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STUDY OF HELICALLY COILED ADIABATIC CAPILLARY TUBES FOR TRANSCRITICAL CO_2 EXPANSION

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ABSTRACT

Transcritical expansion through coiled adiabatic capillary tubes having CO_2 as the refrigerant is numerically simulated and the performance is compared with R22 flow through such tubes. Mori and Nakayama friction factor correlation in combination with Churchill equation, which is fitted into Blasius type equation, is used to calculate the friction factor. Results indicate that reduction in mass flow rate with a coiled capillary tube compared to a straight one is more pronounced in case of CO_2 in comparison to R22. Mass flow rate of refrigerant increases as the coil diameter increases, but changes little beyond a coil diameter (D) of 180 mm. A shorter capillary tube will be required to match the requisite system mass flow rate when coiled capillary tubes are employed.

NOMENCLATURE

A	[m^2]	cross-sectional area
d	[m]	capillary tube diameter
D	[m]	coil diameter
Eve	[-]	evaporator
f	[-]	friction factor
G	[kg m^{-2} s^{-1}]	mass flux
GC	[-]	gas cooler
h	[J kg^{-1}]	specific enthalpy
isr	[mm]	internal surface roughness
L	[m]	capillary tube length
$\dot{m}$	[kg s^{-1}]	mass flow rate
n	[-]	exponent constant in Eq. (16)
p	(Pa)	pressure
Re	[-]	Reynolds number
Ref	[-]	refrigerant
T	[K]	temperature
V	[m s^{-1}]	velocity
x	[-]	dryness fraction

Special characters

α	[-]	proportional coefficient in Eq. (16)
υ	[m^3 kg^{-1}]	specific volume
ρ	[kg m^{-3}]	density
ε	[mm]	tube wall roughness
μ	[Pa s]	dynamic viscosity
φ	[-]	two-phase frictional multiplier
ω	[-]	coefficient in Eq. (13)
ψ	[-]	coefficient in Eq. (13)

Subscripts

1 - 4	state points
con	condenser
ev	evaporator
tp	two-phase

INTRODUCTION

In the hunt for alternative refrigerants, use of natural refrigerants has been advocated, as they are ecologically safe. CO_2, designated as R744, is one of the natural refrigerants that has witnessed a strong renewed interest [1]. As a natural refrigerant, CO_2 is the preferred choice today for its environmentally benign nature and largely beneficial heat transfer and safety characteristics compared to the currently used refrigerants [2, 3].

Capillary tubes are universally accepted and well proven expansion device in small capacity refrigeration and air-conditioning units due to their simple configuration and low cost. In the small capacity systems where charge is limited and load variation is relatively small, capillary tubes have the natural capability to balance the system and equalise the pressure during off-design conditions that keeps the compressor starting torque low. It is a paradox that though the capillary tube is simple in configuration, flow inside the capillary tube is quite complex in nature. In an adiabatic capillary tube pressure falls from high side pressure to low side pressure due to frictional pressure drop and momentum pressure drop. However, in the two-phase region after inception of vaporisation, momentum pressure drop dominates over friction pressure drop due to the existence of vapour phase. Consequently, enthalpy reduces in the latter part of the capillary tube as part of the total energy is converted to kinetic energy. Further, increase in specific volume with increase in quality results in increase in longitudinal pressure gradient as the flow progresses. Numerous combinations of bore and length can be provided to obtain the desired flow restriction, which has a strong influence on the performance of the overall system. Tube geometry (diameter and length) at a given operating condition is the main concern in the design of a capillary tube.

