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Pressure Distribution of Refrigerant Flow in an Adiabatic Capillary Tube

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ABSTRACT This paper presents the results from a numerical study on the local pressure distribution of some common traditional and alternative refrigerants flowing in adiabatic capillary tubes. The present model developed from the basic conservation law of mass, energy and momentum includes various relevant parameters. A homogeneous flow model is used in the two-phase flow region. Numerical results show that the alternative refrigerants used as examples in the present study consistently give higher pressure gradients than the traditional refrigerants.The present model can be used to simulate and compare the flow characteristics of the other refrigerants. It may be also an important tool for selecting the length of the capillary tube used in household refrigerators and freezers for given operating conditions.

KEYWORDS: two-phase flow, local pressure distribution, pressure gradient, refrigerant, capillary tube.

INTRODUCTION

The small bore capillary tube is the most widely used as expansion device in small domestic vapor compression air conditioners and refrigerators. The main concern in practical consideration is to determine the appropriate tube diameter and length at a given operating condition. The investigation on the flow characteristics in the capillary tubes has received the most attention[1-9] Bansal et al[1] developed a homogeneous two-phase flow model, CAPIL, to study the performance of adiabatic capillary tubes. They used the REFPROP data base to calculate the refrigerants' thermodynamic and thermophysical properties. Sami et al[7] proposed a numerical model for predicting the capillary tube performance of pure refrigerants (R12, R22, R134a) and binary mixtures (R410A, R410B, R507, R32/R134a). Wong et al[9] developed a homogeneous two-phase flow model to simulate the flow characteristics of R12 and R134a. The results showed that the differences in flow characteristics are due to minor differences in refrigerant properties. Wongwises[10] provided the results of simulations using an adiabatic capillary tube model. The investigation was concerned about making comparisons of the pressure distributions between various alternative mixtures of refrigerant. Jung et al[3] modified the Stoecker's model[11] to provide simple correlations for sizing the capillary tubes used with R22, R134a, R407C and R410A. Effects of the sudden contraction at capillary tube inlet, degree of subcooling, friction factors and various viscosity models were discussed. Melo[5] investigated experimentally the effects of the condensing pressure, size of adiabatic capillary tube, subcooling and the types of the refrigerant (R12, R134a and R600A) on the mass flow rates.

There is relatively little information in the open literature on comparisons of flow characteristics for traditional and alternative refrigerants flowing in a capillary tube. To be a guide-line in the future for selecting the appropriate refrigerants, in the present study, the main concern is to study on the pressure distribution of various refrigerants along the capillary tube and to compare the flow characteristics between some pairs of refrigerants.

MATHEMATICAL MODEL

The flow of refrigerant in a capillary tube used as an expansion device in the refrigerating system is divided into two regions; a single-phase sub-cooled liquid region and a two-phase vapour-liquid flow region.

Single-Phase Sub-Cooled Liquid Region

The single-phase sub-cooled liquid region is the region from the capillary tube inlet to the position where the saturation pressure corresponds to the temperature at the capillary inlet. For steady and fully-developed incompressible flow, the integral form of the momentum equation at distance dz in a capillary tube is

$$A_o dP + \tau_W(\pi d)dz = 0 \qquad (1)$$

where τ_W is the wall shear stress and defined as

$$\tau_W = f\frac{(\rho_L V_L^2)}{8} \qquad (2)$$

where f is the Moody friction factor and can be determined from Colebrook's correlation as follows;

$$\frac{1}{f^{0.5}} = -2\log\left(\frac{e/d}{3.7}+\frac{2.51}{Re_L f^{0.5}}\right) \qquad (3)$$

Substituting Eq (2) into Eq (1), the single phase length (L_{SC}) of the capillary tube is obtained :

$$L_{SC} = \left(p_i - p_{sat}\right)\Big/\left(f\frac{G^2}{2\rho_L d}\right) \qquad (4)$$

where the total mass flux (G) is the total mass flow rate of fluid divided by total cross-sectional area of the tube (A_o).

