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A. A. Silva Huerta
Universidade de Sao Paulo

O. M. Silvares
Universidade de Sao Paulo

SIMULATION OF THE EFFECTS OF OIL IN CAPILLARY TUBES CONSIDERING A SEPARATED FLOW MODEL

Alex A. Silva Huerta and Otávio de Mattos Silvares
Department of Mechanical Engineering
Escola Politécnica Universidade de São Paulo USP
Brazil

ABSTRACT

A numerical model has been developed to simulate the effects of the oil on the performance of capillary tubes considering a separated flow model for refrigerants R-12 and R-134a. The numerical simulation integrates the one dimensional governing equations for the fluid flow (continuity, momentum and energy), using a step by step numerical method. The presence of oil in the refrigerant modifies the properties of the mixture. The separated flow model considers the two phases to be segregated with different properties and different mean velocities. This numerical model was used to illustrate several effects of the oil presence on the capillary tube design parameters and on value of the flow rates. The results show good agreement with experimental data.

NOMENCLATURE

A	=	area [m^2]
C	=	liquid fraction of oil in the liquid mixture
C_L	=	liquid fraction of oil in the mixture
D	=	tube inner diameter [m]
f	=	friction factor
G	=	mass velocity [kg/s.m^2]
h	=	specific enthalpy [J/kg]
$\dot{m}$	=	mass flow rate [kg/s]
Re	=	Reynolds number
T	=	temperature °C
T_N	=	$T_K/293,15$
T_K	=	temperature K
x	=	vapor quality
W	=	molecular weight
α	=	void fraction
μ	=	dynamic viscosity [Pa.s]
υ	=	specific volume [m^3/kg]
ρ	=	density [kg/m^3]
σ	=	surface tension [N/ m]
ε	=	relative surface roughness

Subscripts

ac	=	acceleration
l	=	liquid
fr	=	friction
l-M	=	liquid mixture (oil+refrigerant)
M	=	mixtures
O	=	oil
R	=	refrigerant
sat	=	saturation
v	=	vapor

INTRODUCTION

Capillary tube expansion devices are widely used in household refrigerators and small refrigerating systems, due to their simplicity, reliability and low cost. The greater number of works found in the literature have focused their attention on the homogeneous flow model, but some studies on the non-equilibrium phenomenon and on separated flow with different velocities for liquid and vapor phases has been developed in the last decade. Li et al. [1] developed a comprehensive drift- flux model that is essentially a separated flow model focused on the relative motion of the phases and recently Wong and Ooi [2] reported a comparison of the homogeneous flow and the separated flow models using the *Miropolskiy's* slip ratio. The effects of oil on performance of capillary tubes has been studied in a small number de published papers. Bolstad and Jordan [3] qualitatively studied the effects of oil entrainment on capillary tube capacity. They conjectured that oil lowered the vaporization pressure of the refrigerant, thus keeping the refrigerant in the liquid phase longer and reducing the overall restriction characteristics of the tube. Whitacre et al.[4] also tested qualitatively the effects of oil content on capillary tube performance showing that higher oil content could increase the flow rate by approximately 8 %. Wijaya [5] observed that the presence of oil did not affected significantly the performance of the capillary tubes tested. No previous study has been found on the simulation of the effects of the oil on the performance capillary tubes. The present study is a numerical model to predict the thermal and fluid-dynamic behavior of the evaporating flow of refrigerant fluid contaminated with oil through capillary tubes considering a separated flow model.

MODEL DESCRIPTION

The mathematical model considers adiabatic and non-homogeneous flow. It is important to mention that metastable flow phenomena are neglected in the model. The oil-refrigerant mixture properties are calculated considering that the below equations (1-7) are used after replacing all pure refrigerant properties with the properties of the liquid mixture of oil and refrigerant. The following simplifying assumptions were made: a) The capillary tube is a straight, horizontal and constant transversal area tube. b) The flow through the capillary tube is one-dimensional and adiabatic. c) The two phases are in local thermodynamic equilibrium. d) The thermophysical properties of the mixture are required only for the liquid phase because, the oil concentration in the vapor phase is negligible e) The refrigerant-oil mixtures are considered homogeneous, i.e., the oil is completely dissolved in the refrigerant in a liquid state f) The model assume that the mixture is an ideal solution where chemical reactions can not occur. The numerical solution has been effected based in a local analysis by discretization, where the conservation equations (continuity, energy and momentum) are solved for small control volumes. Thus the solution process is carried out at each instant, step by step, in the flow direction along capillary tube.