In the past, capillary tubes have fascinated many researchers. Extensive research was carried out on design and flow characteristics aspects of the adiabatic straight capillary tubes with halocarbon and hydrocarbon refrigerants by several researchers [4-16]. However, coiled capillary tubes, which has more practical implications to save space in small capacity systems have not been addressed much in the open literature. For a given condition given by its pressure, temperature, mass flow rate, tube diameter and tube length, the frictional pressure drop of single-phase fluid flow through a curved tube is larger than that of flow through a straight tube due to the centrifugal force. However, for the mean coil diameter more than 300 mm, change in mass flow rate was negligible [17]. Kuehl and Goldschmidt [20] have shown in their experimental study that irrespective of capillary tube length, the mass flow rate of R22 does not change by more than 5%. Kim et al. [21] have proposed a correlation for predicting mass flow rate employing Buckingham-π method based on their experimental study with R22 and its alternatives R407C and R410A. It was reported that mass flow rate with a coil diameter of 40 mm reduces by 9% approximately relative to a straight tube. Wei et al. [22] presented a comparative study between straight and coiled capillary tube in case of R22 and R407C. It is reported that coiling effect on mass flow rate is more pronounced at lower coil diameter due to secondary flow effect caused by the centrifugal force in coiled tubes. Deodhar et al. [23] have investigated an experimental and numerical investigation with R134a through straight and coiled capillary tube. Park et al. [24] developed a generalised correlation based on their experimental study for the flow of R22 and its alternatives R407C and R410A to predict mass flow rate through straight and coiled capillary tube using Buckingham-π theorem. A equivalent length L_e for the capillary tube is introduced to accommodate the coiling effect. Valladares [25] presented a numerical model for coiled capillary tube working with pure fluid and refrigerant mixture taking into account the metastable conditions in both liquid and two-phase. A Newton-Raphson technique based numerical scheme was implemented to calculate the mass flow rate. A comparative study between straight and helical adiabatic capillary tube was presented based on the homogeneous two-phase flow model and predicted that the length of helical capillary tube is 12% shorter than that of the straight capillary tube [26].

Majority of such studies have concentrated on the HFCs, hydrocarbon refrigerants and their mixtures. Relatively, much less information is currently available in the open literature on flow characteristics of capillary tube with CO_2 as a refrigerant. Employing capillary tubes with CO_2 as the refrigerant must recognise the different scenario because of the transcritical nature of the process. Unique thermodynamic and thermophysical properties of CO_2, such as low critical temperature, low density ratio (liquid to vapour) and high pressure operation make CO_2 quite different from other refrigerants [2]. Madsen et al. [27] investigated an adiabatic capillary tube in a transcritical CO_2 refrigeration system. It was reported that COP is superior with a suitably designed capillary tube than that in case of constant gas cooler pressure. Silva et al. [28] experimentally studied the transcritical expansion of carbon dioxide through adiabatic capillary tube and developed a dimensionless correlation to predict the refrigerant mass flow rate as a function of tube geometry and operating conditions. Hermes et al. [29] presented an algebraic model for simulating the transcritical expansion of carbon dioxide through adiabatic capillary tubes considering isenthalpic expansion. Agrawal and Bhattacharyya [30] carried out a series of theoretical studies on flow characteristics of straight adiabatic capillary tubes in a transcritical CO_2 heat pump cycle. Transcritical expansion of CO_2 in a straight adiabatic capillary tube was simulated employing a homogeneous two-phase flow model. Additionally, a comparative study of flow characteristics of a straight adiabatic capillary tube in a transcritical CO_2 heat pump system was investigated employing separated and homogeneous two-phase flow model [31].

To the best of the authors' knowledge, no result has yet been reported in the open literature on expansion of transcritical carbon dioxide through a coiled adiabatic capillary tube. The present work is an attempt to discuss the prediction of mass flow rate of transcritical carbon dioxide through a coiled adiabatic capillary tube employing homogeneous two-phase model in association with the *Mori and Nakayama* friction factor correlation.

MATHEMATICAL MODEL

Gorasia et al. [32] has shown that the coiling effect can be embodied in the calculation of friction factors. Consequently, a straight capillary tube model can be employed to simulate the flow through coiled capillary tube by incorporating an appropriate friction factor equation. Accordingly, as mentioned earlier, Mori and Nakayama friction factor correlation is used in the analysis to capture the coiling effect.

The capillary tube can be divided into three distinct flow regions, namely, supercritical flow region 1-2, transcritical flow region 2-3 and the subcritical flow region 3-4 as shown in Figure 1. Point '2' lies on the critical temperature line (Figure 1). Therefore, in the region 2-3, the fluid is considered to be subcooled. In the supercritical and transcritical single-phase region, temperature does not remain constant unlike in case of subcritical refrigeration cycles due to the unique shape of isotherms. As a result, density is not constant leading

