Experimental investigation and numerical simulation of choked refrigerant flow through helical adiabatic capillary tube

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HIGHLIGHTS

• The results confirm average error of about 5.5% for pressure.
• Drift flux model was applied for choked flow in coiled capillary tube.
• Under the same conditions, critical mass flux through helical tube with coil diameter of 40 mm is about 16% less than of straight one.
• The mass flux in steady and choked conditions through coiled capillary tube are equal approximately.
• The present model can be used as a suitable tool for design and optimization VCRC using helical capillary tubes.

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ABSTRACT

This paper presents a drift flux model and experimental study of choked refrigerant flow through both straight and helical adiabatic capillary tubes. The conservation equations of mass, energy, and momentum are solved using the fourth order Runge-Kutta method. This model is validated by previously published experimental data and also by test results performed and presented in this work for R-134a with average error of 5.5%. The effect of capillary tube inner diameter, length, relative roughness and coil diameter, and also various test conditions such as inlet pressure, inlet temperature, and sub-cooling degree of refrigerants are investigated. Critical mass flux variation, pressure distribution and temperature variation are obtained experimentally as well as vapor quality, vapor velocity and void fraction variation by numerical simulation. The results show that mass flux reaches a maximum amount at a specific value of evaporator pressure in choked conditions and also it is decreased by increasing the length of capillary tube. Moreover, critical mass flux increases by increasing of the tube inner diameter, condensation temperature and refrigerant degree of sub-cooling.

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1. Introduction

Capillary tubes are used as refrigerant controlling devices, expansion devices and also as heart of a small vapor compression refrigeration cycle [1–3]. It connects outlet condenser to the inlet evaporator and balances the refrigeration cycle pressure and controls the refrigerant mass flux [1–9]. Because capillary tube has no moving parts, it cannot be adjusted to varying load conditions, resulting in extensive implementations therefore such as simple expansion devices in small household refrigerators and freezers with nearly constant refrigeration load [2,3]. In general, the inner diameter and length of a capillary tube ranges from 0.5 to 2.0 mm, and 2–6 m, respectively [1–8].

In some vapor compression refrigeration cycle applications (VCRC), capillary tubes are coiled to minimize the space [4–8]. The fluid flow in coiled capillary tubes is subjected to the centrifugal force which causes secondary flow effect. Some researchers have referred to Dean effect to describe this secondary flow [1]. Dean Number is defined as De = Re (d/D)0.5 that affects the amount of heat transfer, momentum, and mass flux in both kinds of coiled tubes [1].

Since in a capillary tube, the flow temperature reduces as it flashes into vapor phase, the flow analysis is complicated. Sub-cooled refrigerant enters the capillary tube and flows as a single phase up to the point where the pressure reaches the saturation pressure of refrigerant at flow temperature [1–5]. In a single-phase flow region, pressure gradient is almost constant [1–3]. Although it
is expected that flashing to the vapor phase starts at the end of single-phase region, it happens with some delay at a point where flow pressure is slightly less than its saturation pressure. This region is called metastable flow region [10–12] which is often ignored in numerical simulations. By decreasing the pressure at the end of the tube, mass flow increases up to the “critical condition” in which mass flow remains stable and “choked phenomenon” happens at the end of the capillary tube. At this point, the pressure drops with a sharp gradient, in the other hands, dp/dz approaches to infinity and as a result fluid rapidly flashes to vapor phase [2,3]. Then, velocity of refrigerant flow increase due to increase in specific volume of the fluid until Mach number (local velocity of sound) reaches to 1.0 [2]. The condition dp/dz approaching infinity is a criterion for critical conditions [12–14]. Once the entropy reaches the maximum amount, the refrigerant velocity is equal to the velocity corresponding to Mach number 1.0, and then choked condition is happened [15–17]. Since, refrigeration systems are designed to work with critical condition of their capillary tube, analysis of the flow through these tubes is a vital step in the design and optimization of the refrigeration systems.

Bansal and Wang [2] proposed a homogenous flow model for R134a and R600a choked flow through adiabatic straight capillary tube. This numerical simulation is established based on first law of thermodynamics, empirical relations and a fresh look of fluid mechanics problem, known as “Fanno flow”. The point of negative entropy change is a criterion for choked flow. In addition, their numerical modeling showed that critical mass flow increases as capillary tube diameter, condensation temperature and degree of sub-cooling increases.

Zhou and Zhang [5,8] have conducted numerical simulations and experimental validation for R22 refrigerant flow through coiled capillary tubes. They also used a homogenous two-phase flow model and three different friction factor correlations. They found that the mass flow through a capillary tube with coil diameter of 40 mm is reduced about 10% in comparison with a straight tube for the same test conditions.

