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THEORETICAL INVESTIGATION OF AN ADIABATIC CAPILLARY TUBE WORKING WITH PROPANE / n-BUTANE / iso-BUTANE BLENDS

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ABSTRACT

In this paper, a theoretical model is developed to predict refrigerant flow characteristics in adiabatic capillary tubes using propane/n-butane/iso-butane mixtures as working fluids in a domestic refigerator. This model is based on mass, energy and momentum conservation equations for a homogeneous refrigerant flow under different inlet conditions such as subcooled, saturated and two-phase flow. Effects of inlet pressure (8-16bar), inlet vapor quality (0.001-15%), inlet subcooling degree (1-15°C), mass flow rate (1-5 kg/h), propane mass fraction (0.5-0.7), capillary tube inner diameter (0.6-1.0mm) and tube surface roughness on the capillary tube length are predicted.

Results showed that the present model predicts data that are very close to the available experimental data in literature with an average error of 2.65%. Pressure of hydrocabon mixture (HCM) decreases while its vapor quality, specific volume and Mach number increase along the capillary tube. Also, the results indicated that capillary tube length is largely dependent on capillary tube diameter. Other parameters such as mass flow rate, inlet pressure, sub-cooling degree (or quality) and relative roughness influence the capillary tube length in that order. Capillay tube length as function of the significant parameters is presented in equation form. Also, capillary tube selection charts either to predict mass flow rates of propane/n-butane/iso-butane mixtures through adiabatic capillary tubes or to select the capillry tube size according to the required applications are developed. The comparison betweenn R12, R134a and the hydrocarbon mixture (HCM) of propane/n-butane/iso-butane indicated that for a given mass flow rate, the pressure drop per unit length is about 4.13, 5.0 and 12.0 bar/m for R12, R134a and HCM, respectively. The ratios of average mass flow rate of HCM with propane mass fraction of 0.6 to those of R12 and R134a are about 0.62 and 0.67, respectively. Average capillary tube length for HCM with propane mass fraction of 0.6 is longer than those of R134a and R12 by about 30% and 48%, respectively.

KEYWORDS; Capillary tube sizing, flow characteristics, alternative refrigerants, hydrocabon mixtures.

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Nomenclature

A	Cross-section area (m^2)
c	Fluid velocity (m/s)
C	Contraction coefficient (-)
d	Capillary tube diameter (m)
f	Friction factor (-)
G	Refrigerant mass flux (kg/m^2s)
h	Specific enthalpy (kJ/kg)
HCM	Hydrocarbon mixture
L	Capillary tube length (m)
m	Mass flow rate (kg/s)
M	Mach number (-)
p	Pressure (MPa)
PMF	Propane mass fraction (-)
Re	Reynolds number
s	Specific entropy (kJ/kg K)
T	Temperature (K)
v	Specific volume (m^3/kg)
x	Dryness fraction (-)

Greek letters

ε	Roughness (mm)
φ	Flow factor (-)
μ	Dynamic viscosity (Pa.s)
ρ	Density (kg/m^3)
Δ	Difference (-)

Subscripts

1-n	Refers to Fig. (1)
c	Contraction
con	Condensation
ch	Chock
ds	Down-stream
in	Inlet
m	Mean
r	Refrigerant
sat	Saturated
sl	Saturated liquid
sp	Single-phase
std	Standard
sv	Saturated vapor
sub	Subcooling
tot	Total
tp	Two-phase
us	Up-stream

1. INTRODUCTION

A capillary tube is made of a small internal diameter copper tube of varying length depending upon the application. It is installed in the liquid line between the condneser and evaporator of a vapor compression system to reduce the condenser pressure to the evaporator pressure and to regulate the refrigerant flow rate based on the evaporator load. It has several advantages such as simple in construction, no moving parts (which wear or stick, i.e. no maintenance is required), no receiver is necessary, low starting torque motor (low cost motor) and less expensive. When the refrigerant expands from the condenser pressure to the evaporator pressure adiabatically, i.e. tube is fully insulated, the capillary tube is called an adiabatic tube. In some refrigeration system, the capillary tube is soldered to the suction line and the combination is called a capillary tube-suction line heat exchanger. This type of capillary tube is known as a non-adiabatic capillary tube.

Sizing of capillary tubes commonly used as expansion devices in the household refrigerators and freezers working with R12 alternatives is carried out by many investigators using either theoretical, empirical [1-2] or experimental approach [3]. Experimental approach is very expensive and the empirical approach depends upon the available experimental data over a wide range of operating conditions. In the present work, theoretical approach is adopted to investigate flow characteristics through adiabaic capillary tubes and to determine their sizing.

Bittle and Pate [4] presented a theoretical model to predict flow rate of R134a through an adiabatic capillary tube. The model is validated with data of previous studies over a range of operating conditions. It was found that the model agrees reasonably well with the experimental data for R134a. A numerical model for predicting capillary tube performance using pure refrigerant or binary mixtures is proposed by Sami and Tribes [5]. The model was established for a homogeneous refrigerant flow under saturated, subcooled and two phase conditions. Numerical results of the proposed model and experimental data showed fair agreement. However, numerical models have been developed to size adiabatic and non-adiabatic capillary tubes by Sami and Maltais [6] for R22 subsitutes (R407C, R410A and R410B) and Sami et al. [7] for R502 alternatives such as azeotropic mixtures (R507, R404A) and quaternary mixture (R32/R125/R134a/R143a). Their results indicated that the system using either R407C or R507C would experience smaller pressure drop across the capillary tube compared to the other alternatives.

An adiabatic capillary tube model is developed by Wongwises et al. [8] and Wongwises and Pirompak [9] to study the flow characteristics in capillary tubes. Numerical results showed that the traditional refrigerants gave lower pressure drops (for both single-phase and two-phase flow) than the environmentally acceptable alternative refrigerants, which resulted in longer tube lengths for them. The model is validated via comparing its simulation results with the published experimental data for R12 and R134a. Fatouh and El-Kafafy [10] have reported that a hydrocarbon mixture (HCM) of propane/iso-butane/n-butane mixture with propane mass fraction of about 0.6 is a promising drop-in refrigertant for R134a in domestic refrigerators. This mixture offers many advantages such as its local availability and low price [11] compared with other refrigerants.

Despite that many investigations are carried out to size the capillary tube working with either ozone depleting substances such as R12, R22 and R502 or ozone safe refrigerants such as R134a, the sizing of capillary tubes working with HCM needs to be investigated. Thus, the main objectives of the present work are sizing adiabatic capillary tubes which use the propane/iso-butane/n-butane mixture and predicting the flow characteristics through them with evaluating the effects of operating parameters on their sizes. In order to achieve these objectives, a computer program based on mass, energy and momentum conservation equations has been developed. The input data are thermophysical properties, inlet conditions (subcooled, saturated or two-phase), refrigerant mass flow rate and capillary tube inner diameter. Thermophysical properties include vapor pressure, specific volume of subcooled and saturated liquid, specific volume of saturtaed vapor, saturated liquid and vapor viscosity. The outputs are flow characteristics and size of capillary tube.

