A Model for Helical Capillary Tubes for Refrigeration Systems

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ABSTRACT

This paper presents a flow model that has been developed to design and study the performance of helical capillary tubes and to mathematically simulate a situation closer to that prevailing in practice. Homogeneous flow of two-phase fluid is assumed through the adiabatic capillary tube. The model includes the second law restrictions. The effect variation of different parameters like condenser and evaporator pressures, refrigerant flow rate, degree of sub-cooling, tube diameter, internal roughness of the tube, pitch and the diameter of the helix on the length of the capillary tube are included in the model. Theoretically predicted lengths of helical capillary tube for R-134a are compared with the length of the capillary tube needed under similar experimental conditions and majority of predictions are found to be within 10% of the experimental value.

1. INTRODUCTION

The components of a vapour compression refrigeration system never work in isolation; changes in one component affect the performance of other components as well. The performance of the system also depends on the type and quantity of refrigerant charged. With the use of new non-CFC refrigerants, the components of the system need to be redesigned. The capillary tube used as expansion device for small capacity systems also deserves proper attention for better operation, control and energy efficiency of the refrigeration system.
The selection of the proper diameter and length of a capillary for a given application is largely a trial and error process. In practical applications the capillary is not used as straight, instead a helical or spiral shape is preferred as this being compact. For a particular diameter and length, the capillary performance varies with the capillary tube configuration as well. Significant efforts have been made to model straight capillary tubes whereas the tubes used in refrigeration systems are coiled. Modeling efforts on coiled capillary tubes are fairly limited. This paper is an attempt to model an adiabatic helical capillary tube and compare it with the experimental data available in the literature for R-134a and those obtained for the purpose.
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2. BACKGROUND INFORMATION

2.1 Flow Through Straight Adiabatic Capillary Tubes

First recognized work in this area is due to Steabler [1948], who performed extensive tests and presented experimental data graphically for Freon-12 and proposed a correlation: $L_1 = L_0(d_1/d_0)^{4.6}$. Later, Bolstad and Jordan [1949], Marcy [1949], Hopkins [1950], and Whitesel [1957] contributed significantly. The work of Whitesel and Hopkins coupled together formed the basis for the ASHRAE charts for capillary tube selection. Churchill[1977] suggested equations for friction factor which are widely referred. Lin et al. [1991] developed correlation to calculate the single-phase and two-phase flow friction factors. Wong et al. [1994] developed a model for R-134a and compared it with available experimental data. Bansal et al. [1998] presented a two-phase flow model CAPIL assuming homogeneous flow through a capillary tube. Melo et al. [1999] suggested empirical correlation for the modeling of R –134a flow through capillary tubes. Melo et al. [1999] conducted extensive experiments using three different refrigerants and presented their data. They also did the dimensional analysis to develop empirical correlations. Chen et al. [1999] suggested some rating correlations for flow of R-134a through capillary tubes.

2.2 Fluid Flow through Coiled Tubes

First theoretical study on flow of incompressible fluids in curved pipes was done by Dean [1927]. He proposed a non-dimensional number $Re\ (d/D)^{0.5}$ known as Dean Number to characterize the flow in coils. There after several researchers worked on curved pipes but the efforts on coiled capillary are quite limited. Ali [2001] presented the state of the art by giving various correlations available for helical coils. Most of the available correlations were for the ratio of friction factor for the coil to the friction factor for the flow of the same fluid in a straight length of the same diameter tube. He also did an experimental study to obtain pressure drop versus flow rate data for different helical coils and made an attempt to better characterize dimensionless group for steady flow of fluids in helically coiled tubes. Paliwal et al. [2004] presented a homogeneous flow model for adiabatic capillary tube of spiral shape considering the effect of all the common parameters. They conducted experiments to compare model with experimental data. Deodhar et al. [2006] performed experiments and gave empirical expression for helical capillary.