Two-Phase Flow Region

In the present study, the model used in the two-phase region is derived from the one dimensional homogeneous two-phase flow assumption. The model is based on that of Wong et al[9] and Wallis.[13] The basic physical equations governing the capillary tube flow are the conservations of mass, energy and momentum.

First, the specific enthalpy at a saturation state of a pure substance having a specific quality can be determined by using the definition of quality (x) as follows;

$$h = h_L(1-x) + h_G x \qquad (5)$$

With no applied works and neglecting the elevation changes, the following form of energy equation for refrigerant flow in a capillary tube is obtained:

$$\frac{d}{dz}\left(xh_G + (1-x)h_L + \frac{V^2}{2}\right) = 0 \qquad (6)$$

where the quality of the mixture in a saturated condition (x) is the ratio of the vapour mass flow rate to total mass flow rate and the velocity of each phase is equal ($V = V_G = V_L$).

For a pure substance in the equilibrium homogeneous two-phase region, the enthalpies and densities are functions of pressure ($h = h(p)$, $\rho = \rho(p)$).

Mass fluxes of vapour and liquid phase (G_G and G_L) are the mass flow rates of the vapour and liquid divided by the cross-sectional area of the capillary tube, so

$$G_G = G_L = \rho V \qquad (7)$$

Void fraction (α) is a terminology in the two-phase flow study, it represents the time-averaged fraction of the cross-sectional area or of the volume which is occupied by the vapour phase. The general equation for determining the void fraction in the homogeneous flow is

$$\alpha = \frac{1}{1+\left(\frac{1-x}{x}\frac{\rho_G}{\rho_L}\right)} \qquad (8)$$

Actual average velocity of vapour and liquid phases (V_G and V_L) can be obtained from

$$V = G\upsilon = G(x\upsilon_G + (1-x)\upsilon_L) \qquad (9)$$

After all above equations are rearranged, the following form of the total pressure gradient is obtained:

$$\frac{dP}{dz} = -\frac{dx}{dz}\left(\frac{A}{B}\right) \qquad (10)$$

where $A = h_{LG} + G^2\upsilon\upsilon_{LG}$

$$B = x\frac{dh_G}{dP} + (1-x)\frac{dh_L}{dP} + G^2\upsilon\left[x\frac{d\upsilon_G}{dP} + (1-x)\frac{d\upsilon_L}{dP}\right]$$

The total pressure gradient $\left(\frac{dP}{dz}\right)$ is often expressed as the sum of the three distinct components, so

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_f + \left(\frac{dP}{dz}\right)_a + \left(\frac{dP}{dz}\right)_g \qquad (11)$$

Three terms on the right hand side represent the frictional, accelerational, and gravitational components of the total pressure gradient, respectively.

Frictional term in Eq (11) can be obtained from

$$\left(\frac{dP}{dz}\right)_f = \frac{-f_{tp}\, G^2 (x\upsilon_G + (1-x)\upsilon_L)}{2d} \qquad (12)$$

Accelerational term in Eq (11) can not be measured directly. However, it can be calculated from the momentum flux as follows;

$$\left(\frac{dP}{dz}\right)_a = = -G^2 \left(\frac{dP}{dz}\right)\left(x\frac{d\upsilon_G}{dP} + (1-x)\frac{d\upsilon_L}{dP}\right) - G^2 \upsilon_{LG}\frac{dx}{dz} \qquad (13)$$

Gravitational term in Eq (11) can be negligible because the flow is horizontal.