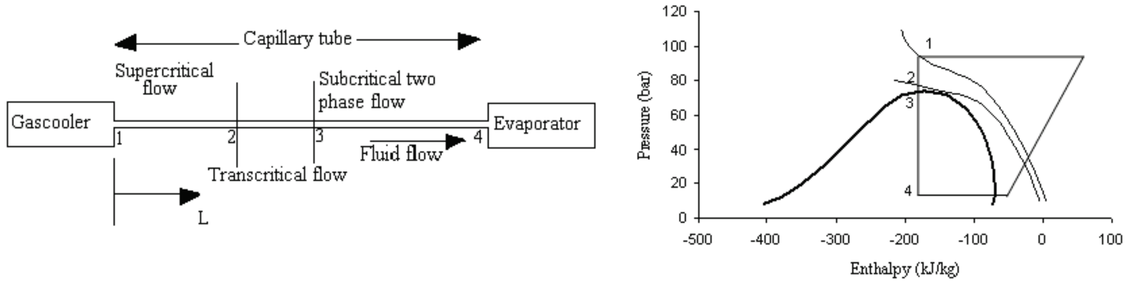


Figure 1. Adiabatic capillary tube showing different flow regions and corresponding cycle plot

to momentum pressure drop in addition to friction pressure drop even in the single-phase region (supercritical and subcooled). Total tube length is expressed as: $L = L_{sup} + L_{subliq} + L_{tp}$. The expansion process from point 1 to 4 is shown in Fig. 1 on the pressure-enthalpy plane cycle plot.

It is assumed that the capillary tube is straight with constant inner diameter and roughness; further, the flow is one-dimensional, steady, with no heat and work interactions. Homogeneous two-phase flow is assumed since Agrawal and Bhattacharyya [31] has shown that homogeneous flow model predicts such phenomena reasonably well. As mentioned earlier, as the flow progresses in the subcooled region, temperature decreases due to the unique shape of the isotherms, unlike the subcritical system where temperature remains constant in the subcooled region. Consequently, probability of occurrence of metastable condition is small and hence thermodynamic equilibrium is assumed to occur. Entrance losses are negligible since there is a large reduction of pressure from inlet to exit of the capillary tube during transcritical CO_2 expansion, and thus entrance losses are negligible as a fraction of the total and this was verified by using the available correlations.

The model is set around fundamental equations of conservation of mass, energy and momentum and it incorporates variation in property values. This latter feature is essential for simulation of transcritical CO_2 systems as property variation is extremely large in the neighbourhood of the critical point. The capillary tube is discretised into a number of longitudinal elements to enable the sharp changes in CO_2 property to be captured adequately in the analysis.

Single-phase flow region

The conservation of mass and energy for steady flow in an element of fluid is given by:

$$d\left(\frac{AV}{\upsilon}\right) = 0 \qquad (1)$$

$$dh + \frac{G^2}{2} d\upsilon^2 = 0 \qquad (2)$$

where h and υ are the enthalpy and specific volume at any point, respectively.

From the conservation of momentum equation, the difference in forces applied to the element of fluid due to drag and pressure difference on opposite ends of the element should be equal to that needed to accelerate the fluid and is given by:

$$-dp - f_{sp}\frac{dL}{d}\frac{V}{2}G = GdV \qquad (3)$$

Hence, $$dL = \frac{2d}{f_{sp}}\left\{-\left(\frac{\rho}{G^2}\right)dp - \left(\frac{dV}{V}\right)\right\} \qquad (4)$$

Two-phase flow region

Principles of mass, energy, and momentum conservation are employed to a discretised element (inlet L_1 and exit L_2) of the capillary tube. The conservation of mass for steady flow in an element of fluid follows the single-phase regime model given by equation (1). Neglecting the elevation difference and the heat transfer in and out of the tube, energy conservation yields:

$$dh + \frac{G^2}{2} d\upsilon^2 = 0 \qquad (5)$$

Two-phase enthalpy and specific volume are expressed as:

$$h_{L_1} = h_{l_{L_1}} + x_{L_1} h_{lg_{L_1}}, \upsilon_{L_1} = \upsilon_{l_{L_1}} + x_{L_1} \upsilon_{lg_{L_1}} \qquad (6)$$

Substituting equation (6) into equation (5):

$$\left(\frac{G^2}{2}\upsilon_{lg_{L_2}}^2\right)x_{L_2}^2 + \left(h_{lg_{L_2}} + G^2 \upsilon_{l_{L_2}} \upsilon_{lg_{L_2}}\right)x_{L_2} + \left\{h_{l_{L_2}} - h_{L_1} + \frac{G^2}{2}\left(\upsilon_{l_{L_2}}^2 - \upsilon_{L_1}^2\right)\right\} = 0 \qquad (7)$$

This quadratic equation is solved to calculate x_{L_2} .

