A mathematical model of an adiabatic helical capillary tube used for R1234yf and R600a refrigerants

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Abstract. This paper presents a mathematical model to simulate the effect of the coiling diameter, subcooling degree and internal diameter on the characteristics flow of the refrigerants R1234yf and R600a, flowing through an adiabatic helical capillary tube. The homogeneous flow model was developed in the two-phase region based on the fundamentals of conservation of mass, momentum and fluid energy in the capillary tube. To validate this model, a comparison with numerical data on helical capillary tubes obtained by previous investigators has been made. It was reported that the coiling diameter, the inner diameter and inlet temperature of the capillary tube significantly influence on the pressure drop and quality and temperature distribution of the refrigerants R1234yf and R600a flowing inside the adiabatic helical tube. To validate this model, a comparison with numerical data on helical capillary tubes obtained by previous investigators has been made. Finally, the outcomes obtained from the present model exhibit satisfactory agreement with the available numerical data of helical capillary tubes.

1. Introduction

The capillary tube is a simple configuration with an inner diameter ranges between 0.5-2 mm and 1-6 m length which is usually used as a device to control the expansion and refrigerant flow rate of a small-scale household and industrial refrigeration systems, heat pump, air conditioning, etc. It has many advantages viz. inexpensive, no needs maintenance requirements and low starting torque as well. These qualities led the manufacturers to use the capillary tube in different refrigeration systems industries [1–4].

The capillary tube (CT) usually regulates the flow of refrigerants from the condenser to the evaporator while lowering pressure. The installation of the CT requires great attention in the application as well as its operating conditions. It is surely possible to know the capillary tube applicability based on the capacity of the system in the range of 10 kW [5]. Recently, the applicability has been expanded to cover larger units such as units of air conditioners with the system's capacity up to 35.2 kW [6], [7].

In order to obtain the desired mass flow through the capillary tube, several groupings of tube length, coil diameter, pitch and the inner diameter of the helical capillary tube can be applied. Selecting suitable tube geometry has a crucial issue to achieve optimum performance.
Capillary tubes are classified according to their shape viz. straight and helical capillary tube HCT. To save the required space and due to the virtue effect of the coiling diameter, HCTs in the refrigeration system were used. Capillary tubes are very simple structures, but the fluid flow process inside them is very complicated [8], [9], which is a very important part of the integrity of any cooling system in terms of temperature and pressure changes [6]. Many researchers begin in the study of the HCT with different boundary conditions due to its wide applications, in addition to that, it is considered one of the best techniques in the field of improving heat transfer [10].