3 EXPERIMENTAL SETUP

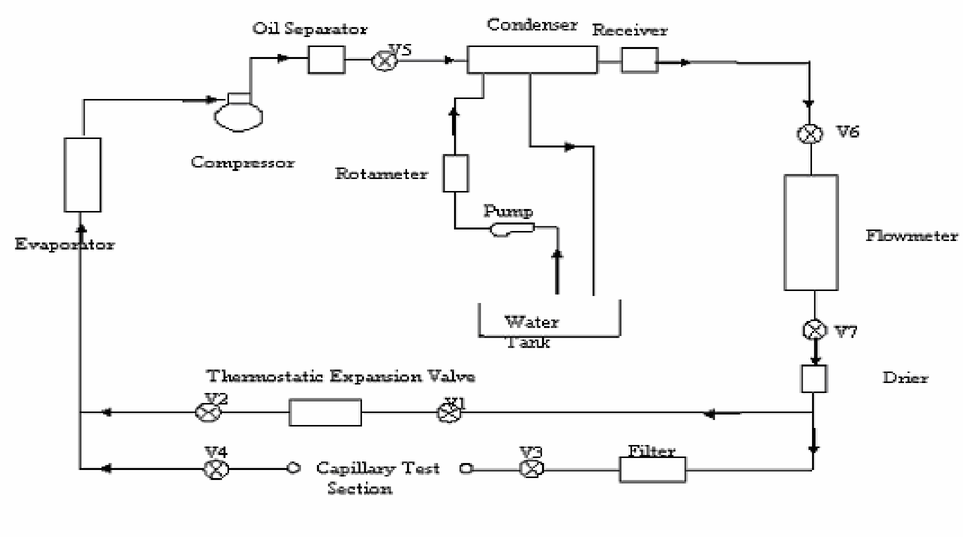

Figure 1. Schematic diagram of the experimental setup

The test rig basically consists of a single stage vapour compression refrigeration system using an open compressor of 2 TR capacity. It was provided with two options of using an expansion valve or capillary tube as expansion device. The refrigerant used was R-134a. The schematic of the test rig is shown in figure 1. The test rig was instrumented to measure the relevant temperatures and pressures at different points and also the flow rate. Pressure distribution along the capillary tube was measured using pressure taps at the pre-decided location.

4. THEORETICAL MODELLING

In a coiled tube, the centrifugal force of a flowing fluid produces a pressure gradient in a cross section. This pressure gradient yields secondary flows. The secondary flows cause a larger amount of pressure drop or heat transfer rate than that for a straight tube.

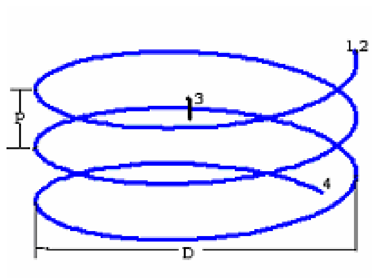

Figure 2 Helical capillary tube.

The flow through a helical capillary is divided into two distinct regions: a liquid single-phase and a two-phase region. In figure 2 point 1 denotes condenser exit and point 2 denotes the capillary inlet. There is a small pressure drop from point 1 to 2 due to sudden contraction to capillary diameter. The refrigerant is sub-cooled between points 2 and 3, saturated liquid at point 3 and is a two-phase mixture between points 3 and 4. Point 4 denotes capillary exit.

Following assumptions are made in the model:

1. Capillary tube is of constant inside diameter and surface roughness.
2. Flow through capillary tube is steady, adiabatic and one dimensional.
3. Metastable flow phenomenon is neglected.
4. Pure refrigerant flowing out of the condenser is either saturated or subcooled.
5. Entrance effects (Entrance length) are neglected.
6. Homogeneous two phase flow is assumed.

The fundamental equations governing flow through the capillary tube are the conservation of mass, momentum and energy. In modeling the flow, these equations along with the second law of thermodynamics have to be satisfied for both the single phase and two phase regions.

4.1 Subcooled Single Phase Region

The pressure loss due to sudden contraction at capillary inlet can be given as

$$p_1 - p_2 = k\, V_2^{\,2} / 2v_2 \tag{1}$$

where, k is contraction factor, due to sudden contraction at the capillary inlet.
For flow of subcooled liquid through capillary tube, for any two points ‘a’ and ‘b’

$$V_a = V_b = V_2 \quad \text{and} \quad h_a = h_b = h_2 \tag{2}$$

The pressure drop for a differential length is given by

$$dp = f\left(V^2/2vd\right)dL + GdV \tag{3}$$

where, f is the Darcy’s friction factor and dV is the change in velocity along length dL, (dV =0 for liquid region)

$$dL = \left(2vd/fV^2\right)dp \tag{4}$$

The transition Reynolds number is increased in a coil and is given as

$$\mathrm{Re}_{trans} = 2300\left[1 + 8.6\left(d/2\rho\right)^{0.45}\right] \tag{5}$$

where, ρ is the radius of curvature given by

$$\rho = \frac{D^2 + \left(\dfrac{P}{\pi}\right)^2}{2D} \tag{6}$$

where, P is the pitch of helical coil and D is the radius of helix.
Above expression takes care of the effect of torsion.
For laminar flow, the ratio of coil to straight pipe friction factor is given by Collier[1972].

