Also, in order to calculate the gas-wall shear stress, Agrawal [14] suggested the Blasius equation for smooth pipe flow by substituting $C_k = 0.079$ and $n = 0.25$ into eq.(4).

Preston [2] proposed a simple method of determining local turbulent skin friction on a smooth surface by means of a round Pitot tube or impact tube resting on the surface. Assuming that the inner law

$$u / u_\tau = f(u_\tau y / \nu) \tag{9}$$

is valid in a region close to the surface in both fully developed pipe flow and boundary layer flow, and the functional relationship is the same for both types of flow, the following equation relating the wall shear stress (τ_w) and the pressure difference between the total pressure recorded by the preston tube and static pressure at wall (ΔP_p) is developed:

$$\log_{10} \frac{\tau_w d^2}{4\rho\nu^2} = A + B \log_{10} \frac{(\Delta P_p) d^2}{4\rho\nu^2} \tag{10}$$

where constants (A = -1.396, B = 0.875) were obtained by calibration. Head and Rechenberg [5] tested and compared the experimental data obtained from Preston and Stanton tubes, both for pipe and boundary layer flows. They confirmed the validity of Preston's method.

Patel [6] suggested that Preston's original calibration was in error. He made a more extensive study of the Preston tube and compared the various proposed correlations. Over a wide range of experimental conditions, Patel's calibration equation is obtained as follows:

$$y^* = 0.8287 - 0.1381 x^* + 0.1437 x^{*2} - 0.0060 x^{*3} \tag{11}$$

where

$$x^* = \log_{10} \frac{(\Delta P_P) d^2}{4\rho\nu^2} \quad \text{and} \quad y^* = \log_{10} \frac{\tau_w d^2}{4\rho\nu^2} \tag{12}$$

The simplicity, low cost and sufficient accuracy of the Preston tube makes it useful and suitable for determining the shear stress distribution at the solid boundaries in two-phase stratified flow.

Results and Discussion

The Preston tubes were calibrated over a wide range of single-phase air flow rates by measuring the air flow rate, the pressure drop along the test section (ΔP_L) and the pressure difference between the Preston and static probes (ΔP_P). To examine the symmetry of the shear stress distribution, the preston tube was rotated for measuring the pressure differences at various circumferential locations around the pipe (θ). The experiment was performed for both sizes of pipe diameters (54 mm. and 29 mm.). From the momentum balance equation (eq.2) for single-phase air flow and substituting the measured pressure drop along the test section (ΔP_L) and α (= 0), the fully developed wall shear stress can be estimated.

Assuming the existence of a region near the surface in which conditions are functions only of the skin friction, the relevant physical constants of the fluid and a suitable length, a universal non-dimensional relationship between the total pressure recorded and the static pressure at the wall in terms of the skin friction proposed by Preston [2], namely

$$\frac{\tau_w d^2}{4\rho v^2} = f\left(\frac{(\Delta P_p)d^2}{4\rho v^2}\right) \quad (13)$$

will be used to form the correlation. On this assumption, this relationship is independent of the pressure gradient. Assuming symmetry about the center of the circular pipe, the wall shear stress terms along the horizontal inner surfaces for each run were curve-fitted by the least squares method to a third degree polynomial as follows

$$y^* = 0.5947 + 0.3975x^* - 0.0168(x^*)^2 + 0.0071(x^*)^3 \quad (14)$$

In the stratified two-phase flow study, the air flow rate was increased by small increments while the water flow rate was kept constant. The smooth and two dimensional wavy flows were obtained in accordance with results obtained from the study. At each flow rate of air and water, the pressure difference between Preston and static probes was recorded at various circumferential location. To do this, the test section which was attached with Preston and static probes was rotated clockwise in small steps towards the interface. Substituting the measured ΔP_p into eq.(14), the gas-wall shear stresses were determined. The measured wall shear stress distributions were found to be strongly influenced by the flow pattern which exists. A number of graphs can be drawn from the experimental results but because of space limitation, only typical results are shown. Figures 5 to 7 show typical wall shear stress distributions encountered in two-phase smooth stratified flow. The measured wall shear stress decreases slightly with circumference distance to a minimum point. As a result of a small wavy interface, it then increases sharply to a specific shear stress value at the interface. Typical gas-wall shear stress distributions encountered in two dimensional wavy flow are shown in Figures. 8 to 10. As a result of wavy interface, the gas-wall shear stress increases slightly and approaches a specific value at the interface. Because the amplitude of the water layer fluctuation increases slightly with the air flow rate, the gas-wall shear stress for higher air flow rate is higher than for lower flow rate. The wall shear stress distributions obtained are the same for both pipe sizes. The literature contains considerable data on wall shear in horizontal two-phase flow, but only the works of Kowalski [9] and Newton et al. [11] were performed in circular pipes. The present experimental results are also compared with those and some qualitative agreement is noted. Kowalski did

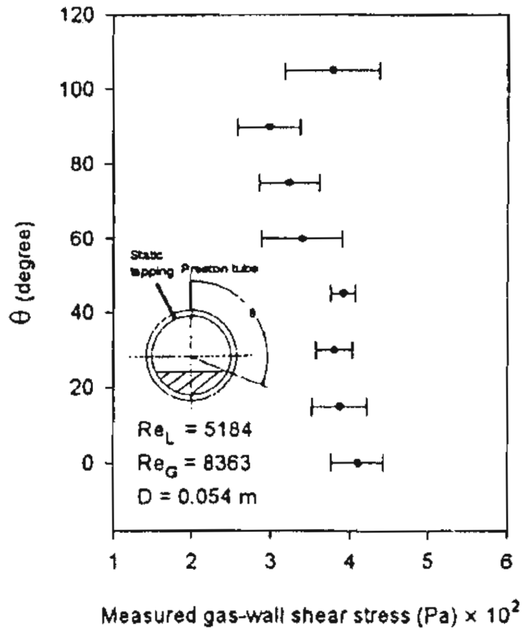

FIG. 5
Gas-wall shear stress distribution around circumference of pipe for stratified smooth flow

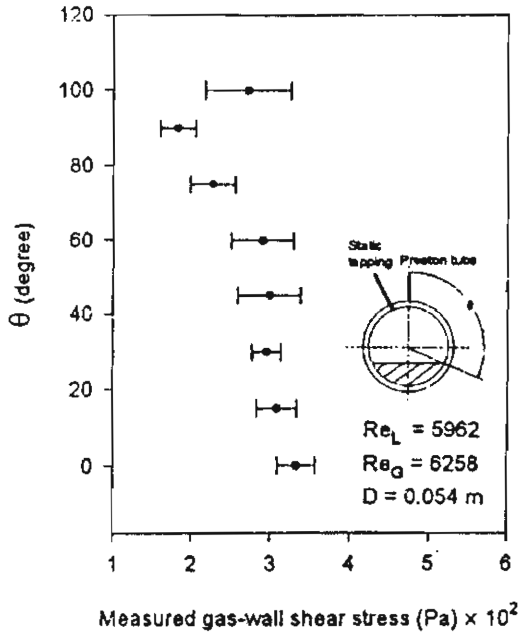

FIG. 6
Gas-wall shear stress distribution around circumference of pipe for stratified smooth flow

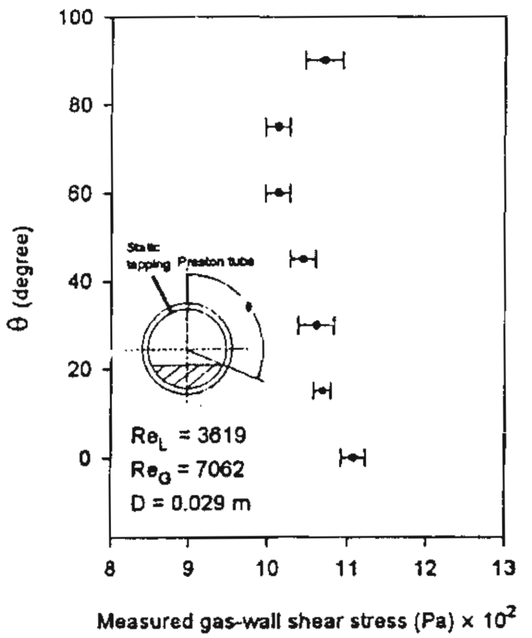

FIG. 7
Gas-wall shear stress distribution around circumference of pipe for stratified smooth flow

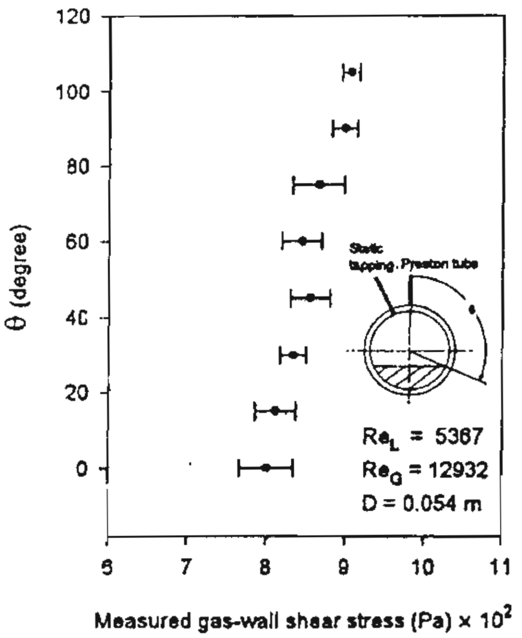

FIG. 8
Gas-wall shear stress distribution around circumference of pipe for stratified wavy flow

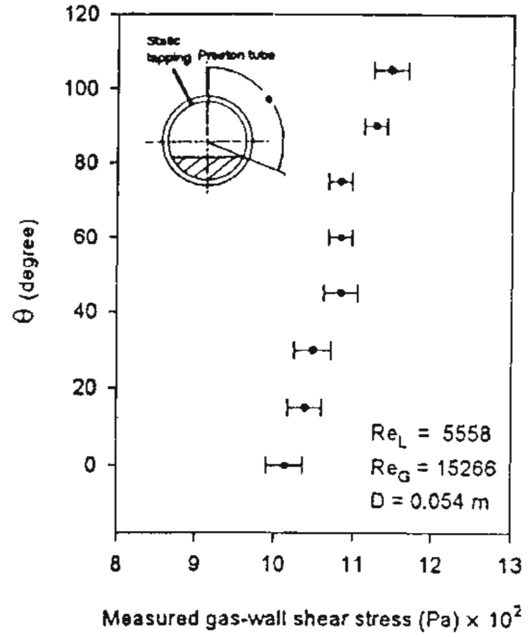

FIG. 9
Gas-wall shear stress distribution around circumference of pipe for stratified wavy flow

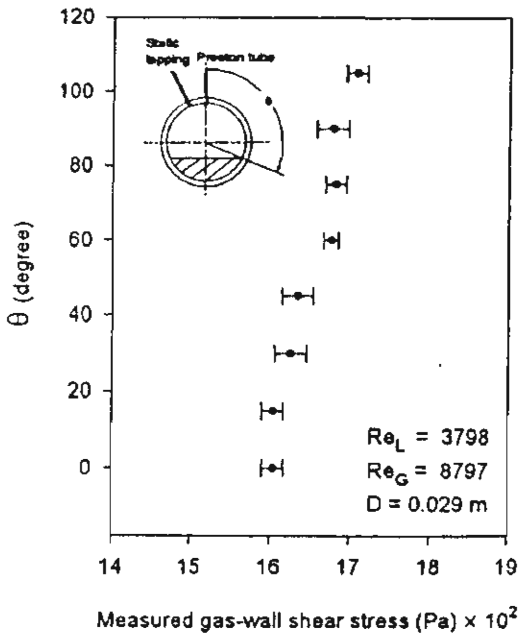

FIG. 10
Gas-wall shear stress distribution around circumference of pipe for stratified wavy flow

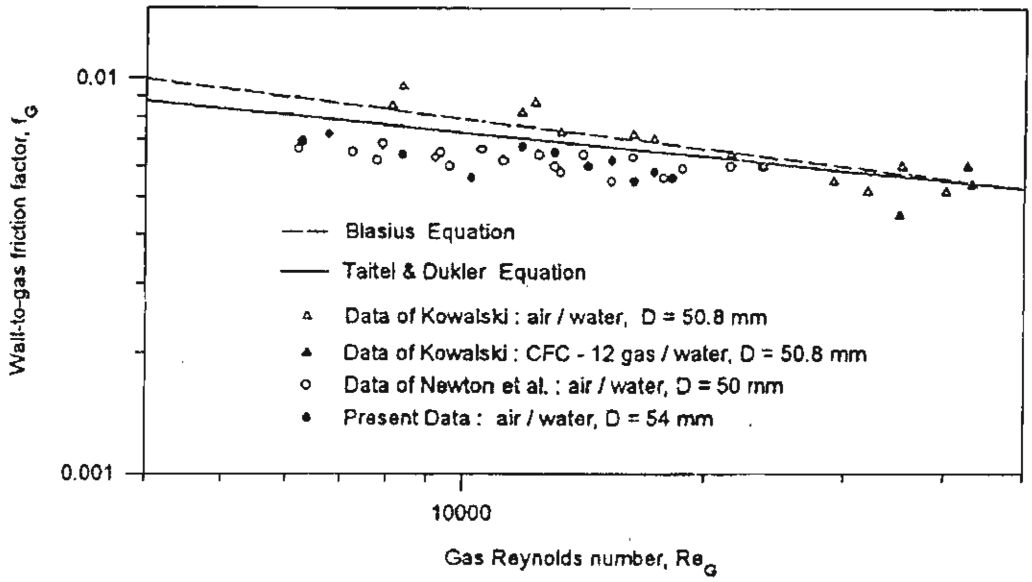

FIG. 11
Comparison of present measured gas-wall friction factor with results reported in the literature

not measure the shear stress at the region very close to the liquid-gas interface. However his existing data show that some major variations occured close to the interface. The measured wall shear stress distribution is different from those that were obtained by Davis [7] because of the difference in pipe configuration. The average gas-wall friction factors were calculated from eq. (3) and were plotted against the gas Reynolds number (Re_G) obtained from eq.(5). It can be observed that the friction factor decreases slightly with increasing Re_G. Figure 11 compares the experimental values of the gas-wall friction factor with theoretical values for smooth pipe flow given by the Blasius and Taitel & Dukler equations. Concerning the gas flow rates, the Blasius and Taitel et al. relationships overpredict the friction factor for Re_G by less than 20,000. The friction factor data obtained by Kowalski [9] and Newton et al. [11] are also compared with the present results. Kowalski's data points were taken from a log scale, thus were a cause of some uncertainties. The present measurements agree well with those from Newton et al. for D = 50 mm.

Conclusion

This paper presents the results of the experimental work on the gas-wall shear stress for the co-current air-water stratified flow in pipes. The measurement of wall shear stress by a calibrated Preston tube appears to be accurate and convenient. Preston and static probes are installed on the dry walls of the circular pipe and are calibrated by measuring the pressure drops along the test section and pressure differences between the Preston and static probes for single-phase air flow over a wide range of flow rates. The non-dimensional relationship between the Preston probe reading and wall shear stress is reported in a practically more convenient form. The probes are used to measure the gas-wall shear stress distribution up to positions close to the air-water interface. The gas-wall friction factor is determined and compared with other reported models.

