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Improved Refrigerant Characteristics Flow Predictions in Adiabatic Capillary Tube

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Abstract: This study presents improved refrigerant characteristics flow predictions using homogenous flow model in adiabatic capillary tube, used in small vapor compression refrigeration system. The model is based on fundamental equations of mass, momentum and energy. In order to improve the flow predictions, the inception of vaporization in the capillary tube is determined by evaluating initial vapor quality using enthalpy equation of refrigerant at saturation point and the inlet entrance effect of the capillary tube is also accounted for. Comparing this model with experimental data from open literature showed a reasonable agreement. Further comparison of this new model with earlier model of Bansal showed that the present model could be use to improve the performance predictions of refrigerant flow in adiabatic capillary tube.

Key words: Compression, flow, homogenous, model, refrigeration system, two-phase

INTRODUCTION

Capillary tube is a low capacity refrigeration equipment use exclusively in small vapor compression refrigeration and window air-conditioning system for expansion of refrigerant between outlet of condenser and inlet of evaporator. It is usually made from hollow pipe of drawn copper with inner bore between 0.0005 to 0.002 m and length between 1.5 to 6 m. Capillary tube is simple in construction, cheap in cost, no maintenance required and the starting torque of the compressor is reduced. Though, the geometry of the tube is simple, but, the analysis of the refrigerant flow is complex because of the change in phase from liquid single-phase to two-phase liquid/vapor as the refrigerant flows along the tube.

Literature showed that there are three common types of numerical models been used to study the behavior of refrigerant flow in adiabatic capillary tube - two-phase homogenous flow, two-phase separated flow and two-phase drift flux models. However, most of these researches used two-phase homogenous flow model (Bansal and Rupasinghe, 1998; Kritsadathikarn *et al*., 2002; Liang and Wong, 2001; Sami and Tribes, 1998; Wongwises and Pirompak, 2001; Wongwises and Suchatawut, 2003). Bansal and Rupasinghe (1998) developed a mathematical model using conservation equations of mass, momentum and energy. The equations were solved simultaneously using iterative step and Simpson's rule. A numerical model based on homogenous model was also proposed by Sami and Tribes (1998). The model was tested under different input conditions using refrigerant R-12, R-22 and its alternatives and result shows good agreement with measured data. Wongwises and Suchatawut (2003) developed a model to study pressure distribution along the capillary tube with some conventional refrigerants and their alternatives. The equations were solved using Runge-kutta method.

Even though, the slip velocity and drift flux velocity as it applies to separated and drift flux models respectively are not considered in homogenous model, Escanes *et al*. (1995) and Bittle and Pate (1996) regard it (homogenous model) as a good numerical estimator for flow of refrigerant in capillary tubes. As a result, up to date, many researchers (Fatouh, 2007; Kim *et al*., 2011; Shodiya *et al*., 2011; Trisaksri and Wongwises, 2003) use homogenous model for simulating refrigerant flow in adiabatic capillary tubes. Since the model is widely accepted, there is need for improvement of the model in order to have better refrigerant flow predictions. Most of the earlier models determined their inception of vaporization (two-phase) by selecting their initial enthalpy directly from the standard thermodynamic table (REFPROP, 2007). The objective of this study is to improve upon the refrigerant flow characteristics

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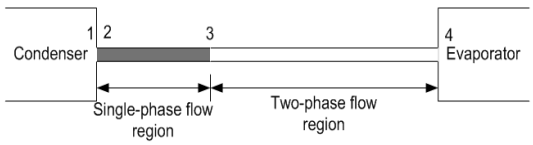

Fig. 1: Schematic diagram of capillary tube between condenser and evaporator

predictions in adiabatic capillary tube by determining the vaporization inception at saturation point using enthalpy equation. The capillary tube entrance correction is also included.

MATHEMATICAL MODELLING

Figure 1 shows capillary tube which is connected between outlet condenser and inlet evaporator. The flow in the tube is modeled by dividing it into liquid single phase and two-phase liquid-vapor section. There is a linear pressure decrease in the single-phase while pressure drop per unit length increases as the length increases in the two-phase.

The flow in the tube is based on one dimensional two-phase homogenous flow assumptions. The present model is also based on that of Wongwises and Chingulpitak (2010) with modifications. The governing equations that were used to describe the single-phase and the two-phase are discussed in the following subsection.