$$f/f_s = \left\{1 - \left[1 - \left(11.6/K\right)^{0.45}\right]^{2.22}\right\}^{-1} \tag{7}$$

where $K = \mathrm{Re}\left(d/2\rho\right)^{0.5}$, and f_s is the friction factor for a straight tube of the same diameter.
And for turbulent flow, the ratio of coil to straight pipe friction factor for $\left[\mathrm{Re}\left(d/2\rho\right)^2\right] > 6$ is given by

$$f/f_s = \left[\mathrm{Re}\left(d/2\rho\right)^2\right]^{0.05} \tag{8}$$

To calculate the value of f_s, Churchill’s equation [1977] is used as given below

$$f_s = 8\left[\left(8/\mathrm{Re}\right)^{12} + 1/\left(A+B\right)^{3/2}\right]^{1/12} \tag{9}$$

where, $\mathrm{Re} = Gd/\mu$,

$$A = \left[2.457 \ln\left(\frac{1}{\left(7/\mathrm{Re}\right)^{0.9} + 0.27\varepsilon/d}\right)\right]^{16} \tag{10a}$$

and,

$$B = \left(\frac{37530}{\mathrm{Re}}\right)^{16} \tag{10b}$$

4.2 Two Phase Region

From conservation of mass for any point ‘a’ between ‘3’ and ‘4’

$$V_a/v_a = V_3/v_3 \tag{11}$$

The energy equation applicable to any point 'a' between '3' and '4' in terms of mass flux is

$$h_a + v_a{}^2 G^2/2 = h_3 + V_3{}^2/2 \tag{12}$$

where, h_a is given by $h_a = h_f + x h_{fg}$

Substituting the value of h_a in "Eq.(12)" and solving it for x we obtain,

$$x = \frac{\left(-h_{fg} - v_f v_{fg} G^2\right) + \sqrt{\left(h_{fg} + v_f v_{fg} G^2\right)^2 - 2 v_{fg}{}^2 G^2 A_x}}{v_{fg}{}^2 G^2} \tag{13}$$

where,

$$A_x = \left(h_f + v_f{}^2 G^2/2 - h_3 - V_3^2/2\right) \tag{13b}$$

Using the dryness fraction x and the thermodynamic properties, velocity and friction factor at every point is determined and used in further calculations.

The pressure drop for a differential length is given by

$$dp = f_{tp}\, V^2/2vd\; dL + G dV \tag{14}$$

where, f_{tp} is the Darcy's friction factor for two phase flow based on homogeneous model and dV is the change in velocity along length dL.

Rearranging above equation, we get

$$dL = 2vd/f_{tp}V^2\left(dp - GdV\right) \tag{15}$$

The two-phase friction factor applicable to "Eq.(14)" is calculated as suggested by Lin et al [1991], using the following equation:

$$f_{tp} = \phi_{fo}{}^2 f_{fo}\, v_f/v_{tp} \tag{16}$$

where, f_{tp} and f_{fo} are given as

$$f_{tp} = 8\left[\left(\frac{8}{\mathrm{Re}_{tp}}\right)^{12} + \frac{1}{\left(A_{tp} + B_{tp}\right)^{3/2}}\right]^{1/12} \tag{17a}$$

$$f_{fo} = 8\left[\left(\frac{8}{\mathrm{Re}_{fo}}\right)^{12} + \frac{1}{\left(A_{fo} + B_{fo}\right)^{3/2}}\right]^{1/12} \tag{17b}$$

Substituting the values of f_{tp} and f_{fo} from "Eq.(17a)" and "Eq.(17b)" in the "Eq.(16)", we obtain

$$\phi_{fo}{}^2 = \left[1 + x\left(\frac{v_g}{v_f} - 1\right)\right]\left[\frac{\left(\frac{8}{\mathrm{Re}_{tp}}\right)^{12} + \frac{1}{\left(A_{tp} + B_{tp}\right)^{3/2}}}{\left(\frac{8}{\mathrm{Re}_{fo}}\right)^{12} + \frac{1}{\left(A_{fo} + B_{fo}\right)^{3/2}}}\right] \tag{18}$$

where, A_{tp}, A_{fo}, B_{tp}, and B_{fo} are given by "Eq.(10a)" and "Eq.(10b)" by replacing Re by Re_{tp} and Re_{fo} defined by

$$\mathrm{Re}_{tp} = \frac{Gd}{\mu_{tp}} \qquad \text{and} \qquad \mathrm{Re}_{fo} = \frac{Gd}{\mu_{fo}} \tag{19}$$

The two phase flow multiplier for calculating frictional pressure drop in coils is taken same as that for straight tube flow given by "Eq.(18)".The value of f_{fo} for a helical coil is calculated by the method described earlier for single phase flow. Equations (4) and (15) are integrated numerically to determine the single phase length L_{sp} and two phase length L_{tp} respectively. L_{sp} and L_{tp} are added to get the total length of the capillary tube.