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Nomenclature

A_G, A_L	crossectional area of gas and liquid phase, m^2	C	constant
d	external diameter of Preston tube, m	D	pipe diameter, m
D_G, D_L	hydraulic diameter of gas and liquid phase, m	f_G, f_L	gas-wall and liquid-wall friction factor
f_i	interfacial friction factor	g	gravitational acceleration, m/s^2
n	constant	P	pressure, N/m^2
ΔP_L	pressure drop along the test section, Pa	S	perimeter, m

ΔP_P	pressure difference between Preston and static tubes reading, Pa		
Re	Reynolds number	u_τ	friction velocity, m/s
u	velocity, m/s	V_G	average velocity of gas, m/s
V_L	average velocity of liquid, m/s	x^*	group of variables in eq.(12)
y	vertical position measured from bottom, m	y^*	group of variables in eq.(12)

Greek Symbols

β	angle in eq. (7), radian	α	inclination angle from the horizontal, deg.
θ	circumferential location, degree	ρ	density, kg/m^3
ν	kinematic viscosity, m^2/s	τ	shear stress, N/m^2

Subscripts

k	gas or liquid	G	gas phase	L	liquid phase
i	interface	w	wall		

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Dr. Somchai Wongwises,
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Dear Dr. Somchai,

Thank you very much for submitting the manuscript entitled: Pressure Distribution of Refrigerant Capillary Tubes (Code 0012-152, received 7 December 2000) for consideration for publication in *ScienceAsia.*

The manuscript has been read by two independent referees, whose reports are enclosed for your information. Although the referees have found the work to be of interest, there are a number of queries and comments which require clarification from you. In addition, the manuscript needs to be revised in light of their comments. Please reply to every point of the referees' comments or queries, and send 3 copies of the revised manuscript, together with the diskette, back to me as soon as possible.

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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

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

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

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

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

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

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;

$$h = h_L(1-x) + h_G x \tag{5}$$

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

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

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

$$G_G = G_L = \rho V \tag{7}$$

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

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

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

$$V = G\upsilon = G(x\upsilon_G + (1-x)\upsilon_L) \tag{9}$$

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

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

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

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

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

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

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

Frictional term in Eq.(11) can be obtained from

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

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

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

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

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

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

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:

$$Re = \frac{Gd}{\mu_{tp}} \tag{15}$$

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

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

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.

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 word, 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.

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.

ACKNOWLEDGMENTS

This work was given financial supported by the Thailand Research Fund whose guidance is gratefully acknowledged. The first author wish to acknowledge his lovely undergraduate students; Mr. Pakawat Kritsadathikarn, Mr.Tirawat Songnetichaovalit and Mr.Noppadon Lokathada for their generous assistance.

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

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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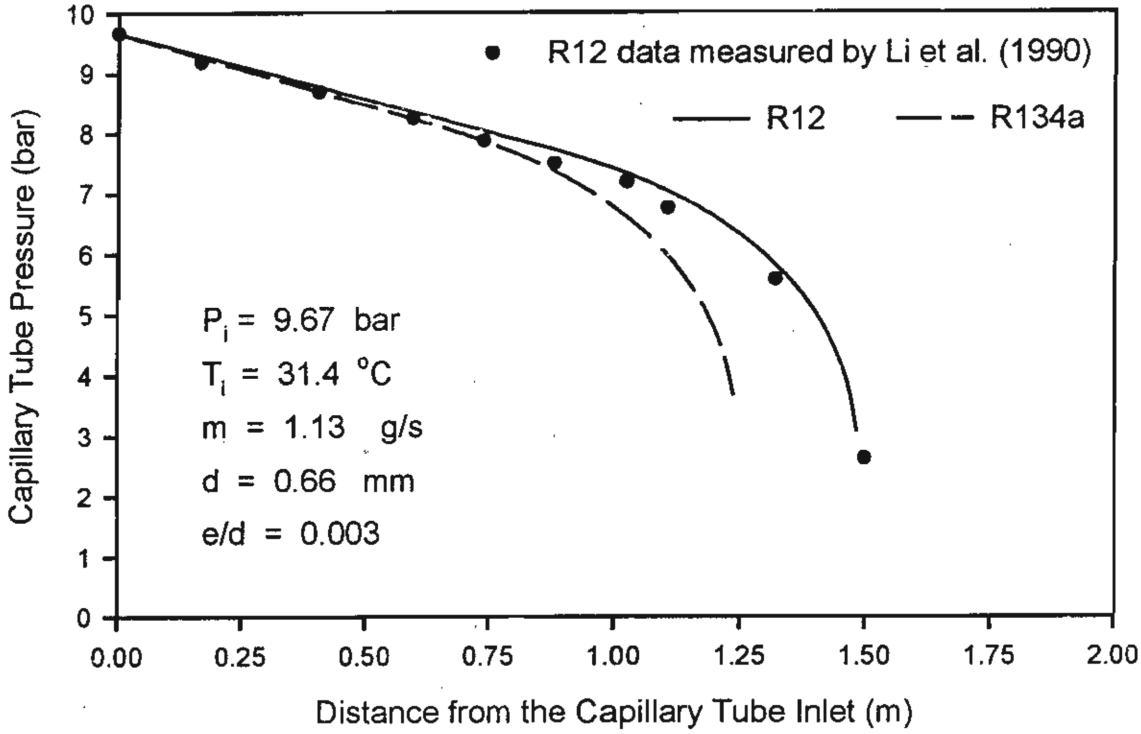

Fig.1 Comparison of pressure distributions along the capillary tube for R12 and R134a

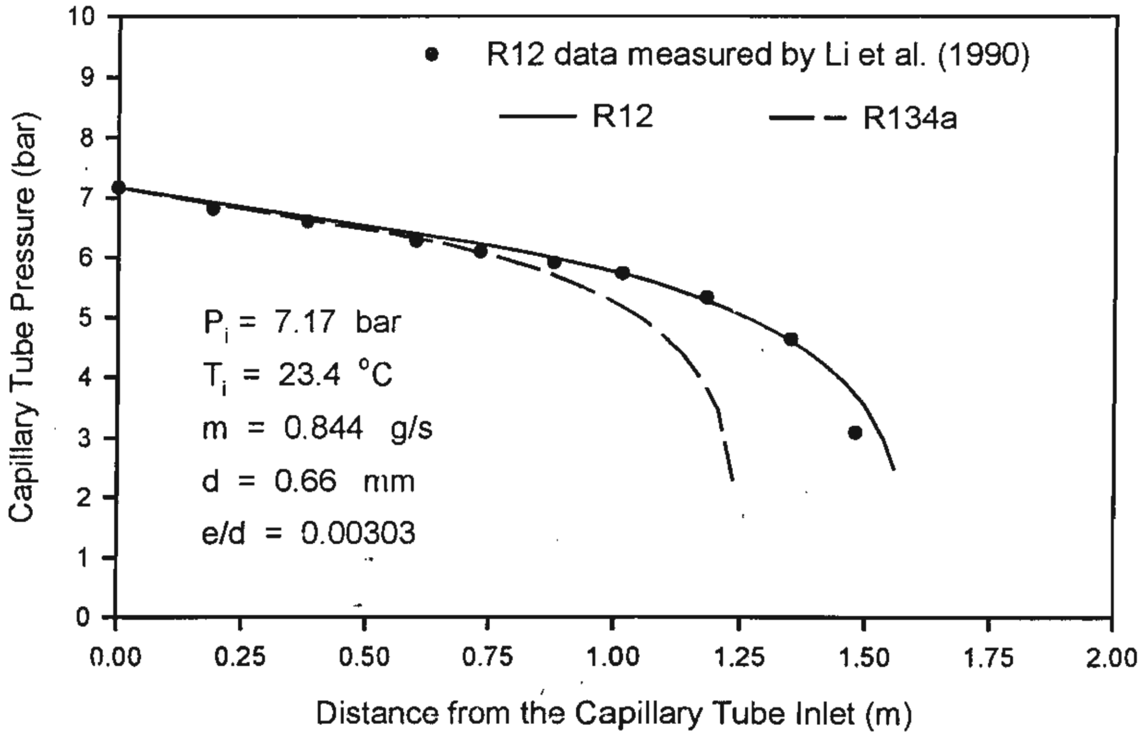

Fig.2 Comparison of pressure distributions along the capillary tube for R12 and R134a

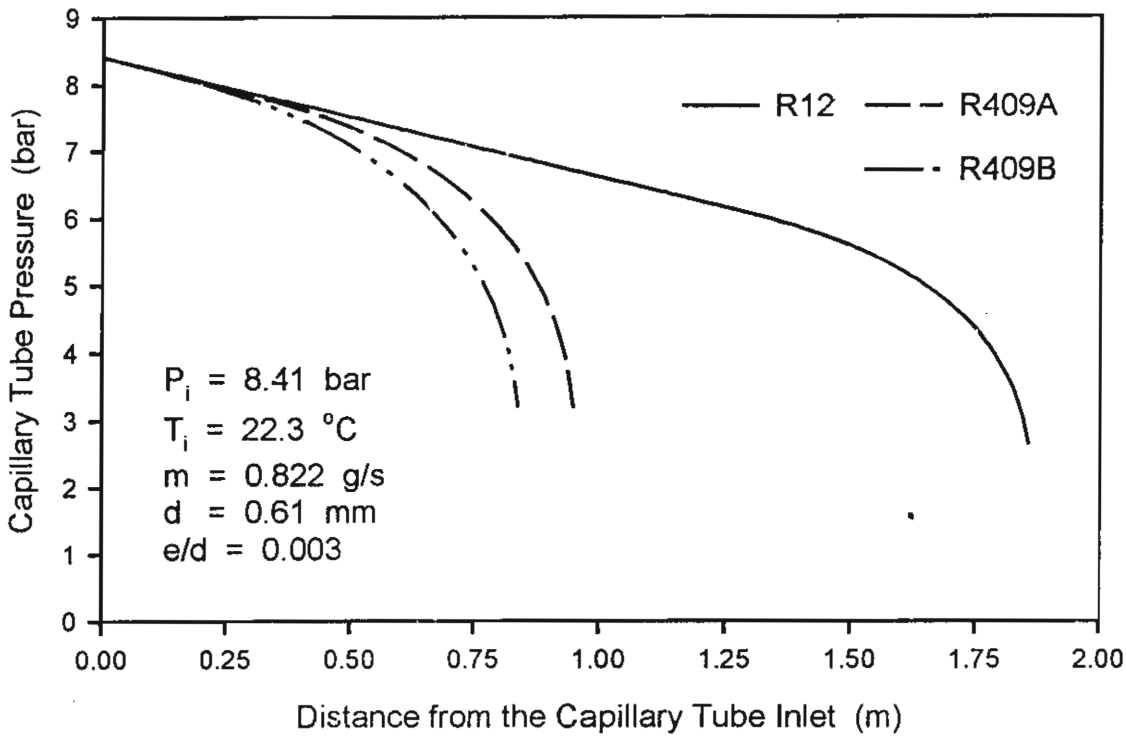

Fig. 3 Comparison of pressure distributions along the capillary tube for R12, R409A and R409B

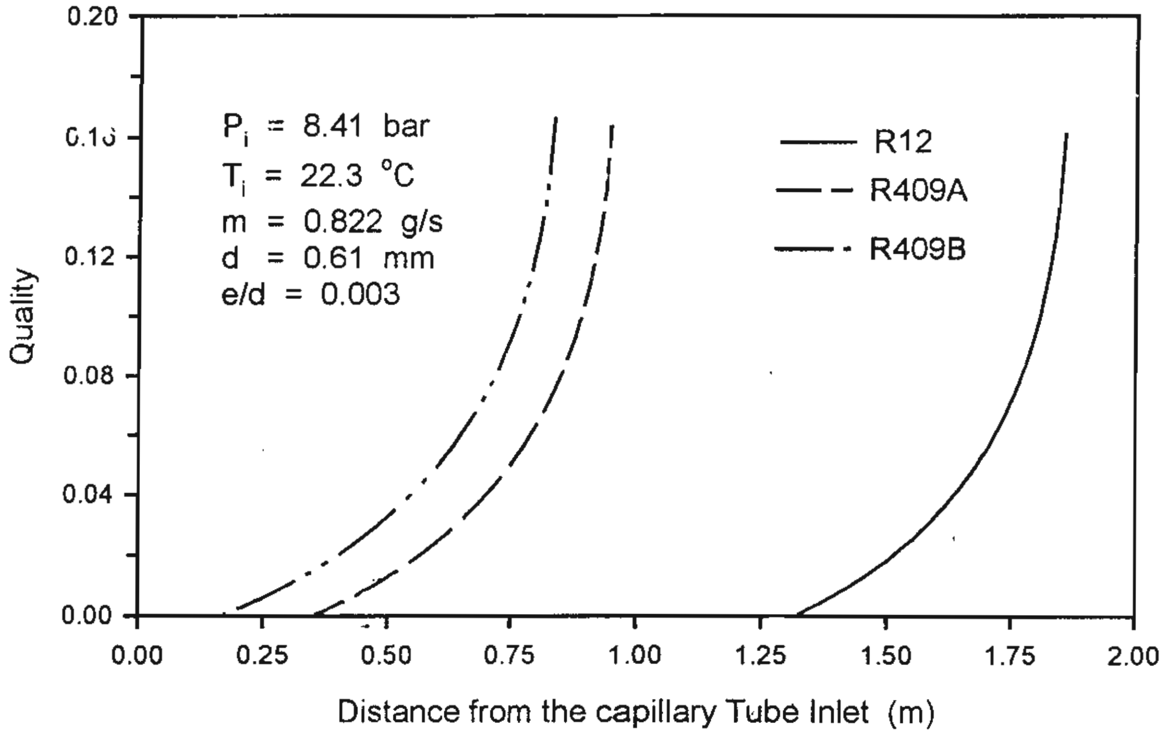

Fig. 4 Comparison of quality distributions along the capillary tube for R12, R409A and R409B

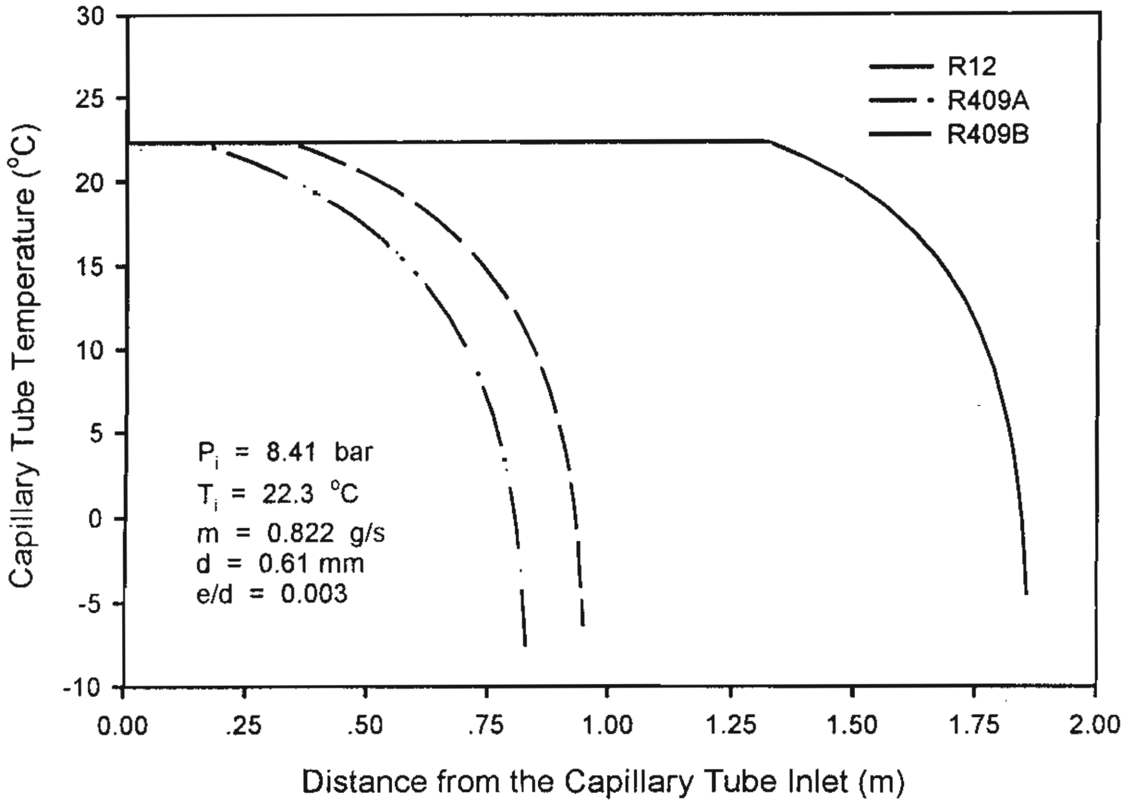

Fig. 5 Comparison of temperature distributions along the capillary tube for R12, R409A and R409B