Substituting Eqs (12) and (13) into Eq (11), gives

$$\frac{dx}{dz} = \frac{\left(\frac{dP}{dz}\right)_f - C\frac{dP}{dz}}{D} \qquad (14)$$

where

$$C = 1 + G^2\left(\frac{x d\upsilon_G}{dP} + \frac{(1-x)\, d\upsilon_L}{dP}\right)$$

$$D = G^2 \upsilon_{LG}$$

The two-phase friction factor (f_{tp}) can be calculated from Colebrook's equation with the Reynolds number being defined as:

$$Re = \frac{Gd}{\mu_{tp}} \qquad (15)$$

The following Dukler 's equation[14] is used to determine μ_{tp}:

$$\mu_{tp} = \frac{x\upsilon_G \mu_G + (1-x)\upsilon_L \mu_L}{x\upsilon_G + (1-x)\upsilon_L} \qquad (16)$$

where μ_L and μ_G are absolute viscosity of liquid and gas, respectively.

Solution Methodology

In the present study, the following pairs of refrigerants whose properties are very similar are chosen as examples and used in the present simulation;

- R12 vs R134a,
- R12 vs R409A (R22/124/142b; 60/25/15)
- R12 vs R409B (R22/124/142b; 65/25/10)
- R501 (R22/12; 75/25) vs R402A (R125/290/22; 60/2/38)
- R501 (R22/12; 75/25) vs R402B (R125/290/22; 38/2/60)

All thermodynamic and transport properties of refrigerants are taken from REFPROP[12] and are developed as a function of pressure. The calculation is divided into two steps; sub-cooled single-phase region and two-phase vapour liquid region. Initial conditions required in the calculation are temperature and pressure of refrigerant at capillary tube inlet, roughness and diameter of the capillary tube and mass flow rate of refrigerant. In the single-phase flow region, after substituting the friction factor calculated from Colebrook equation and the saturation pressure at the capillary inlet temperature into Eq (4), the single-phase region length is obtained. The end condition of the single phase flow region is used to be an inlet condition of the two-phase flow region. The Runge-Kutta method is used to solve Eqs (10) and (14) in the two-phase flow region. The calculation in two-phase flow region is terminated when the flow is at the critical flow condition. Total capillary tube length is the sum of the single and two-phase length.

Results and Discussion

The refrigerant mass flow rate, temperature and pressure at the inlet of the capillary tube, diameter and relative roughness of the tube were each varied in turn to investigate the effect on the total length of capillary tube. The results from the simulation are properties at each position along the capillary tubes. Figures 1, 2, 3 and 6 show the variation of the local pressure of all refrigerants with position along the capillary tube. In the sub-cooled liquid region, due to friction, the pressure of refrigerant drops linearly. After the position of the inception of vaporization due to both friction and acceleration, the pressure of refrigerant drops rapidly and more rapidly as flow approaches the critical flow condition. However, in real situation, the actual point of inception of vaporization may not occur at the end of the sub-cooled liquid region because of the delay of vaporization. In order to validate the present model, comparisons are made with limited available measured data of Li et al[4] which were obtained from 10 pressure transducers installed along the capillary tube. Figures 1 and 2 also compare the simulation results obtained from the present model with the R12 data measured by Li et al.[4] The model is shown to fit the data quite well.

The experimental conditions used by Li et al[4] are given in Table 1.

Table 1. Experimental conditions of Li et al.[4]

Case	Ti (°C)	Pi (bar)	m (g/s)	d (mm)	e/d
1	31.40	9.67	1.13	0.66	0.003
2	23.40	7.17	0.844	0.66	0.003

Comparison on the pressure distributions of R12 and R134a (Figures 1 and 2), in general, the flow of R12 in the capillary tube gives a lower pressure gradient than that of R134a. In the other words, the total tube length for R134a is shorter. Comparisons on the pressure drop characteristics for the rest of each pair of refrigerant (R12 vs R409A and R409B; R501 vs R402A and R402B) show that for all cases in the single-phase region, the traditional refrigerant gives a slightly lower pressure gradient than the alternative refrigerants. In the two-phase flow region, the traditional refrigerant gives a momentous lower pressure gradient than the alternative refrigerant.