Due to environmental concerns, the usage of CFCs and HFCs have been banned. Thus, capillary tube flow behaviour with new alternative refrigerant has achieved noticeable attention [11]. R600a and R1234yf are an excellent substitute to conventional refrigerants due to their eco-friendly properties. Studies on HCTs are only two decades old, and much work still to be done to show the effect of the coiling on the performance of the refrigerant flow. The performance of the cooling system may be significantly reduced due to the use of the inappropriate size of the capillary tube [12], and therefore, with the emergence of new eco-friendly refrigerants, the studies on HCTs with alternative refrigerant, it has become more important select the suitable capillary tubes for different refrigeration applications. Numerous researches have done on the adiabatic HCTs with different refrigerants. One of the researches, Naphon and Wongwises [13] has been reported on their literature review regarding the helical tube with a single and two-phase region. That the capillary length using a conventional refrigerant was less than of the alternative refrigerant and the helical coil diameter was more effective in less of 300 mm. Zhou and Zhang [7] had a numerical and experimental study to compare the performance of helical coiled capillary tubes with the straight capillaries under various operating conditions. The mass flow rate increased as the coiled diameter of the CT increased, but the changes were less beyond $D_c = 300$ mm. For a coiled diameter 40 mm, the mass flow rate was 10% lower than a straight capillary tube. Zareh et al. [14] developed a numerical drift flux model and experimentally validated the performance of refrigerant flow, using R12, R22 and R-134a, in an HCT. It was found that, for the same length of the tube and test conditions, the mass flux decreased through HCT by 11% with 40 mm coil diameter, whereas, there was a 14% decrease in the length of the helical tube to obtain the same refrigerant mass flux compared to a straight tube one. A capillary equivalent length parameter was introduced by Park et al. [15] to investigate the flow characteristics and prediction the mass flow rate of the refrigerant inside coiled CTs. The mass flow rates of the coiled CTs decreased by 5-16% higher than that of the straight CTs at the same operating conditions. The correlation exhibited good agreement with the available database for R410A, R407C and R22 in the straight and coiled CTs. Chingulpitak and Wongwises [16] concluded in their experiments, that the pressure drop along the helical coiled capillary tube was more than of the straight, and the mass flow rate had less. Dubba and Kumar [5] in their experimental study using R600a in a straight and HCT, it was found that the mass flow rate had decreased in a HCT by more than 3-12% of a straight CT one. Khan et al. [17] reported that the mass flow of refrigerant R-134a through an adiabatic HCT was less about 5-15% than the straight one under the same boundary conditions of parameters. Experimental investigations performed by Zhou and Zhang [2] on a split-type air conditioner using HC290 to replace R22 with matching different coiled adiabatic CTs. It was found that the mass flow rate tended to increase slightly with increased the CT coil diameter. The mass flow rate for R290 was only 47% compared to R22, due to the much lower density. Mittal et al. [18] experimentally investigated in an adiabatic helical and straight CTs for the flow of R-407C and the results were compared to indicate the coiling effect on the mass flow rate in the capillary tube. It has been concluded that the mass flow rate in the straight CTs is about 5–10 per cent higher than those of coiled one. In a separate investigation, Rasti and Jeong [9] developed a dimensionless empirical correlation to predict the mass flow rate of the refrigerant flowing inside adiabatic HCTs as well as straight CTs. It was found that the new correlation presented a good agreement when it was verified with experimental database for R-134a, R-22, R-410A, R-407C and LPG. To save space in household refrigerators applications, the CTs are commonly coiled and the coiling shape leads to increases the inside fluid flow pressure drop [2] (see figure 1). As presented above, many researchers focused upon their works on flow characteristics of conventional refrigerants.
flowing inside the CT and there is a lack of information in the literature about the flow characteristics of alternative refrigerants. Thus, in this study, a new model will be developed to study the flow characteristics of alternative refrigerants as R1234yf and R600 flowing inside the HCT used in small refrigeration systems. The effect of inner diameter, degree of subcooling and coil diameter on pressure profile, vapour quality and temperature profile are also presented.

2. Mathematical modelling of the capillary tube

In the vapour compression system, not their heat exchange from/to the refrigerant flowing through the adiabatic CTs. The mathematical model is based on conservation equations of mass, energy and momentum. As shown in figure 1, the flow of refrigerant flowing in a CT can be divided into two distinctive regions: single-phase and two-phase regions. As shown in figure 1, the region between points 1 and 2 is the inlet of the capillary tube which has a drop in pressure due to sudden contraction, where the pressure decreases linearly until the flashpoint. Likewise, the place between points 2 and 3 is the single-phase region consists of the subcooled liquid region and the places between points 3 and 4 are the two-phase region consisted of a liquid–vapour two-phase region, where the pressure drop as the length increases.

The following assumptions of the expansion device were adopted for the present mathematical model. Horizontal constant internal diameter capillary tube, coil diameter of the HCT is constant, the uniform surface roughness of along the capillary tube, constant the length of the CT, flow-through the capillary tube is one-dimensional, incompressible flow in single-phase region, steady-state flow, homogeneous two-phase flow, no heat exchange with ambient air and thermodynamic equilibrium (metastable flow phenomena are neglected).