Zhou and Zhang [7] have also conducted experimental investigation on the hysteretic effect of the coiled adiabatic capillary tube performance. The reported results indicate that the hysteresis effect decreases, as the coil diameter increases.

An experimental investigation and a homogenous numerical simulation were performed by Park et al. [9] for R22 flow through a straight and coiled capillary tube. They found that under the same test conditions, the mass flow through the coiled capillary tubes is decreased by about 5–16% compared to the straight capillary tubes.

The effects of sub-cooling degree for R22 flow through the coiled capillary tube were studied by Garcia–Valladares et al. [12,13]. They used a separated two-phase flow model for simulation of refrigerant flow through coiled tube. Their results indicated that the mass flux of capillary tube increases with increases in degree of sub-cooling as well as condensation pressure. Moreover, the entropy equations were used for detecting critical flow conditions.

Melo et al. [18] performed an experimental study on adiabatic straight capillary tubes with three refrigerants, namely HFC-134a, CFC-12 and HC-600a, under choked flow conditions in order to investigate the effect of tube diameter, length, degree of subcooling and condensation pressure. In addition, they reported a dimensional analysis for predicting the mass flux for different refrigerants. Their results showed that diameter affects the mass flux more significantly than the other parameters.

Chinguptak and Wongwises [15–17] also developed a homogenous two-phase flow model for refrigerant flow through helical capillary tube, where metastable liquid region was ignored. They found for the same length, mass flux through a coiled capillary tube with coil diameter of 40 mm is reduced by 9% compared to the straight tube.

A one-dimensional numerical simulation for R22, R12 and R-134a through straight adiabatic capillary tubes, using a drift flux model was reported by Liang and Wong [19]. They used the criteria pressure gradient criteria when dp/dz approaches to infinity in the exit of capillary tube.

In the past, there have been limited numerical or experimental works dedicated to choked flow phenomenon in capillary tubes [2,18]. Therefore, considering the scarcity of devoted literature on choked phenomenon in helically capillary tube, main objects of this work are choked phenomenon investigation and determination of the critical mass flow rate through coiled capillary tube under choked conditions. In most of the previous studies reviewed here, the flow in straight and coiled capillary tubes is simulated either by homogenous or by separated two-phase flow model for refrigerants R12 and R22. So, in the present work, a numerical model simulation is presented along with experimental investigation that predict the choked flow through coiled adiabatic capillary tube with various refrigerants, especially R134a under critical conditions.

To this end, a drift flux flow model using proper friction factor equations is numerically developed for helical capillary tube choked flow modeling and the generated simulation results are verified using present experimental data included herein as well as previously published relevant data.

2. Numerical modeling and resolution

Ignoring of the metastable flow passing through straight and coiled capillary tubes, it could be divided into single liquid phase and liquid–vapor flow regions [15–17 and 19]. Single-phase flow continues up to the point where flow pressure reaches the saturation pressure corresponding to its temperature [19]. Therefore, a model is first proposed for straight capillary tube and is then extended to coiled capillary tube, using proper friction factor correlation.

This proposed model is based on the following assumptions: fixed capillary tube geometry (inner diameter and relative roughness) through its length, oil-free refrigerant, adiabatic refrigerant flow, one-dimensional and steady state flow, and neglect of metastable flow. In drift flow flux model, the conservation of mass, momentum, and energy equations are applied by considering the two-phase region as a mixture of two liquid and vapor phases with different velocity [19]. Therefore, the formulation of the two-phase flow is constructed in terms of three equations for the mixture (continuity, momentum, and energy) and one drift velocity equation only for one of the vapor or liquid phases. In what follows, development of the governing equations for single phase and two-phase refrigerant flow are presented.

2.1. Single phase region in straight and coiled capillary tubes

The equation of conservation momentum for refrigerant flow in liquid phase going steady and one-dimensional flow is expressed in Eq. (1). Integration of the Eq. (1) along the length of capillary tube from inlet to the saturation point of the incompressible refrigerant fluid flow results in:

\[
\frac{dp}{dz} = \frac{f_D G^2}{2D \rho_\ell} \tag{1}
\]

\[
\frac{p_{in} - p_{sat}}{L_{sub}} = \frac{f_D G^2}{2D \rho_\ell} \tag{2}
\]

where \(p_{sat}\) is defined as the saturated pressure corresponding to the refrigerant temperature at inlet. From Eq. (2) the sub-cooled liquid’s length through capillary tube is obtained as:
\[ L_{\text{sub}} = \frac{2d\rho_l_1 \left( \rho_m - P_{\text{sat}} \right)}{f_0 G^2} \]  

