCTION

Due to its major influence in refrigeration equipments, there has been a great amount of literature related to capillary-tube expansion devices. The flow in such a device offers several challenges for a phenomenological description: turbulence, phase-change, sonic and metastability effects all occur in the flow through capillary tubes. Some representative contributions on both analysis and experiments related to such flow can be found in [1] and [2].

A literature review indicates that all works that have been reported assume homogeneous flow in modeling capillary tubes. In the homogeneous flow model no distinction is made between the liquid and vapor phases and the governing equations are written for a pseudo fluid having properties whose values are mean values for the flow [3]; velocity and temperature for both phases are assumed to be equal. In this respect several features of the phenomenon cannot be disclosed through the homogeneous flow model.

The present work presents, for the first time, a numerical analysis of the HFC 134a flow through capillary tubes using a two-fluid model. For validation, results from the model are compared with the experiments performed in [2].

PROBLEM FORMULATION

In the present model the flow along the capillary tube is divided in two regions: subcooled liquid and two-phase region. The tube is taken to be straight with constant inner diameter and the flow is assumed to be one-dimensional, steady and adiabatic. Additionally, the refrigerant is free from oil and both phases are assumed to be at the same pressure, that is, surface tension effects are neglected. Metastable flow effects are also disregarded, and the vapor is assumed to be saturated at the local pressure.

Liquid flow region:

In this region the flow is fully developed and pressure drop is due to friction; also, because of the adiabatic assumption, temperature can only be changed due to viscous dissipation effects. The flow is then governed by the momentum and energy conservation equations,

$$\frac{dp}{dz} = -\frac{f_L G^2 V_L}{2D} \quad , \quad \frac{dT}{dz} = -\frac{V_L}{c_v}\frac{dp}{dz} \tag{1}$$

where z is the axial location along the tube, p is pressure, f_L is the single-phase friction factor, G is the volumetric mass flow rate, V_L is the velocity of the liquid, D is the tube inner diameter, T is temperature and c_v is the specific heat at constant volume.

Two-phase flow region:

As the refrigerant flows along the capillary tube, friction causes a decrease in pressure and eventually saturation is reached. After that, the decrease in pressure is associated to liquid changing into vapor, and the two-phase flow is then established. The equations for the two-phase region are:

(i) Overall Mass Conservation:

$$\frac{d}{dz}[(1-\alpha)\rho_L V_L + \alpha\rho_V V_V] = 0 \tag{2}$$

where α is the void fraction, ρ is the density and V is the phase velocity; the subscripts L and V refer, respectively, to the liquid and vapor phases.

(ii) Momentum Conservation for Liquid Phase:

$$\frac{d[(1-\alpha)\rho_L V_L^2]}{dz} = -(1-\alpha)\frac{dp}{dz} - F_{WL} + F_{LV} - V_i\frac{d}{dz}[\alpha\rho_V V_V] \tag{3}$$

where F_{WL} is the friction force per unit volume between the liquid phase and the tube wall; F_{LV} is the interfacial force per unit volume between liquid and vapor and V_i is the interfacial velocity given by $(V_V+V_L)/2$.

(iii) Momentum Conservation for Vapor Phase:

$$\frac{d(\alpha\rho_V V_V^2)}{dz} = -\alpha\frac{dp}{dz} - F_{WV} - F_{LV} + V_i\frac{d}{dz}[\alpha\rho_V V_V] \tag{4}$$

where F_{WV} is the friction force per unit volume between the vapor phase and the tube wall.

(iv) Overall Energy Conservation:

$$\frac{d}{dz}[(1-\alpha)\rho_L V_L(h_L + \frac{V_L^2}{2}) + \alpha\rho_V V_V(h_V + \frac{V_V^2}{2})] = 0 \tag{5}$$

where h is the specific enthalpy of either the liquid (subscript L) or the vapor phase (subscript V).

(v) Energy Conservation for Vapor Phase:

$$\frac{d}{dz}\left[\alpha\rho_V V_V\left(h_V + \frac{V_V^2}{2}\right)\right] = q_i + \frac{1}{2}(V_V^2 - V_L^2)\frac{d}{dz}[\alpha\rho_V V_V] \tag{6}$$

where q_i is the heat transfer per unit volume between the liquid and vapor phases due to both temperature difference and phase change.