2. CAPILLARY TUBE SIMULATION MODEL

Figure (1) shows a schematic diagram of a capillary tube connecting the condenser and evaporator. In order to develop the model, the following assumptions are made:

- steady flow, one dimensional, adiabatic and homogeneous flow through the capillay tube.
- thremodynamic equilibrium through the capillary tube.
- metastable effect is neglected ,
- the capillary tube is straight and horizontal with constant inner diameter and uniform surface roughness.

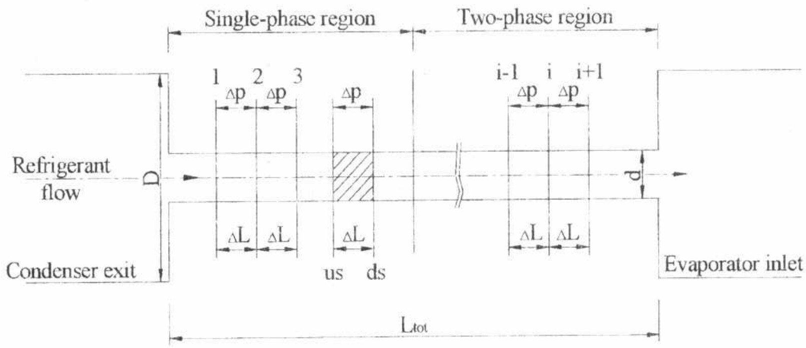


Figure (1) Schematic diagram of a capillary tube

Capillay tube length (L_{tot}) can be determined by assuming step decrements of the pressure through a particular inner diameter (d) as shown in Fig. (1), then calcualting the corresponding required increments of the length (ΔL). These increments can be accumulated to give the capillary tube length required for the specified pressure drop to reach the downstream pressure, i.e.

$$L_{tot} = \Delta L_1 + \Delta L_2 + \ldots\ldots\ldots \Delta L_n \tag{1}$$

In the next section, governing equations are developed to describe the flow through each element of the capillary tube.

2.1 Governing Equations

For steady and uniform flow, mass conservation law for each element in the capillary tube can be expressed as;

$$m_r = \frac{Ac_{us}}{v_{us}} = \frac{Ac_{ds}}{v_{ds}} \tag{2}$$

or

$$G = \frac{m_r}{A} = \frac{c_{us}}{v_{us}} = \frac{c_{ds}}{v_{ds}} \tag{3}$$

Where m_r is the refrigerant mass flow rate, c is the fluid velocity, v is the specific volume, A is the cross-section area of the capillary tube and G is the refrigerant mass flux. Subscripts "us" and "ds" denote to upstream and downstream, respectively.

For steady flow, one dimensional, adiabatic and homogeneous flow without external work through a straight, horizontal and constant inner diameter with uniform relative roughneuos capillay tube, energy and momentum conservation laws are given by Eqs. (4) and (5), respectively.

$$h_{us} + \frac{c^2{}_{us}}{2} = h_{ds} + \frac{c^2{}_{ds}}{2} \tag{4}$$

$$(p_{us} - p_{ds}) - \frac{f_m c^2{}_m \Delta L}{2d v_m} = \frac{m_r}{A}(c_{ds} - c_{us}) \tag{5}$$

Where v_m, c_m and f_m are the arthimatic average of upstream and downstream specific volume, fluid velocity and friction factor, respectively.

As the refrigerant passes through the elements (increments) of the capillary tube, its pressure and temperature drop, thereby the vapor quality (x) continuously increases. Hence, in the present simulation model, the flow within the capillary is divided into two flow regions; namely single phase and two-phase flow regions as shown in Fig. (1). Two phase flow region starts when the saturated temperature of the liquid approaches the initial subcooled liquid temperature. Wongwise and Pirompak [9] and Liang and Wong [12] reported that the friction factor of either single-phase flow (f_{sp}) or homogenous two-phase flow (f_{tp}) can be calculated from Colebrook correlation as follows;

$$\frac{1}{\sqrt{f}} = 1.14 - 2\log\left(\frac{\varepsilon}{d} + \frac{9.3}{\mathrm{Re}\sqrt{f}}\right) \tag{6}$$

where

$$\mathrm{Re}_{sp} = \frac{4\, m_r}{\mu_{sp} \pi\, d} \tag{7}$$

$$\mathrm{Re}_{tp} = \frac{4\, m_r}{\mu_{tp} \pi\, d} \tag{8}$$

Where μ_{sp} and μ_{tp} are the single-phase and two-phase dynamic viscosities.

2.2 Refrigerant Properties

It is well known that refrigerant status through the capillary tube changes from either subcooled or saturated state to two-phase state. In order to conduct the present research, refrigerant properties, at these conditions, are required. Based on published data by SUPERTRAPP [13] for all the refrigerants under consideration, subcooled liquid in addition to saturated liquid and vapor refrigerant properties were correlated to facilitate the computation work. For subcooled conditions, refrigerant specific enthaply (h), specific entropy (s) and specific volume (v) are given as function of subcooled temperature (T_{sub}) and saturated pressure (p_{sat}) while for saturated conditions, these properties are obtained as a function of the saturated pressure. Refrigerant specific enthaply, specific entropy and specific volume for two-phase flow can be written as follows;

$$h = x\, h_{sv} + (1-x)\, h_{sl} \tag{9}$$

$$s = x\, s_{sv} + (1-x)\, s_{sl} \tag{10}$$

$$v = x\, v_{sv} + (1-x)\, v_{sl} \tag{11}$$

A number of viscosity models is available in literature to calculate the two-phase dynamic viscosity (μ_{tp}). Xu and Bansal [14] reported that McAdam's model is suitable for prediction of two-phase dynamic viscosity for a wide range of refrigerants, i.e.

$$\frac{1}{\mu_{tp}} = \frac{x}{\mu_{sv}} + \frac{(1-x)}{\mu_{sl}} \tag{12}$$

Saturated liquid and vapor dynamic viscosities are computed based on the saturated pressure. Subscripts "sl" and "sv" stands for saturated liquid and vapor, respecively.

2.3 Solution Method

Figure (1) indicates that the capillary tube is divided into many elements in single phase and two-phase flow regions. The pressure at any point in the capillary tube can be calculated from

$$p_i = p_{in} - i\Delta p \tag{13}$$

Where i is the element number and p_{in} is the capillary tube inlet pressure, which can be expressed as;

$$p_{in} = p_{con} - \Delta p_c \tag{14}$$

Where p_{con} is the saturated pressure at the condenser exit and Δp_c is the pressure drop due to area contraction. The pressure drop due to aera contraction can be determined according to Stoecker and Jones [15] by the following equation:

$$\Delta p_c = (G)^2 \left(\frac{1}{2\rho_{sl}}\right)\left(\frac{1}{C_c} - 1\right)^2 \tag{15}$$

Where ρ_{sl} is the density of saturated liquid refrigerant, C_c is the contraction coefficient. It is necessary to obtain the contraction coefficent in equation form to facilitate the required computations. Hence, based on reported data of Stoecker and Jones [15], contraction coefficient is correlated as function of area ratios (A_2/A_1) in the following form;

$$C_c = 0.6083 + 0.5476\left(\frac{A_2}{A_1}\right) - 0.3266\left(\frac{A_2}{A_1}\right)^2 + 0.1687\left(\frac{A_2}{A_1}\right)^3 \tag{16}$$

Where A_1 and A_2 are the cross-section areas of condenser outlet tube and capillary tube, respectively.