(3)

For a straight capillary tube, the friction factor for turbulent flow \( f_0 \) is calculated from Colebrook’s correlation:

\[ \frac{1}{\sqrt{f_0}} = 1.14 - 2 \log \left( \frac{1}{d} \frac{f_0 G^2}{\nu} \right) \]  

(4)

The friction factor of helical capillary tube is evaluated from the following formula proposed by the Garcia–Valladares correlation [14] by taking the relative roughness and curve ratio of capillary tube into account:

\[ f_0 = \frac{c_1 \left( d/D \right)^{0.5}}{Re \left( d/D \right)^{2.5}} \left( 1 + \frac{c_2}{Re \left( d/D \right)^{2.5}} \right)^{1/6} \]  

(5)

Substituting \( f_0 \) into Eq. (3), the sub-cooled liquid region’s length is obtained for adiabatic helical capillary tube. Coefficients of \( c_1 \) and \( c_2 \) are presented in Appendix A.

2.2. Two-phase flow region in straight and coiled capillary tubes

The governing equations for one-dimensional steady state two-phase flow include continuity equation and the conservation of momentum and energy which are respectively represented by Eqs. (6)–(8).

\[ \frac{d}{dz} \left( \rho_l u_l (1 - \alpha) \right) + \frac{d}{dz} \left( \rho_m u_m \alpha \right) = 0 \]  

(6)

\[ \frac{d}{dz} \left( \rho_l u_l^2 \right) + \frac{d}{dz} \left( \left( 1 - \alpha \right) \rho_l u_l^2 \right) = -\frac{dp}{dz} - \frac{1}{2} f_0 G^2 \]  

(7)

\[ \frac{d}{dz} \left( \rho_l u_{l} \left( h_{v} + \frac{u_{l}^2}{2} \right) \right) + \frac{d}{dz} \left( \rho_m u_m (1 - \alpha) \left( h_{l} + \frac{u_{l}^2}{2} \right) \right) = 0 \]  

(8)

The drift velocity of the vapor phase is defined by:

\[ u_{v} = u_{v} - j \]  

(9)

where volumetric flux is defined as:

\[ j = \frac{\alpha \rho_l u_v + (1 - \alpha) \rho_l u_l}{\rho_m} \]  

(10)

and density of the liquid–vapor mixture is defined by Eq. (11):

\[ \rho_m = \alpha \rho_l u_v + (1 - \alpha) \rho_l u_l \]  

(11)

Zuber and Findlay [20] presented the following correlation for drift velocity of two-phase flow through horizontal tubes:

\[ u_{v} = c_{0} j + 1.48 \frac{\rho_l}{\rho_m} \left[ \left( \frac{\rho_l - \rho_v}{\rho_l} \right) \sigma_{g} \theta_{g} \right]^{1/2} \]  

(12)

Liang and Wong [19] have found reasonable results with \( c_{0} = 1 \) for critical flow inside capillary tubes. Using \( c_{0} = 1 \) and substituting Eqs. (10) and (11) into Eq. (12), the following equation is obtained:

\[ u_{l} = u_{v} - \frac{1.48}{1 - \alpha} \left[ \left( \frac{\rho_l - \rho_v}{\rho_l} \right) \sigma_{g} \theta_{g} \right]^{1/2} \]  

(13)

Hence, wall friction effect for two-phase flow, is calculated by multiplying the wall friction gradient of overall single-phase flow by two-phase multiplier.

\[ F_{w} = -\frac{1}{2} f_0 G^2 \]  

(14)

For coiled capillary tube, the friction factor is also determined using Eq. (5). The pressure drop friction coefficient [19] in two-phase flow, \( \lambda_{t} \), is expressed as:

\[ \lambda_{t} = \frac{\lambda_{t0} \rho_{m}}{2} \]  

(15)

where \( \lambda_{t0} \) is frictional pressure drop factor for liquid phase, which is calculated by Churchill [21] equation and \( \rho_{m} \) is the multiplier for two-phase flow region was presented by Lin et al. [22] as:

\[ \rho_{m} = \left[ \frac{A_{t0} + B_{t0}}{A_{t0} + B_{t0}} \right] \left[ 1 + x \left( \frac{\mu_{l}}{\mu_{v}} - 1 \right) \right] \]  

(17)

and

\[ A_{t0} = \left[ -2.457 \ln \left( \frac{7}{\left( \frac{Re_{t0}}{Re_{m0}} \right)^{0.9}} + 0.27 \right) \right]^{16} \]  