The two-phase model represent a set of five unknowns V_L, V_V, p, α and h_L (or T_L). The thermodynamic properties ρ_L, ρ_V, α and h_V were calculated using curves adjusted from data obtained from [4]. To closure the models, constitutive equations for f_L, F_{WL}, F_{WV}, F_{LV}, and q_i are required and will be explored next.

Constitutive equations

According to [5], the flow along capillary tubes can vary from bubble to annular flow. In the present work it is assumed bubble flow up to void fractions of 0.3, churn flow from void fractions between 0.3 and 0.8, and above 0.8 the flow is taken as annular. For bubble and annular flows constitutive equations are somehow available in the literature. This is not the case for churn flow, and in the present work the constitutive equations for such regime were obtained from interpolation between bubble and annular flow regimes.

(i) Single phase friction factor, f_L

It has been shown in [2], that Churchill's equation, [6], yields good agreement with experiments, and, therefore, it will be employed here.

(ii) Friction between tube wall and fluid, F_{WL}, F_{WV}

For the flow regimes considered here, there is no flow of vapor adjacent to the tube wall and, therefore, F_{WV} is taken as zero. The friction force between the liquid phase and the tube surface is obtained from [7]. In this model, F_{WL}, is given by,

$$F_{WL} = A_{WL} B_{WL} V_L \tag{7}$$

where A_{WL} and B_{WL} are, respectively, the contact area and the friction coefficient. For bubble and churn flow $A_{WL}=4(1-\alpha)/D$, whereas for annular flow $A_{WL}=4/D$. The friction coefficient for all three flow regimes is given by $B_{WL}=f_L\rho_L V_L/8$, where f_L is obtained from Churchill's equation, [6].

(iii) Interfacial force, F_{LV}

In the present work the interfacial force is modeled considering two effects: virtual mass and interphase friction. The expression for F_{LV} is then given by,

$$F_{LV} = \frac{2C_{fi}\rho_V\alpha^{0.5}}{D} V_i(V_V - V_L) + C_{mv}\left(V_V\frac{dV_V}{dz} - V_L\frac{dV_L}{dz}\right) \tag{8}$$

where C_{fi} is the interfacial friction factor calculated using the correlations given in [8] and [9] for the bubble and annular flow regimes, respectively. The virtual mass coefficient is obtained from [8] as $C_{mv}=0.5\alpha\rho_L$.

(iv) Interfacial heat transfer, q_i

The heat transfer between the phases are due to both temperature difference and phase change and can be expressed as,

$$q_i = a_i h_i (T_L - T_V) + \frac{d}{dz}(\alpha_V\rho_V V_V)(h_V - h_L) \tag{9}$$

where a_i is the interfacial area and h_i is the interfacial heat transfer coefficient. For bubble flow regime $a_i=6\alpha/d$ where d is the bubble diameter obtained from $d=(6\alpha/\pi N)^{1/3}$ in which N is the density of vapor nucleus in the fluid ($N=10^{11}$ nucleus/m^3, [10]); for annular flow regime $a_i=4\alpha^{0.5}/D$. The interfacial heat transfer coefficient for the bubble flow regime is obtained using Whitaker's correlation [11] with the Reynolds number defined as $\rho_L V_L d/\mu_L$. For the annular regime h_i is obtained from the Dittus-Boelter correlation [12] using $D\alpha^{0.5}$ as the characteristic diameter.

As mentioned before, all information related to the churn flow regime was obtained through interpolation between bubble and annular regimes. For the interfacial heat transfer coefficient the interpolation of a heat transfer parameter, as suggested by [13], allowed a smooth transition between the phases and eliminated some instabilities otherwise observed in the numerical solution.

Boundary Conditions

At the entrance of the capillary tube, z=0, the pressure is that measured at the wall of the capillary suction line minus a pressure drop (friction factor coefficient, K=0.8) associated with the entrance loss [14]. The temperature is obtained from the saturation temperature at the given pressure minus the degree of subcooling.

At the beginning of the two-phase regime, non-slipping is assumed between the phases and $V_L=V_V$; the pressure is that corresponding to saturation at the liquid temperature, and the void fraction is obtained from the vapor nucleus density N, assuming that the bubbles are spherical,

$$\alpha = N\pi d_o^3 / 6 \tag{10}$$

where d_o is the bubble initial diameter (d_o=2.5 x 10^{-5} m) taken from [10]. Along the bubble flow regime, the vapor nucleus density is kept the same and the bubble diameter is obtained from the void fraction using equation (10). The bubble diameter is required in evaluating the interfacial area a_i needed to calculate the heat transfer between the phases.