Conservation of momentum may be written in the same manner following the single-phase region model, as expressed in equation (4), and the differential length expression for the capillary tube is obtained similarly:

$$dL = \frac{2d}{f_{tp}}\left\{\frac{d\rho}{\rho} - \frac{\rho}{G^2}dp\right\} \qquad (8)$$

This needs to be integrated first between 1 and 2 for the supercritical zone and then between 3 and 4 for the two-phase region to obtain the required length of the tube.

McAdams [33] model is employed to obtain two-phase viscosity to facilitate an estimation of the two-phase friction factor at various quality conditions.

$$\frac{1}{\mu_{tp}} = \frac{(1-x)}{\mu_l} + \frac{x}{\mu_g} \qquad (9)$$

Calculation of friction factor

For a straight capillary tube, single-phase friction factor, f_{sp}, can be calculated from Churchill correlation [34]:

$$f = 8\left[\left(\frac{8}{\text{Re}}\right)^{12} + \left(A^{16}+B^{16}\right)^{-\frac{3}{2}}\right]^{\frac{1}{12}} \quad (10)$$

where

$$A = 2.457\ln\frac{1}{\left(\frac{7}{\text{Re}}\right)^{0.9} + 0.27\,\varepsilon/d}, \qquad B = \frac{37530}{\text{Re}},$$

$$\text{Re} = \frac{Gd}{\mu}$$

Lin [35] and Churchill friction factors are used to calculate the two-phase friction factor. Lin correlation for friction factor is given by:

$$f_{tp} = \phi_{tp} f_{sp}\left(\frac{\upsilon_{sp}}{\upsilon_{tp}}\right) \quad (11)$$

where

$$\phi_{tp} = \left[\frac{\left(\frac{8}{\text{Re}_{tp}}\right)^{12} + \left(A_{tp}^{16}+B_{tp}^{16}\right)^{-\frac{3}{2}}}{\left(\frac{8}{\text{Re}_{sp}}\right)^{12} + \left(A_{sp}^{16}+B_{sp}^{16}\right)^{-\frac{3}{2}}}\right]^{\frac{1}{12}}\left[1 + x\left(\frac{\upsilon_g}{\upsilon_l} - 1\right)\right] \quad (12)$$

For coiled capillary tubes, the Mori and Nakayama friction factor equation, as shown below, is used:

$$f_{coil} = \frac{\omega\left(d/D\right)^{0.5}}{\left[\text{Re}\left(d/D\right)^{(n/2)}\right]^{1/(n+1)}}\left\{1 + \frac{\psi}{\left[\text{Re}\left(d/D\right)^{(n/2)}\right]^{1/(n+1)}}\right\} \quad (13)$$

with

$$\ln\omega = \frac{1}{n+1}\left\{\frac{1}{4}\left\langle \begin{matrix} -3\ln(2n+1) + (16n-7)\ln(2n-1) \\ -(8n-3)\left[\ln n + \ln(4n-1)\right] + (6n-1) \\ +9\ln 2 \end{matrix}\right\rangle + n\ln\alpha\right\}$$

(14)

$$\ln\phi = \frac{1}{n+1}\left\{\frac{1}{4}\left\langle \begin{matrix} 3n\ln(2n+1) - (15n+4)\ln n \\ +(19n-4)\ln(2n-1) - (7n-4)\ln(4n-1) \\ -n\ln(6n-1) - 9n\ln 2 \end{matrix}\right\rangle + n\ln\alpha\right\}$$

(15)

where the value of n is taken as 5 (for $\text{Re} \ge 10^5$) [19] and α is calculated using the general friction factor formula expressed as:

$$f_s = \alpha\,\text{Re}^{-(1/n)} \quad (16)$$

where f_s is the friction factor for a straight capillary tube.

SOLUTION METHODOLOGY

The governing equations presented here are coupled equations and are solved simultaneously by an iterative technique to calculate the capillary tube length using an implicit step-by-step numerical scheme. A special procedure is implemented to accommodate the inter-region transitions for supercritical region, subcooled region and finally two-phase region. Using the new equation of state for CO_2 and its transport property correlations available in the literature, a property code CO2PROP was developed to calculate sub-critical and super-critical thermodynamic and transport properties of carbon dioxide [36] with pressure as the marching parameter. The entire length is divided into three flow regions as stated before and length is calculated separately for each section. In mass flow rate simulation studies, mass flow rate is guessed initially. The solution process continues until the calculated capillary tube length matches the input length within a given tolerance limit.