(18)

\[ B_{t0} = \left[ \frac{37530}{Re_{t0}} \right]^{16}, \quad \frac{B_{t0}}{A_{t0}} = \left( \frac{Re_{m0}}{Re_{t0}} \right) \]  

(20)

\[ \frac{Re_{m0}}{Re_{t0}} = \frac{D}{\mu_{l}}, \quad \frac{Re_{m0}}{Re_{m0}} = \frac{D}{\mu_{l}} \]  

(21)

\[ x = \frac{\rho_{l} u_{v} \alpha}{G} \]  

(22)

\[ \mu_{l} = \frac{\mu_{g} \mu_{l} + (1 - \chi) \mu_{l} \mu_{l}}{\mu_{g} \left( 1 - \chi \right) \mu_{l}} \]  

(23)

Substituting \( u_{l} = f(u_{l}) \) and its \( du_{l}/dz \) from Eq. (13) and it’s derivative into Eqs. (6)–(8), and replacing \( F_{w} \) from Eq. (14) into Eq. (7) and Lin’s [22] two-phase multiplier from Eq. (14), general form of the system of linear differential equations (Eq. (24)) are achieved. Definitions of coefficients \( a_{1}, b_{1}, c_{1}, \) and \( d_{1} \) for \( j = 1, 2, 3 \) are given in Appendix B. These coefficients contain properties of flow which could be calculated using thermodynamical charts. The fourth order Runge–Kutta method is employed to determine the critical mass flux of the choked refrigerant flow through the helical capillary tube. Using this numerical simulation by iteration method, system of equations for \( j = 1, 2 \) and 3 are solved simultaneously, then unknown variables \( u_{l}, \alpha, \) and \( p \) under specified initial conditions are obtained, and mass flux is increased up to the point where choked flow occurs at the end of the tube when \( dp/dz \) approaches infinity. Finally, pressure variations, vapor quality and void fraction variations through helical capillary tubes are obtained.
3. Experiments

3.1. Experimental apparatus

The schematic feature of test apparatus used herein is shown in Fig. 1. This closed circuit refrigerant flow’s set up can be used for different refrigerants, such as the R134a flow, under adiabatic and diabatic flow condition with or without oil in different capillary tubes, using oil separator. The experimental test section, made of copper capillary tubes, is shown in Fig. 2. From capillary tube, refrigerant enters evaporator. A tank with capacity of 100 L of glycol ethylene was designed for heating load of the evaporator. Ethylene glycol was circulated by a centrifugal pump. Then, the two-phase flow from the evaporator is sent into the accumulator allowing only vapor to enter the reciprocating compressor.

The oil separator are installed downstream of the reciprocating compressor in the bypass circuit. The oil-free vapors from oil separators are condensed in the shell and tube condenser. A 200 L open loop cooling water tank is used to obtain steady condition cooling water flow. Three sight glasses are installed: 1) after the condenser, 2) at the inlet, and 3) at the outlet of capillary tube, in order to visualize the state of refrigerant flow. A compact type sub-cooler and an electrical heater are used after the water cooled condenser to fine tune the refrigerant flow sub-cooling degree. The high pressure liquid from condenser is collected in the receiver. To continuously supply refrigerant flow through capillary tube, saturated or sub-cooled liquid is collected in the drier-receiver. A filter is used to collect and remove the moisture, impurities, and others particles. A manual needle expansion valve is also placed after the condenser and in parallel with the test section to adjust the excessive refrigerant flow in the capillary tube.

3.2. Test procedure and conditions

Temperature at different locations of the main components, such as condenser, sub-cooler, and compressor is measured by K-type thermocouples. Moreover, pressure at the inlet and outlet of the compressor and test section is measured using four high accuracy Bourdon pressure gauges.

The pressure along the helical capillary tube is measured using 11 online pressure transducers connected to a computer. The pressure distribution along the both types of straight and helical capillary tubes was measured with a special “T” connection. In order to minimize the disturbance to the normal refrigerant flow, the holes with a diameter of 0.15–0.25 mm, were made by a micro "supper drilling machine" in desired positions along the capillary tubes. Holes with diameter a little bit more than outer diameter of capillary tube was made in copper tube with 4 mm inner diameter. Then, capillary tube was passed through them. One end of the tubes was closed with solder and at the other end a pressure tap was installed which was connected to the pressure transducer by high pressure flexible rubber tube. The details also were illustrated in Fig. 3.

