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A STUDY ON CAPILLARY TUBE FLOW

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ABSTRACT

This paper studies the effects of various design parameters on the performance of the capillary tube. A homogeneous two phase flow model was used to illustrate various effects of the design parameters such as: tube diameter, tube roughness, tube length, degrees of sub-cooling and the refrigerant flow rates on capillary tube performance. The validation of the prediction is confirmed by comparing predicted pressure distribution and critical mass flow with available measured values.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Dimension</th>
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<tbody>
<tr>
<td>A</td>
<td>Area</td>
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<tr>
<td>C</td>
<td>sonic velocity</td>
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</tr>
<tr>
<td>d</td>
<td>nominal pipeline diameter</td>
<td>m</td>
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<tr>
<td>e</td>
<td>relative surface roughness</td>
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<tr>
<td>f</td>
<td>friction factor</td>
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<tr>
<td>G</td>
<td>mass velocity</td>
<td>kJ/kg</td>
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<tr>
<td>h</td>
<td>specific enthalpy</td>
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<tr>
<td>M</td>
<td>mass flow rate</td>
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<tr>
<td>P</td>
<td>pressure</td>
<td>N/m²</td>
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<tr>
<td>Uₘ</td>
<td>mixture velocity</td>
<td>m/s</td>
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<tr>
<td>x</td>
<td>mass dryness fraction (quality)</td>
<td></td>
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<tr>
<td>Z</td>
<td>axial direction or length</td>
<td>m</td>
</tr>
<tr>
<td>(dP/dZ)ₘ</td>
<td>pressure gradient due to wall friction</td>
<td>kN/m³</td>
</tr>
<tr>
<td>(dP/dZ)ₚ</td>
<td>total pressure gradient</td>
<td>kN/m³</td>
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INTRODUCTION

The capillary tube is one of the most important component in a vapour compression refrigeration system. It is specially used in small systems with capacity below 10kW. Although it is the simplest device in the refrigeration cycle, the flow phenomena involved is not simple. Two phase refrigerant vapour and liquid usually exist in the tube and make the study more complicated. In the design of refrigeration systems, an incorrect matching of the capillary tube with the compressor and evaporator may penalise the performance of the whole system. Various adiabatic capillary tube models are available in the open literature. The flow parameters such as pressure drop, mass flow rate, critical mass flow rate and the delay of vapourisation were studied by numerous researchers.

In studies of such systems, the refrigerant flow in the capillary tube is generally divided into a single phase sub-cooled liquid region and a two phase liquid-vapour flow region. Visual and photographic studies on the two phase flow patterns on the capillary tube by Mikol [1], Mikol and Dudley [2] and Koizumi and Yokoyama [3] show that the flow condition can be assumed as homogeneous. To model the capillary tube flow, homogeneous flow assumption were made by Goldstein [4], Koizumi and Yokoyama [3] and Melo [5].
Recent studies on the effects of metastable flow on the capillary tube performance was carried out by Li et al [6,7] and Chen [8]. The model includes the effects of the thermodynamic non-equilibrium vaporization and relative velocity between the liquid and vapour phase.

To study on the effects of new refrigerant, Wijaya [9] presented experimental test data of an adiabatic capillary flow using HFC-134a as the working fluid.

Previous study on the two phase flow by the author [10] showed that the homogeneous flow model worked reasonable well for pipe flow of a refrigerant R113 liquid-vapour system. This paper attempts to use a simple homogeneous two phase flow model [10] to study the effects of the design parameters such as, the tube diameter, the tube roughness, the tube length, the degrees of sub-cooling on the performance of the capillary tube.