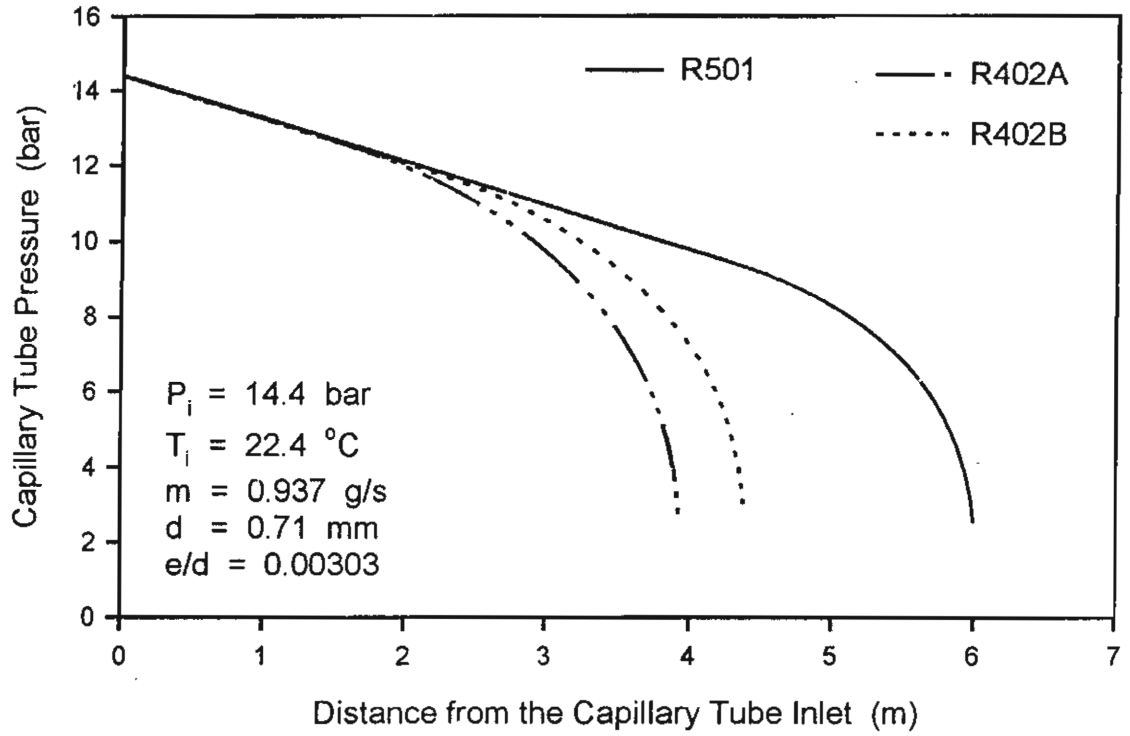

Fig. 6 Comparison of pressure distributions along the capillary tube for R501, R402A and R402B

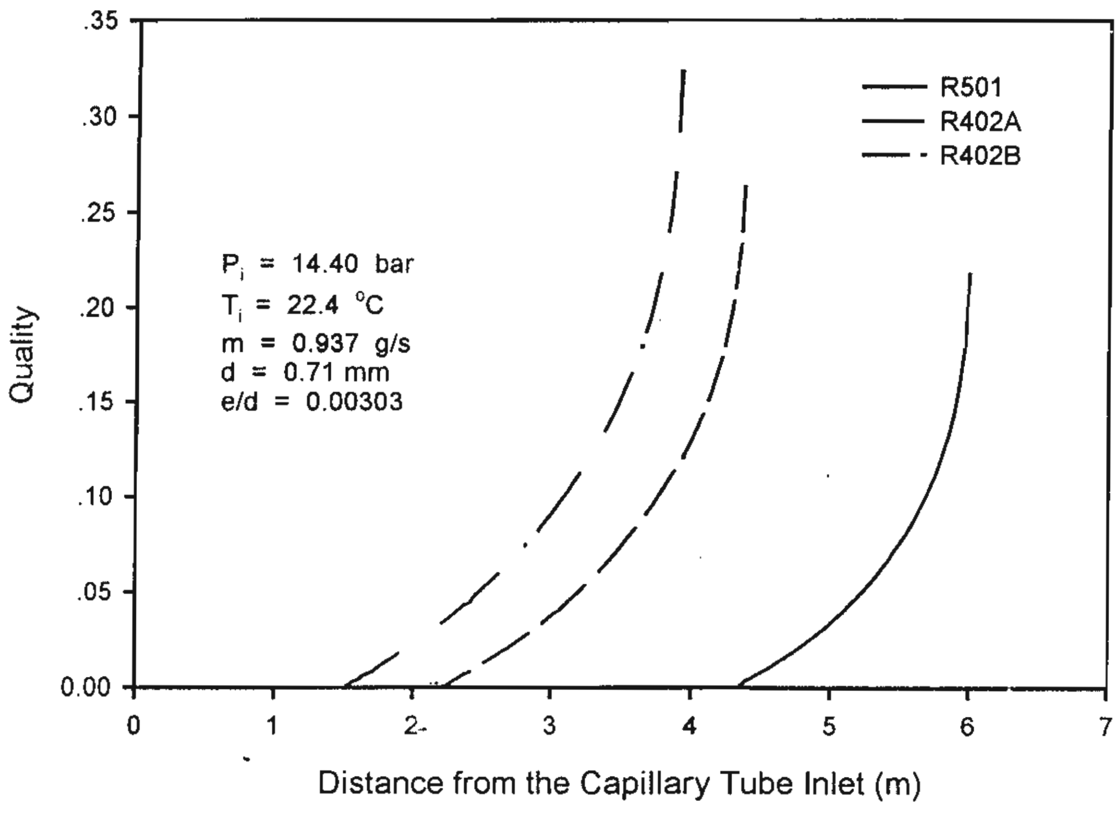

Fig. 7 Comparison of quality distributions along the capillary tube for R501, R402A and R402B

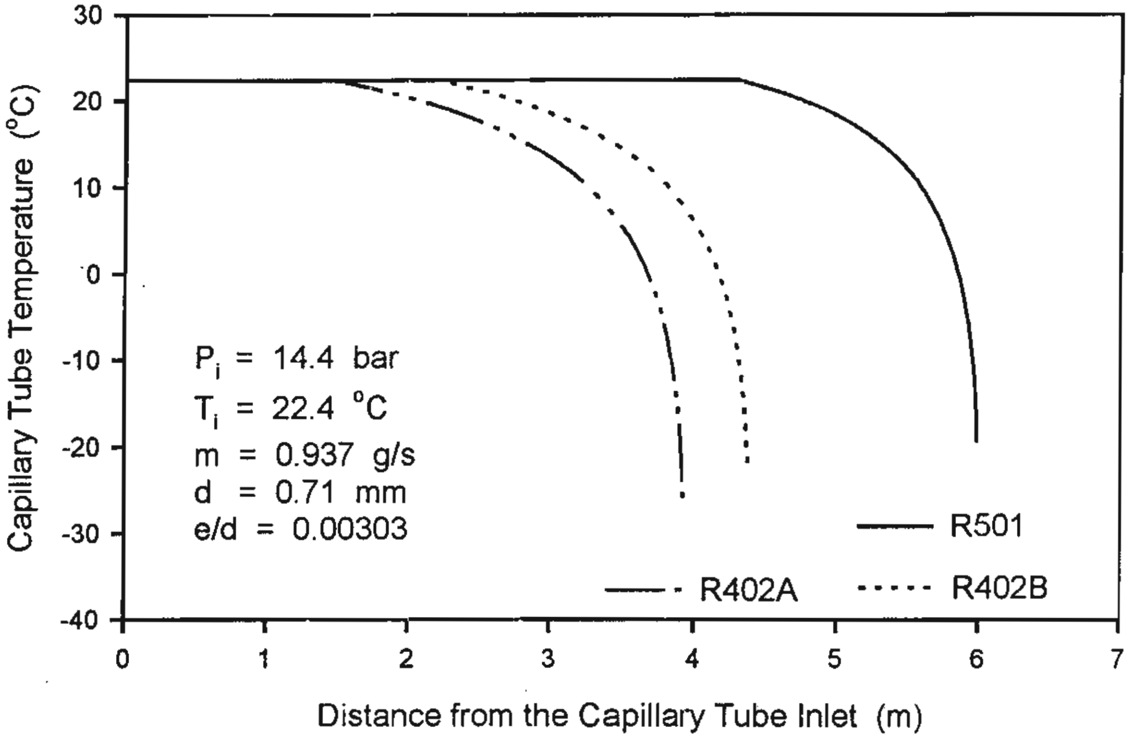

Fig. 8 Comparison of temperature distributions along the capillary tube for R501, R402A and R402B

Wongwises, S., Songnetichavarit, T., Lokathada, N., Kritsadathikarn, P., Investigation on Adiabatic Flows of Traditional and Alternative Refrigerants Through Capillary Tubes, *J.of Energy Heat and Mass Transfer* (in press).

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REGIONAL CENTRE FOR ENERGY HEAT AND MASS TRANSFER
ASIA AND THE PACIFIC

May 11, 2001

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Re: *Manuscript Number JEHMT/25 entitled "Investigation on Adiabatic Flows of Alternative Refrigerants Through Capillary Tubes" by S. Wongwises, P. Kritsadathikarn, T. Songnetichaovalit and N. Lokathada.*

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Re: Manuscript Number JEHMT/25 entitled "Investigation on Adiabatic Flows of Alternative Refrigerants Through Capillary Tubes" by S. Wongwises, P. Kritsadathikarn, T. Songnetichaovalit and N. Lokathada.

Dear Dr. Wongwises:

Thank you for your revised version of the above paper. I am pleased to inform you that it has been accepted for publication in the Journal of Energy Heat and Mass Transfer.

You will receive the proofs shortly.

With regards,

Yours sincerely,

Prof. A. R. Balakrishnan
Editor, JEHMT

Investigation on Adiabatic Flows of Traditional and Alternative Refrigerants Through Capillary Tubes

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ABSTRACT

In this paper, the local pressure, temperature and quality of some common traditional and environmentally acceptable alternative refrigerants flowing through adiabatic capillary tubes are numerically determined. The mathematical model is developed from the basic law of mass, energy and momentum conservations. Homogeneous flow is assumed for the two-phase liquid-vapor flow region. Numerical results reveal that, in general for both single-and two-phase regions, the alternative refrigerants used in this study consistently give higher pressure drops per unit length than the traditional refrigerants.

INTRODUCTION

The capillary tube is a kind of expansion devices used in small vapour-compression refrigerating and air conditioning systems. In practical consideration, the main concern is to determine the appropriate length and diameter of the tube at a given refrigeration capacity. The design and analysis of capillary tubes have been received the most attention, both analytically and experimentally [1-12]. Bansal et al.[1] presented a homogeneous two-phase flow model to study the performance of adiabatic capillary tubes The REFPROP data base [14] which is based on the Carnahan-Starling-DeSantis equation of state was used to determine the refrigerant's properties. Sami et al.[10] presented a numerical model for predicting capillary tube performance using new alternative refrigerants, both pure and binary mixtures. Wong et al.[11] developed a homogeneous two-phase flow model to simulate and compare the flow characteristics of CFC12 and HFC134a. The results showed that even with minor differences in thermophysical properties of both refrigerants, the difference in pressure, temperature, mixture velocity and quality distributions in capillary tube may be significant. More recently, Jung et al.[4] modified the Stoecker's basic model [13] and provided simple correlations for sizing the capillary tubes for practicing engineers. Various effects due to subcooling, area contraction, different equations for viscosity and friction factor were considered. Melo [8] studied the effects of diameter and length of capillary tube, refrigerant subcooling, condensing pressure and type of the refrigerant on the mass flow rates in adiabatic capillary tubes. The experiments were performed with R12, R134a and R600a. Wongwises et al. [12] provided the results of simulation using a developed adiabatic capillary tube model. The investigation was concerned about making comparisons between various alternative mixtures of refrigerant. Although some information is currently available on flow characteristics in adiabatic capillary tubes, there still remains room to discuss. The present study is concerned with making comparisons between some pairs of traditional refrigerants with alternative ternary refrigerant mixtures.

MATHEMATICAL MODEL

The flow of refrigerant through a capillary tube can be divided into two distinct regions as shown in Fig.1; a single-phase sub-cooled liquid region and a two-phase liquid-vapor region. In Fig.1, point 1 represents the capillary tube inlet at the condenser side, point 2 represents the capillary tube inlet at the capillary tube side, point 3 is the end of the single- phase subcooled liquid (saturation point) or the beginning of the two-phase region, point 4 is the capillary tube exit at the capillary tube side and point 5 represents the capillary tube exit at the evaporator side. The typical pressure-enthalpy relationship from point 1-5 is shown in Fig.2

The mathematical model developed is based on the assumptions as follows;

- one-dimensional flow,
- adiabatic and homogeneous two-phase flow,
- straight horizontal and constant inner diameter and roughness capillary tube,
- thermodynamic equilibrium through the capillary tube
- no metastable liquid region

SUB-COOLED LIQUID REGION

The sub-cooled liquid region is the region from the inlet of the capillary tube to the position where the saturation pressure corresponds to the inlet temperature. For steady 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;

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

Substituting Eq. (2) into Eq. (1), the single phase subcooled liquid length, L_{SC}, of the capillary tube can be obtained

$$L_{sc} = \frac{\left(p_i - p_{sat}\right)}{\left(f\,\frac{G^2}{2\rho_L d}\right)} \tag{4}$$

where the total mass flux, G, is the total mass flow rate divided by total cross-sectional area of channel, A_o.

TWO-PHASE FLOW REGION

In the two-phase flow region, the model is derived from the one dimensional homogeneous two-phase flow assumption. The model is based on that of Wong et al.[11] and Wallis [15]. In the modeling, the basic physical equations governing the capillary tube flow are the continuity and conservations of energy and momentum.

The specific enthalpy of a substance at a saturation state having a given quality can be calculated by utilizing the definition of quality, x,

$$h = x\,h_G + (1-x)h_L \tag{5}$$

where the quality is the ratio of the mass flow rate of vapour to total mass flow rate when a substance is in a saturation state.