Temperature along the capillary tube’s wall was also measured by 20 K-type thermocouples connected to the computer. The helical capillary tubes of two different diameters (0.7874 mm and 1.397 mm), four different lengths (4.36 m–6.2 m), and two coil diameters (40 mm and 50 mm) were used in the experiments. The range of inlet sub-cooling degree was adjusted from 3 °C to 12 °C, while condensation temperature was set at 38 °C, 40 °C, and 46 °C for each tube length, tube diameter, and coil diameter of the capillary tubes.

In order to reach the steady conditions in each test run, the exit pressure of the capillary tube was adjusted to less than 150 kPa for all test runs that to ensure the choked condition. This process takes about 50 min to complete. A steady condition is achieved when inlet pressure and condensed temperature remain steady during 5 min within 0.05 bars and 0.2 °C, respectively. A counter was used to measure the critical mass flux in the bypass circuit located next to the horizontal test section. The pressure and temperature variation along the coiled capillary tube were recorded using online pressure transducers and online thermocouples connected to a computer, with an accuracy ±0.25% of full scale (20 bars) and ±0.3 °C, respectively.

![Fig. 1. The schematic diagram of experimental set up.](image-url)
3. Measurement of inner diameters and roughness of the capillary tubes

The inner diameter of capillary tube was measured by an optical measuring device. The internal cross sectional areas of each capillary tube was magnified by 40–55 times, then by computerized process and integrating, correct cross section area was calculated. Therefore, average inner diameter is obtained by uncertainty ±0.025 mm.

For measuring of capillary tube’s roughness, at first, each capillary tube was straightened and embedded longitudinally in a fixture, and then it was sliced into two halves. According to British Standard 1134 (part 1), inner surface roughness of the capillary tube was measured by a profile meter. At least, this measurement has done about 15–20 times for each capillary tube, and then average roughness was found by uncertainty ± 0.15 μm. Finally, uncertainty of all measured parameters presented in Table 1.

4. Results and discussion

Considering the scarcity of data for helical capillary tube under choked conditions, the numerical results for straight capillary tubes are compared with the numerical simulation results reported by Bansal and Wang [2], and also with experimental data of Zhou and Zhang [5] and Garcia–Valladares [14] for helical capillary tubes.

Table 1

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Instruments</th>
<th>Uncertainty</th>
<th>Remarks</th>
</tr>
</thead>
<tbody>
<tr>
<td>Capillary tube length</td>
<td>Steel rule</td>
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<td></td>
</tr>
<tr>
<td>Capillary tube diameter</td>
<td>Optical measuring</td>
<td>±0.025 mm</td>
<td></td>
</tr>
<tr>
<td>Roughness of capillary tube</td>
<td>Vernier calipers</td>
<td>±0.1 mm</td>
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</tr>
<tr>
<td>Roughness of capillary tube</td>
<td>Surface profilometer</td>
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<td>Mass flow rate</td>
<td>Digital counter</td>
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</tr>
<tr>
<td>Temperature</td>
<td>Thermocouples (K-Type)</td>
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<td>Online data acquisition system</td>
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<tr>
<td>Pressure</td>
<td>Pressure transducer</td>
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<td>Online data acquisition system, max. ± 0.04 bar</td>
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<tr>
<td>Pressure</td>
<td>Pressure gauge</td>
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</table>

4.1. Validation of the drift flux model for adiabatic straight capillary tube

All parameters and factors that affect the critical mass flux through straight capillary, such as tube diameter and length, degree of sub-cooling and condensation temperature are compared in Table 2. This comparison shows that critical mass flux increases by increasing of the tube inner diameter, condensation temperature and refrigerant degree of sub-cooling. Moreover, this table displays comparison between the present results and numerical data reported by Bansal and Wang [2]. It is clearly seen that maximum error for mass flow rate are about 12%. Therefore, this comparison shows a reasonable agreement between the presented numerical results and the numerical data reported by Bansal and Wang [2], providing enough confidence for utilizing this model to predict the choked flow characteristics through the adiabatic straight capillary tube.

4.2. Verification of this model for adiabatic helical capillary tube

To validate this drift flux model for coiled adiabatic capillary tube, our numerical results are also compared with previous experimental data of Zhou and Zhang [5], Garcia–Valladares [14] and also with present experimental results.

Fig. 4 compares the experimental data from Zhou and Zhang [5], and the results of numerical drift flux model mass flow rate varying with capillary tube length and condensation temperature for tube with 1.6 mm inner diameter, 40 mm coiled diameter and refrigerant R22 under choked conditions. The mass flow rate decreases rapidly as capillary tube length increases, and it increases as the condensation temperature increases. The small error of 4.5% also indicates that the model prediction is in agreement with measured data.