The program is developed using MATLAB. Conditions for chocked flow are taken care and the second law restrictions are imposed. Thermo physical property values for the refrigerant are taken from ASHRAE tables.

5. DISCUSSION

Figure 3 shows the pressure variation along the capillary length for various coil diameters. The values of other parameters are p_{cond} = 0.93 MPa, p_{evap} = 0.19 MPa, degree of subcooling = 3.4 °C, d = 0.77 mm, tube roughness = 0.75µm, k = 0.5, mass flow rate = 3.32 kg/h, and the axial pitch of the coil = 2 mm. The figure shows that with the increase in the coil diameter the length of the capillary tube needed for the required expansion increases. The rate of increase is smaller at the larger values of the coil diameter.

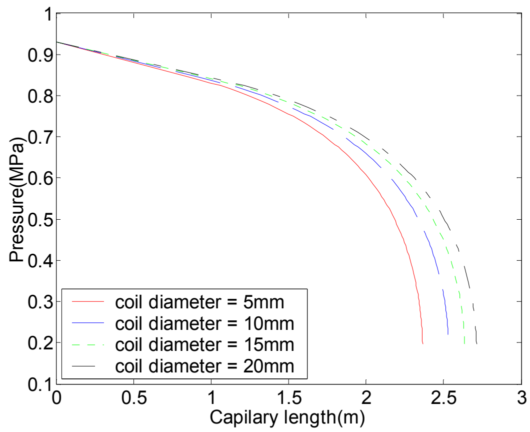

Figure 3: Effect of coil diameter on the pressure distribution along the capillary length.

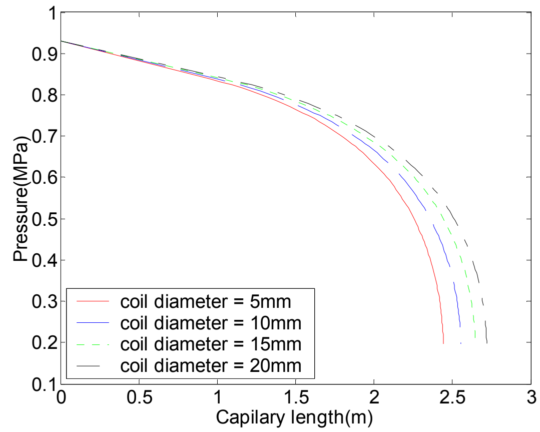

Figure 4: Effect of coil diameter on the pressure distribution along the capillary length at a larger value of axial pitch.

Figure 4 also shows the pressure variation along the capillary length for various coil diameters. The value of other parameters is the same as that for the figure 1, except the axial pitch which is 10mm in this case. Apart from the conclusions made from figure 1 it can also be observed that the effect of change in coil diameter on the required capillary length is smaller at larger value of axial pitch

Figure 5 shows the pressure variation along the capillary length for various values of axial pitches. The values of other parameters are p_{cond} = 0.93 MPa, p_{evap} = 0.19 MPa, degree of subcooling = 3.4 °C, d = 0.77 mm, tube roughness = 0.75µm, k = 0.5, mass flow rate = 3.32 kg/h, and the coil diameter = 5mm. It can be observed that with the increase in the axial pitch the required length of the capillary tube increases. The rate of increase is slightly higher at the larger values of the coil diameter.

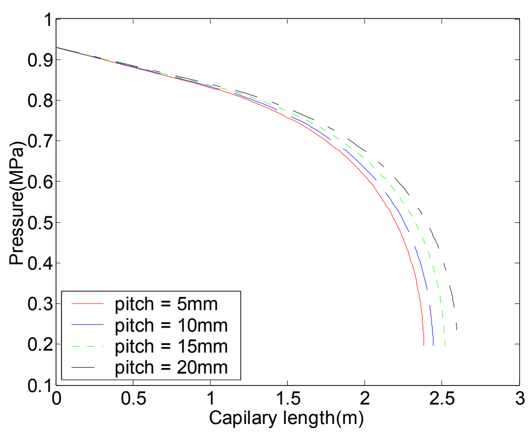

Figure 5: Effect of axial pitch on the pressure distribution along the capillary length.