For single-phase, upstream properties (h_{us}, c_{us}, v_{us}, and μ_{us}) of the first element are calculated based on inlet pressure and subcooled temperature. With the pressure p_i and subcooling temperature, v_{ds} and μ_{ds} are determined then Eqs. (2) and (4) are solved for c_{ds} and h_{ds}. If p_i is reached the saturated pressure corresponding to the initial subcooling temperature, two-phase flow region starts. Hence, for each element in the two-phase flow region, substituting Eqs. (2) and (9) into Eq. (4) yields

$$h_{us} + \frac{c^2{}_{us}}{2} = h_{sl,ds} + x_{ds}\left(h_{sv,ds} - h_{sl,ds}\right) + \frac{G^2}{2}\left[v_{sl,ds}\left(1 - x_{ds}\right) + v_{sv}\, x_{ds}\right]^2 \qquad (17)$$

Expanding the right hand side of Eq. (17) and rearranging yields

$$\left[\frac{1}{2}\left(\Delta v_{ds} G\right)^2\right]x^2{}_{ds} + \left[G^2\, v_{sl,ds}\, \Delta v_{ds} + \Delta h_{ds}\right]x_{ds} + \left[\frac{1}{2}G^2\, v^2{}_{sl,ds} - h_{us} - \frac{c^2{}_{us}}{2} + h_{sl,ds}\right] = 0 \qquad (18)$$

Where Δh is the latent heat (h_{sv}-h_{sl}) and Δv is the specific volume difference (v_{sv}-v_{sl}) through each element in the two-phase region. Equation (18) is a quadratic equation from which the quality (x) can be exactly determined.

In general, for given upstream conditions (h_{us}, c_{us}, v_{us}, and μ_{us}) and pressure drop through the considered element, the above equations have to be solved for h_{ds}, v_{ds}, c_{ds}, and μ_{ds} according to flow zone. As a result the mean specific volume (v_m), fluid velocity (c_m) and friction factor (f_m) can be obtained, then the substituted into Eq. (5) to evaluate ΔL. It may be stated that same procedure has to be repeated for the next element considering that the downstream conditions of the previous element are the entering (upstream) conditions to the next element, then all incremental lengths can be accumulated to obtain the total length of the capillary tube. The calculation is terminated when either specific entropy reduces or the flow is chocked, i.e. velocity of the two phase fluid equals the sound velocity (Mach number =1). Jung et al. [16] determined the two-phase Mach number by the following eqtaution;

$$M_{tp} = \left\{-G^2\left[x\frac{dv_{sv}}{dp} + (1-x)\frac{dv_{sl}}{dp} + \Delta v\left(\frac{dx}{dp}\right)_h\right]\phi\right\}^{1/2} \qquad (19)$$

where

$$\phi = \left[1 + \frac{G^2\, \Delta v\left(v_{sl} + x\,\Delta v\right)}{\Delta h}\right]^{-1} \qquad (20)$$

A computer program based on equations from (1) to (20) with necessary subroutines of refrigerant thermo-physical properties is developed for sizing the capillary tube and for predicting flow characteristics of propane/n-butane/isobutene mixtures. The input data for simulation are working fluid type, inlet pressure, inlet subcooling dergree, inlet vapor quality, capillary tube inner diameter, and surface roughness. Also, thermodynamic and transport properties of the considered working fluids are needed for computation. The program calculates flow characteristics along the capillary tube, capillary tube length, refrigerant mass flow rate, and flow factors over the entire range of the considered operating conditions.

3. RESULTS AND DISCUSSIONS

The present work deals with capillary tube working with hydrocarbon mixture (HCM) of propane, iso-butane and n-butane. The results are presented under five titles; (i) model verification, (ii) flow characteristics of the hydrocarbon mixture (HCM) of propane, iso-butane and n-butane, (iii) effect of various parameters, (iv) rating charts for selection of the capillary tube working with the blend of propane, iso-butane and n-butane, and (v) R12, R134a and HCM comparison.

3.1 Model Verification

Model verification is based on available experimental data for R12 and R134a. Experimental data for pressure distribution along an adiabatic capillary tube working with R12 up to chock flow conditions are presented by Li et al. [2]. Measured mass flow rates of R134a under chock flow conditions as function of sub-cooling degrees at inlet of the adiabatic capillary tube are reported in literature [3]. The pressure distribution of R12 along the adiabatic capillary tube calculated by the present model is compared with experimental data of Li et al. [2] in Fig. (2.A) while mass flow rate of R134a predicted by the present model under chock flow conditions is compared with the experimental data of Melo et al. [3] for R134a in Fig. (2.B). Experimental data are presented by symbols while predicted results are shown by solid lines. Clearly, the present model predictions are very close to the experimental data with an average error of 2.65%. Hence, the present model can be used to size capillary tubes and predict flow characteristics of new refrigerants along them.

After validation of the present model with literature data for R12 and R134a, results are generated for the hydrocarbon mixture (HCM) of propane, iso-butane and n-butane under the follwing conditions:

- capillary tube inner diameter (0.6 -1.0 mm),
- refrigerant mass flow rate (1-5 kg/h),
- inlet pressure (8-16 bar),
- relative roughness (0.002-0.004),
- sub-cooling degrees (0 -15°C),
- quality (5% to 15%), and
- propane mass fraction (0.5, 0.6 and 0.7).

Subcooling degrees and quality are used to specify the inlet conditions of the capillary tube. Due to limited space, samples of results are presented in graphical forms and all results are given in a polynomial form in section (3.3).

3.2 Flow Characteristics of HCM

Profiles of flow characteristics of the hydrocarbon mixture (HCM) of propane, iso-butane and n-butane along the capillary tube are ploted in Fig. (3) for three inlet pressures of 8, 12 and 16 bar, refrigerant mass flow rate of 2.0kg/hr, subcooling degree of 10°C, capillary tube inner diameter of 0.8mm and relative roughness of 0.003.

Figure (3.A) shows the pressure distribution along the capillary tube up to chock conditions for HCM with propane mass fraction of 0.6 for three inlet pressures. It appears from this graph that the refrigerant pressure decreases along the capillary tube. Due to friction effects, the refrigerant pressure drops linearly through the first section (single-phase region) of the capillary tube. Then, most of the pressure drop occurs in the second section (two-phase region) of the capillary tube. This can be attributed to combined effects of friction and acceleration losses which cause the refrigerant pressure to decease sharply and more rapidly as the flow approaches the chock conditions. With reference to Fig. (3.A), it can be noted that chock pressure (at the end point of each curve) of 1.25, 1.5 and 1.65 bar occurs at capillary length of 5.26, 7.99 and 10.65m for inlet pressures of 8, 12 and 16 bar, respectively. These results revealed that inlet pressures of 12 and 16 bar yield higher chock pressure than that of 8 bar by about 20% and 32%, respectively.