Fig. 5 compares our numerical results with experimental data by Garcia–Valladares [14] for mass flow rate variation with degree of sub-cooling for two different tube diameters (1.4 mm and 1.6 mm), refrigerant R22, 2 m tube length and 80 mm coil diameter. It is observed that mass flow rate of helical capillary tube increases as the tube diameter increases because of pressure drop decrease and increases as the degree of sub-cooling is increased. The model prediction displays error of 3.7% for this set of measured data.
The effect of coiled diameter for present model and measured data by Zhou and Zhang [5] for capillary tubes with 1.0 mm inner diameter, 1 m length and refrigerant R22 are shown in Fig. 6. It is clearly seen that mass flow rate increases linearly as degree of subcooling increases. Moreover, this comparison displays an average deviation of around 5.0%. Although, it verifies that the proposed model can be utilized to predict the flow characteristics through the helical adiabatic capillary tube.

To validate the present drift flux model for cooled adiabatic capillary tube, numerical results are compared with our experimental data. Fig. 7 illustrates the simulated and present experimental pressure profile through the helical tube for a tube with 1.397 mm inner diameter, 4.36 m length, 40 mm coil diameter, 6 °C degree of subcooling and refrigerant R134a for two different inlet pressure and mass flux under choked conditions. In liquid phase region, pressure gradient is almost constant, but the pressure drops rapidly flashes to vapor phase. It is also observed that there is small deviation and a reasonable agreement between present measured data and numerical results. In addition, there is slight deviation in two-phase region, due to the fact that in this model two-phase metastable flow region is ignored. Moreover, comparison between the present measured data and the developed drift flux model displays a maximum error of about 5.5% for pressure distribution.

4.3. Present numerical and experimental results

After validation of the present numerical model, some simulation results are presented in this section.

Mass flux variation of the present numerical simulation results versus degree of subcooling for 1.5 m helical capillary tube length, 40 mm coil diameter and refrigerant R134a are described in Fig. 8. It is clearly observed that, similar to the straight tube, the mass flux through helical capillary tube increases with increases in the degree of subcooling as well as increase in the tube diameter. As is expected, critical mass flux increases with increasing of the coil diameter.

Table 2

<table>
<thead>
<tr>
<th>Capillary tubes dimension, inlet condition</th>
<th>Bansal and Wang [2]</th>
<th>Present numerical results</th>
</tr>
</thead>
<tbody>
<tr>
<td>L (m)</td>
<td>d (mm)</td>
<td>Tcond (°C)</td>
</tr>
<tr>
<td>1</td>
<td>4.5</td>
<td>0.66</td>
</tr>
<tr>
<td>2</td>
<td>5.5</td>
<td>0.66</td>
</tr>
<tr>
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<tr>
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<td>0.5</td>
</tr>
<tr>
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<td>5.5</td>
<td>0.77</td>
</tr>
<tr>
<td>8</td>
<td>5.5</td>
<td>0.77</td>
</tr>
</tbody>
</table>

Fig. 4. Experimental (Zhou and Zhang [5]) and numerical mass flow rate for different condensation temperature.

Fig. 5. Comparison of the measured data (García–Valladares [14]) and numerical simulation results for different capillary tube diameter.

Fig. 6. Comparison of the helical capillary tube simulation results with the experimental data of Zhou and Zhang [5] for different coiled diameter.

Fig. 7. Comparison of the present simulation data and numerical results given by Bansal and Wang [2] for R134a choked flow through adiabatic straight capillary tubes.
change. Close to the capillary tube exit, it increases notably and produces a large amount of the vapor due to sharp pressure drop, since the refrigerant flow approaches the critical condition and maximum flow mass flux. The void fraction increases rapidly after the flash point and reaches to the maximum value of about 1.0 in the capillary tube exit. Comparison between void fraction and vapor quality profiles show that void fractions gradients are bigger than vapor quality. Moreover, variation of void fraction and vapor quality for different capillary tube lengths indicate the same trend.

The numerical mass flux and pressure variations for both straight capillary tube for 1.397 mm tube diameter, 4.36 m tube length, refrigerant R134a, 6.9 °C degree of sub-cooling and helical tube with coiled diameter 40 mm under the same conditions are illustrated in Fig. 11. It is noticed that as exit pressure of both type of capillary tubes is decreased, the mass flux increases and remains

Fig. 7. Comparison of present experimental data and numerical results for R134a choked flow.

Fig. 8. Mass flux variation versus degree of sub-cooling for different coiled diameter and tube diameter.