RESULTS AND DISCUSSION

Figure 2 shows a schematic diagram of a helically coiled capillary tube under consideration. Simulation results are reported here for capillary tubes of 40 mm, 60 mm, 100 mm and 200 mm coil diameters. Simulation is carried out for two capillary tubes; tube 1: d = 1.42 mm, L = 1 m, ε = 5.76 μm and tube 2: d = 1.71 mm, L = 2.95 m, ε = 3.92 μm. Capillary tubes are selected based on test conditions. Gas cooler pressure and temperature are taken as 100 bar and 313 K, respectively. The evaporator temperature is taken as 273 K to get unchoked flow condition for the chosen capillary tube specifications. Mass flow rate is evaluated at various coil diameters and subsequently results are compared with R22 results. Table 1 shows the capillary tube specifications used in simulation.

The model is validated with test results [21] since the experimental data for adiabatic coiled capillary tube in a transcritical CO_2 system is not available. Adiabatic coiled capillary tube model with transcritical CO_2 is developed on the same principles as with R22, so validation of R22 model establishes a reasonable accuracy for the R744 model. Validation results (Figure 3) show that the agreement is excellent and the prediction accuracy of the present model is greater than the previously published one [19].

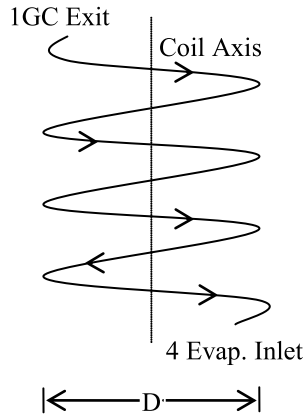


Figure 2. Schematic diagram of a helically coiled capillary tube

Table 1: Capillary tube specifications

Refrigerant	d (mm)	ε (μm)	D (mm)	L (m)	P_{gc}/P_{con} (bar)	T_{gc}/T_{con} (K)	ΔT_{sub} (K)	T_{ev} (K)
R744	1.42	5.76	40, 60,100, 200	1.0	100	313	-	273
	1.71	3.92	40, 60,100, 200	2.95	100	313	-	273
R22	1.42	5.76	40, 60,100, 200	1.0	16.53	313	3.1	283

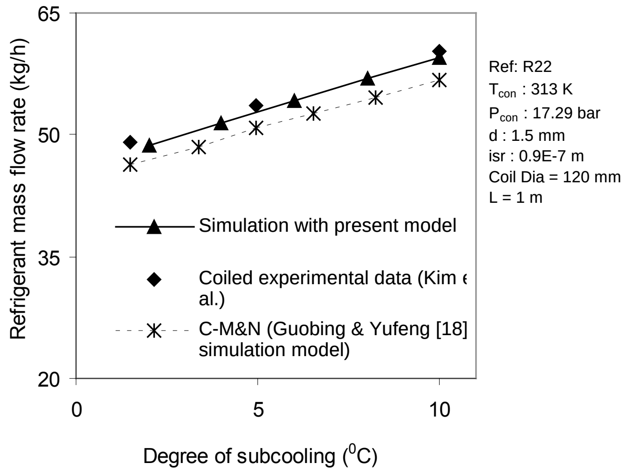


Figure 3. Validation of coiled capillary tube model

Figure 4 exhibits the mass flow rate variation with coil diameter for R744 and R22 for a set of capillary tube diameter. It is noted that as the coil diameter increases, mass flow rate of refrigerant increases. This may be attributed to the reduction in centrifugal force effect which ultimately reduces the secondary flow in case of larger coil diameter. However, the rising trend diminishes with coil diameter and beyond D = 180 mm the mass flow rate changes little. The secondary flow which is also known as *Dean effect*, affects the transfer of heat, momentum and mass in coiled tubes. Dean [37] had proposed a dimensionless number as the ratio of viscous force acting on a fluid flowing in a curved pipe to centrifugal force, which is equal to the Reynolds number times the square root of the ratio of the radius of the pipe to its radius of curvature. Coiling effect is more pronounced at higher capillary tube diameter due to higher mass flow rate. Reduction in mass flow rate due to coiling is more in case of R744 than for R22 for the chosen capillary tube diameter (d = 1.42 mm).