For homogeneous flow, the velocity of each phase can be expressed as

$$V = V_G = V_L \tag{6}$$

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

$$\frac{d}{dz}\left(x h_G + (1-x)\,h_L + \frac{V^2}{2}\right) = 0 \tag{7}$$

In the homogeneous two-phase region for a pure substance in equilibrium, the enthalpies (and also densities) may be arranged in the function of pressure,

$$h = h(p), \qquad \rho = \rho(p) \tag{8}$$

Vapour mass flux, G_G, and liquid mass flux, G_L, are the vapour and liquid mass flow rates divided by total cross-sectional area of channel, so

$$G_G = G_L = \rho V \tag{9}$$

In liquid-vapour flows, void fraction,α, usually represents the time-averaged fraction of the cross-sectional area (or the volume) which is occupied by the vapour phase. The homogeneous flow model takes into account the fact that the two phases physically flow with same velocities. This general correlation is then

$$\alpha = \frac{1}{1+\left(\frac{1-x}{x}\frac{\rho_G}{\rho_L}\right)} \tag{10}$$

$$= \alpha(x,p) \tag{11}$$

Actual average velocity of vapour and liquid can be determined from

$$V = G\upsilon = G(x\upsilon_G + (1-x)\upsilon_L) \tag{12}$$

Eventually, after rearrangement using above correlations, an expression for the total pressure gradient is obtained as follows;

$$\frac{dP}{dz} = -\frac{dx}{dz}\left(\frac{A}{B}\right) \tag{13}$$

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

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

The momentum equation is often rewritten as an explicit equation for the pressure gradient. The total pressure gradient $\left(\frac{dP}{dz}\right)$ is, therefore, expressed as the sum of the three different components, so

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_f + \left(\frac{dP}{dz}\right)_a + \left(\frac{dP}{dz}\right)_g \tag{14}$$

The three terms on the right hand side are regarded as frictional, accelerational, and gravitational components of the total pressure gradient. Gravitational term in Eq.(14) is negligible because the flow is horizontal. Accelerational pressure gradient can not be measured directly. However. it can be calculated from momentum flux as follows;

$$\left(\frac{dP}{dz}\right)_a = -\frac{m}{A_o}\frac{d(V)}{dz} \tag{15}$$

and, so

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

Frictional pressure gradient can be obtained from

$$\left(\frac{dP}{dz}\right)_f = \frac{-f_{tp}\, G^2(x\upsilon_G + (1-x)\upsilon_L)}{2d} \tag{16}$$

Substituting Eqs. (15) and (16) into Eq. (14), gives

$$\frac{dx}{dz} = \frac{\left(\frac{dP}{dz}\right)_f - C\frac{dP}{dz}}{D} \tag{17}$$

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 friction factor for the homogeneous two-phase flow, f_{tp}, can be determined from Colebrook's equation with the Reynolds number being defined as:

$$Re = \frac{Gd}{\mu_{tp}} \tag{18}$$

Dukler 's equation [16] is used to calculate μ_{tp} as follows:

$$\mu_{tp} = \frac{x\upsilon_G\mu_G + (1-x)\upsilon_L\mu_L}{x\upsilon_G + (1-x)\upsilon_L} \tag{19}$$

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

RESULTS AND DISCUSSION

The calculation is divided into two parts; sub-cooled single-phase region and two-phase vapour-liquid region. The thermodynamic and transport properties of all refrigerants are taken from REFPROP [14]. All properties are developed as a function of pressure. Initial conditions required in the calculation are pressure and temperature of refrigerant at capillary tube inlet (or condition at outlet of condenser), mass flow rate of refrigerant, roughness and diameter of the capillary tube. In the single-phase flow region, after substituting the friction factor calculated from Colebrook equation (eq.(3)) and the saturation pressure of refrigerant at the temperature at the capillary tube inlet, P_{sat}, into eq.(4), the single-phase region length is obtained. The exit condition of the single phase flow region is used to be an initial condition of the two-phase flow region. In the two-phase flow region, the Runge-Kutta method is used to solve equations (13) and (17). The calculation in two-phase flow region is terminated when the flow is at the choked flow condition ($dP/dz \rightarrow \infty$).

In the present study, the comparisons are concerned between the following pairs of refrigerants;

- R12 vs R134a,
- R12 vs R414B (R22/R124/R600a/R142b; 50/39/1.5/9.5)
- R12 vs R411A (R1270/R22/R152a; 1.5/87.5/11.0)
- R12 vs R411B (R1270/R22/R152a; 3/94/3)
- R22 vs R407A (R32/R125/R134a; 20/40/40)
- R22 vs R407B (R32/R125/R134a; 10/70/20)
- R22 vs R407D (R32/R125/R134a; 15/15/70)
- R500 (R12/R152a; 73.8/26.2) vs R401A (R22/R152a/R124; 53/13/34)
- R500 (R12/R152a; 73.8/26.2) vs R401B (R22/R152a/R124; 61/11/28)
- R500 (R12/R152a; 73.8/26.2) vs R401C (R22/R152a/R124; 33/15/52)

Figures 3-5 show the relationship between the pressure and temperature at the saturated conditions for each group of the refrigerant. As shown in the figures, the saturated conditions of refrigerant in each group are similar. However even with the small differences in property in each group, it is not necessary that performances of the system are similar and this issue will be investigated in the present study. The mass flow rate, pressure and temperature at the capillary tube inlet, diameter and relative roughness of tube were each varied in turn to investigate their effect on the total length of capillary tube. The results from the calculation by the present model developed are pressure, temperature and quality at each position along the capillary tubes. In order to validate the present model, comparisons are made with limited available measured data of Li et al. [7].

Figures 6-8 compare the simulation results of the present model with the R12 measured data obtained by Li et al.[7] . The experimental conditions used by Li et al. [7] are given in Table 1. The present model is shown to agree with the measured data.

Table 1 Experimental conditions of Li et al. [7]

Case	T_i (°C)	P_i (bar)	m (g/s)	d (mm)	e/d
1	30.00	8.85	4.35	1.17	0.003
2	31.40	9.67	1.13	0.66	0.003
3	23.40	7.17	0.844	0.66	0.003

Figures 6-8, 9, 11, 13 and 15 show variation of local pressure of all refrigerants with position along the capillary tube. In the subcooled liquid region, due to frictional effects in fully developed flow in a constant-area tube, the pressure of refrigerant drops linearly along the capillary tube. After the position of the inception of vaporization, due to frictional and accelerational effects, the pressure of refrigerant decreases relatively fast and, more and more rapidly as the flow approaches the critical flow condition. But in fact, due to the delay of vaporization, from a single-phase subcooled liquid to a two-phase mixture, the actual starting point of vaporization may not occur at the end of the single-phase liquid region or at the saturated liquid condition.

It should be noted that comparing the local pressure distribution of R12 and of R134a as shown in Figure 9, the flow of R12 through the capillary tube gives a lower pressure drop per unit length than that of R134a. In the other word, at the same total pressure drop, the total tube length by using R134a is shorter. Except Figs. 15-16, comparisons of the pressure drop characteristics for the rest of each pair of refrigerant type show that for all cases in the single-phase region, the elder traditional refrigerants flowing through the capillary tube give longer single-phase region and give a slightly lower pressure drop per unit length than the newer alternative refrigerants. In the two-phase flow region, the traditional refrigerants give a meaningfully lower pressure drop than the alternative refrigerant which results in a longer total tube length. In Fig.15, R500 gives lower pressure drop than R401A and R401B but gives higher pressure drop than R401C. This may be due to the difference in the composition of each refrigerant. R401A and R401B compose of the similar component composition. While, amount of R22 and R124 in R401A and R401B is quite difference from those in R401C. It is also interesting to note that the comparison between the pressure drop characteristics for R22 and R407D in Fig.13, R407D gives a little bit longer single phase region and slightly shorter two-phase flow region. The result shows that although both refrigerants have a difference in composition, the pressure distributions along the capillary tube are almost the same.

Figures 10, 12 and 14 show the distributions of the temperature along the capillary tube for each pair of refrigerant type. In all cases, in the single phase region, the refrigerant temperature remains constant along the capillary tube. After the position of the inception of vaporization, the temperature drops rapidly as the flow approaches the critial condition. In general, the traditional refrigerants show a slightly lower temperature drop along the capillary tube, which corresponds to the lower pressure drop. Figure 16 shows the quality distribution along the capillary tube. In the single phase region, the quality is zero til the flash point which the two-phase region begins and then increases more rapidly as the choked flow condition is approached. It is also shown that, in general, traditional refrigerants vaporize later than their corresponding alternative refrigerants.

CONCLUSIONS

A homogeneous flow model has been applied to determine the characteristics of refrigerants flowing through adiabatic capillary tubes. The basic physical governing equations are established from the conservations of mass, energy and momentum. The differential equations derived are solved by using the Runge-Kutta method. By varying the model input parameters, in general, it has been found that the traditional refrigerants consistently give lower pressure drops per unit length for both single-phase and two-phase regions. The present model includes the various relevant parameters and is a tool for sizing the capillary tubes used in household refrigerators and freezers, especially to select the capillary tube length for given operating conditions.

ACKNOWLEDGMENTS

The present study was supported financially by the Thailand Research Fund (TRF) and National Energy Policy Office (NEPO) whose guidance and assistance are gratefully acknowledged.

NOMENCLATURE

A_o	cross-sectional area of tube, m^2	d	tube diameter, 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	dynamic viscosity, Pa s	υ	specific volume, m^3/kg
ρ	density, kg/m^3	τ	shear stress, N/m^2

Subscripts

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

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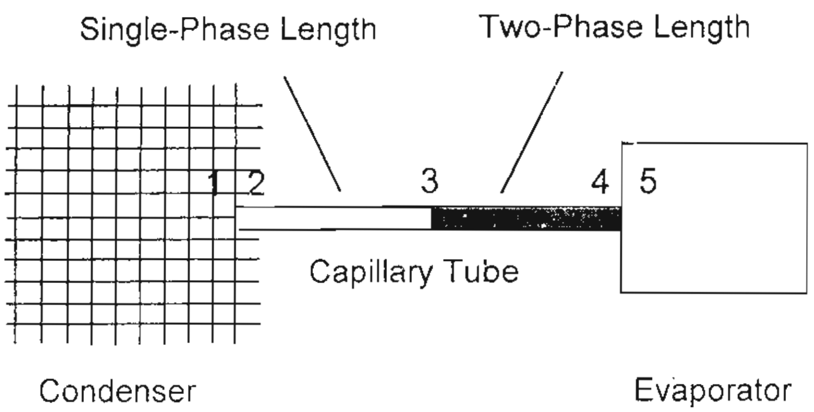

Fig. 1 Schematic diagram of an adiabatic capillary tube

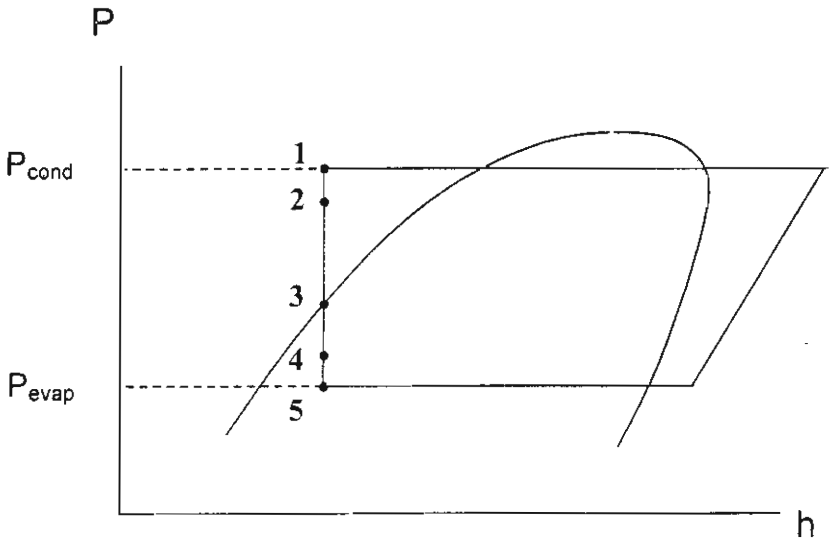

Fig. 2 Pressure – Enthalpy diagram for the vapor – compression cycle

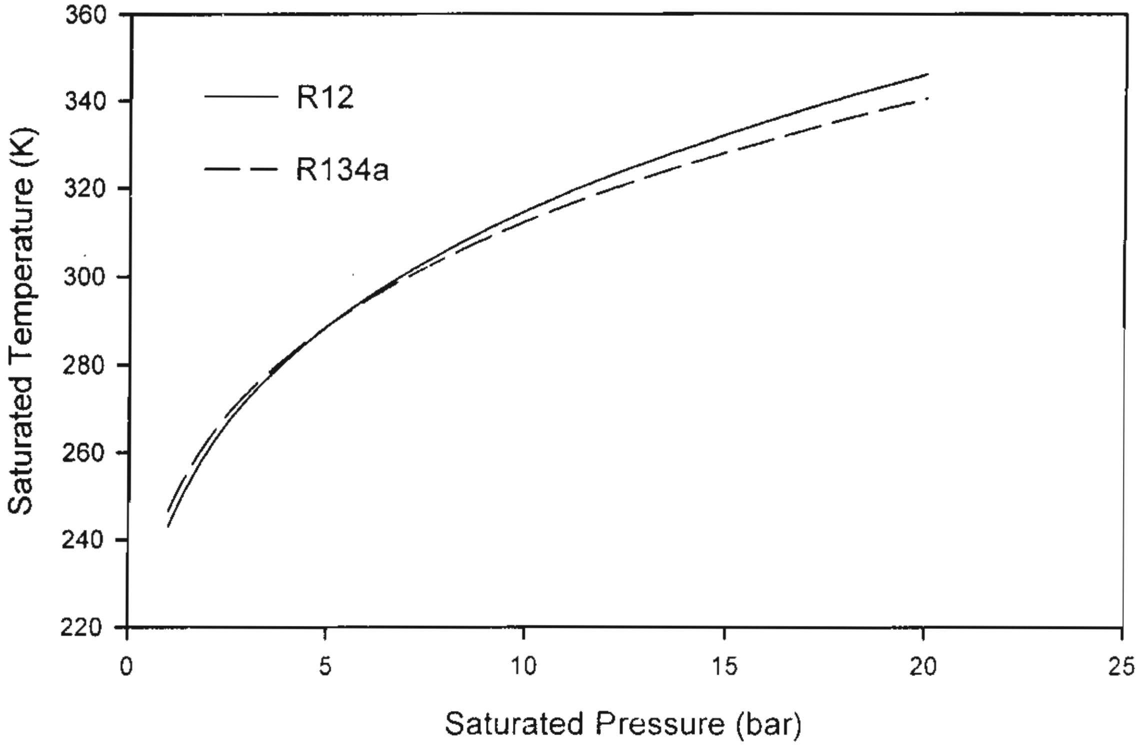

Fig. 3 Comparison of saturated pressure and temperature for R12 and R134a

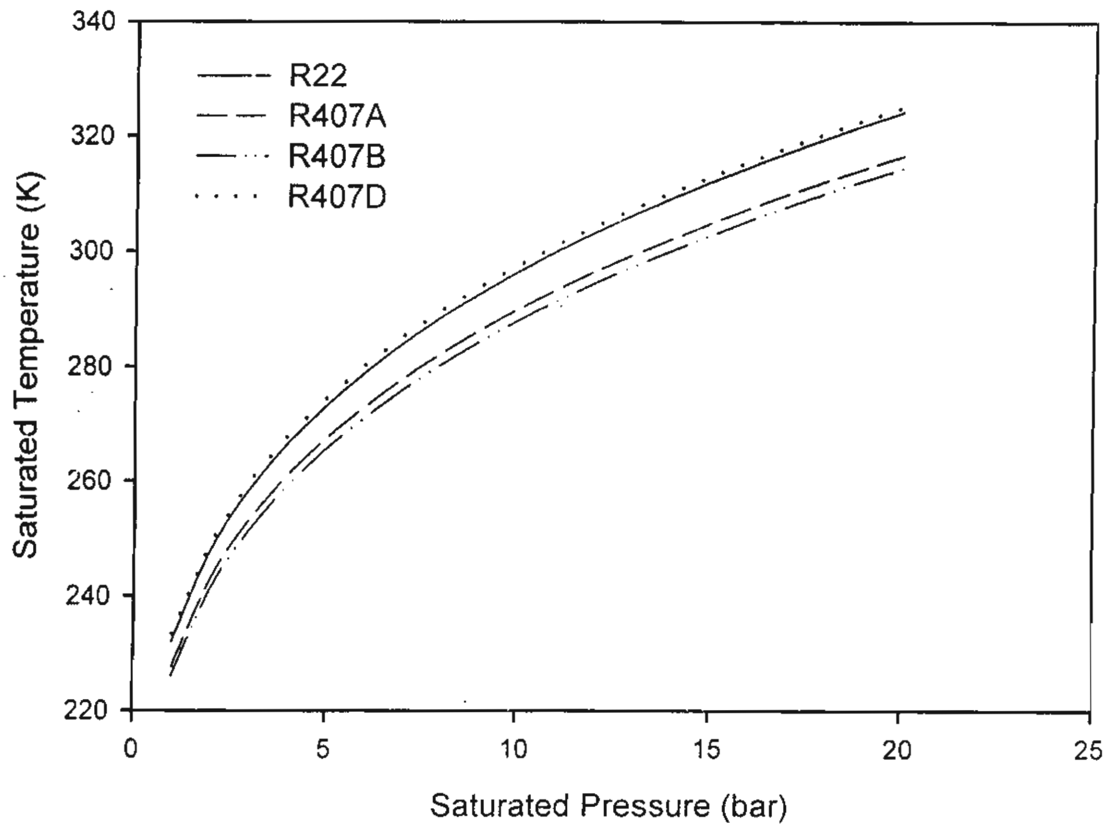

Fig. 4 Comparison of saturated pressure and temperature for R22, R407A, R407B and R407D