Oil Effects

A common problem in vapor-compression refrigeration systems is the migration of lubricating oil from the compressor into different system components, such as the condenser, the evaporator and the capillary tube. This migrated oil becomes a part of the working fluid in the system and can affect the pressure drop, the flash point and the choked flow conditions. Also, the refrigerant-oil mixture will have properties that differ from those of a pure refrigerant. Properties such as density, viscosity, specific heat, surface tension and pressure vapor are all variables that could affect the performance of the capillary tubes. In order to consider the oil effects on their calculation, it is necessary to know, with acceptable accuracy the thermodynamic and transport properties of the mixture of the lubricating oil with the refrigerant. The concentration of the oil in the mixture and the vapor quality are defined as:

$$C = \frac{\dot{m}_O}{\dot{m}_O + \dot{m}_{Rl} + \dot{m}_{Rv}} \quad (1)$$

The vapor quality is given by:

$$x = \frac{\dot{m}_{Rv}}{\dot{m}_{Rv} + \dot{m}_{Rl} + \dot{m}_O} \quad (2)$$

The concentration of oil in the liquid phase of the mixture is define as:

$$C_L = \frac{\dot{m}_O}{\dot{m}_{Rl} + \dot{m}_O} \quad (3)$$

With the above equations, it can be shown that

$$C_L = \frac{C}{C + (1-x)(1-C)} \quad (4)$$

Vapor pressure

The presence of oil reduces the vapor pressure of the refrigerant because its low vapor pressure. Sur and Azer, [6] applied *Raoult's* law to calculate the saturated vapor pressure of the refrigerant-oil mixtures.

$$p_{sat-M} = p_{sat-R}\left[\frac{1-C_L}{(1-C_L)+(W_R/W_O)\,C_L}\right] \quad (5)$$

Viscosity

The viscosity of oil for refrigeration use is two orders of magnitude greater than the refrigerant viscosity. Therefore, it is very important consider the viscosity effects on the mixture, Jensen and Jackman, [7] give the following equation:

$$\mu_{l-M} = \mu_{Rl}\exp\left[C_L\left(\mu_{Rl}/\mu_O\right)^{0,3}\right] \quad (6)$$

Density

ASHRAE [8] gives a predictive relationship for the densities of oil-refrigerant liquid mixtures.

$$\rho_{l-M} = \frac{\rho_{Rl}}{1-C_L(1-\rho_{Rl}/\rho_O)} \quad (7)$$

MATHEMATICAL FORMULATION

Single Phase Flow Region

For sub-cooled single-phase flow, the conservation of momentum gives:

$$-\frac{dP}{dz}=f_l\frac{G^2}{2D\rho_{l-M}}+G^2\frac{d}{dz}\left(\frac{1}{\rho_{l-M}}\right) \tag{8}$$

In this region the refrigerant can be assumed to be an incompressible fluid, so pressure drop is only caused by friction represented by the first term in right side of the equation. To determine the friction factor Haaland's [9] equation is used.

$$f_l=\left\{-1.8\log\left[\frac{6.9}{\mathrm{Re}_{l-M}}+\left(\frac{\varepsilon}{3.7D}\right)^{1.11}\right]\right\}^{-2} \tag{9}$$

Two-Phase Flow Region

The two-phase pressure drop along the tube can be expressed as the sum of the pressure drop due to tube wall friction and fluid acceleration. The separated flow model considers the phases to be artificially segregated with constant but not necessarily equal velocities for the vapor and liquid phases. The two-phase flow in the capillary tube requires the knowledge of the physical properties of the liquid and vapor phase. Therefore, these phases are considered completely segregated with different properties and mean velocities. Thus:

$$-\left(\frac{dP}{dz}\right)=-\left(\frac{dP}{dz}\right)_{fr}+\frac{d}{dz}\left[\frac{G^2x^2\upsilon_v}{\alpha}+\frac{G^2(1-x)^2\upsilon_{l-M}}{(1-\alpha)}\right]_{ac} \tag{10}$$