Choked flow is assumed at the tube exit and dp/dz is taken as minus infinity according to [15]. Numerically this condition is implemented evaluating dp/dz at each location along the tube until it becomes positive indicating that the maximum absolute value has been reached.

SOLUTION METHODOLOGY

The flow in consideration is parabolic and the solution of the governing equations is obtained using a fourth order Runge-Kutta method [16]. For a given mass flow rate the solution yields the tube length or, given the tube length, 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 such that corresponds to choked flow. When the tube length 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.

RESULTS AND DISCUSSIONS

Comparisons between the results obtained from the numerical model and the experimental data of [2] are presented in Figures 1 and 2 for two situations. As can be seen from the figures the two-fluid model yields a better agreement with the experiments than the homogeneous model. In Figure 1 the maximum deviation from the experimental values was 6% for the two-fluid and 17% for the homogeneous model. In Figure 2 the deviations are 4% and 7% for the two-fluid model and the homogeneous model, respectively.

Pressure profiles along the capillary tube are explored in Figures 3 and 4. The comparisons between computation and experiments are performed in two different ways. First the mass flow rate and the operating conditions are kept constant and the tube length is obtained from the two-fluid model (dotted line) and the homogeneous model (dashed line). Alternatively, the tube length can be provided and the mass flow rate can be calculated from the two-fluid model (solid line). The two-fluid model yielded better results than the homogeneous model for the prediction of the tube length, mass flow rate as well as for the pressure distribution.

Numerical results for the void fraction along the tube are shown in Figure 5, and in Figure 6 the local variation of the vapor quality is explored. It should be noted that near the tube exit the void fraction is very close to unity.

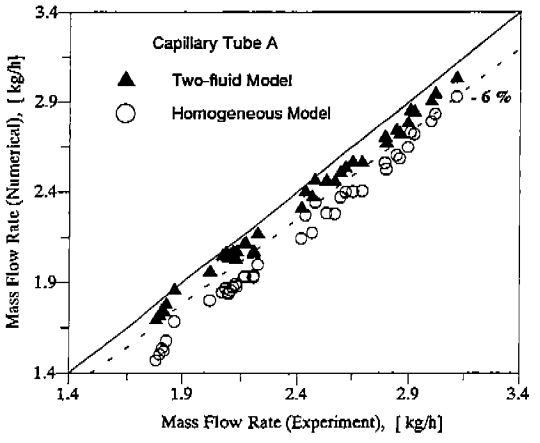

Fig. 1 - Comparison between measured and predicted mass flow rate for capillary tube A.

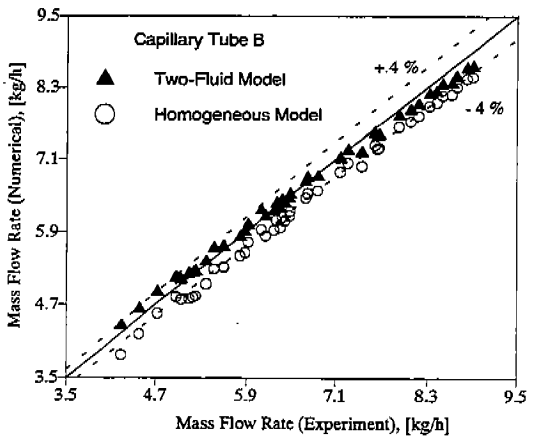

Fig. 2 - Comparison between measured and predicted mass flow rate for capillary tube B.

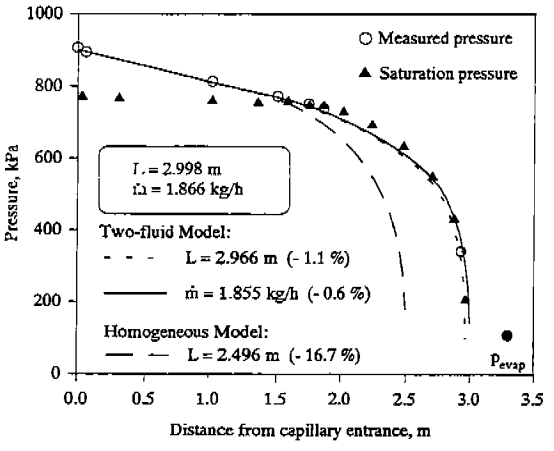

Fig. 3 - Pressure distribution along capillary tube A; p_{ent} = 905 kPa and ΔT_{sub} = 5.5 °C.