Fig. 9. Mass flux variation versus coil diameter for different capillary tube diameter.

Fig. 10. Void fraction and vapor quality variation of R134a choked flow through helical capillary tubes.

Fig. 11. Comparison of the mass flux variation through straight and coiled capillary tubes for R134a flow (L = 4.36 m, d = 1.397 mm).

Fig. 12. Experimental mass flux variation through helical capillary tube for R134a flow (L = 6.2 m, d = 0.7874 mm).
fairly constant about 300 kPa due to reaching the critical or choked condition. Moreover, it is observed that under the same test conditions and tube length, the critical mass flux through helical tube with coil diameter of 40 mm is about 16% less than that through the straight tube.

**Fig. 12** displays experimental mass flux variation versus evaporator pressure of helical capillary tube for a tube with 0.7874 mm inner diameter, 6.2 m length, 20 mm coil diameter, 40 °C condensation temperature and refrigerant R134a. As is expected, the mass flux increases gradually as the evaporator pressure decreases, then, the rate of mass flux remains stable at 180 kPa. At this point, mass flux reaches a maximum value through coiled capillary tubes, in other words, choked phenomenon has occurred. In addition, when the evaporator pressure decreases from that in the unchoked region to that in the choked region by about 8.2%, mass flux increases by only 0.4%, indicating that there is no major difference in mass flux for the steady condition and choked condition.

**Fig. 13** describes mass flux and exit pressure variation obtained from present experimental study versus evaporator pressure of helical capillary tube for a tube with 1.397 mm inner diameter, 4.36 m length, 40 mm coil diameter and refrigerant R134a. It is also observed that mass flux increases gradually with decrease in evaporator pressure, and then the rate of mass flux remains fairly constant at 270 kPa. Moreover, in the unchoked region, exit pressure and evaporator pressure are roughly equal until the choked region is reached at which point, mass flux reaches a maximum value through coiled capillary tubes in the choked phenomenon.

**Fig. 14** shows experimental mass flux variation versus evaporator pressure of a helical capillary tube with inner diameter 1.397 mm, 4.36 m length, 40 mm coil diameter, 40 °C condensation temperature and refrigerant R134a. It is observed that mass flux
increases gradually as the evaporator pressure decreases, hence mass flux variation remains steady at 125 kPa. At this point, mass flux reaches a maximum value through coiled capillary tubes, in other words, choked phenomenon has occurred.

Fig. 15 illustrate distributions of the experimental measured pressure, saturated pressure and measured temperature of refrigerant flow through helical adiabatic capillary tube. It can be seen that refrigerant flow may be divided two regions without considering the metastable flow. The saturated pressure and measured temperature remain nearly constant before vaporization has occurred at their flashing point and mass increases gradually as the evaporator pressure decreases, hence mass flux variation remains steady at 125 kPa. At this point, mass flux approaches the critical or choked condition.

Variations in mass flux versus helical capillary tube lengths for different tube diameters according to the present numerical results are shown in Fig. 16. It is seen that the critical mass flux decreases with increasing helical capillary tube length and increases with increase in the tube inner diameter.

\begin{align}
  c_1 &= 1.88411 \times 10^{-1} + 8.524721 \times 10^1 (\varepsilon/d) - 4.6303062 \times 10^4 (\varepsilon/d)^2 \times 1.31570014 \times 10^7 (\varepsilon/d)^3 \\
  c_2 &= 6.79778 \times 10^{-2} + 2.538803 \times 10^1 (\varepsilon/d) - 1.0613314 \times 10^6 (\varepsilon/d)^2 + 2.545534 \times 10^6 (\varepsilon/d)^3
\end{align}

5. Conclusions

A numerical simulation with proper accuracy, and no refrigerant-oriented advantages was introduced. The experimental investigation of the choked (and unchoked) flow through coiled capillary tube was also presented. The critical mass flux determination through coiled capillary tube under choked conditions is the main subject of this study. Therefore, a drift flux model was developed for analysis of choked refrigerant flow in a helical capillary tube, using proper friction factor equations. Present model was validated by the previously published data for R12, R22, and R-134a, and also was compared to the present measured data for R134a with average error of about 9%.

These experimental results indicate that when evaporator pressure decreases from unchoked region to the choked region by about 8.2%, mass flux increases by only 0.4%, indicating there is no major difference in mass flux between steady and choked conditions. It is also observed that for the same test conditions and tube length, the critical mass flux through helical tube with coil diameter of 40 mm is about 16% less than that of straight capillary tube. Finally, the present model can be used as a suitable tool for design and optimization of the vapor compression refrigeration systems (VCRS) with helical capillary tube, thus avoiding a vast amount of repetitive experiments.