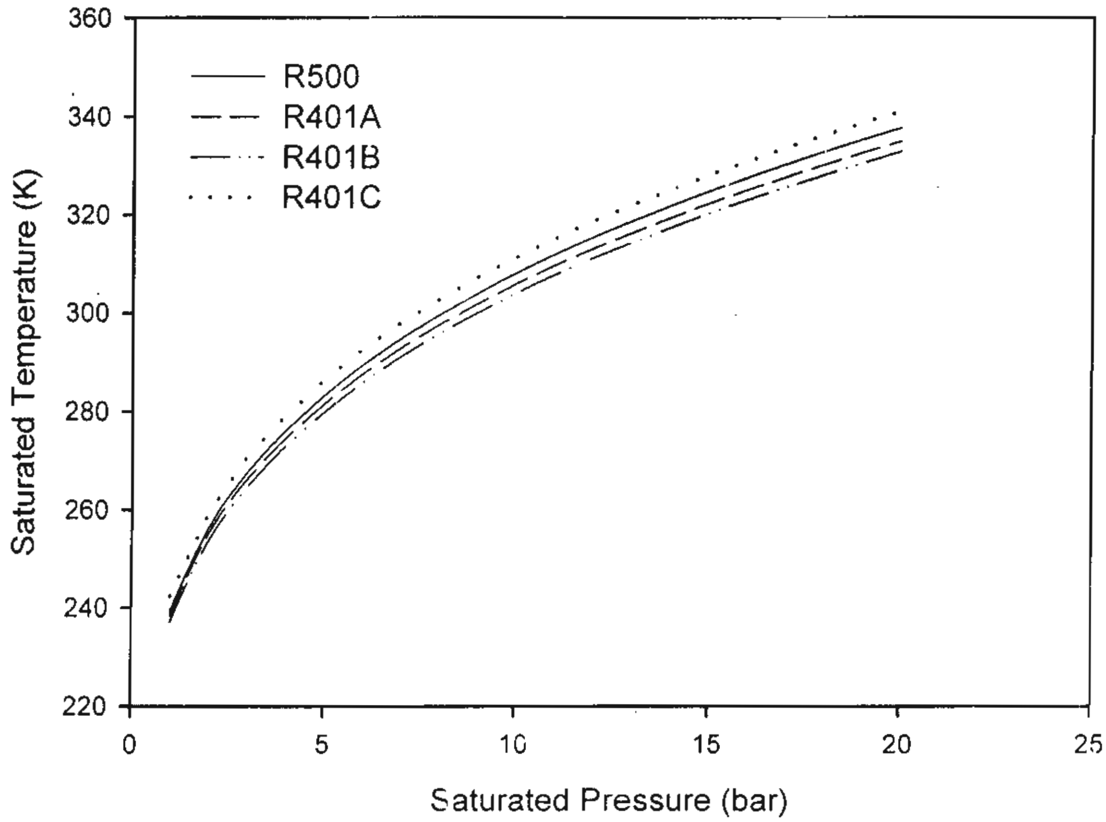

Fig. 5 Comparison of saturated pressure and temperature for R500, R401A, R401B and R401C

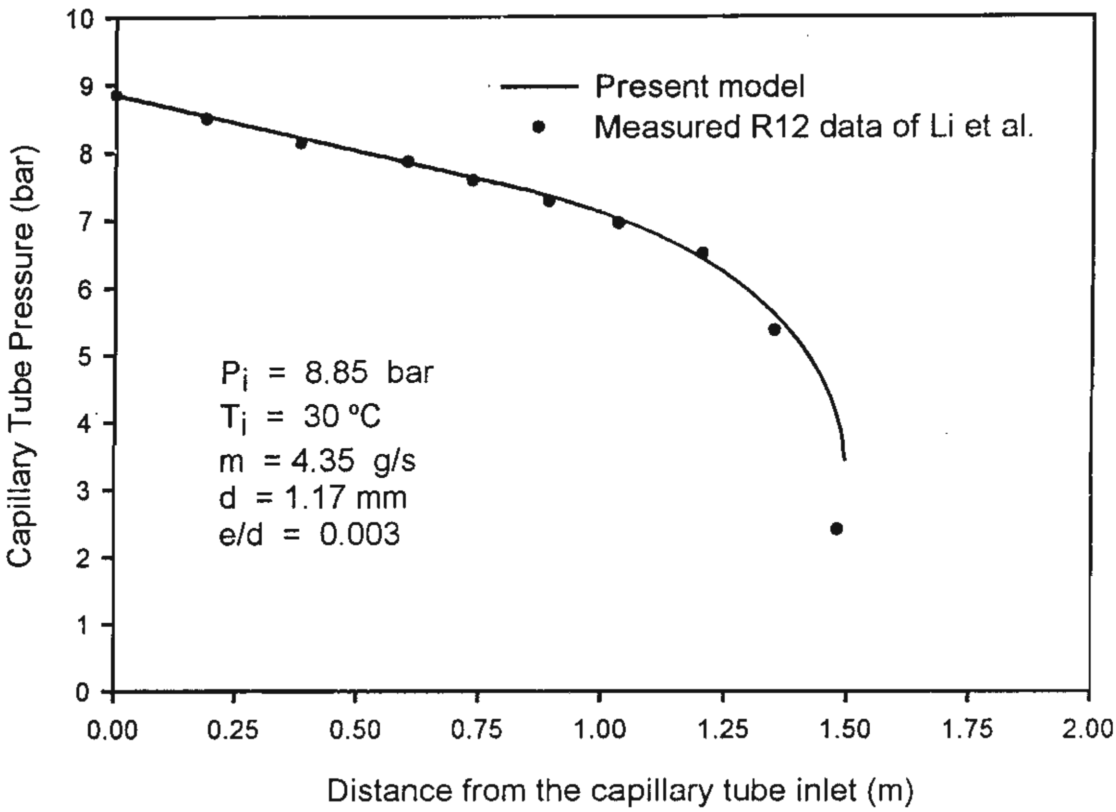

Fig. 6 Comparison of measured pressure distributions with present numerical results

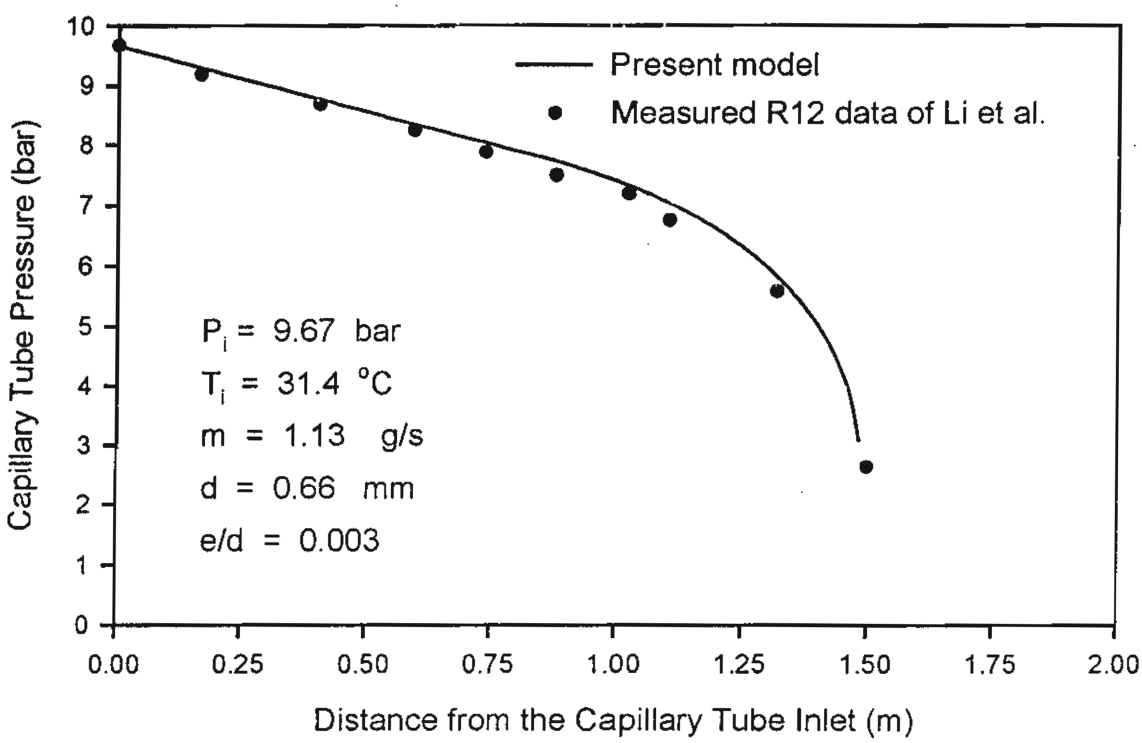

Fig. 7 Comparison of measured pressure distributions with present numerical results

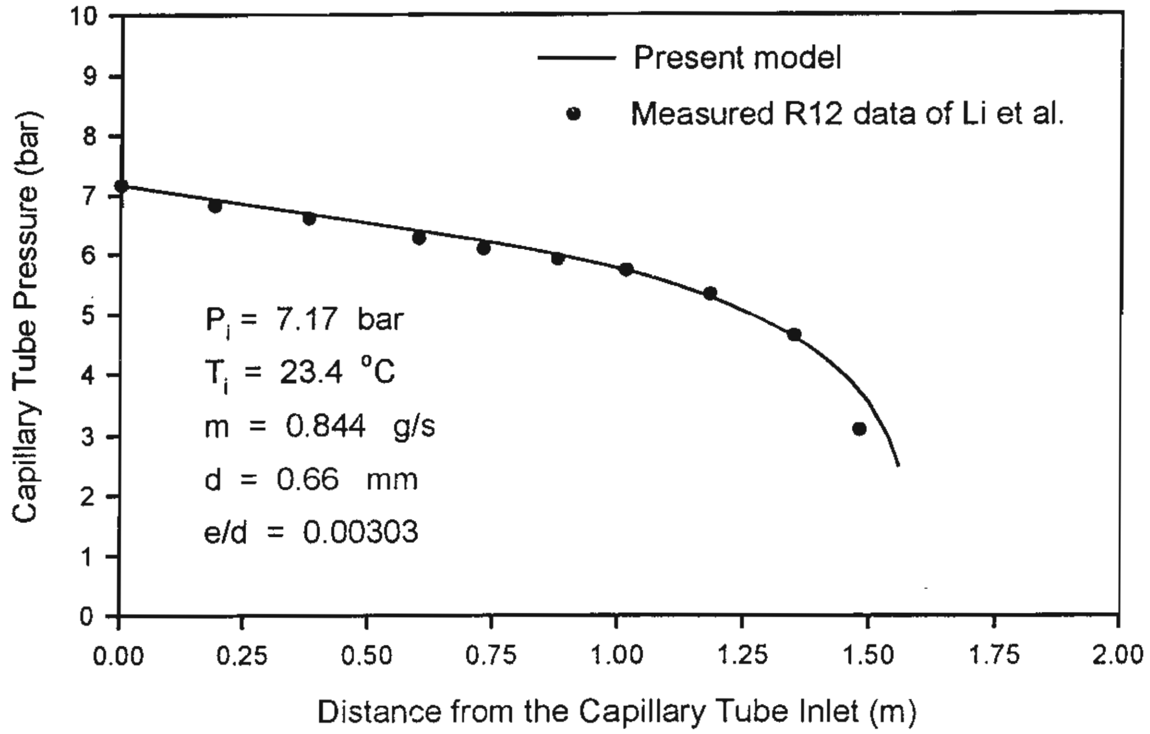

Fig. 8 Comparison of measured pressure distributions with present numerical results

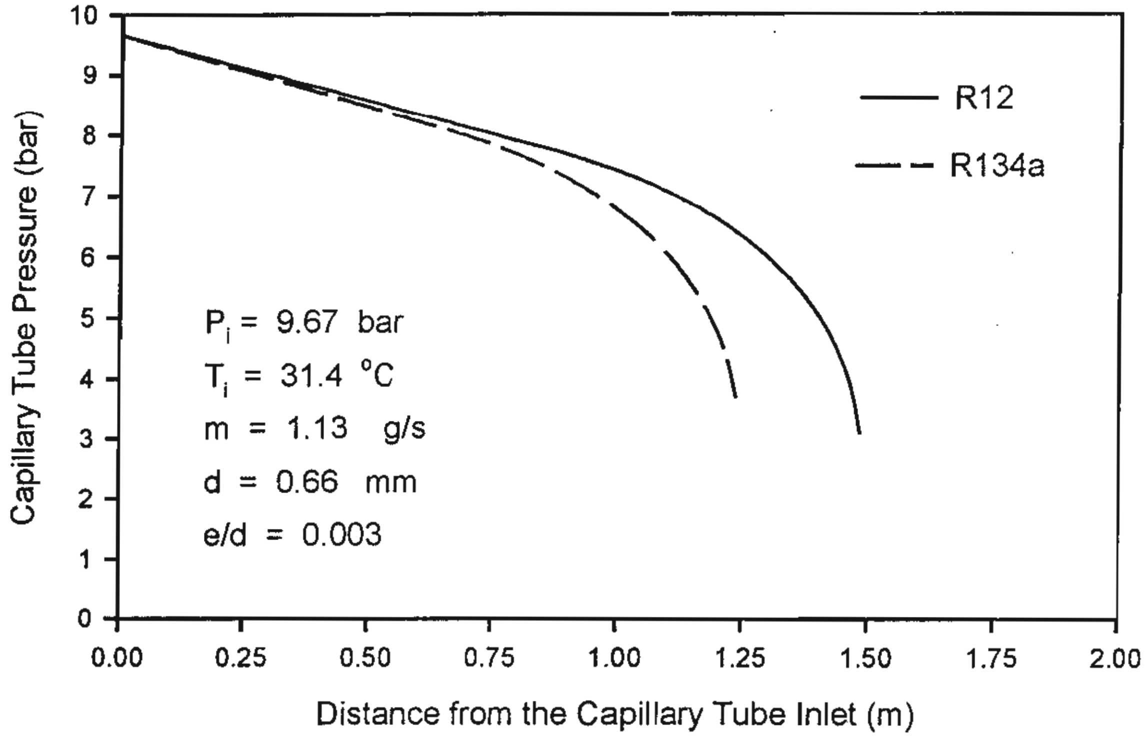

Fig. 9 Comparison of pressure distributions along the capillary tube for R12 and R134a