THEORETICAL MODELLING

In present analysis, the flow in the capillary tube is divided into a sub-cooled single phase liquid and saturated two phase flow regions. In the two phase flow region the flow is modelled by assuming one dimensional, adiabatic and thermodynamic equilibrium homogeneous flow. Liquid flashing arising purely from the reducing pressure. The unknown flow parameters in the model are quality, velocity, pressure and void fraction. It is important to mention that metastable flow phenomena are neglected in the model. The governing equations used in describing the flow are presented below:

Single phase flow region

For a horizontal, sub-cooled single phase flow in tube, from the conservation of momentum, the pressure drop along the tube may be expressed as:

$$\left( \frac{dP}{dZ} \right)_p = -\frac{f G^2}{2d} \nu_p$$

(1)

the friction factor $f$ can be obtained from Colebrook single phase friction factor curve,

$$\frac{1}{\sqrt{f}} = 1.14 - 2.0 \log \left( \frac{\epsilon / d}{3.7} + \frac{9.3}{Re \sqrt{f}} \right)$$

(2)

As the sub-cooled liquid refrigerant flows along the tube, the fluid pressure drops along the line, hence the saturated temperature of the liquid decreases. The length of the single phase flow region can be determined when the saturated temperature of the liquid approaches the initial sub-cooled liquid temperature.

Two phase flow region

As the saturated temperature reaches the initial sub-cooled temperature of the liquid, two phase flow situation occurred. The two phase flow is assumed a homogeneous mixture in this case. The mixture velocity of the vapour-liquid refrigerant may be obtained from the consideration of mass conservation. It can be expressed as:

$$U_m = \frac{M}{A} \nu_m = \frac{M}{A} [x \nu_x + (1-x) \nu_f]$$

(3)

The change of quality of the fluid along the tube may be derived from the conservation of energy, see eqn (4). The two-phase pressure drop along the tube can be expressed as the sum of the pressure drop due to tube wall friction as well as the fluid acceleration effects as shown in eqn (5).

$$\frac{dx}{dZ} = \frac{\left( \frac{dP}{dZ} \right)_p}{B} \frac{A}{D - A} C \frac{\partial x}{\partial z}$$

(4)

and

$$\left( \frac{dP}{dZ} \right)_f = \frac{\left( \frac{dP}{dZ} \right)_p}{D} C \frac{\partial x}{\partial z}$$

(5)

where

$$A = x \frac{d \nu_x}{dP} + (1-x) \frac{d \nu_f}{dP} + \frac{G \nu_m}{dP} \left( x \frac{d \nu_x}{dP} + (1-x) \frac{d \nu_f}{dP} \right)$$

$$B = h_x + G \nu_m \nu_{sf}$$

$$C = G \nu_{sf}$$
\[ D = I + G \left[ x \frac{d\nu_s}{dP} + (1 - x) \frac{d\nu_f}{dP} \right] \]

\[ \left( \frac{dP}{dZ} \right)_f = -\frac{G^2}{2d} [\nu_m] \]

\[ h_v = h_i - h_f \quad \nu_v = \nu_i - \nu_f \]

Notice that in the two phase flow region, the single phase Moody's friction factor expressed by Colebrook's equation as shown in eqn (2) is employed, with the Reynolds number being defined as:

\[ Re = \frac{U_m d}{\mu_v \nu_m} \]

and Dukler [11] mixture viscosity is defined as:

\[ \mu_v = \mu_f (1 - \beta) + \mu_v \beta \]

Where

\[ \beta = \frac{x \nu_e}{x \nu_e + (1 - x) \nu_f} \]

**Sonic velocity in homogeneous two phase flow region**

The fluid velocity increases in the flow direction due to pressure drop. A situation will reach such that the fluid velocity reaches the local sonic velocity, and the flow is said to have reached the critical flow condition or choked.

The sonic velocity may be obtained from eqn (5). Choking occurs as the denominator of eqn (5) approaching zero, i.e \((dP/dZ)_f \to 0\), hence

\[ C = \sqrt{\frac{[\nu_f (1 - \beta) + \nu_e \beta]}{\nu_i (1 - \beta) + \nu_e \beta}} \] (6)

In the above analysis, equation (1) is applied to the single phase sub-cooled liquid region. Once the saturated condition is reached, the governing equations (4) and (5) for homogeneous two phase flow are solved using a standard 4th order Runge-Kutta technique with a small length increment. During the computation, equations (3) and (6) are computed to determine if the critical flow condition is reached. The computation terminates when the flow is choked. The tube length corresponds to the choked condition is termed the critical tube length. In the above analysis, the Moody's single phase friction factor with Dukler [11] mixture viscosity expression has been applied in the two phase flow region.