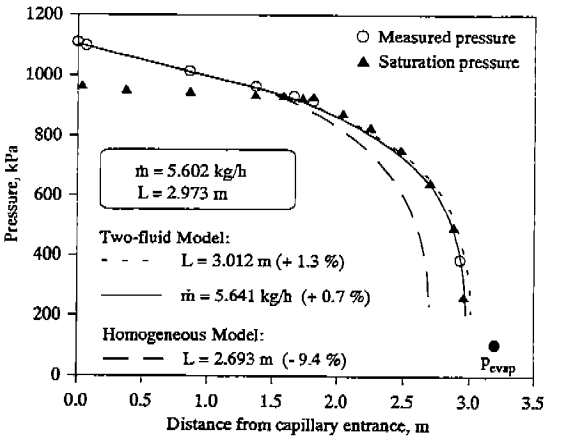

Fig. 4 - Pressure distribution along capillary tube B; p_{ent} = 1110 kPa and ΔT_{sub} = 5.2 °C.

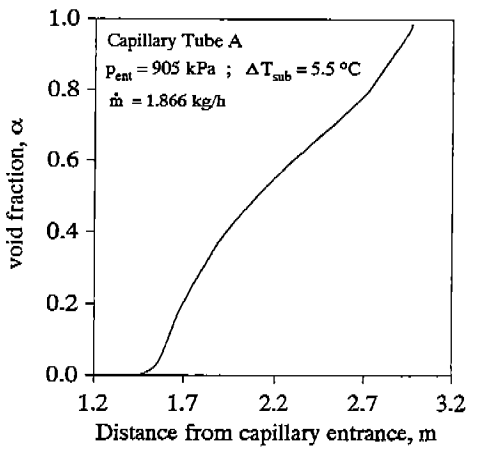

Fig. 5 - Void fraction along capillary tube A.

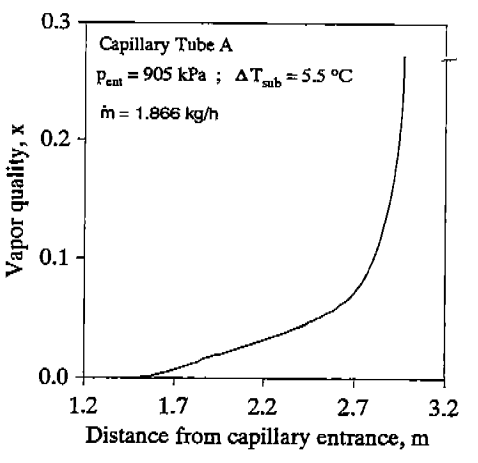

Fig. 6 - Vapor quality along capillary tube A.

Results for the phase velocities and temperatures have shown that the departures from equilibrium (both hydrodynamic and thermodynamic) are small. The maximum difference between liquid and vapor velocities occurred at the tube exit and was 2.4 %. Furthermore, the maximum temperature difference between the phases was 7 °C. The main outcome from this finding is that the refrigerant flow through capillary tubes can be considered as homogeneous. However, the better agreement with the experiments accorded to the two-fluid model is due to a better representation of the flow through the appropriate constitutive equations, and, at the moment, it appears that only through the two-fluid model the important features of the flow can be incorporated into the analysis.

CONCLUSIONS

A two-fluid model is employed for the first time to represent the HFC-134a flow through capillary tubes. Five equations are needed to solve for the velocity of both phases, pressure, void fraction and enthalpy of the liquid phase. From that and the thermodynamic relationship all other properties are determined. The set of five differential equations was solved using a fourth order Runge-Kutta algorithm.

Comparisons between the two-fluid model and experimental results indicated that the present model yields better results than the commonly adopted homogeneous model. These better results are not due to the nonequilibrium between the phase velocities and temperatures, which were both very small, but due to a better representation of the flow through more accurate constitutive equations afforded by the two-fluid model.

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