Figures 4 and 7 show the quality distributions along the capillary tube. For all cases, the quality in the single phase region is zero till the flash point at which the two-phase region begins and then increases more rapidly in a non-linear fashion as the critical flow condition is approached. It is also shown that in general, traditional refrigerants vaporize later than their corresponding alternative refrigerants. Figures 5 and 8 show the distributions of temperature along the capillary tube for each pair of refrigerant type. In all cases, in the single phase region, because the flow is incompressible, the refrigerant temperature along the capillary tube remains constant. After the position of the inception of vaporization, the temperature drops rapidly as the flow approaches the critial flow condition. In general, the traditional refrigerants give longer total capillary tube length.

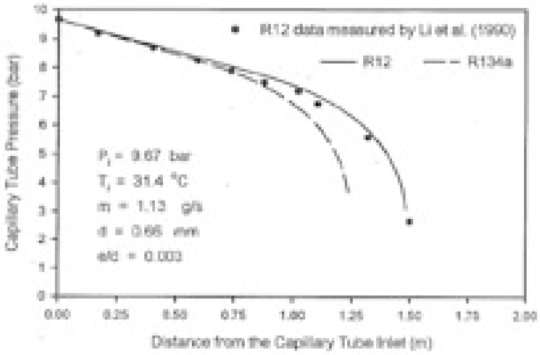

Fig 1. Comparison of pressure distributions along the capillary tube for R12 and R134a.

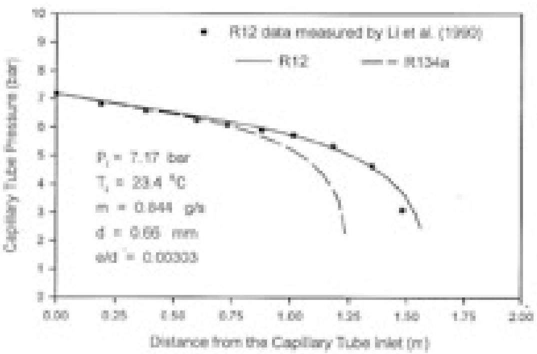

Fig 2. Comparison of pressure distributions along the capillary tube for R12 and R134a.

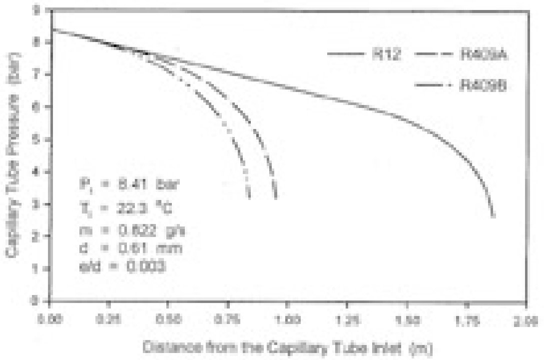

Fig 3. Comparison of pressure distributions along the capillary tube for R12, R409A and R409B.

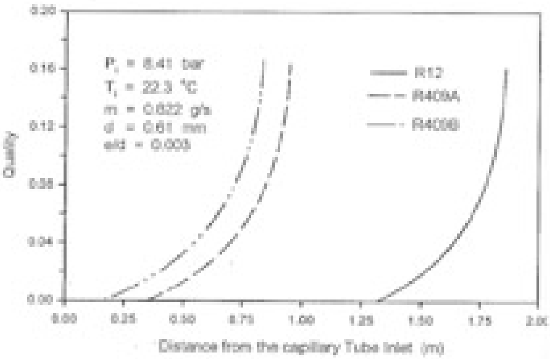

Fig 4. Comparison of quality distributions along the capillary tube for R12, R409A and R409B.

Conclusions

A homogeneous two-phase flow model is modified to study the flow characteristics of some refrigerants flowing in adiabatic capillary tubes. The basic governing equations are based on the conservations of mass, energy and momentum. The differential equations obtained are solved simultaneously by the Runge-Kutta method. It is found that even the differences in properties of each pairs of the refrigerants is small, the differences on the overall system performance may be meaningful. By varying various input parameters, it is found that the traditional refrigerants consistently give lower pressure gradients and give longer total length of the capillary tube.