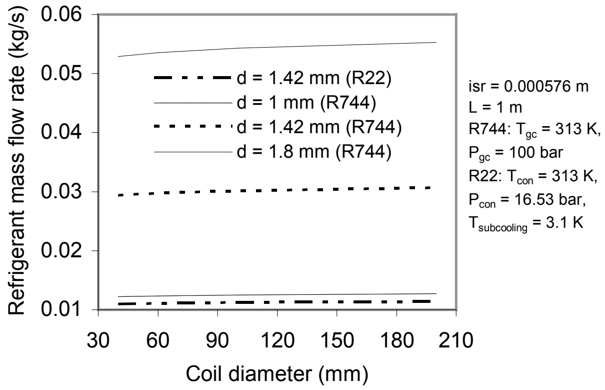


Figure 4. Mass flow rate versus coil diameter

A comparison of percentage reduction in mass flow rate as compared to that of a corresponding straight capillary tube is depicted in Figure 5. It may be inferred that at all coil diameter, percentage reduction is more significant for R744 compared to R22 for a given capillary tube specification. However, as the coil diameter increases, percentage reduction decreases for both refrigerants due to the diminishing coiling effect. Mass flow rate in the specified capillary tube (d = 1.42 mm, L = 1 m and ε = 5.76 μm) with coil diameter 40 mm is approximately 8.5% less than that of straight capillary tube for R744 in comparison to R22 for which it is approximately 7% less.

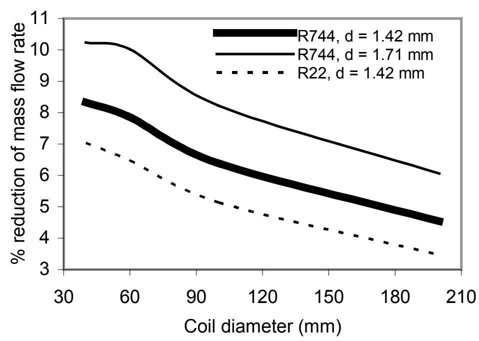


Figure 5. Variation of percentage reduction of mass flow rate in comparison to respective straight tube of R744 and R22

Mass flow rate of the adiabatic coiled capillary tube is non-dimensionalised with respect to the mass flow rate of a straight capillary tube. Figure 6 shows the variation of mass flow rate ratio with coil diameter of R744 and R22 for the chosen capillary tube. As the coil diameter increases, mass flow rate increases for both R744 and R22. However, mass flow rate ratio is substantially lower for R744 in comparison to R22. It implies that coiling effect on mass flow rate is more distinct in case of R744. The trend in variation of mass flow rate ratio with coil diameter is similar for both the refrigerants. A comparison of mass flow rate ratio of the three capillary tube diameters in case of R744 (Figure 7) shows that reduction in mass flow rate in comparison to the corresponding straight capillary tube is greater for a larger diameter capillary tube. This may be attributed to the fact that at a larger capillary tube diameter mass flow rate is higher correspondingly coiling effect is more pronounced.