The governing equations applying to describe the refrigerant flow characteristics in the helical capillary of the single-phase and two-phase regions are illustrated below.

2.1. The single-phase flow region

The refrigerant flow in the HCT can be divided into single-phase and two-phase regions (see figure 1). The refrigerant flows through the CT is in steady-state and consequently, the mass flow rate remains constant. Therefore, for an incompressible fluid:

\[ \rho_2 = \rho_3 = \rho, \]
\[ V_2 = V_3 = V. \]

The continuity equation for one-dimensional flow between points 2 and 3 is given below:

\[ \dot{m} = \rho_2 V_2 A = \rho_3 V_3 A = \rho V A. \]  \hfill (1)

In the relationship (1) \( \rho \) stands for density, \( V \) stands for the velocity, \( \dot{m} \) is the mass flow rate and \( A \) cross-sectional area of the capillary tube.

The pressure loss caused by the sharp entrance into the HCT between points 1 and 2 is given in the following equation:
where \( \rho_{\text{loss}} \) is the entrance loss factor takes the value of 0.5 for square-edged [15].

By applying the steady-state flow energy equation between points 2 and 3,

\[
p_2 + \frac{V_2^2}{2} + gZ_2 = \frac{P_1}{\rho_3} + \frac{V_3^2}{2} + gZ_3 + h_{\text{loss}}.
\]

(3)

Total head loss \( h_{\text{loss}} \) from the entrance region to end of the subcooled liquid region at flashpoint (1-3) can be determined from, [19].

\[
h_{\text{loss}} = f_{\text{sp},h} \frac{L_{\text{sp}}}{d_{\text{cap,in}}} \frac{V^2}{2 \rho},
\]

(4)

where \( L_{\text{sp}} \) and \( f_{\text{sp},h} \) are the single-phase region length and single-phase friction factor of the HCT, respectively. By neglecting the elevation difference \( Z_2 = Z_3 \) (horizontal tube) and substituting equation (4) in equation (3) can be expressed as:

\[
p_2 = p_1 + \left( f_{\text{sp},h} \frac{L_{\text{sp}}}{d_{\text{cap,in}}} \right) \left( \frac{V^2}{2 \rho} \right).
\]

(5)

Assuming the liquid specific volume as constant, and adding equations (5) and (2), the single-phase region length \( L_{\text{sp}} \), from the capillary tube entrance until the flashpoint (point 3), (see figure 1), can be determined from [20]:

\[
L_{\text{sp}} = \left[ (p_1 - p_3) \frac{2 \rho}{G^2} - \rho_{\text{loss}} - 1 \right] \frac{d_{\text{cap,in}}}{f_{\text{sp},h}}.
\]

(6)

The pressure at a flashpoint (3) is saturation pressure corresponding to the inlet temperature of the refrigerant at the entrance to the CT. \( G \) in equation (6) is the mass flux of the refrigerant [kg/(s·m²)].

For HCTs, the friction factor is estimated from the formula proposed by Mori and Nakayama [21] as reported by the author [22]. This correlation takes the relative roughness of the CT into account:

\[
f_{\text{sp},h} = C_1 \left( \frac{d_{\text{cap,in}}}{D_c} \right)^{0.5} \left[ \frac{Re}{D_c} \left( \frac{d_{\text{cap,in}}}{D_c} \right)^{2.5} \right]^{-1/6} \left[ 1 + C_2 \left( \frac{Re}{D_c} \left( \frac{d_{\text{cap,in}}}{D_c} \right)^{2.5} \right) \right]^{-1/6},
\]