Deriving the acceleration term and neglecting the compressibility of the liquid phase, the total pressure gradient in the separated flow is expressed as:

$$-\left(\frac{dP}{dz}\right)=\frac{-\left(\frac{dP}{dz}\right)_{fr}+G^2\frac{dx}{dz}\left\{\left[\frac{2x\upsilon_\upsilon}{\alpha}-\frac{2(1-x)\upsilon_{l-M}}{(1-\alpha)}\right]+\frac{d\alpha}{dx}\left[\frac{(1-x)^2\upsilon_{l-M}}{(1-\alpha)^2}-\frac{x^2\upsilon_\upsilon}{\alpha^2}\right]\right\}}{1+G^2\left\{\frac{x^2}{\alpha}\left(\frac{d\upsilon_\upsilon}{dP}\right)+\left(\frac{d\alpha}{dP}\right)\left[\frac{(1-x)^2\upsilon_{l-M}}{(1-\alpha)^2}-\left(\frac{x^2\upsilon_\upsilon}{\alpha^2}\right)\right]\right\}} \tag{11}$$

To compute the frictional component of pressure drop in two-phase flow using the separated flow model, the *Friedel's* correlation is recommended by Whalley [10] for the two-phase multiplier. This equation was developed using a large data base of measurements and can be expressed as:

$$\left(-\frac{dP}{dz}\right)_{fr}=\left(\frac{-dP}{dz}\right)_{lo}\Phi^2{}_{lo} \tag{12}$$

The two-phase multiplier in the above equation can be expressed as follows:

$$\Phi^2{}_{lo}=(1-x)^2+x^2\frac{\rho_{l-M}}{\rho_v}\frac{f_{go}}{f_{lo}}+\frac{3{,}24[x^{0{,}78}(1-x)^{0{,}224}][\left(\frac{\rho_{l-M}}{\rho_v}\right)^{0{,}91}\left(\frac{\mu_v}{\mu_{l-M}}\right)^{0{,}19}\left(1-\frac{\mu_v}{\mu_{l-M}}\right)^{0{,}7}]}{Fr^{0{,}045}We^{0{,}035}} \tag{13}$$

The Froude and Weber numbers can be expressed as:

$$Fr=\frac{G^2}{gD\rho_h{}^2} \qquad We=\frac{G^2D}{\rho_h\sigma_{l-M}} \qquad \rho_h=\left(\frac{x}{\rho_v}+\frac{1-x}{\rho_{l-M}}\right)^{-1}$$

where f_{lo} and f_{go} are respectively the friction terms considering uniquely the liquid phase and vapor phase.

To evaluate the pressure drop due to acceleration it is necessary to know the void fraction with can be well correlated using the following equation given by Whalley [10]

$$\alpha = \frac{x\upsilon_\upsilon}{x\upsilon_\upsilon + S(1-x)\upsilon_{l-M}} \tag{14}$$

The differentiation of void fraction with respect to quality and pressure can be expressed as:

$$\left(\frac{\partial\alpha}{\partial x}\right)_P = \frac{\alpha}{x} - \frac{\alpha(\upsilon_\upsilon - S\upsilon_{l-M})}{x\upsilon_\upsilon + S(1-x)\upsilon_{l-M}} \tag{15}$$

$$\left(\frac{\partial\alpha}{\partial P}\right)_x = \frac{\dfrac{d\upsilon_\upsilon}{dP}\left[\dfrac{x^2\upsilon_v}{\alpha} - S(1-x)x\upsilon_v\right]}{\dfrac{x\upsilon_v}{\alpha}\left[x\upsilon_v + S(1-x)\upsilon_{l-M}\right]} \tag{16}$$

And, the derivative of quality with respect the length, to express the change of quality along tube:

$$\frac{dx}{dz} = \frac{1}{h_{lv}}\frac{dh}{dz} + \left(\frac{\partial x}{\partial P}\right)_h \frac{dP}{dz} \tag{17}$$