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Appendix A

Comparison of numerical results obtained from proposed model and those of experiments, suggests that our model can be confidently used for design, simulation and optimization of straight and particularly coiled capillary tubes.

Appendix B

\begin{align}
  a_1 &= \rho_v \alpha + \rho_f (1 - \alpha) \\
  b_1 &= \rho_v \alpha + \rho_f \alpha - 2.61 e^{0.55 d_f} \left( \rho_f - \rho_v \right)^{0.25} \\
  c_1 &= u_v \alpha \frac{d_1}{df} + 0.6525 \alpha^{0.25} \rho_f^{0.5} \left( \rho_f - \rho_v \right)^{0.75} \frac{d_1}{df} \\
  d_1 &= 0 \\
  a_2 &= 2 \alpha \rho_v + 2 (1 - \alpha) \rho_f \alpha \\
  b_2 &= \rho_v \alpha \frac{u_f}{u_t} + 5 \frac{2250.5 \sigma^{0.5}}{1 - \sigma} \left( \rho_f - \rho_v \right)^{0.25} \\
  c_2 &= 1 + a u_v^{2} \frac{d_1}{df} + 1.305 \rho_f^{0.5} \sigma^{0.5} \left( \rho_f - \rho_v \right)^{-0.75} \frac{d_1}{df} \\
  d_2 &= F_w
\end{align}

\begin{align}
  a_3 &= \alpha \rho_v h_v + 1.5 \alpha \rho_v u_t^2 + (1 - \alpha) \rho_f h _t + 1.5 (1 - \alpha) \rho_f u_t^2 \\
  b_3 &= \rho_v \alpha h_v + \rho_v \frac{u_v^2}{2} + \rho_f h _t + \rho_f \frac{u_t^2}{2} - 2.61 e^{0.55 d_f} \left( h_t + 1.5 u_t^2 \right) \left( \rho_f - \rho_v \right)^{0.25} \\
  c_3 &= a u_v \frac{d_1}{df} + a \rho_v \rho_f \frac{d_2}{df} + \alpha \frac{u_v^2}{2} \frac{d_1}{df} + (1 - \alpha) \rho_f \frac{u_f}{u_t} \frac{d_1}{df} + 0.6525 \alpha^{0.25} \rho_f^{0.5} \left( h_t + 1.5 u_t^2 \right) \left( \rho_f - \rho_v \right)^{-0.75} \frac{d_1}{df} \\
  d_3 &= 0
\end{align}
References


Glossary

\[ \begin{align*}
A_0 & : \text{coefficient} \\
B_{ij} & : \text{coefficient} \\
C_{ij} & : \text{constant} \\
d & : \text{tube diameter, m} \\
D & : \text{coil diameter, m} \\
& : \text{Darcy's friction factor} \\
F_w & : \text{wall friction} \\
g & : \text{acceleration gravity, m s}^{-2} \\
G & : \text{mass flux, kg s}^{-1} \text{m}^{-2} \\
h & : \text{enthalpy, J kg}^{-1} \\
j & : \text{volumetric velocity, m s}^{-1} \\
L & : \text{length, m} \\
p & : \text{pressure, Pa} \\
Re & : \text{Reynolds number (\(\rho v d/\mu\))} \\
T & : \text{temperature, K} \\
u & : \text{velocity, m s}^{-1} \\
v & : \text{vapor quality} \\
w & : \text{axial coordinate, m} \\

& : \text{Greek letters} \\
\rho & : \text{two-phase multiplier} \\
\mu & : \text{dynamic viscosity, kg s}^{-1} \text{m}^{-1} \\
\nu & : \text{kinematic viscosity, m}^2 \text{s}^{-1} \\
\rho_f & : \text{fluid density, kg m}^{-3} \\
\rho_m & : \text{mixture density, kg m}^{-3} \\
\sigma & : \text{surface tension, N m}^{-1} \\
r & : \text{roughness, mm} \\
\alpha & : \text{void fraction} \\
\beta & : \text{pressure drop coefficient} \\
t & : \text{shear stress, N m}^{-2} \\

& : \text{Subscripts} \\
D & : \text{Darcy friction factor} \\
ex & : \text{external} \\
f & : \text{liquid} \\
L & : \text{liquid overall} \\
in & : \text{inlet of capillary tube} \\
sub & : \text{sub-cooled} \\
\text{tp} & : \text{two-phase} \\
v & : \text{vapor} \\

\end{align*} \]

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