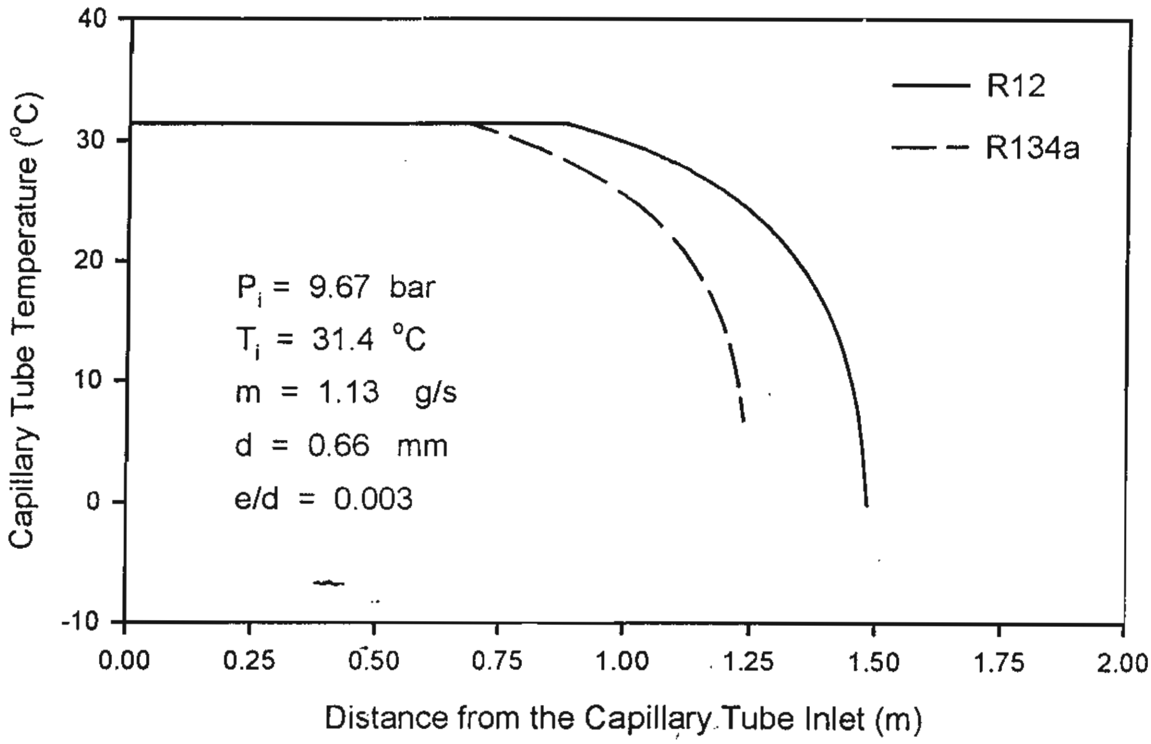

Fig. 10 Comparison of temperature distributions along the capillary tube for R12 and R134a

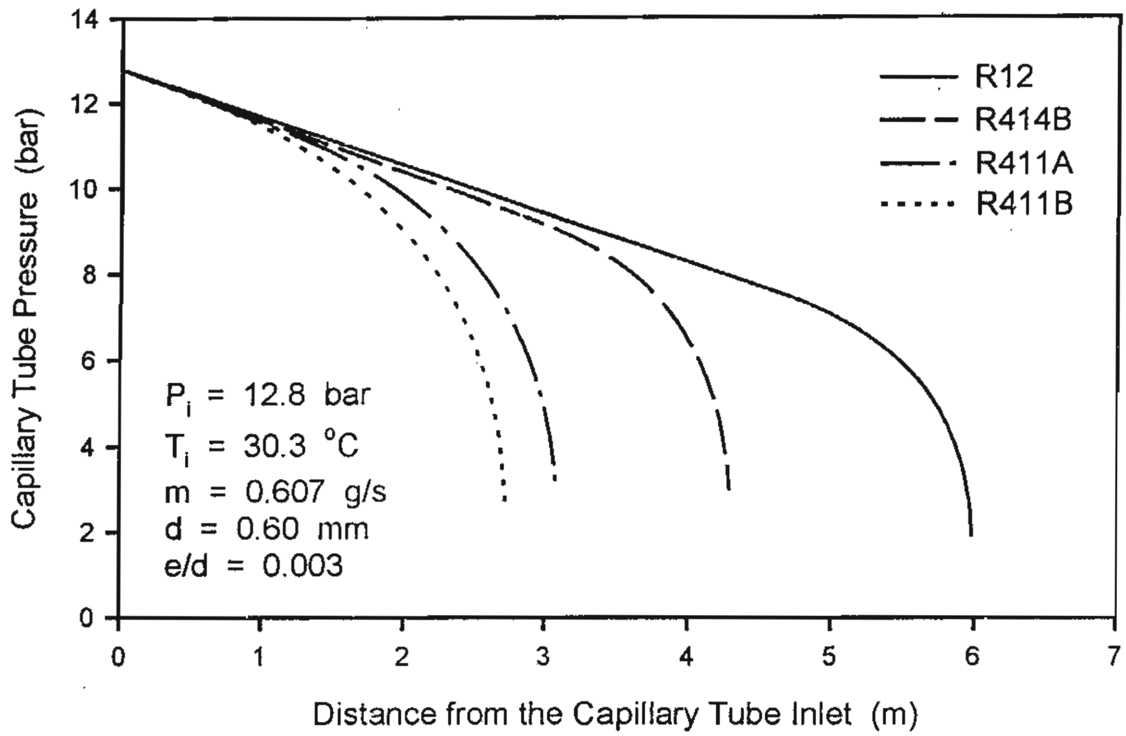

Fig. 11 Comparison of pressure distributions along the capillary tube for R12, R414B, R411A and R411B

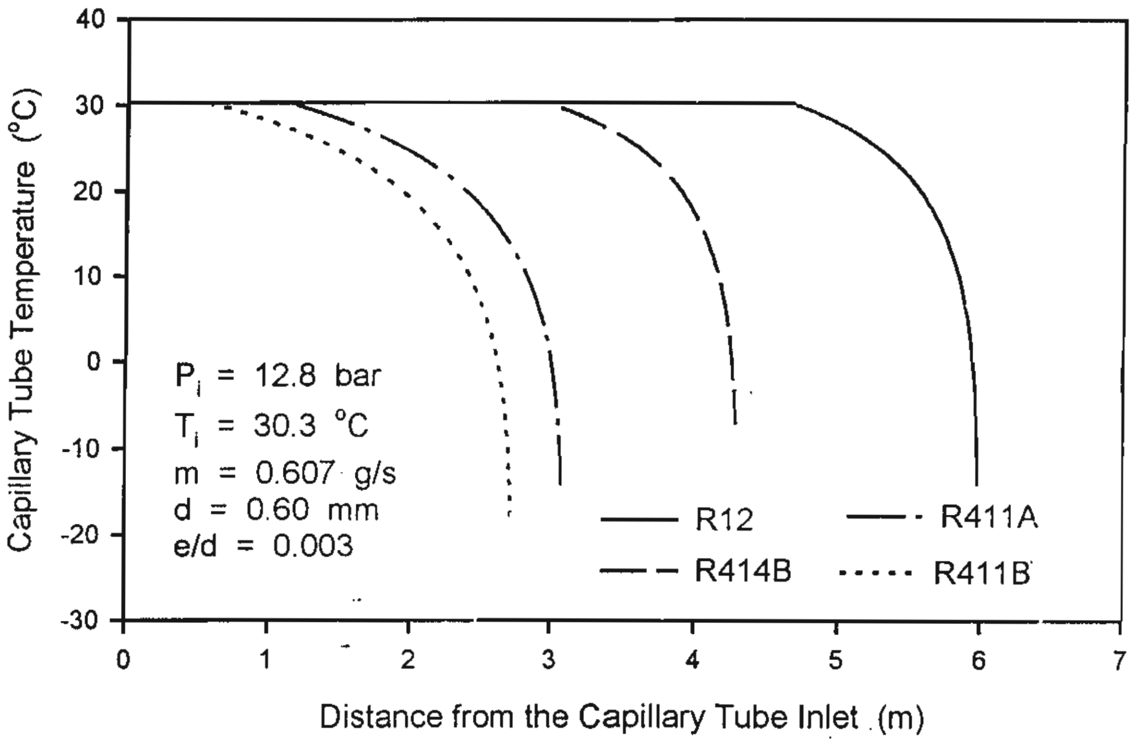

Fig. 12 Comparison of temperature distributions along the capillary tube for R12, R414B, R411A and R411B

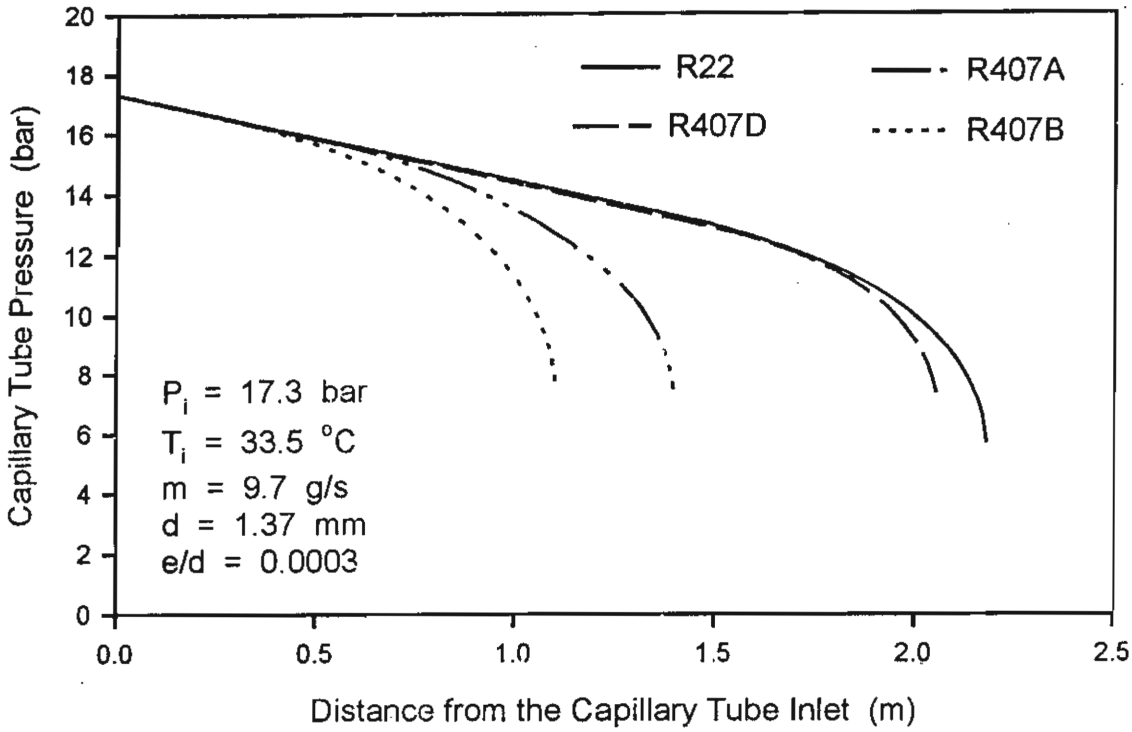

Fig. 13 Comparison of pressure distributions along the capillary tube for R22, R407A, R407B and R407D

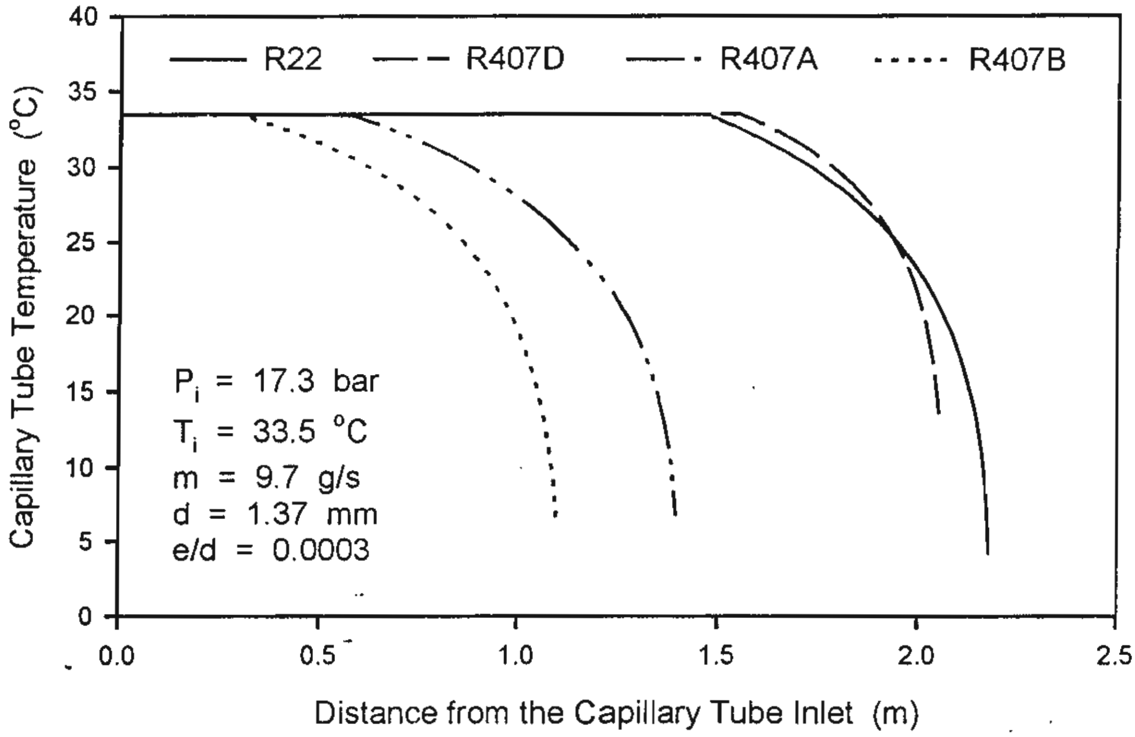

Fig. 14 Comparison of temperature distributions along the capillary tube for R22, R407A, R407B and R407D

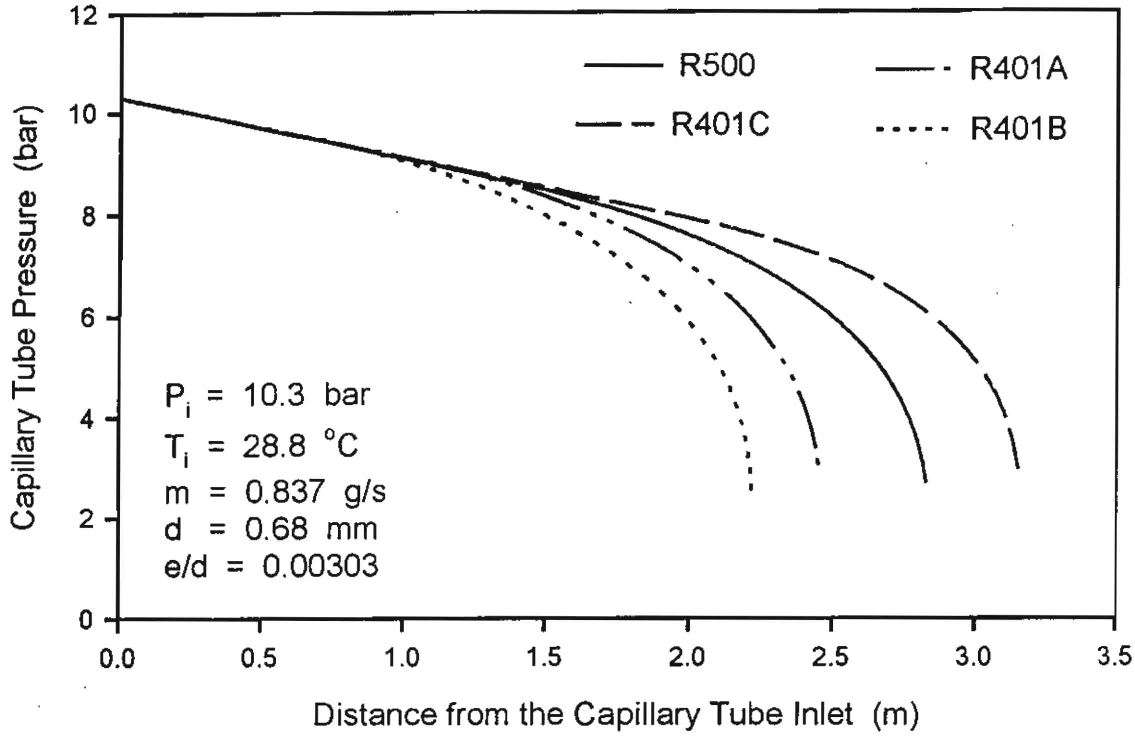

Fig. 15 Comparison of pressure distributions along the capillary tube for R500, R401A, R401B and R401C