(7)

\[
C_1 = 1.8841177 \cdot 10^{-1} + 85.2472168 \frac{\varepsilon}{d_{\text{cap,in}}} - 4.6303063 \cdot 10^4 \left( \frac{\varepsilon}{d_{\text{cap,in}}} \right)^{2} + 1.31570014 \cdot 10^{-7} \left( \frac{\varepsilon}{d_{\text{cap,in}}} \right)^{3}, \quad (7a)
\]

\[
C_2 = 6.79778633 \cdot 10^{-2} + 25.388038 \frac{\varepsilon}{d_{\text{cap,in}}} - 1.0613314 \cdot 10^7 \left( \frac{\varepsilon}{d_{\text{cap,in}}} \right)^{2} + 2.5455343 \cdot 10^6 \left( \frac{\varepsilon}{d_{\text{cap,in}}} \right)^{3}.
\]

(7b)

\( D_c \) in equation (7) is the coil diameter [m], \( d_{\text{cap,in}} \) is the capillary tube internal diameter [m], where \( \varepsilon \) in equations (7a) and (7b) is the relative surface roughness of the tube [m], for a smooth tube is chosen equal to 0.0000005 m [19].

2.2. The two-phase flow region

The refrigerant changes from liquid to two-phase state in the position of the flashpoint (point 3). The incremental length of each control volume in the two-phase region can be calculated by [23]:

\[
p_1 - p_2 = k_{\text{loss}} \left( \frac{V^2 \rho}{2} \right),
\]

(2)
\[
\Delta L_{tp,i} = \frac{2d_{cap,in}}{f_{tp,i}} \left( \frac{\Delta \rho f_{tp,i} \Delta p_{tp,i}}{\rho_i G^2} \right),
\]

where the subscripts \(tp, i\) and \(cap\) denote to two-phase region, the outlet point of each new section and capillary tube, respectively. And \(\rho_i\) is the specific density at the outlet the section of the capillary tube, \(\Delta p_{tp,i}\) is the two-phase capillary tube pressure drop and \(f_{tp}\) denote to the two-phase friction factor and the total length of the two-phase region calculated as the summation of incremental length as shown below:

\[
L_{tp} = \sum_{i=1}^{n} \Delta L_{tp,i}.
\]

The two-phase flow friction factor was proposed in [1] and is used in the present work as shown below:

\[
f_{tp,h} = \phi f_{tp,h} \left( \frac{\rho_{tp}}{\rho_f} \right),
\]

where \(f_{sh,p}\) is expressed by equation (7) and the multiplier, \(\phi^2\), is formed by:

\[
\phi^2 = \left[ \frac{A_{tp} + B_{tp}}{A_{tp}} \right]^{1/8} \left[ 1 + x \left( \frac{\rho_f}{f_{sp}} - 1 \right) \right],
\]

where subscripts \(f\) and \(g\) are the liquid phase and gas phase of the refrigerant, respectively, and \(x\) is the vapour quality.

\[
A_{tp} = 2.457 \ln \left( \left( \frac{7}{Re_{tp}} \right)^{0.9} + 0.27 \left( \frac{\varepsilon}{d_{cap,in}} \right)^{1/3} \right)^{16},
\]

\[
B_{tp} = \left( \frac{37530}{Re_{tp}} \right)^{16},
\]

\[
A_{sp} = 2.457 \ln \left( \left( \frac{7}{Re_{sp}} \right)^{0.9} + 0.27 \left( \frac{\varepsilon}{d_{cap,in}} \right)^{1/3} \right)^{16},
\]

\[
B_{sp} = \left( \frac{37530}{Re_{sp}} \right)^{16},
\]

\[
Re_{tp} = \frac{G d_{cap,in}}{\mu_f},
\]

\[
Re_{sp} = \frac{G d_{cap,in}}{\mu_{sp}}.
\]

In relationships (11e and 11f), \(Re\) is the Reynolds number and \(\mu\) is the dynamic viscosity (kg m\(^{-1}\) s\(^{-1}\)).
The dynamic viscosity of a two-phase mixture, $\mu_{tp}$, was suggested by many investigators. The model proposed in [1] was used in this paper as follows [24]:

$$\mu_{tp} = \frac{\mu_f \mu_g}{\mu_g + x^{1.4}(\mu_f - \mu_g)}.$$  \hspace{1cm} (12)