Vapor quality distribution of HCM with propane mass fraction of 0.6 along the capillary tube is shown in Fig. (3.B) which indicates that variation of the vapor quality along the capillary tube is non-linear for the considered inlet pressures. As the refrigerant flows, rapid pressure drop occurs which produces a large quantity of flashing vapor through the capillary tube. This leads to the increase in quality along the capillary tube as shown in Fig. (3.B). For given mass flow rate and inlet conditions, flow was chocked at qualities of 0.266, 0.349 and 0.425 for inlet pressures of 8, 12 and 16bar, respectively. Clearly, the length of subcooled part (at vapor quality =0) is about 3.57m, 4.64m and 5.64m for inlet pressure of 8, 12 and 16 bar, respectievly. Thus, the length of subcooled part is about 0.678, 0.581 and 0.530 of the capillary tube length for inlet pressure of 8, 12 and 16 bar, respectievly.

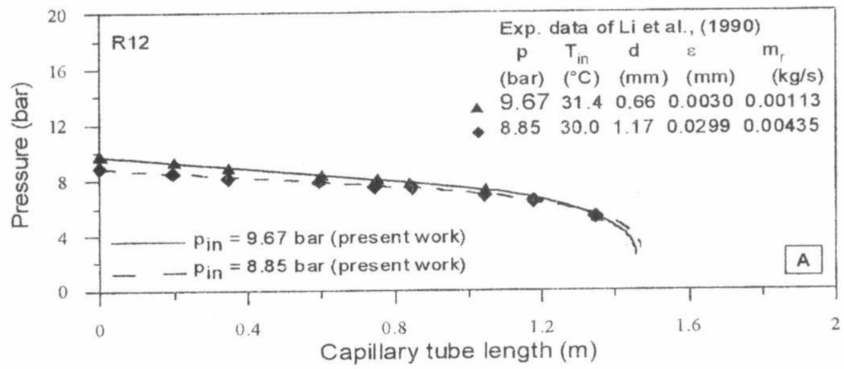


Fig. (2.A) Pressure distribution along capillary tube for R12

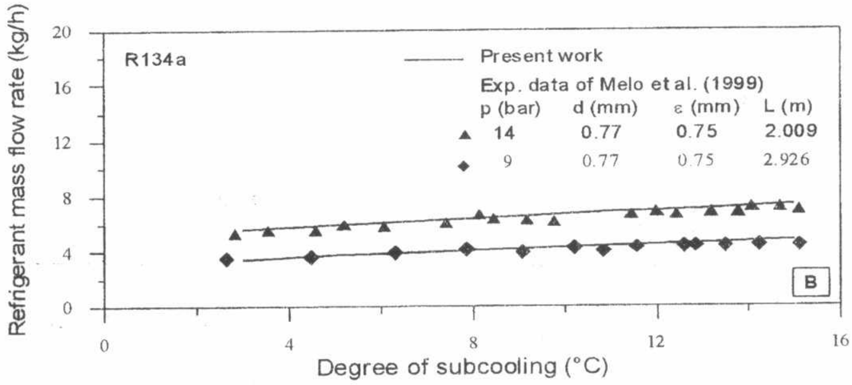


Fig. (2.B) Mass flow rate of R134a against subcooling degree

Variation of specific volume of HCM with propane mass fraction of 0.6 along the capillary tube is presented in Fig. (3.C). It is clear that specific volume increases along the capillary tube. When refrigerant flows through the capillary tube, saturated pressure decreases (Fig. 3.A) causing the vapor quality to increase (Fig. 3.B) and the specific volume of both saturated liquid and vapor refrigerant to increase. But the increase in specific volume of saturated vapor is larger than that of saturated liquid refrigerant. Thus, due to increase in both vapor quality and the specific volume of saturated vapor refrigerant, specific volume of two-phase flow increases along capillary tube.

Figure (3.D) illustrates variations of two-phase Mach number along the capillary tube. Mach number increases with the capillary tube length. It is evident from Fig. (3.C) that the slope of specific volume line for inlet pressure of 8 bar is larger than that of 12 bar or 16 bar. As a refrigerant flows, friction and acceleration losses increase causing the vapor quality to increase, thereby specific volume of two-phase flow increases. This causes refrigerant velocity and then Mach number to increase when the refrigerant mass flow rate was kept constant. It is clear that chock conditions (M=1) occurs at shorter tube length in the case of 8 bar than other cases of 12 and 16 bar as shown in Fig. (3.D).

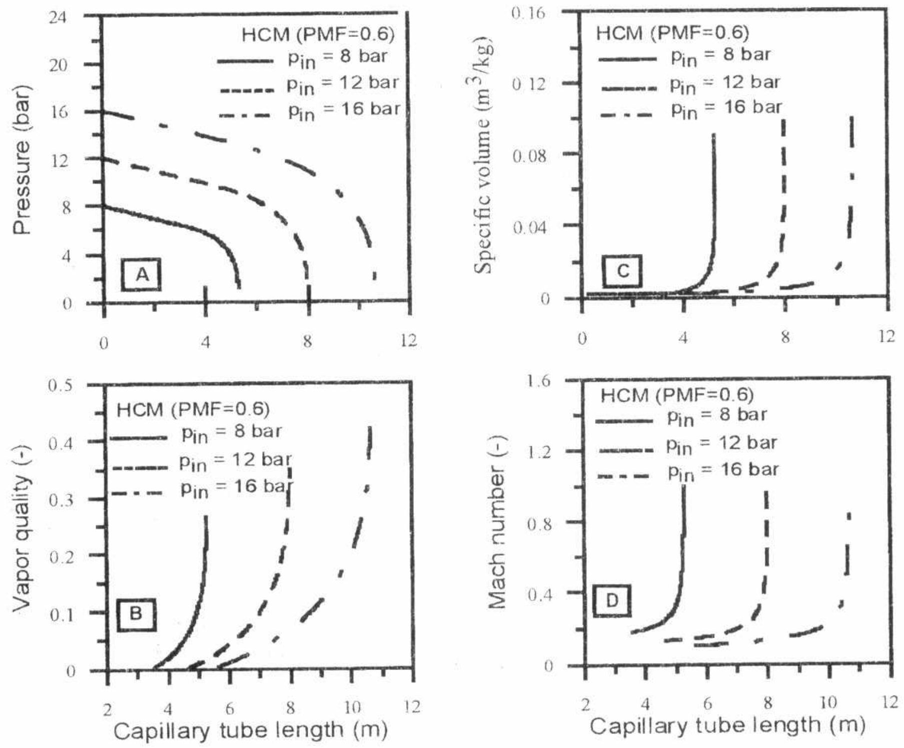


Fig. (3) Variation of flow characteristics along the capillary tube

3.3 Factors Affecing Capillary Tube Length

As mentioned earlier, inlet pressure (p_{in}), subcooling dergree (ΔT_{sub}), vapor quality (x), refrigerant mass flow rate (m_r), propane mass fraction (PMF), surface roughness (ε) and capillary tube inner diameter (d) are varied to predict their effects on the length of capillary tube working with hydrocarbon mixtures (HCM) of propane, iso-butane and n-butane as shown Fig. (4).