The separated model assumes that both liquid and vapor phases move at different velocities. Therefore, is very important to determine the slip ratio (S) due to differences in the thermophysical and flow properties of the refrigerant-oil mixtures in the liquid and vapor phase. A number of alternative correlation's are available for slip ratio (S). The correlation reported by Premoli et. al [11] takes account the mass flux effects and also other physical property effects such as viscosity and density, and can be expressed as:

$$S = E_1\left(\frac{y}{1+yE_2} - y[E_2]\right)^{0,5} \tag{18}$$

$$E_1 = 1 + 1{,}578\,\mathrm{Re}^{-0{,}19}\left(\frac{\rho_{l-M}}{\rho_v}\right)^{0{,}22} \qquad E_2 = \left(\left[0{,}0273 We\,\mathrm{Re}^{-0{,}51}\left(\frac{\rho_{l-M}}{\rho_v}\right)^{-0{,}08}\right]\right)$$

where:

$$y = \frac{\beta}{1-\beta} \qquad \beta = \frac{x\,\rho_{l-M}}{x\,\rho_{l-M} + \rho_\upsilon(1-x)} \qquad \mathrm{Re} = \frac{GD}{\mu_{l-M}} \qquad We = \frac{G^2 D}{\sigma\rho_{l-M}}$$

Substituting equations (12) (14), (15), (16), (17) and (18) into equation (11), gives an expression for the separated flow model. The critical condition is determined as the denominator of equation (11) approaching zero. For the simulation in which mass flow rate is unknown, this is continuously adjusted until the calculated tube length equals to the prescribed tube length.

RESULTS AND DISCUSSIONS

A comparison with available experimental data in figure 1. show that the pressure drop along capillary tube considering the effects of oil can be expressed using the separated flow model in the performance of capillary tubes. The mass flow rates determined using the separated flow model are compared with experimentally obtained values given in the literature by Wijaya [5] and Dirik [4] for R-134a. The deviations approximately of 7% may be considered to be reasonable and satisfactory. Figure 2. shows a comparison of the pressure distribution along the capillary tube for the homogenous and separated flow model. Can be seen that the homogeneous flow model differs extremely at the region near the choked condition, the main reason may be the effects of the two-phase pressure drop due to fluid acceleration. Compared with Paiva's et al [13] experimental date for the tube length of L=2.85 the figure 2. show that the separated flow model give better results.

Figure 3. shows the mass flux of refrigerant as a f unction of the condensing temperature. Can be seen that the mass flow rate increases when the condensing temperature increases. The presence of oil into the flow of refrigerant produces a reduction in the flow mass rate of approximately 3% for oil concentrations of 5% as showed in the figure 3. below. These effects can be compared with experimental data by Wijaya [5] for subcooling temperatures of 5.67 °C [10 °F] and 16.67 °C [30 °F].

The presence of oil into the flow of refrigerant produces a reduction in the length of the capillary tube showed in figure 4. This reduction is originated for the influence of viscosity, density and pressure vapor in the refrigerant-oil properties and express the decrease of the flow mass rate. From the different situations tested for the mass flow rate the results indicate that the presence of oil cause an increase of the length of the liquid phase of approximately 5% but the total length of capillary tube is reduced.