By ignoring the variance in elevation, and applying the conservation equation of energy for steady-state adiabatic conditions without external work is given as below:

$$h + \frac{V^2}{2} = h_i + \frac{V^2_i}{2} = ct.$$  \hspace{1cm} (13)

The energy balance between point 3 (flashing point) where starting two-phase region and any control volume ‘i’ downstream, along the CT in the two-phase flow region, is determined by equation (14):

$$h_3 + \frac{V^2_3}{2} = h_i + x_i(h_{g,i} - h_{f,i}) + \frac{G^2_i}{2}(v_{f,i}(1-x_i) + v_{g,i}x_i),$$  \hspace{1cm} (14)

where

$$h_i = h_{f,i}(1-x_i) + h_{g,i}x_i.$$  \hspace{1cm} (15)

The local vapour quality is determined by applying the energy equation to a fluid element, as mentioned by references [19], [20], [23], [25], thus:

$$x_i = -h_{g,i} - G^2v_{f,i}v_{g,i} + \left(2^2v_{g,i}v_{g,i}^2 - h_i - h_{f,i} - \frac{V^2_i}{2}ight)\cdot \left(\frac{G^2v_{g,i}}{2} - h_{g,i} + h_{f,i}\right)^{-1},$$  \hspace{1cm} (16)

where $h_{g,i} = h_g - h_i$ is the difference between the liquid phase and gas phase-specific enthalpies (J/kg), respectively, and $v_{g,i} = v_g - v_i$ is the difference between the liquid phase and gas phase-specific volume (m$^3$/kg), respectively.

3. Validation of the Mathematical model

In order to verify the proposed mathematical model developed in Engineering Equation Solver program - EES, a comparison is made with the available numerical model data of Khan et al. [17] for R134a.

![Figure 2. Comparison of the present numerical results with numerical results of Khan et al. [17].](image-url)
Figure 2 shows the input data of the parametric study used in the model developed by EES which are corresponding to the input data of Khan et al. [17] using R134a as working fluid flowing through HCT. The results of the present model are in a good agreement with the numerical data of Khan et al., with the error was ± 6.1 %. From figure 2, as the length of HCT increases, the pressure decreases linearly along the single-phase region until the flashpoint (point 3), beyond this point the vaporization process starts and the rate of change of drops of pressure increases. It was found that the percentage difference between the present model and the numerical data of Khan et al. [17] increased with increases the HCT length, then before the exit point of the tube, the percentage difference begins to decreases.

4. Results and discussion
All properties and results illustrated in this work, have been obtained using the EES software [26]. The geometric and working condition are displayed in Table 1 and Table 2, respectively. The geometric parameters were extracted directly from a small refrigerator in the laboratory.

| Table 1. Presents the geometric parameters used in this model. |
|---------------------------------------------------------------|
| Properties | value       |
|--------------|-------------|
| Total HCT length $L_{cap}$                                  | 2.26 m      |
| Capillary tube internal diameter $d_{cap,in}$               | 0.5, 1.0, 1.5 mm |
| Coil diameter of the HCT $D_c$                              | 40, 100, 140 mm |
| The internal surface roughness of the capillary tube $\varepsilon$ | $0.5 \cdot 10^{-6}$ m [20] |
| The entrance loss factor $k_{loss}$                         | 0.5 [20]    |

To better study the fluid’s behaviour flowing inside the HCT, according to the refrigerant state at the inlet of the capillary tube area, the related equations were solved, and the results will be analysed individually.

| Table 2. Presents the working conditions used in this model. |
|-------------------------------------------------------------|
| Properties | value |
|-------------|-------|
| Sub-cooling at the capillary tube inlet $\Delta T_{sub}$   | 5, 7, 9 K |
| Condensing temperature $T_{con}$                           | 318.15 K |