Figures (4.A) and (4.B) show expansion process through the capillary tube from upstream (inlet) pressure to downstream (chock) pressure on pressure-enthalpy diagram for different inlet pressures and subcooling degrees, respectively. Figure (4.A) illustrates that total and two-phase pressure drops along the capillary tube increase with inlet pressure while Fig. (4.B) shows that when degree of subcooling increases, total pressure drop through the capillary tube remains nearly constant but single-phase pressure drop increases and two-phase pressure drop decreases. These two figures indicate that effects of downstream pressure on total pressure drop and on capillary tube length are insignificant.

Dependency of capillary tube length upon the inlet pressure is shown in Fig. (4.C) for various subcooling degrees. The figure shows that the length of capillary tube is directly proportional to the inlet pressure. This can be attribuated to increase in total pressure drop with increasing inlet preesure as illustrated in Fig. (4.A), which requires longer capillary tube for a given mass flow rate. Clearly, capillary tube length at inlet pressure of 16 bar is about double that at inlet pressure of 8 bar for various subcooling degrees. Sensitivity of capillary tube length to inlet pressure increases as subcooling degrees increases.

It is evident from Fig. (4.C) that the capillary tube length increases with increasing of subcooling degrees for a given inlet pressure. As the subcooling degrees at the capillary tube entrance increase, single-phase pressure drop increases while two-phase pressure drop decreases as ahown in Fig. (4.B), thereby the single-phase length increases whereas two-phase length decreases. But, the rate of length increase in single-phase is greater than rate of length reduction in two-phase, so longer capillary tube is needed. This reveals the dependency of capillary tube length upon the subcooling degrees. The effect of subcooling degrees on the capillary tube length at high inlet pressure is lower than that at low inlet pressure. This is evident by the fact that lengths of capillary tube for inlet pressure of 16bar with Δt_{sub} of 15°C and 10°C are higher than that at Δt_{sub} of 5°C by about 43.1% and 22.1%, respectively, whereas these ratios become 60.2% and 31.3% at inlet pressure of 8 bar.

Effect of inlet quality on the capillary tube length can be depicted from Fig. (4.D). For a given inlet pressure, the capillary tube length is inversely proportional to quality. This is because large quality results in a large specific volume, which leads to increase in refigerant velocity and pressure drop. Thus, the capillary tube length must decrease to maintain a constant pressure drop. The influence of inlet pressure on the capillary tube length increases as the quality decreases., e.g. when the inlet pressure increases from 8bar to 16bar, the capillary tube length increases by about 4.54m, 3.96m and 3.5m with inlet qualities of 5%, 10% and 15%, respectively. In other words, the ratios of the average capillary tube length with inlet qualities of 10% and 15% to that with inlet quality of 5% are about 80% and 67%, respectively.

Figure (4.E) shows capillary tube length dependency upon HCM mass flow rate. Clearly, the capillary tube length is inversely proportional to mass flow rate when other operating parameters are held constant. This is due to the fact that as the mass flow rate decreases, refrigerant velocity decreases and then the pressure drop reduces, thereby the capillary tube length must increase to cause the same pressure drop. Based on the results in this figure, line slope for mass flow rates of 1.5, 2.0 and 2.5 kg/h are 0.6, 0.345 and 0.23, respectively, indicating that the dependency of capillary tube length upon the inlet pressure increases when mass flow rate decreases. This is confirmed because the average capillay tube length at mass flow rate of 1.5kg/h is longer than those of mass flow rates of 2.0 and 2.5 kg/h by about 24% and 62.1%, respectively.

Effect of propane mass fraction of HCM on capillary tube length can be predicted from Fig. (4.F), which reveals that as porpane mass fraction increases from 0.5 to 0.7, the length of capillay tube increases by 3.17% and 2.25% at inlet pressure of 8bar and 16 bar, respectively. This indicates that effect of propane mass fraction on the length of capillary tube is insiginficant. However, average capillary tube length for PMF of 0.5 is nearly 1% shorter while that length for PMF of 0.7 is about 1.7% longer than that of PMF of 0.6. This means that PMF of 0.6 can be used to determine the capillary tube length for PMF from 0.5 to 0.7 with adequate accuracy.

Figure (4.G) illustrates effect of inlet pressure on capillary tube length for various relative roughnesses. Clearly, capillary tube length is inversely proportional to relative roughness. This is due to the fact that as relative roughness increases, friction factor (Eq. 6) increases causing capillary tube length (Eq. 5) to decrease for given operating conditions. It is evident from Fig. (4.G) that effect of inlet pressure on the length of capillry tube increases when relative roughness reduces. It can be concluded that relative roughness of 0.003 and 0.004 yield shorter average capillary tube length by about 4.2% and 9.5% compared to that length of 0.002.

Variation of the capillary tube length with the inlet pressure for different capillary tube diameters is presented in Fig. (4.H), which shows that larger diameter yields longer capillary tube length. As the capillary tube diameter increases, fluid velocity decreases, thereby capillary tube length should be increased in order to maintain the required pressure drop for a given refrigerant mass flow rate. It should be stated that as the diameter increases the slope of its line increases, indicating that capillary tube length is more sensitive to inlet pressure for larger diameter than small diameter. This figure domenstrates that average capillary tube lengths for inner diameters of 0.7 and 0.8 mm are about 2.2 and 3.1 times, respectively compared to that for inner diameter of 0.6mm.

In summary, the capillary tube length is inversely proportional to mass flow rate, dryness fraction and relative roughness whereas directly proprtional to inlet pressure, subcooling degrees and inner tube diameter. Actually, the relation between the capillary tube length and these operating paramters is very useful for design engineers. Thus, in order to generalize predicted data by the present capillary tube model, the capillary tube length for inlet sub-cooled or two-phase flow conditions is correlated with the relevant parameters in the following form;

$$L = k_0 (d)^{k1} (m_r)^{k2} (p_{in})^{k3} (\varepsilon)^{k4} (\Delta T_{sub})^{k5} (x)^{k6} \tag{21}$$