Single-phase (liquid) flow region: The single phase liquid region is the region from entrance of the capillary tube to where the pressure decreased to saturated pressure. Energy equation from point 2 to 3 (Fig. 1) is given in Eq. (1):

$$\frac{p_2}{\rho_2 g} + \frac{V_2^2}{2g} + Z_2 = \frac{p_3}{\rho_3 g} + \frac{V_3^2}{2g} Z_3 + h_{IT} \tag{1}$$

where, p_2 and p_3 are pressures, V_2 and V_3 are velocities, Z_2 and Z_3 are horizontal height ρ_2 and ρ_3 are densities at points 2 and 3 respectively and g is the acceleration due to gravity.

The total head loss (h_{IT}) is shown in Eq. (2):

$$h_{IT} = f_{sp} \frac{L_{sp} V^2}{D 2g} + k \frac{V^2}{2g} \tag{2}$$

where f_{sp} is friction factor of single phase, L_{sp} is single-phase length, D is inner diameter of capillary tube, k is coefficient of entrance loss and its value is 1.5 as given by Zhou and Zhang (2006).

The continuity equation for an incompressible fluid is given in Eq. (3):

$$m = \rho_2 V_2 A_2 = \rho_3 V_3 A_3 = \rho V A \tag{3}$$

where m is the mass flow rate, A_2 and A_3 are cross sectional area at points 2 and 3 respectively.

The pressure drop as a result of the sharp entrance of the refrigerant into the capillary tube is determined from Eq. (4) (Chung, 1998):

$$P_2 - P_1 = \frac{G^2 \upsilon_f}{2} \left[\left(\frac{1}{C_c} - 1 \right)^2 + \left(1 - \frac{A^2}{A_1^2} \right) \right] \left[1 + \left(\frac{\upsilon_{fg}}{\upsilon_f} \right) x \right] \tag{4}$$

where G is mass flow per unit area, υ_f and υ_{fg} are specific volumes of liquid and liquid/vapor phases respectively, x is the vapor quality and coefficient of contraction C_c is a function of the area ratio A_2/A_1given in Eq. (5):

$$C_c = 0.544 \left(\frac{A_2}{A_1} \right)^3 - 0.242 \left(\frac{A_2}{A_1} \right)^2 + 0.111 \left(\frac{A_2}{A_1} \right) + 0.585 \tag{5}$$

Since the tube is horizontal, $Z_2 = Z_3$ and G $=\rho V$. Combining Eq. (1), (2) and (3), the length of single-phase, L_{sp} is calculated from Eq. (6):

$$L_{sp} = \left[(p_1 - p_3) \frac{2\rho}{G^2} - k - 1 \right] \frac{D}{f_{sp}} \tag{6}$$

The saturated pressure at point 3, p_3, is determined by using the level of subcooling, ΔT_{sub} at tube inlet i.e., $T_3 = T_1 - \Delta T_{sub}$

where T_1 and T_3 are temperatures at points 1 and 3, respectively.

Friction factor of single-phase f_{sp} is determined using Colebrook equation, given below:

$$\frac{1}{\sqrt{f_{sp}}} = 1.14 - 2 \log \left[\frac{\varepsilon}{D} + \frac{9.3}{\mathrm{Re} \sqrt{f_{sp}}} \right] \tag{7}$$

where,

$$\mathrm{Re} = \frac{\rho V D}{\mu} \tag{8}$$

ε and μ are wall roughness and refrigerant viscosity, respectively.

Two-phase (liquid/vapor) region: The flow is treated as two-phase homogeneous flow and the three conservation equations are applied. The conservation of mass equation can be expressed as:

$$m = \frac{V_3 A}{\upsilon_3} = \frac{V_4 A}{\upsilon_4} \tag{9}$$

Considering the steady state adiabatic and ignoring the difference in elevation, the equation of conservation of energy is written as:

$$h + \frac{V^2}{2} = \text{Constant} \tag{10}$$

where, h is the enthalpy.