4.1 Effect of inner diameter on the helical capillary tube
As shown in figure 3, the pressure distribution along the HCT with a different inner diameter between R1234yf and R600a as alternative refrigerant of R134a in small scale refrigeration system were compared. From figure 3, as the distance from HCT inlet grows the pressure drops linearly through single-phase region and the rate of change of drop of pressure increases in two-phase regions of both refrigerants, with single-phase length is slightly lower of R1234yf than that of R600a. The pressure drop of R1234yf is stronger than in the case of R600a. For both refrigerants, when inner diameter increases, the pressure at the outlet of the capillary tube increases due to raising the friction factor value, this means large inner diameter results higher pressure, higher evaporating temperature. The pressure difference at the inlet of the tube is big because the refrigerants have different thermodynamic properties but at the end of the tube the difference is closer, this means that both refrigerants can be used one instead of the other. At the same length, the mass flow rate increases with increasing of the inner diameter of the CT, the high diameter means a high mass flow rate. Meanwhile, the mass flow
rates of R600a were found to be slightly lower than R1234yf, except in case the inner diameter is 1.5 mm the mass flow rate of R600a is significantly lower than R1234yf.

Figure 4 shows the change in the quality at different inner diameters along the HCT of R1234yf and R600a. In the case of the inner diameter is small, the quality is higher and the quality decreases with the increases of the internal diameter for both refrigerants. From figure 4, it can be observed that when the distance from the entrance of the capillary tube grows, the quality increases at the same boundary condition for each case. Furthermore, the vapour quality for both refrigerants is lower at the same capillary length, when the inner diameter of the HCT increases. The decreases in the vapour quality at the evaporator inlet leading to increases the refrigeration capacity, as mentioned by Seixlack and Barbazelli [27]. At the flashpoint, the quality is zero and then grows in a nonlinear pattern till the exit point of the capillary tube.

4.2 Effect of degree of subcooling on the helical capillary tube

Figure 5 presents a comparison of pressure distribution at different degree of subcooling along the HCT for R1234yf. It can be noted, at the same mass flow rate and same capillary tube length, the pressure drops linearly at single-phase region and the pressure continues to decrease until the end of the capillary tube. In addition, as the degree of subcooling increases, the pressure obtained at the outlet of tube increases, thus, the higher evaporating temperature at the same HCT length.
Figure 6 presents the quality variation at different degree of subcooling with a HCT length of R1234yf. It can be observed in figure 6, the HCT has greater refrigerant vapour quality for the same length of the CT. In addition, the quality at the exit of the CT is higher as the degree of subcooling is lower and vice versa.

Figure 5. Comparison of pressure distribution along the helical capillary tube for R1234yf.

Figure 6. Quality variation with a helical capillary tube length of R1234yf.

Figure 7. Temperature distributions along the helical capillary tube for R1234yf.
The refrigerant vapour quality increases as the distance from capillary tube inlet grow. Furthermore, it can be noticed that the high degree of subcooling increases the length of the single-phase region. Figure 7 shows the temperature distributions of refrigerant with different degree of subcooling along the HCT for R1234yf. As previously mentioned in assumptions of this work, the flow inside the CT is adiabatic, then the temperature along the single-phase region remains constant. In the two-phase region, the saturation temperature is a function of the saturation pressure. From figure 7, it can be noticed that the temperature lowers as the length of the CT increases due to decreases of the pressure inside the tube. In addition, the temperature falls down as the degree of subcooling smaller.

4.3. Effect of coil diameter on the helical capillary tube

Figure 8 presents the pressure distribution at a different coil diameter along the HCT of R1234yf. From figure 8, the pressure decreases as the distance from capillary tube inlet grow, and the pressure increases as the coil diameter increases. The friction factor can be calculated by Lin et al. [1] because their correlation involves coil diameter in the calculation. As shown in figure 8 the capillary tube with $D_c = 100\, \text{mm}$ and $D_c = 140\, \text{mm}$ will have close behaviour.