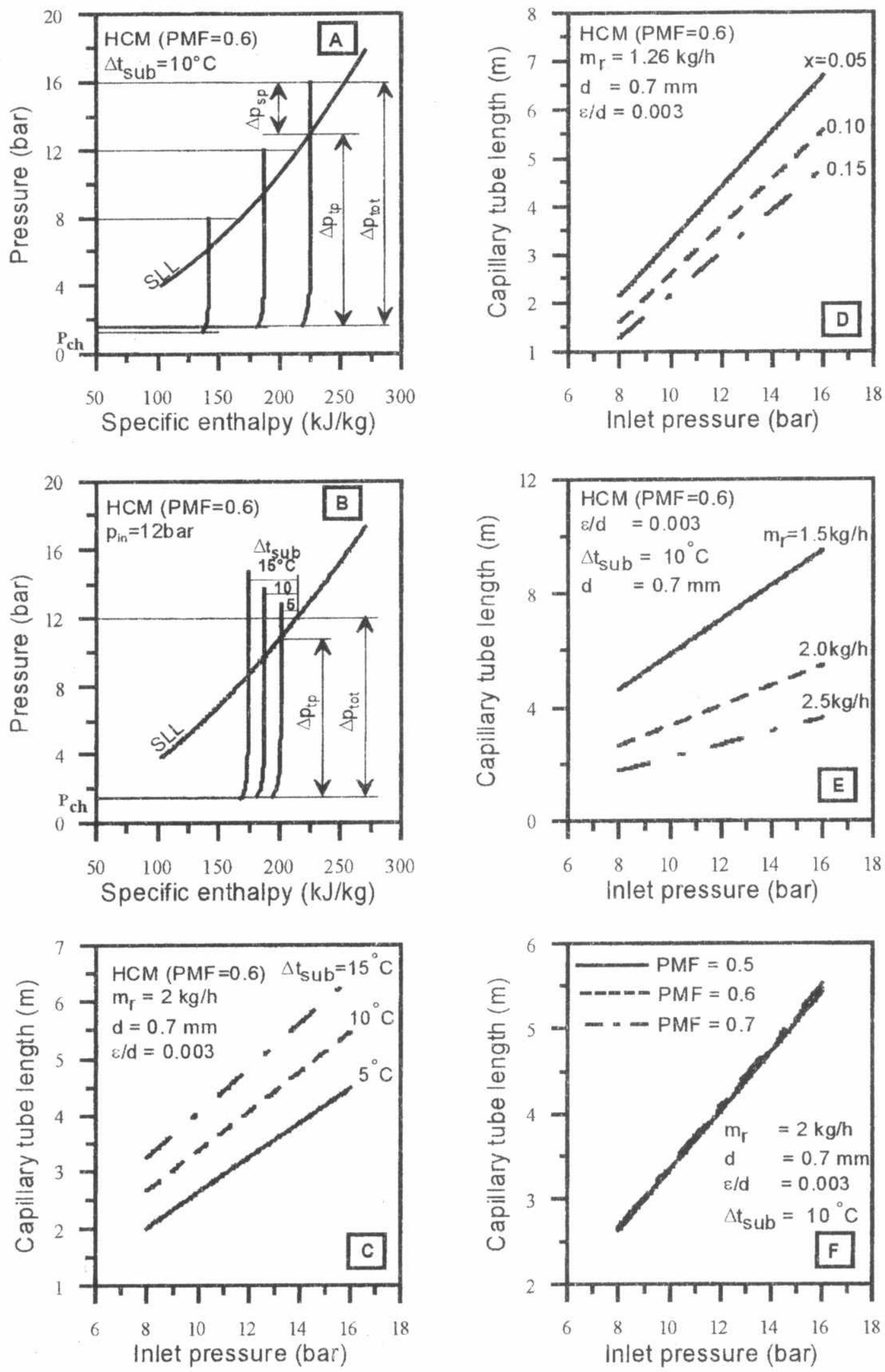


Fig. (4) Effect of operating parameters on capillary tube length

Where L is the capillary tube length (m) and the numerical constants (k_0, k1, k2, k3, k4, k5 and k6) in Eq. (21) are given in Table 1 along with multiple correlation factors. Equation (21) is valid for capillary tube inner diameter (0.6-0.8mm), refrigerant mass rate (1.5-2.5 kg/h), inlet pressure (8-16bar), tube surface roughness (0.0014-0.0024mm), inlet sub-cooling degrees (0-15°C), inlet quality (5-15%) and propane mass fraction (0.5-0.7). It may be concluded, based on the power of each term in equation (21), that the capillary tube length is highly sensitive to capillary tube diameter followed by mass flow rate, inlet pressure, sub-cooling degrees or quality and surface roughness in that order. It should be noted that the effects of inlet pressure and surface roughness are more pronounce in two phase region compared to subcooled flow region.

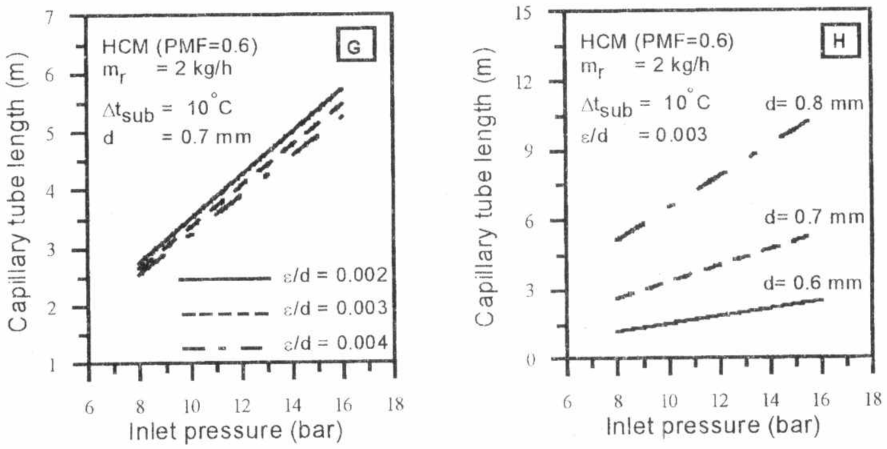


Fig. (4) Effect of operating parameters on capillary tube length (Cont.)

Table 1 Numerical constants for capillary tube length, Eq. (21)

Inlet conditions	Constants							
	k_0	k1	k2	k3	k4	k5	k6	CF
Subcooled	1.5125	5.124	-1.900	1.050	-0.115	0.366	0.0	0.999
Two-phase	0.1536	5.214	-1.979	1.703	-0.147	0.0	-0.124	0.988

3.4 Rating Charts for HCM

The capillary tube size (inner diameter and length) must be selected for drop-in replacements for ozone depleting substances and for new designs of domestic refrigerators/freezers to obtain the balance point between the compressor and the capillary tube. In order to facilitate a selction of appropriate capillary tube inner diameter and length for domestic refrigerators/freezers, rating charts for the hydrocarbon mixture (HCM) of propane, iso-butane and n-butane with propane mass fraction of 0.6 are developed based on the obtained results from the present model for steady adibatic flow through capillary tube and chocked flow conditions. Such those charts, which are similar to those charts reported by ASHRAE [17] for R134a are shown in Fig. (5).

Figure (5.A) illustrates standard mass flow rate (m_{std}) of HCM with propane mass fractio of 0.6 through a reference capillary tube of 0.8 mm inner diameter and 3.3m long as a function of inlet pressure for different inlet conditions such as sub-cooled, saturtaed and two-phase flows. Figure (5.B) presents a correction (flow) factor as a function of capillary tube geometry (d and L) for sub-cooled flow conditions. The flow factor (φ) is defined as the ratio of actual (m_{act}) to standard (m_{std}) mass flow rates.