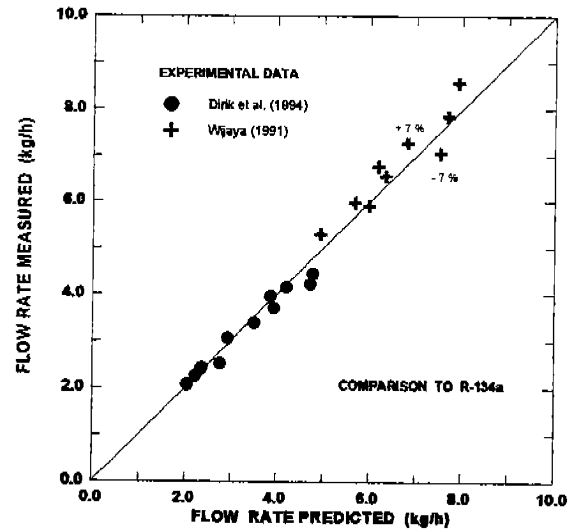

Figure 1. Comparison with experimental data

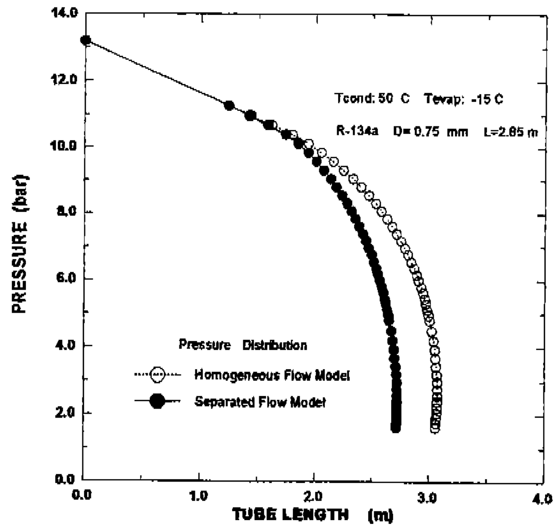

Figure 2. Comparison with the homogeneous flow model

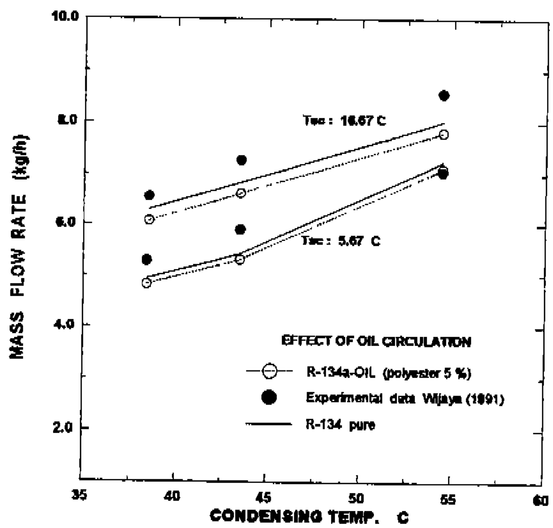

Figure 3. Predicted and experimental results of the effect of oil on mass flow rate

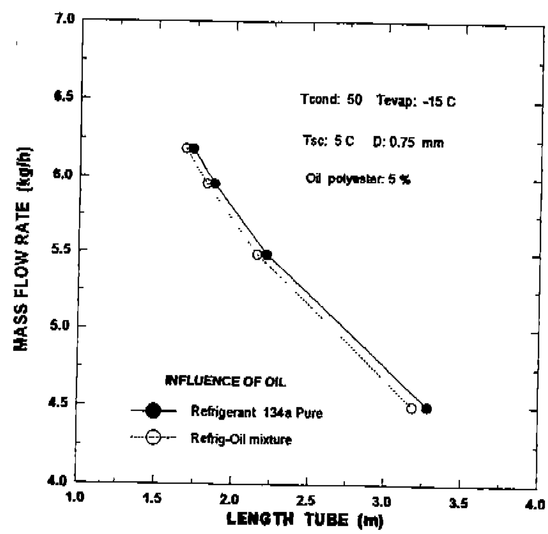

Figure 4. The influence of oil concentration on the length of capillary tube.

CONCLUSIONS

A numerical method for analyzing the influence of oil on the performance of capillary tubes has been developed using the separated flow model. A comparison with experimental data available in the literature shows good agreement. The results indicate that even a small amount of oil added to the refrigerant flow cause an increase of the length of the liquid phase but the total length of capillary tubes is reduced due that the increase in liquid length produces a corresponding decrease in overall resistance to flow. The resulted show that the presence of oil cause a decrease in the refrigerant mass flow rate in capillary tubes of approximately 3% for oil concentrations of 5 %. These results show that the separated two-phase flow model is adequate to predict mass flow rate for pure refrigerant and refrigerant-oil mixtures.

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APPENDIX

This appendix is concerned with the determination of physical properties of oil developed by Huerta et al.[14]. All thermodynamics properties for pure refrigerants are evaluated through REFPROP 5.12 software.

	POLYESTER	NAPTHENIC
VISCOSITY	$\mu_O = [1{,}44E5 - 5{,}84E3 T_K + 89{,}58 T_K^2 - 4{,}77 T_K^3] 10^{-6}$	$\log \mu_O = [10{,}0 - 11{,}948 T_N + 3{,}565 T_N^2] 10^{-3}$
DENSITY	$\rho_O = 100{,}884E4 + 0{,}75352\ T_K^3$	$\rho_O = 1{,}0625E3 - 0{,}1666E3\ T_N$
SUF.TENSION	$\sigma_O = 29{,}1\ \ 10^{-3}$	$\sigma_O = 31{,}0\ \ 10^{-3}$
MOL. WEIGHT	$W_O = 550{,}0$	$W_O = 325{,}0$