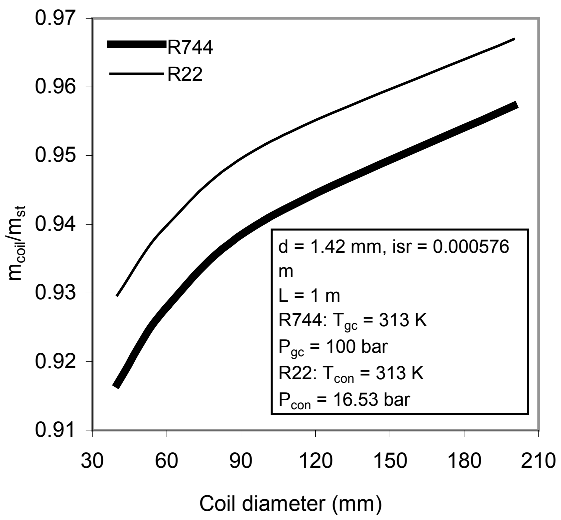


Figure 6. Variation of mass flow rate ratio with coil diameter

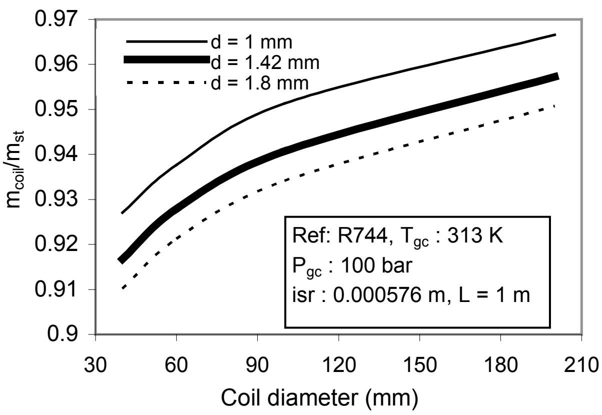
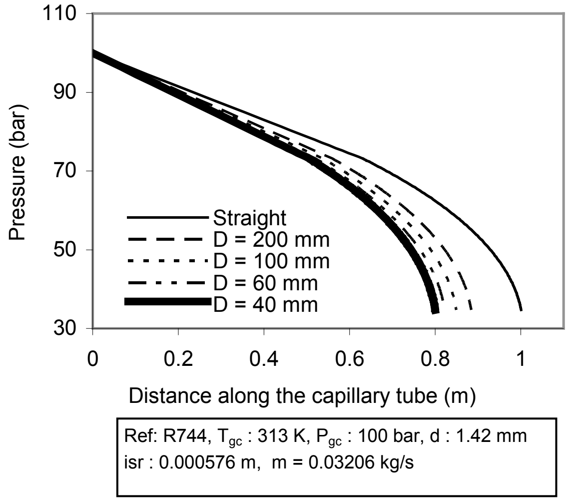
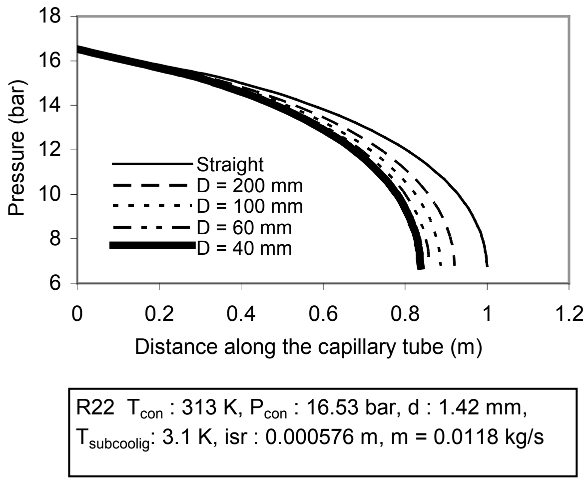


Figure 7. Comparison of mass flow rate ratio for three tube diameters with R744



(a)



(b)

Figure 8. Variation of pressure along the length of the coiled capillary tube with (a) R744 and (b) R22 refrigerant

Pressure variation along the coiled capillary tube considering McAdams correlation as the viscosity model for R744 and R22 is shown in Figures 8(a) and 8(b), respectively. As the capillary length increases, the pressure decreases almost linearly up to saturation point (i.e. for the single-phase region). On inception of vaporization, pressure drop increases rapidly an in a non-linear trend. Length of the specified capillary tube with coil diameter 40 mm is approximately 20% shorter than that of the straight capillary tube for R744 while for R22 length is shorter by 16% approximately.

CONCLUSIONS

Performance of straight and coiled adiabatic capillary tubes for CO_2 transcritical expansion is numerically investigated. Owing to transcritical expansion flow, simulation of a capillary tube using CO_2 as a refrigerant is different from other refrigerants. A special procedure has been implemented to accommodate the transition between supercritical region, subcooled region and then two-phase region. Mass flow rate of transcritical carbon dioxide flow through coiled adiabatic capillary tube is predicted employing homogeneous two-phase model with the Mori and Nakayama friction factor correlation along with Churchill equation that is fitted into the Blasius type equation.

Two capillary tubes with different inner diameter, length and coil diameter are simulated under various operating conditions. Results are compared with R22 as the refrigerant using a specified coiled capillary tube. Mass flow rate with a coil diameter of 40 mm is approximately 8.5% less than that of straight capillary tube for R744 compared to R22 for which it is approximately 7% less. Higher penalty on mass flow rate of transcritical CO_2 flow is because of its low liquid to vapour density ratio. Reduction in mass flow rate due to coiling is more pronounced at a relatively higher inner diameter due to higher mass flow rate for the given operating conditions. A shorter capillary tube will be required to match the requisite system mass flow rate with a coiled capillary tube.

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