These rating charts can be used to determine either capillary tube size (inner diameter and length) or mass flow rate of HCM with propane mass fraction of 0.5-0.7. If the capillary tube inner diameter and length are the same as those of the reference capillary tube, the refrigerant flow rate through it will be the standard refrigerant mass flow rate which can be determined from Fig. (5.A) using the upstream conditions (inlet pressure and sub-cooling degrees or quality). On the other side, if the capillary tube size is different from the reference capillary tube, actual refrigerant flow rate can be calculated as follows;

$$m_{act} = \varphi \, m_{std} \qquad (22)$$

where m_{std} is the standard mass flow rate (Fig. 5.A) and φ is the flow factor (Fig. 5.B). It is interesting to note that the ratio of two-phase flow factor to that of sub-cooled liquid is about 0.97 for HCM of propane, iso-butane and n-butane. Thus, Fig (5.B) can be used also for two phase-flow.

In order to select the inner diameter and length of capillary tube for a given refrigerant mass flow rate, the standard mass flow rate has to be determined from Fig. (5.A) by knowing the inlet pressure and the subcooling degree. Then, flow factor ($\varphi = m_{act} / m_{std}$) has to be calculated. As a final point, capillary tube geometry (inner diameter and length) can be selected from Fig. (5.B) according to the value of flow factor.

3.5 Comparison between Different Refrigerants

Comparison between R12, R134a and a hydrocarbon micture (HCM) of propane, iso-butane and n-butane with propane mass fraction of 0.6 is made over wide ranges of operating parameters namely; mass rate (1-5 kg/h), capillary tube inner diameter (0.6-0.8 mm), sub-cooling degree (0-15°C), inlet vapor quality (5 to 15%) and inlet pressure (8-16 bar). Samples of this comparison between the considered refrigerans are presented in Fig. (6).

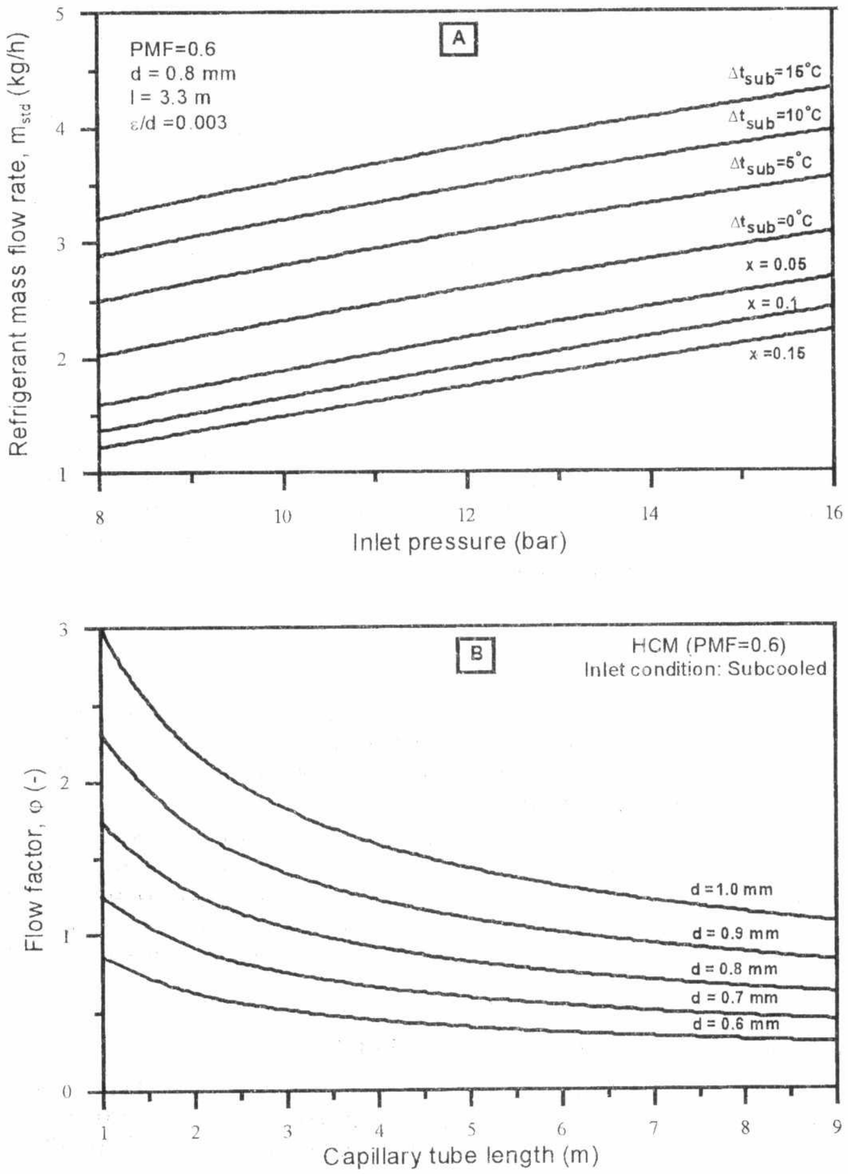
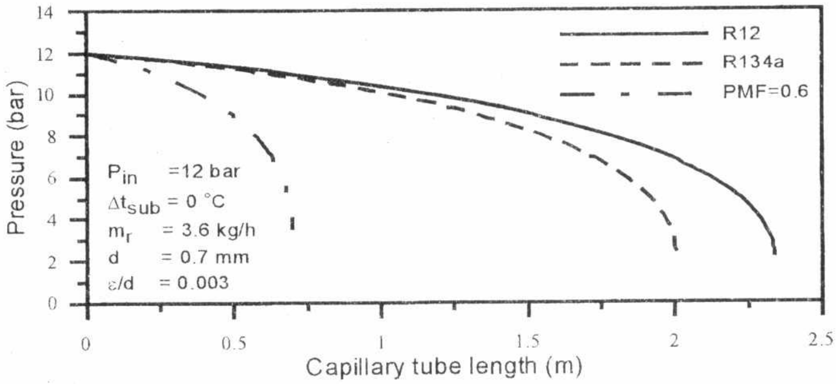


Fig. (5) Rating charts for the hydrocarbon mixture of 60% propane and 40% commercial butane