As the refrigerant flows in the capillary tube the pressure drops and there is an increase in vapor quality, x. At any point:

$$h = h_f (1 - x) + h_g x \tag{11}$$

$$\upsilon = \upsilon_f (1 - x) + \upsilon_g x \tag{12}$$

From Eq. (3):

$$V = \frac{m}{\rho A} = \frac{G}{\rho} \tag{13}$$

By substituting Eq. (11), (12) and (13) into (10), expanding it and rearranging gives:

$$h_3 + \frac{V_3^2}{2} = h_f + x(h_g - h_f) + \frac{G^2 \upsilon_f^2}{2} + x^2 (\upsilon_g - \upsilon_f)^2 \frac{G^2}{2} + G^2 \upsilon_f (\upsilon_g - \upsilon_f)^2 x \tag{14}$$

The vapor quality, x, can now be expressed as:

$$x = \frac{-h_{fg} - G^2 \upsilon_f \upsilon_{fg} + \sqrt{(h_{fg} + G^2 \upsilon_f \upsilon_{fg})^2 - \left(2G^2 \upsilon_{fg}^2\right)\left(h_f + \frac{G^2 \upsilon_f^2}{2} - \frac{V_3^2}{2} - h_3\right)}}{G^2 \upsilon_{fg}^2} \tag{15}$$

where,

$$h_{fg} = h_g - h_f \text{ and } \upsilon_{fg} = \upsilon_g - \upsilon_f$$

In this study, since a finite amount of superheat is required for the inception of vaporization (formation of first bubble), the first vapor quality x, is evaluated by calculating enthalpy h_3 at saturated point using the enthalpy formulation in Eq. (16) and substituting in Eq. (15), thus:

$$h_3 = u_3 + p_3 \upsilon_3 \tag{16}$$

The values of refrigerant properties including u_3 (internal energy) and v_3 are taken from REFPROP (2007).

Considering an elemental fluid, the force applied as a result of inner tube shear force and difference in pressure difference on opposite ends on an element are equal to the time rate of change in linear momentum. The conservation of linear momentum is written as:

$$PA - (P + dP)A - \tau_w \pi D dL = m dV \tag{17}$$

where, τ_w is the shear stress at the wall defined as:

$$\tau_w = \frac{f_{tp} \rho V^2}{8} \tag{18}$$

Substituting Eq. (18) into (17) and assuming constant mass flow rate, gives:

$$dL = \frac{2D}{f_{tp}} \left[\frac{-\rho dp}{G^2} + \frac{d\rho}{\rho} \right] \tag{19}$$

The friction factor of the two-phase, f_{tp} is calculated using Lin *et al*.(1991) formulation as:

$$f_{tp} = \varnothing^2 f_{sp} \left(\frac{\upsilon_{sp}}{\upsilon_{tp}} \right) \tag{20}$$

where, $\varnothing^2$, the multiplier is given by:

$$\varnothing^2 = \left\{ \frac{\left(\frac{8}{\mathrm{Re}_{tp}}\right)^{12} + \frac{1}{(A_{tp} + B_{tp})^{1.5}}}{\left(\frac{8}{\mathrm{Re}_{sp}}\right)^{12} + \frac{1}{(A_{sp} + B_{sp})^{1.5}}} \right\}^{\frac{1}{8}} \left[1 + x \left(\frac{\upsilon_g}{\upsilon_2} - 1 \right) \right] \tag{21}$$

where,

$$A_{sp} = \left\{ 2.457 \ln \left[\frac{1}{\left(\frac{7}{\mathrm{Re}_{sp}}\right)^{0.9} + 0.27\varepsilon} \right] \right\}^{16} \tag{22}$$

$$B_{sp} = \left(\frac{37530}{\mathrm{Re}_{sp}} \right)^{16} ; \mathrm{Re}_{sp} = \frac{mD}{\mu_{sp} A}$$

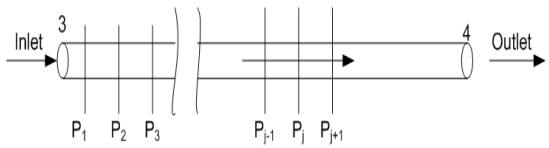

Fig. 2: The grids of the numeric solution

A_{tp}, B_{tp} and Re_{tp} is calculated in a similar way as Eq. (22).