![Figure 8. Pressure distribution along the helical capillary tube of R1234yf.](image)

![Figure 9. Quality variation with a helical capillary tube length of R1234yf.](image)
Figure 9 illustrates the quality variation at different coil diameter with a HCT length of R1234yf. The vapour quality of the refrigerant as the CT length increases, as well as the quality decreases as the coil diameter increases for the same length of the CT. Higher coil diameter leads to lower vapour quality.

Figure 10 presented the temperature distributions with different coil diameter along the HCT for R1234yf. It can be noted that the temperature lowers as the length of the CT increases and raises whenever the coil diameter increases. The temperature at the end of the CT with increasing the coil diameter is lower than when changing the degree of subcooling of the refrigerant except that of the CT with diameter is 0.5 mm and degree of subcooling is 5 °C and $D_c$ is 40 mm where the temperature is lowest.

![Temperature distributions along the helical capillary tube of R1234yf.](image)

5. Conclusions
This work presents the effect of inner diameter, degree of subcooling as well as the coil diameter on the flow characteristics of R1234yf and R600a as alternative refrigerants flowing inside an adiabatic HCT. The present mathematical model developed in Engineering Equation Solver program - EES was verified by comparing it with the available numerical model data of Khan et al. [17] for R134a and was found to give an average deviation by about ±6%. The present model was developed to study the performance of an adiabatic HCT in the domestic refrigeration system. In the present model, the friction factor can be determined by Lin et al. [1] because their correlation involves coil diameter in the calculation. The obtained results report that the inner diameter strongly affects the mass flow rate and pressure drop of the HCT. To obtained closer pressure approach of the capillary tube for the same length should be the inner diameter with R1234yf less than that of R600a and the mass flow is lower. For the same length, the pressure and temperature values at the outlet of the capillary tube are higher and the vapour qualities are lower for all cases tested. Moreover, the it is found that the pressure decreasing linearly as distance from the capillary tube inlet grows through single phase region and the rate of change of drop of pressure increases in two phase regions of both refrigerants, with single phase length is slightly lower of R1234yf than that of R600a. The higher pressure at the exit of the capillary tube means higher evaporating pressure and then higher evaporating temperature, this may be affecting on the performance of the refrigeration system.