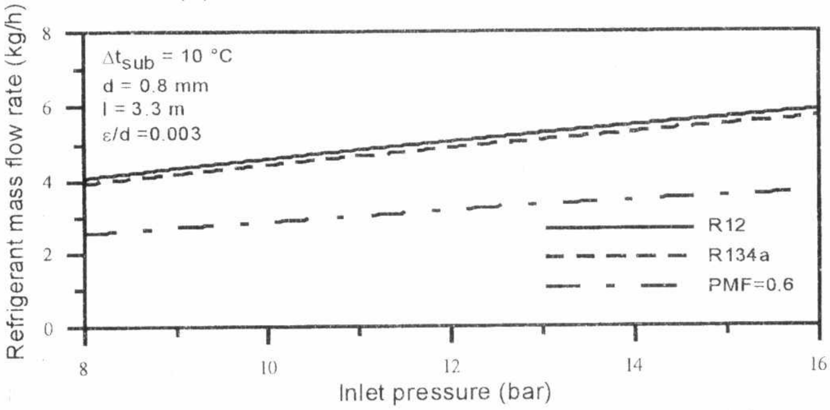
Pressure distribution along the capillary tube up to chock flow conditions for R12, R134a and HCM with propane mass fraction of 0.6 is shown in Fig. (6.A), which indicates that saturated pressure decreases as the capillary tube length increases. Cleraly, rate of decrease in the saturated pressure of HCM is higher than that of R12 and R134a. This is due to the fact that the specific volume of HCM is higher than those of R12 and R134a at any section of the capillary tube. Thus, HCM velocity along the capillary tube is higher than those of R12 and R134a. This leads to increase in Mach number of HCM, thereby its chock flow occurs earlier than those of R12 and R134a for given mass flow rate and inner diamter of the capillary tube. Figure (6.A) reveals that chock pressures of 2.2, 2.4 and 3.6bar occur at caipllary tube lengths of 2.34, 2.00 and 0.70m for R12, R134a and HCM, respectively. Hence, pressure drop per unit length is 4.13, 5.0 and 12.0 bar/m for R12, R134a and HCM, respectively. Thus, the mixture of propane, iso-butane and n-butane with propane mass fraction of 0.6 yields higher pressure drop per unit length compared to either R12 or R134a for a given mass flow rate. As a result, to maintain the same pressure drop (Δp), the mass flow rate has to be reduced significantly when the hydrocarbon blend is used.

Figures (6.B and 6.C) show variations of standard mass flow rate (m_{std}) of R12, R134a and HCM through the reference capillary tube (0.8 mm inner diameter and 3.3m long) with inlet pressure under chock conditions for inlet subcooled and two phase conditions, respectively. Clearly, the blend of propane, iso-butane and n-butane with propane mass fraction of 0.6 yields the lowest refrigerant mass flow rate for both flow conditions. Figure (6.B) reveals that average mass flow rate of HCM is about 0.62 for R12 and 0.67 for R134a.

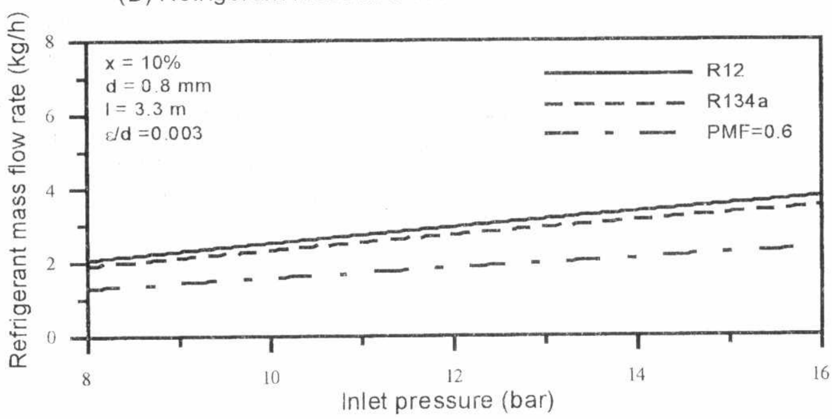
Actual refrigerant mass flow rate can be calcutated based on the compressor volumetric effeicincy and compressor specifications [18] such as stroke volume (6.64cm^3) and speed (2900 rpm). Considering the condensation and evaporation pressures and sub-cooling degree are 14 bar, 2 bar and 10°C, respectively. Consequently, the standard mass flow rate for inlet pressure of 14bar and sub-cooling degrees of 10°C is nearly 5.508, 5.326 and 3.472 kg/h while actual mass flow rate through the compressor is approximately 6.001, 5.018 and 2.345 kg/h for R12, R134a and HCM, respectively. This gives a correction factor of 1.09, 0.94 and 0.68 for R12, R134a and HCM, respectively. Hence, for capillary tube inner diameter of 0.7mm, the required capillary tube lengths are about 2.2, 2.5 and 3.25m for R12, R134a and HCM, respectively. Average capillary tube length for HCM of 0.6 propane mass fraction is longer than those of R134a and R12 by about 30% and 48%, respectively.



(A) Pressure distribution along capillary tube



(B) Refrigerant mass flow rate for inlet subcooled condition



(C) Refrigerant mass flow rate for inlet two phase condition

Fig. (6) Comparison between R12, R134a and HCM of PMF=0.6

4. CONCLUSION

In the present work, a theoretical investigation is carried out to size a capillary tube of a domestic refrigerator working with a hydrocarbon mixture (HCM), consists of propane, iso-butane and n-butane. The considered parameters include refrigerant mass flow rate, capillary tube inner diameter, relative roughness, inlet sub-cooling degrees, inlet vapor quality, propane mass fraction (PMF), inlet pressure and refrigerant type. Based on the simulation results of HCM, the following conclusions are presented:

- Average pressure drops per unit length are nearly 1.28, 1.31 and 1.35 bar/m whereas the length of subcooled part is about 0.678, 0.581 and 0.530 of the capillary tube length for inlet pressures of 8, 12 and 16, respectievely.
- Capillary tube length increases either as inlet pressure, subcooling degrees, propane mass fraction and capillary tube inner diameter increase or as dryness fraction, tube surface roughnous and mass flow rate decrease.
- Capillary tube length at inlet pressure of 16 bar is about double that at inlet pressure of 8 bar for various subcooling degrees.
- Average lengths of capillary tube with Δt_{sub} of 15°C and 10°C are higher than that at Δt_{sub} of 5°C by about 51.7% and 26.7%, respectively
- Ratios of the average capillary tube length with inlet quality of 10% and 15% to that one with inlet quality of 5% are about 80% and 67%, respectively.
- Average capillay tube length of mass flow rate of 1.5kg/h is longer than those of mass flow rate of 2.0 and 2.5 kg/h by about 24% and 62.1%, respectively.
- Average capillary tube length for PMF of 0.5 is nearly 1% shorter while that length for PMF of 0.7 is about 1.7% longer than that of PMF of 0.6.
- Relative roughness of 0.003 and 0.004 yield shorter average capillary tube length by about 4.2% and 9.5% compared to that of 0.002.
- Average capillary tube lengths for inner diameters of 0.7 and 0.8 mm are more than two and thre folds, respectively compared to that for inner diameter of 0.6mm.
- Capillary tube length correlaion for the blend of propane, iso-butane and n-butane with propane mass fraction of 0.5-0.7 has been developed for inlet sub-cooling and two-phase conditions.
- Rating charts for the blend of propane, iso-butane and n-butane valid for propane mass fraction of 0.5-0.7 are developed for sizing the capillar tube.

However, comparison between R12, R134a and the hydrocarbon mixture (HCM) of propane, iso-butane and n-butane with propane mass fraction of 0.6 reveals that;

- HCM of propane, iso-butane and n-butane with PMF of 0.5-0.6 yields higher pressure drop per unit length compared to either R12 or R134a for a given mass flow rate.
- Velocity of HCM along the capillary tube is larger compared to those of R12 and R134a.
- Chock flow of HCM occurs faster than that of R12 or R134a for given mass flow rate and inner diamter of the capillary tube.
- HCM requires low refrigerant mass flow rate and long capillary tube compared to those for R12 and R134a.

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