**RESULTS AND DISCUSSIONS**

**Comparison with existing experimental data**

To validate the simulation model, comparisons have been made with available experimental results by Wijaya [9] and Li et al [17].

Fig 1 shows variation of critical mass flow rate against the critical tube length. The predicted results compare reasonable well with measured data [9]. Fig 2 shows that pressure distribution along the tube for both predicted and measured results [7]. Good agreement is shown.

In the following analysis, refrigerant HFC-134a was used as the working fluid. The inlet pressure at the capillary tube entrance is fixed at 13.27 bar, while the exit pressure is computed until the flow reaches the critical flow condition and the computation is terminated.

**Effect of tube diameters.**

Fig 3 shows the temperature distribution along the capillary tube with various tube diameters. It shows a constant temperature distribution in the single phase liquid region and followed by a sudden drop in temperature indicating that the point which vaporization begins. The results show that bigger tube diameters delay the occurrence of the vaporisation.

Fig 4 shows the critical mass flow rates with various tube diameters. As the diameter increases, the critical mass flow rate increases for a given tube length. The results also show that, for a given mass flow rate, increases in the tube diameter increase the tube length. In practice, an optimum diameter is required to provide the necessary restriction for the desired flow condition. However, as depicted in Fig 4 the optimum diameter is affected by the length of the tube available and the mass flow rate.
Effects of degree of sub-cooling

Fig 5 shows the variation of the critical mass flow rate with various degrees of sub-cooling. The results show that for a given tube length, a higher degree of inlet sub-cooling produces a higher mass flow rate. This is because as the inlet sub-cooling increases, it allows the refrigerant to remain in the single phase liquid over a longer portion of the tube, thus the two phase flow region is shortened. This illustrates that the restriction to the critical mass flow rate depends very much on the length of the two phase flow region which is affected by the degree of sub-cooling. Whereas, at a given critical mass flow rate an increase of inlet sub-cooling increases the critical tube length.

Fig 6 shows the temperature distribution along the capillary tube with various degrees of sub-cooling. The result shows that the higher the degree of sub-cooling the later the onset of vaporisation. Clearly, the single phase liquid region is extended at a higher degree of sub-cooling. Fig 6 can be used to determine the location for the onset of vaporization.

Effects of tube roughness

As shown in Fig 7, the results demonstrate that for a given mass flow rate, a decrease in tube roughness resulted in a longer tube before the choked condition is reached.

Fig 8 shows the variation of quality at the tube exit with various tube roughness. As indicated in the figure, higher tube roughnesses increase the refrigerant quality at the tube exit for a given critical tube length.

CONCLUSIONS

The results reconfirmed that a simple homogeneous flow model may be adequately used to predict the performance of the two phase capillary tube flow. The effects on various parameters are shown. The results also show that, the single phase Moody’s friction factor expressed by Colebrook equation with the Dukler mixture viscosity expression can be used to predict the flow of two phase refrigerant through capillary tube. It is believed that, the model, at its current state, may be used to assist the selection of the capillary tube in refrigeration systems.

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REFERENCES

**Fig. 1** Comparison of predicted and measured critical mass flow rate with tube length

**Fig. 2** Comparison of predicted and measured pressure along the tube

**Fig. 3** Temperature distribution along the tube

**Fig. 4** Mass flow rate at choked condition

**Fig. 5** Mass flow rate at choked condition with various degrees of subcooling (d=0.6mm)

**Fig. 6** Temperature distribution along the tube
Variation of mass flowrate at choked condition

Fig. 7  Mass flow rate at choked condition

Variation of quality at choked condition

Fig. 8  Quality at choked condition