The viscosity model used in calculating the two-phase viscosity μ_{tp} is given by Cicchitti *et al*. (1960) as follow:

$$\mu_{tp} = x_{\mu g} + (1 - x)\, \mu_f \tag{23}$$

SOLUTION METHODOLOGY

A computer program written in MATLAB (2009b) version 7.02 was used to solve the governing equations for the present two-phase homogenous flow model. The capillary tube in the two-phase is divided into several sections as shown in Fig. 2 (between points 3 and 4). To calculate the pressure P_j, the vapor quality at that point must be known. The initial enthalpy h_3 is calculated using Eq. (16), which is substituted in Eq. (15) to evaluate the first vapor quality, x. With the vapor quality known, the first value of μ_{tp}, Re_{tp}, A_{tp}, B_{tp} and f_{tp} are calculated. The pressure, P_j in the capillary tube is now evaluated from Eq. (24):

$$P_j = P_3 - j\Delta P \tag{24}$$

Subsequent vapor qualities are calculated by evaluating the enthalpy h, using Eq. (11) and substituting in Eq. (15). The entropies in these sections are express as:

$$s_j = s_{jf}\,(1 - x) + s_{jg}\, x \tag{25}$$

In each elemental point, P_j, T_j, x_j, s_j and f_{tpj} are computed. Entropy must increase as the refrigerant flows through the capillary tube. To make sure that the entropy increases, the computed entropy s_j, is compared with the earlier one s_{j-1}. The evaluation is done element by element along the capillary tube until a region where the entropy reached maximum, that is, critical flow condition achieved. The pressure, $(P_j)_{max.}$, where the entropy attained maximum, is compared with pressure of evaporator, P_{evap}. If $(P_j)_{max}$ is greater than P_{evap}. , the pressure at point 4, P_4, is taken as $(P_j)_{max}$ and it is use for the computation. But, if pressure, $(P_j)_{max,}$ is lower than pressure, P_{evap}., the pressure at point 4 is taken as P_{evap}.

To calculate the two-phase length, Eq. (19) was discretized so that each elemental length is calculated thus:

$$\Delta L_j = \frac{2D}{f_{tpj}}\left[\frac{-\rho \Delta P}{G^2} + \frac{\Delta \rho}{\rho_j}\right] \tag{26}$$

Total two-phase length is evaluated thus:

$$L_{tp} = \sum_{j=1}^{n} \Delta L_j \tag{27}$$

Total capillary tube length, L, is evaluated thus:

$$L = L_{sp} + L_{tp} \tag{28}$$

All thermophysical and thermodynamics properties are from REFPROP (2007) a computer program, version 8.0 which is created based on pressure function.

RESULTS AND DISCUSSION

This section discusses the validity of the model with experimental data of Wijaya (1992) and Fiorelli *et al*. (2002). Figure 3 shows the comparison and relations between mass flow rate and tube length for D = 0.84 mm and subcooling, ΔT_{sub} = 16.7 K, for different condensing temperatures. The Figure shows clearly that the mass flow rate decreases with decrease in temperature of condenser and also decreases with increase in tube length. The decrease in mass flow rate as the tube length is increasing could be attributed to increase in resistance to the refrigerant flow. This suggests that increasing the capillary tube length will definitely raise the pressure drop. In order to retain the pressure drop, the mass flow rate must be brought down. The predicted results have an error within ±6% compare with measured data of Wijaya (1992).

Figure 4 shows the comparison of the mass flow rate with the tube length for subcooling between 5.5 to 16.7 K and diameter of 0.84 mm. As it can be seen in the Figure, for the subcooling of 5.5 K, the new model result over-predicts the Wijaya (1992) result having a percentage of about 8% and also, the subcooling of 16.7 K under-predict

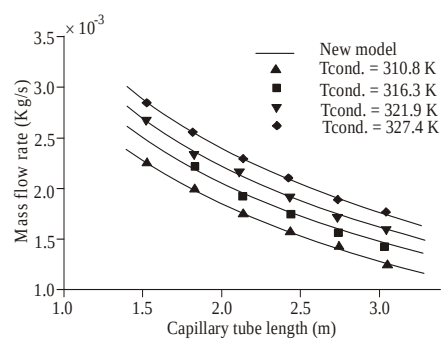

Fig. 3: Mass flow rate against capillary tube length with different condensing temperature