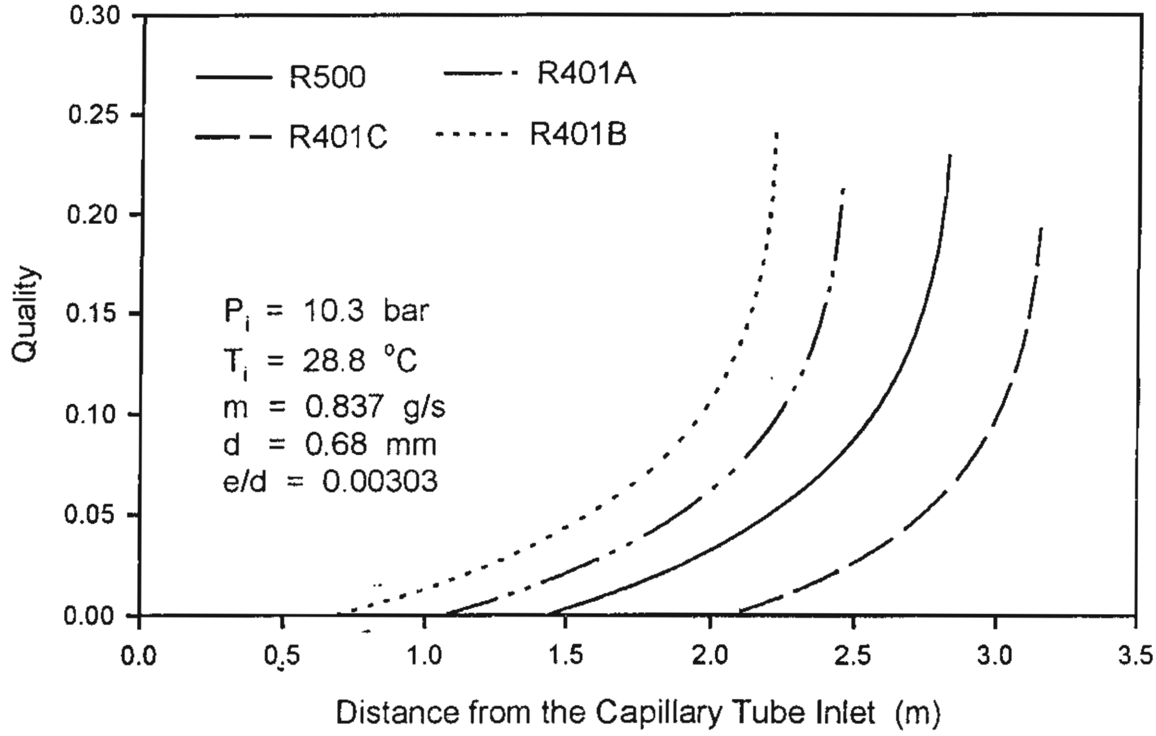

Fig. 16 Comparison of quality distributions along the capillary tube for R500, R401A, R401B and R401C

Wongwises, S., Lokathada, N., Kritsadathikarn, P., Songnetichavarit, T. , A Simulation of Refrigerant Flow Through Capillary Tube Expansion Device, *Asian J. of Energy and Environment*, 2001; 2(1):69-88.

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Ref.: AJEE/046/2001

18 April 2001

Dear Dr. Somchai Wongwises,

Your paper **"A Simulation of Refrigerant Flow Through Capillary Tube Expansion Device " (our ref.: AJEE290101)** has been read by two reviewers. They both recommended acceptance subject to revisions in accordance with the details given on the attached sheets.

We, therefore, hope that you will resubmit the paper after making revisions based on these recommendations. We suggest that if you disagree with any of recommendations, you may wish to rewrite the relevant passages to clarify why you present the information in the way you do.

Yours sincerely,

(R.H.B. Exell)
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16 August, 2001

Dear Prof. Dr. Somchai Wongwises,

AJEE290101 : "A Simulation of Refrigerant Flow Through Capillary Tube Expansion Device"

From your submission of the revised above paper, for publication in the Asian Journal of Energy & Environment (AJEE), we are please to inform you that your revised paper has been accepted for publication in the next issue (Volume 2 Issue 1).

In order for publication of your paper to proceed, your manuscript is being sent to the copy editor and then to the publisher for typesetting. Due to a large backlog of accepted manuscripts with the copy editors, there will be a delay in your receipt of your galley proofs. However, please be assured that you will receive "master proofs" for your approval from the typesetters via e-mail prior to publication.

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Yours sincerely,

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Ref.: AJEE/159/2001

November 20, 2001

Dear Dr. Somchai Wongwises,

AJEE290101 -- A Simulation of Refrigerant Flow Through Capillary Tube Expansion Device

From your accepted paper to be published the Asian Journal of Energy & Environment (AJEE) Volume 2 Issue 1/2001, enclosed here are master proof for your approval from the typesetters prior to publication.

Please be informed that galley proofs will be sent to the corresponding author for minor corrections. The corrections are restricted to printer's error or misprints only. Major alterations are not accepted and should be returned to the Production Manager within <u>one week</u>. If the master proof is not be returned within the specific time, we will regard that the draft is all correct.

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Thank you for submitting your manuscript to AJEE for publication.

Yours sincerely,

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A Simulation of Refrigerant Flow Through Capillary Tube Expansion Device

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Abstract: A homogeneous flow model is applied to study the flow characteristics of refrigerants in adiabatic capillary tubes. The basic physical equations governing the flow are established from the conservation of mass, energy and momentum. The obtained differential equations are solved simultaneously by the Runge-Kutta method. The model input parameters are pressure and temperature at capillary tube inlet, mass flow rate of refrigerant, roughness and diameter of the capillary. The simulation can be used to determine the appropriate size of the capillary tubes used in household air conditioners and refrigerators, especially to select the capillary tube length for given operating conditions.

Keywords : two-phase flow, homogeneous flow, capillary tube, adiabatic flow, alternative refrigerant

Introduction

The capillary tube is the most widely used as expansion device in small domestic vapor compression air conditioners and refrigerators. It is made from a small-bore hollow copper tube (in the order of 0.5×10^{-3} to 1.5×10^{-3} m. diameter) of about 2 to 5 m. in length [1]. It is used as an automatic flow rate controller for the refrigerant when varying load conditions and varying condenser and evaporator temperatures are to be encountered. Its simplicity, low initial cost and low starting torque of compressors are compelling reasons for its use.The capillary tube's physical configuration is very simple, the design and analysis of flow and heat transfer characteristics inside the tube are however complex ones. The design of capillary tubes has been studied both analytically and experimentally, mostly for pure refrigerants. Bansal et al. [1] presented a homogeneous two-phase flow model, CAPIL to study the performance and design aspects of adiabatic capillary tubes. The REFPROP data base which is based on the Carnahan-Starling-DeSantis equation of state was used to determine the thermodynamics and transport properties of the refrigerants. Melo [5] investigated experimentally the effects of diameter and length of adiabatic capillary tube, refrigerant subcooling, condensing pressure and the type of the refrigerant (R12, R134a, R600A) on the mass flow rates. Sami et al. [7] presented a numerical model for predicting capillary tube performance using new alternative refrigerants, both pure refrigerants (R12, R22, R134a) and binary mixtures (R410A, R410B, R507, R32/R134a). Numerical results revealed that the proposed model fairly simulated their experimental data and those of other researchers. Wong et al. [8] used a homogeneous two-phase flow model to simulate and compare the flow characteristics of R12 with those obtained from the separated flow model.The results showed that the separated flow model using Lin's pressure gradient correlations [4] and the Miropolskiy's slip ratio [6], gave better prediction. Jung et al.[2] modified the Stoecker's model [10] to provide simple correlations for sizing the capillary tubes used with R22, R134a, R407C and R410A. Various effects due to the degree of subcooling, sudden contraction at capillary tube inlet, various viscosity models and friction factors were considered. Wongwises et al.[9] provided the results of numerical simulation for R12, R22, R134a, R502, R404A, R407B, R407C, R410A, R410B, R507A. An example of capillary tube selection chart developed from the simulation is shown.

Although some information is currently available on flow characteristics of refrigerant in a capillary tube, there still remains room to discuss. In the present study, the main concern is to develop the flow model and study the flow characteristics of some pure refrigerants (R12, R22, R134a) and refrigerant mixtures (R407E (R32/R125/R134a ; 25/15/60 %), R410B (R32/R125 ; 45/55 %), R502 (R22/R115 ; 48.8/ 51.2), R408A (R125/R143a/R22 ; 7/46/47 %)).

Mathematical Modelling

As shown in Fig. 1, the flow of refrigerant through a capillary tube can be divided into two different regions; a sub-cooled liquid region (the region from the inlet of the capillary tube to the position where the saturation pressure corresponds to the capillary inlet temperature) and a liquid-vapor two-phase region.

The following assumptions are used to formulate the model:

- adiabatic flow
- one-dimensional flow
- homogeneous two-phase flow
- thermodynamic equilibrium
- straight horizontal, constant inner diameter and roughness capillary tube
- no metastable effects

Sub-cooled Liquid Region

For steady fully-developed incompressible flow in a capillary tube, the integral form of the momentum equation at differential distance dz is

$$\tau_W(\pi d)dz + A_O dP = 0 \qquad \text{------------------------(1)}$$

where the shear stress at wall, τ_w, can be determined from

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

where f is the friction factor determined from Colebrook's equation as follows;

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

Substituting Eq. (2) into Eq. (1) and rearranging, the sub-cooled liquid length, L_{SC}, is obtained as follows;

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

where the mass flux, G, is the total mass flow rate divided by cross-sectional area of tube.

Two-phase Flow Region

The one dimensional homogeneous two-phase flow model based on that of Wong et al. [8] and Wallis [12] is used in the present study. In the model, the basic physical equations governing the flow are the continuity and conservations of energy and momentum.

As the refrigerant flows along the capillary tube, its pressure gradually drops and the liquid flashes into vapour arising purely from the reduced pressure. So at any point

$$h = x h_G + (1-x)h_L \quad \text{--------------------(5)}$$

where the quality, x, is the ratio of the mass flow rate of vapour to total mass flow rate when a substance is in a saturation state.

For homogeneous flow, the velocity of each phase is equal, so

$$V = V_G = V_L \quad \text{-----------------------------(6)}$$

For flow in a capillary tube with no applied works and neglecting the elevation changes, the following form of energy equation is obtained

$$\frac{d}{dz}\left(xh_G + (1-x)h_L + \frac{V^2}{2}\right) = 0 \quad \text{--------------(7)}$$

or $$\left(x\frac{dh_G}{dz} + h_G\frac{dx}{dz}\right) + \left((1-x)\frac{dh_L}{dz} - h_L\frac{dx}{dz}\right) + \left(V\frac{dV}{dz}\right) = 0 \quad \text{-----(8)}$$

For a pure substance in equilibrium, the enthalpies and densities at saturation state can be arranged in the function of pressure.

From elementary calculas we know that if h is a differentiable function of P, and P is a differentiable function of z, then

$$\frac{dh}{dz} = \frac{dh}{dP}\frac{dP}{dz} \quad \text{--------------------------(9)}$$

On rearranging, we get

$$\left(\frac{dP}{dz}\right)\left(x\frac{dh_G}{dP}+(1-x)\frac{dh_L}{dP}\right)+\left(h_{LG}\frac{dx}{dz}\right)+\left(V\frac{dV}{dz}\right) = 0 \quad \text{----(10)}$$

Average velocity of refrigerant flowing along the capillary tube is

$$V_G = V_L = V = G\upsilon \quad \text{------------------------(11)}$$

where mass flux, G, is the mass flow rate of mixture divided by the cross-sectional area of tube and specific volume of the mixture, υ, is determined from

$$\upsilon = x\upsilon_G + (1-x)\upsilon_L \quad \text{----------------------(12)}$$

Determining the differential term of Eq.(11), we get

$$\frac{dV}{dz} = G\left(x\frac{d\upsilon_G}{dz}+(1-x)\frac{d\upsilon_L}{dz}+(\upsilon_G-\upsilon_L)\frac{dx}{dz}\right) \quad \text{----------(13)}$$

Substituting Eq. (13) into Eq.(10) and rearranging gives

$$\frac{dP}{dz} = -\frac{dx}{dz}\left(\frac{A}{B}\right) \quad \text{---------------------------(14)}$$

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 momentum equation is often rewritten as an explicit equation for the pressure gradient. The total pressure gradient $\left(\frac{dP}{dz}\right)$ can be expressed as follows

$$\frac{dP}{dz} = \left(\frac{dP}{dz}\right)_a + \left(\frac{dP}{dz}\right)_g + \left(\frac{dP}{dz}\right)_f \text{ -----------(15)}$$

The different three components are regarded as accelerational, gravitational and frictional terms of the total pressure gradient.

Accelerational pressure gradient cannot be measured directly. It can be, however, obtained from the momentum flux as follows

$$\left(\frac{dP}{dz}\right)_a = -\frac{m}{A_o}\frac{d(V)}{dz} \text{ ---------------------(16)}$$

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

Gravitational pressure gradient is negligible in this case because the flow is horizontal.

Frictional pressure gradient can be calculated from

$$\left(\frac{dP}{dz}\right)_f = \frac{-f_{tp}\, G^2((1-x)\upsilon_L + x\upsilon_G)}{2d} \text{ -------------(17)}$$

Substituting Eqs. (16) and (17) into Eq. (15), gives

$$\frac{dx}{dz} = \frac{\left(\frac{dP}{dz}\right)_f - C\frac{dP}{dz}}{D} \quad \text{-----------------(18)}$$

where

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

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

Eq. (3) used for the single-phase will be modified to calculate the two-phase friction factor, f_{tp}, with the Reynolds number being defined as:

$$\text{Re} = \frac{Gd}{\mu_{tp}} \quad \text{------------------------------(19)}$$

The two-phase viscosity, μ_{tp}, is calculated from Dukler 's equation [13] as follows:

$$\mu_{tp} = \frac{x\upsilon_G\mu_G + (1-x)\upsilon_L\mu_L}{x\upsilon_G + (1-x)\upsilon_L} \quad \text{------------(20)}$$

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

Solution Method

The properties of refrigerent are taken from REFPROP [11] and are developed as a function of pressure. The calculating the friction factor by Eq.(3) and substituting P_{sat} into Eq.(4) with the saturation pressure of refrigerant at the inlet temperature of the capillary tube, the length of the single-phase region is obtained. The exit condition of the single phase flow region is used to be an inception of vaporization of the two-phase flow region. In the two-phase flow region, the Runge-Kutta method is used to solve Eqs. (14) and (18) and the calculation is terminated when the flow is at the choked flow condition.

Results and Discussion

Figures 2 and 3 show the relation between saturation pressure and saturation temperature for each pairs of the refrigerants. It can be seen that the differences of the pressure-temperature profiles for each pairs are small. The results from the simulation obtained by using the mathematical model are properties along the capillary tubes. The effect of mass flow rate, pressure and temperature at the capillary tube inlet, diameter and relative roughness, on the total length of capillary tube were investigated. The present model is validated by comparing with the R12 data measured by Li et al. [3]. Figures 4 and 5 show the comparison of the simulation results with the R12 data of Li et al for inlet temperatures of 23.4 °C and 31.4 °C respectively. The experimental data agree quite well with the model.

Figures 4 and 5 also show how pressure varies with position along the capillary tube. In the single-phase region, due to frictional effects in fully developed flow in a constant-area tube, the pressure of refrigerant decreases linearly along the capillary tube. After the position of the inception of vaporization, due to frictional and accelerational effects, the pressure of refrigerant drops relatively fast and more rapidly as the flow approaches the choked flow condition. However, in real situation, due to the delay of vaporization, the onset of vaporization may not occur at the end of the single-phase region.

The comparison of the local pressure distribution of R12 and of R134a is shown in Figure 6, the flow of R12 through the capillary tube gives a lower pressure drop per unit length (dP/dz) than that of R134a. In the other word, at the same pressure drop, R134a needs shorter capillary tube length. Figures 7 and 11 show the temperature distributions along the capillary tube for R12, R134a and R502, R408A respectively. In the single phase region, the temperature of refrigerant along the capillary tube remains constant as expected. Once the inception of vaporization has taken place, the temperature drop will be accelerated as the flow approaches the choked condition.