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Nomenclature

A_o cross-sectional area of tube, m^2
d diameter of the capillary tube, m
e roughness, m
f friction factor
G mass flux, kg/s m^2
h specific enthalpy, kJ/kg
m mass flow rate, kg/s
P pressure, MPa
Re Reynolds number
T Temperature, °C
V velocity, m/s
x quality
z axial; direction or length, m
α void fraction
μ absolute viscosity, Pa s
υ specific volume, m^3/kg
ρ density, kg/m^3
τ shear stress, N/m^2

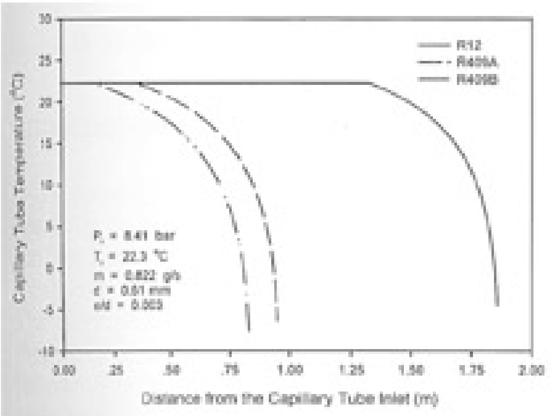

Fig 5. Comparison of temperature distributions along the capillary tube for R12, R409A and R409B.

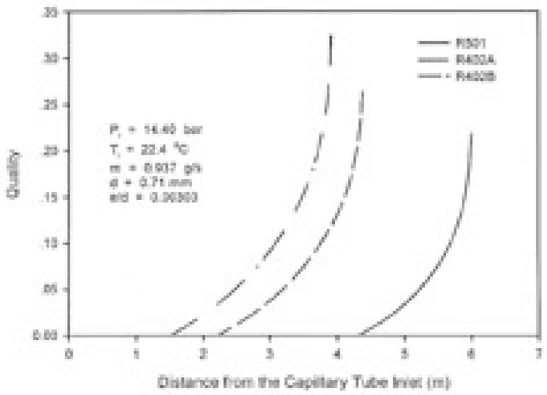

Fig 7. Comparison of quality distributions along the capillary tube for R501, R402A and R402B.

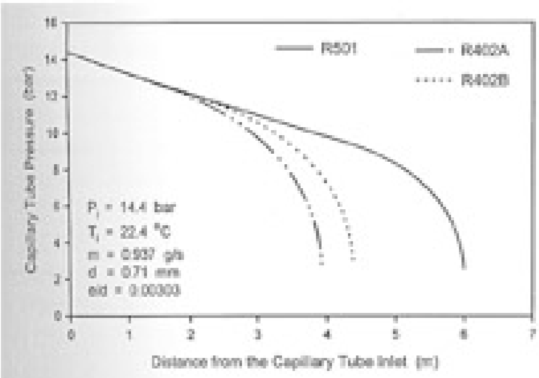

Fig 6. Comparison of pressure distributions along the capillary tube for R501, R402A and R402B.

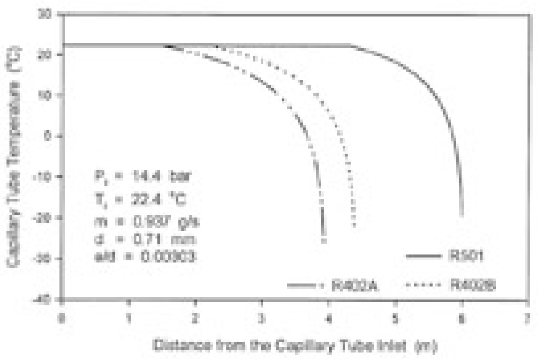

Fig 8. Comparison of temperature distributions along the capillary tube for R501, R402A and R402B.

Subscripts

a	accelerational	f	frictional
g	gravitational	G	vapour
i	capillary tube inlet	L	liquid
sat	saturation		
SC	single-phase sub-cooled		
tp	two-phase	w	wall

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