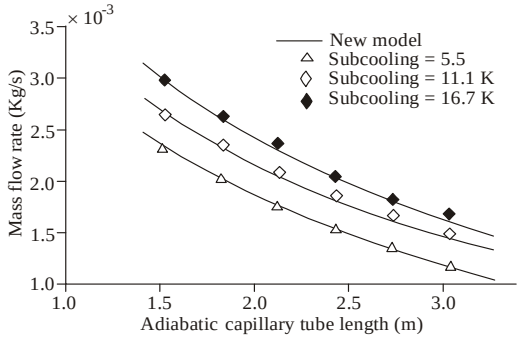

Fig. 4: Mass flow rate against capillary tube length with different subcooling

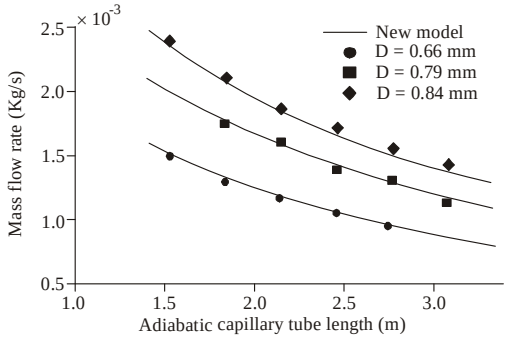

Fig. 5: Mass flow rate against capillary tube length with different diameters

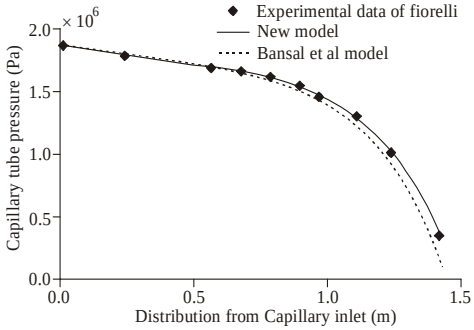

Fig. 6: Comparison of new model results and experimental data of Fiorelli with earlier model of Bansal

the experimental results by about 7.5%. In general, there is an increase in mass flow rate as the subcooling increases. This can be attributed to the fact that the increased subcooling increases the length of the liquid phase refrigerant and this liquid refrigerant offers less resistance to refrigerant flow compare to the two-phase liquid/vapor.

Figure 5 shows the relationship between the new model result and the measured data of Wijaya (1992) for mass flow rate against capillary tube length with different diameters. For D = 0.66 mm the percentage error was between +2 and +5%. For D = 0.79 mm, the new model results predicts the experimental results between -5 and +2% and D = 0.84 mm was between -4 and +2%. In general, the mass flow rate increases with increase in inner diameter of the capillary tube which can be attributed to increase in flow capacity of the tube. This also leads to decrease in pressure drop as the diameter increases. For desired operating conditions, the rise in pressure drop needs relatively reduced mass flow rate.

Figure 6 shows the pressure distribution along the capillary tube and comparing the new model result with the experimental data of Fiorelli *et al*. (2002) and also with the earlier model of Bansal and Rupasinghe (1998). As can be seen in the Figure, the pressure decreases linearly from 0 to 0.75 m (single-phase), thereafter, the inception of the two-phase give rise to pressure drops to increase. The error between the new model result and experimental data of Fiorelli *et al*. (2002) was about ±5%. The error between new model result and Bansal and Rupasinghe (1998) model result was about -2 to +7%.

CONCLUSION

This study presents a modified two-phase homogeneous flow model used to improve the predictions of the effects of the design parameters of adiabatic capillary tube use in small vapor compression refrigerating and window air conditioning system. In an attempt to solving the problem of determining the position of inception of vaporization and evaluating the initial two-phase friction factor, the enthalpy formulation was used. When the new model was compared with the experimental data of Wijaya (1992) and Fiorelli *et al*. (2002), it was found that the present modified homogenous flow model gave reasonable agreement. It was also compared with the previous homogeneous model of Bansal and Rupasinghe(1998). With this present model, it can be concluded that the characteristics flow predictions of refrigerants in adiabatic capillary tube can be enhanced, hence, improved design for capillary tube designers.

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