Comparisons on the pressure drop characteristics for the rest of each pair of refrigerant (R22 vs R407E and R410B; R502 vs R408A) show that for all cases the traditional refrigerant flowing through a capillary tube gives a slightly lower pressure drop per unit length in the single-phase region and gives a sinificantly lower pressure drop per unit length in the two-phase region than the alternative refrigerants. The traditional refrigerants also give longer single-phase region which resulted in a longer total tube length. Figure 9 shows the distribution of quality along the capillary tube. For all cases, the quality in the single phase region is zero up to the flash point and then increases in a non-linear fashion, rising more rapidly as the choked flow condition is approached. It is also shown that in general, traditional refrigerants vaporize later than their corresponding alternative refrigerants.

Conclusions

The distributions of local pressure, temperature and quality of some common traditional and alternative refrigerants flowing through adiabatic capillary tubes are numerically investigated. The mathematical model is developed from the basic law of mass, energy and momentum conservations. Homogeneous flow is assumed for the two-phase liquid-vapor flow region. Numerical results reveal that, it is possible to use the present calculation to predict the flow characteristics in capillary tubes. The model includes various relevant parameters and can be used to determine the size of the capillary tubes used in household refrigerators and freezers.

Acknowledgement

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Nomenclature

A_o	cross-sectional area of tube, m^2	d	tube diameter, 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	dynamic viscosity, Pa s	υ	specific volume, m^3/kg
ρ	density, kg/m^3	τ	shear stress, N/m^2

Subscripts

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

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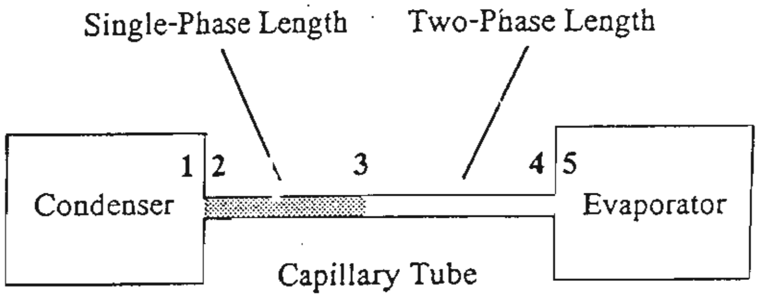

Figure 1. Schematic diagram of an adiabatic capillary tube.

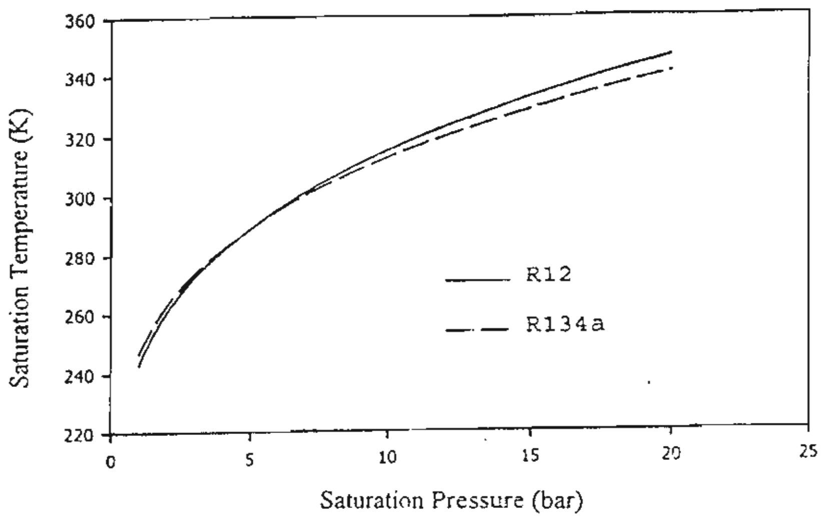

Figure 2. Saturation pressure and temperature for R12 and R134a.

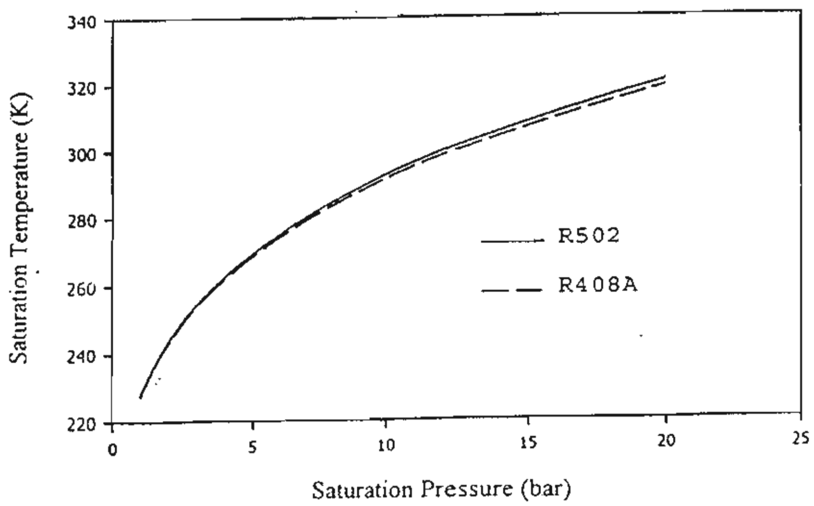

Figure 3. Saturation pressure and temperature for R502 and R408A.

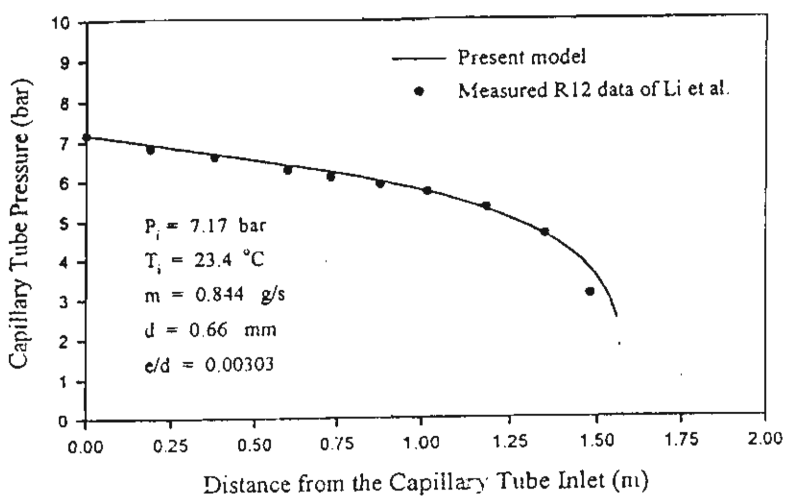

Figure 4. Comparison of pressure distributions along the capillary tube.

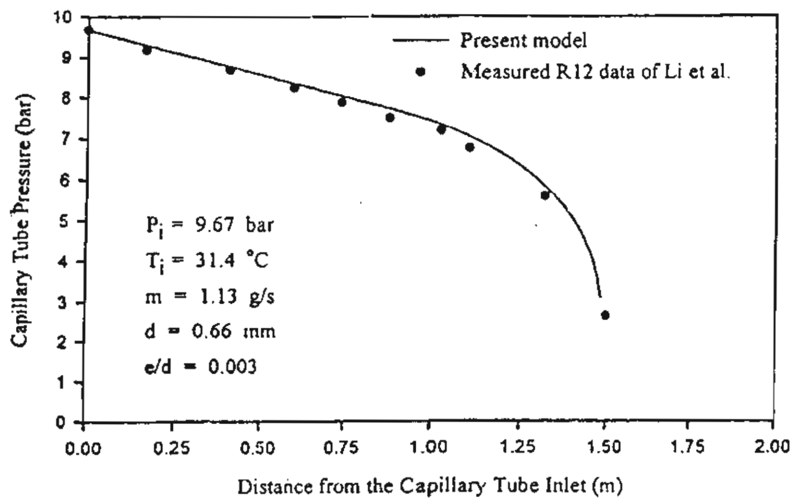

Figure 5. Pressure distributions along the capillary tube.

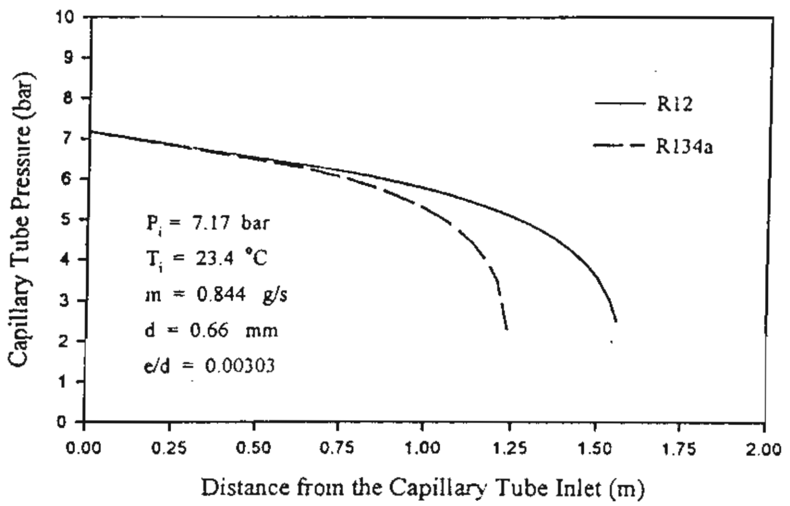

Figure 6. Comparison of pressure distributions along the capillary tube for R12 and R134a.

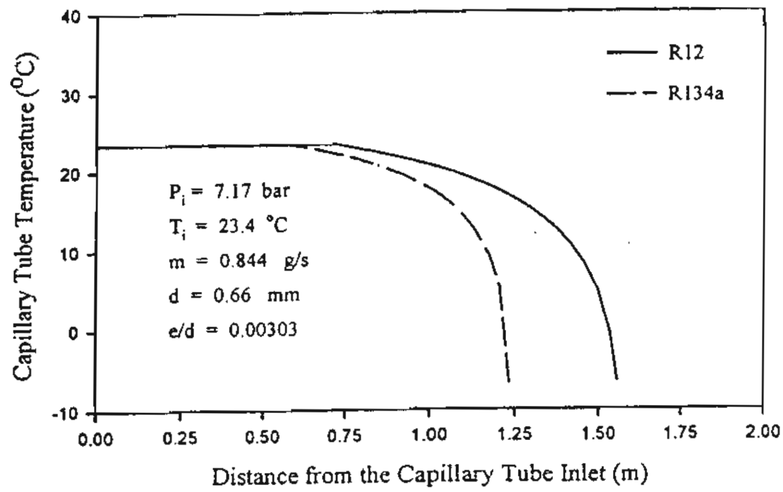

Figure 7. Comparison of temperature distributions along the capillary tube for R12 and R134a.

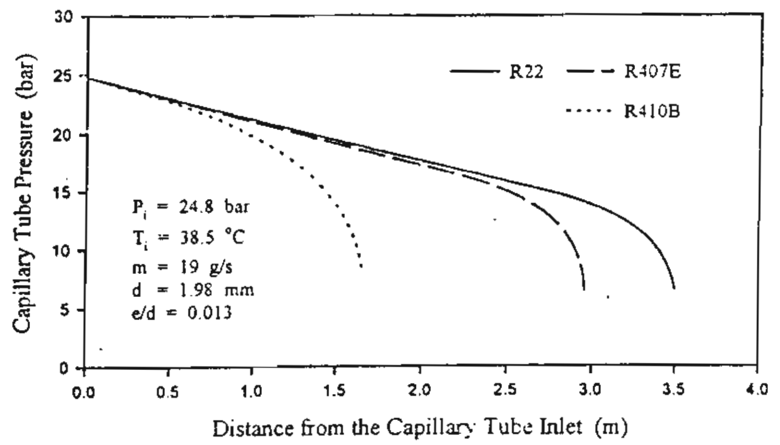

Figure 8. Comparison of pressure distributions along the capillary tube for R22, R407E and R410B.

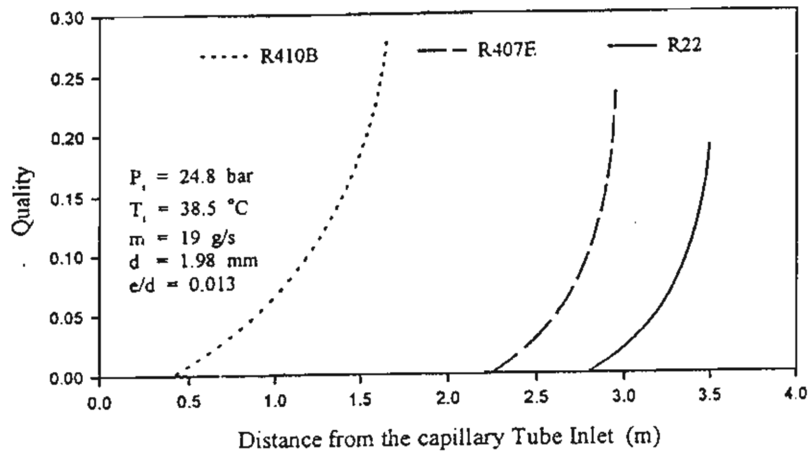

Figure 9. Comparison of quality distributions along the capillary tube for R22, R407E and R410B.

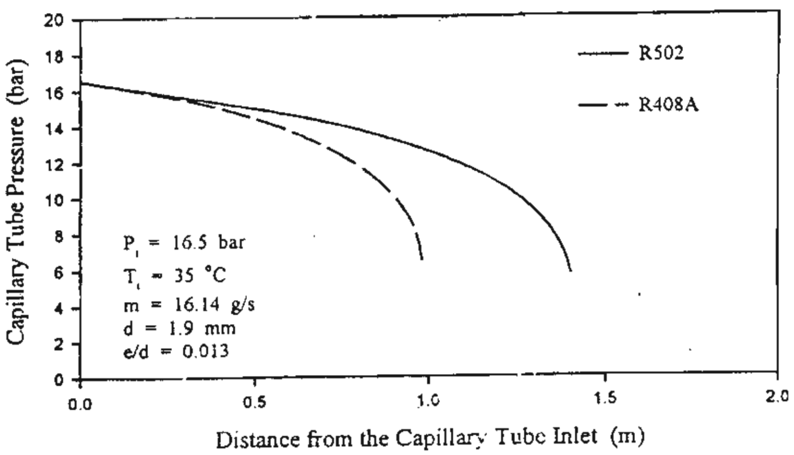

Figure 10. Comparison of pressure distributions along the capillary tube for R502 and R408A.

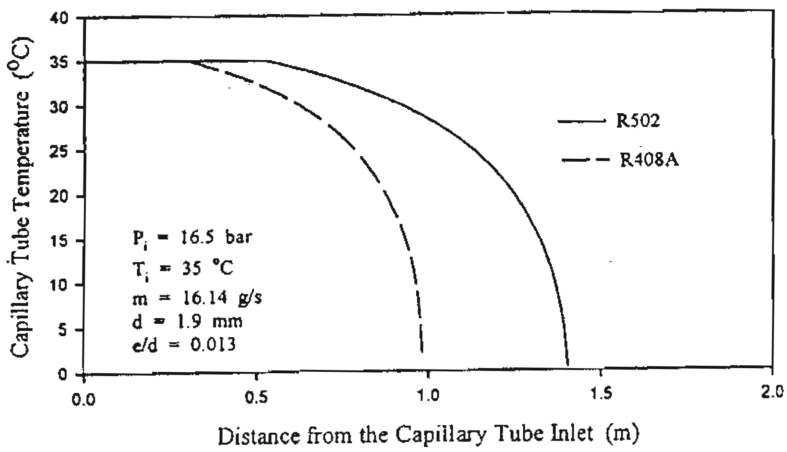

Figure 11. Comparison of temperature distributions along the capillary tube for R502 and R408A.