6. References
[1] Lin S, Kwok C C K, Li R Y, Chen Z H and Chen Z Y 1991 Local frictional pressure drop during vaporization of R-12 through capillary tubes Int. J. of Multiphase Flow 17 pp 95-102
[2] Zhou G and Zhang Y 2010 Performance of a split-type air conditioner matched with coiled adiabatic capillary tubes using HCFC22 and HC290 Appl. Energy 87 pp 1522-8
[3] Heimel M, Lang W, Berger E and Almbauer R 2012 A Homogeneous Capillary Tube Model - Comprehensive Parameter Studies Using Isobutane As Refrigerant In Int. Refrig. Air Conditioning Conf. p 2255
[4] Javidmand P, Zareh M and Hoffmann K A 2014 An Experimental Comparison of the Refrigerant Flow through Adiabatic and Non-Adiabatic Helical Capillary Tubes in Int. Refrig. Air Conditioning Conf. p 1549
[5] Dubba S K and R. Kumar 2018 Experimental investigation on flow of R-600a inside a diabatic helically coiled capillary tube: Concentric configuration Int. J. Refrig. 86 pp 186-195
[6] Liang S M and Wong T N 2001 Numerical modeling of two-phase refrigerant flow through adiabatic capillary tubes Appl. Therm. Eng. 21 pp 1035-48
[7] Zhou G and Zhang Y 2006 Numerical and experimental investigations on the performance of coiled adiabatic capillary tubes Appl. Therm. Eng. 26 pp 1106-14
[8] Hermes C J L, Melo C and Knabben F T 2010 Algebraic solution of capillary tube flows. Part I: Adiabatic capillary tubes Appl. Therm. Eng. 30 pp 449-457
[9] Rasti M and Jeong J H 2018 A generalized continuous empirical correlation for the refrigerant mass flow rate through adiabatic straight and helically coiled capillary tubes Appl. Therm. Eng. 143 pp 450-60
[10] Zheng L, Xie Y and Zhang D 2018 Numerical investigation on heat transfer and flow characteristics in helically coiled mini-tubes equipped with dimples Int. J. Heat Mass Transf. 126 pp 544-70
[11] Jadhav P and Agrawal N 2018 Numerical Study on Choked Flow of CO2 Refrigerant in Helical Capillary Tube Int. J. Air-Conditioning Refrig. 26 pp 1-11
[12] Choi J, Kim Y and Kim H Y 2003 A generalized correlation for refrigerant mass flow rate through adiabatic capillary tubes Int. J. Refrig. 26 pp 881-8
[13] Naphon P and Wongwises S 2006 A review of flow and heat transfer characteristics in curved tubes Renew. Sustain. Energy Rev. 10 pp 463-90
[14] Zareh M, Shokouhmand H, Salimpour M R and Taeibi M 2014 Numerical simulation and experimental analysis of refrigerants flow through adiabatic helical capillary tube Int. J. Refrig. 38 pp 299-309
[15] Park C, Lee S, Kang H and Kim Y 2007 Experimentation and modeling of refrigerant flow through coiled capillary tubes Int. J. Refrig. 30 pp 1168-75
[16] Chingulpitak S and Wongwises S 2010 Effects of coil diameter and pitch on the flow characteristics of alternative refrigerants flowing through adiabatic helical capillary tubes Int. Commun. Heat Mass Transf. 37 pp 1305-11
[17] Khan M K, Sahoo P K and Kumar R 2007 Flow characteristics of HFC-134a in an adiabatic helical capillary tube 5th Int. Conf. on Heat Transfer, Fluid Mechanics and Thermodynamics (Sun City, South Africa) p KM1
[18] Mittal M K, Kumar R and Gupta A 2010 An experimental study of the flow of R-407C in an adiabatic helical capillary tube Int. J. Refrig. 33 pp 840-7
[19] Deodhar S D, Kothadia H B, Iyer K N and Prabhu S V 2015 Experimental and numerical studies of choked flow through adiabatic and diabatic capillary tubes Appl. Therm. Eng. 90 pp 879-94
[20] Melo C 1992 Modelling Adiabatic Capillary Tubes: A Critical Analysis Proc. IIR-Purdue Refrig. Conf. pp 113-23
[21] Mori Y and Nakayama W 1967 Study of forced convective heat transfer in curved pipes (2nd report, turbulent region) Int. J. Heat Mass Transf. 10 pp. 37-59
[22] García V O 2007 Numerical simulation and experimental validation of coiled adiabatic capillary tubes Appl. Therm. Eng. 27 pp 1062-71
[23] Chingulpitak S and Wongwises S 2011 A comparison of flow characteristics of refrigerants
flowing through adiabatic straight and helical capillary tubes *Int. Commun. Heat Mass Transf.* 38 pp 398-404

[24] Sarker D, Kim L, Son K, Jeong J and Chang K 2010 An Evaluation of Constituent Correlations for Predicting Refrigerant Characteristics in Adiabatic Capillary Tubes *Int. J. Air-Conditioning Refriger.* 18 pp 131-9

[25] Bansal P K and Rupasinghe A S 1998 An homogeneous model for adiabatic capillary tubes *Appl. Therm. Eng.* 18 pp 207-19

[26] Engineering Equation Solver 2019 Academic commercial V.10.111-3D #4487 Faculty of Mechanical Engineering University Politehnica of Bucharest

[27] Seixlack A L and Barbazelli M R 2009 Numerical analysis of refrigerant flow along non-adiabatic capillary tubes using a two-fluid model *Appl. Therm. Eng.* 29 pp 523-31

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