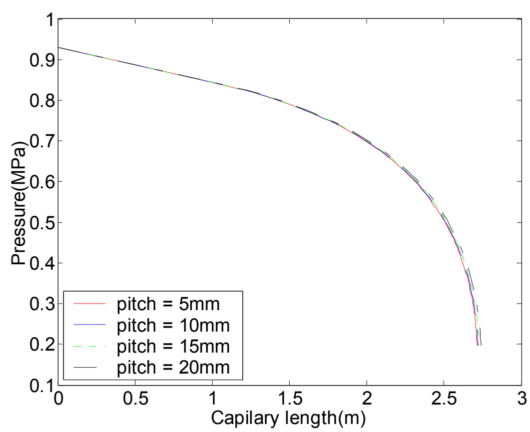

Figure 6: Effect of axial pitch on the pressure distribution along the capillary length at a larger value of coil diameter.

Figure 6 also shows the pressure variation along the capillary length for different values of axial pitch. The value of other parameters is the same as that for the figure 3, except the coil diameter which is 20mm in this case. Similar conclusions as figure 3 can be made from figure 4. It can also be observed that the effect of change in axial pitch on the required capillary length is negligibly small at larger value of coil diameter.

Figure 7 shows the variation of required capillary length with coil diameter for two representative values of axial pitches. The values of other parameters are p_{cond} = 0.93 MPa, p_{evap} = 0.19 MPa, degree of subcooling = 3.4 °C, d = 0.77 mm, tube roughness = 0.75µm, k = 0.5, mass flow rate = 3.32 kg/h. The figure shows that with increase in the coil diameter a larger capillary tube is needed and the rate of increase decreases with increase in values of the coil diameter. At large values of coil diameter, the effect of axial pitch is very small.

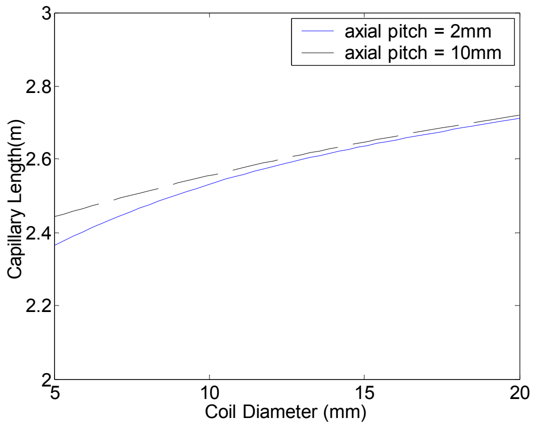

Figure 7: Effect of coil diameter on capillary length for different values of axial pitch.

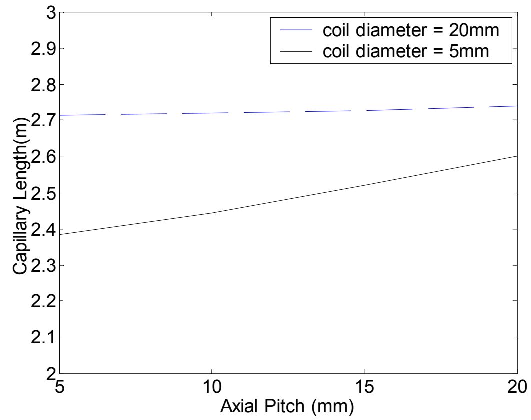

Figure 8: Effect of axial pitch on capillary length for different values of coil diameter

Figure 8 shows the variation of required capillary length with axial pitch for two representative values of coil diameters. The other parameters are p_{cond} = 0.93 MPa, p_{evap} = 0.19 MPa, degree of subcooling = 3.4 °C, d = 0.77 mm, tube roughness = 0.75µm, k = 0.5, mass flow rate = 3.32 kg/h. From figure it may be concluded that with the increase in the axial pitch a larger capillary tube is needed and the rate of increase is higher at larger values of the axial pitch. At large values of coil diameter the effect of change in the axial pitch is very small.

Figure 9 shows the pressure variation along the capillary length using experimental data as well as theoretical model for a capillary tube of length 5.5m under two operating conditions. The geometric parameters for the capillary are diameter =1.62m with a typical surface roughness of 1.2 micron, pitch = 10mm and helix diameter = 66mm. The values of other parameters in case 1 are p_{cond} = 0.69 MPa, p_{evap} = 0.41 MPa, subcooling = 1.0 °C, mass flow rate = 11.19 kg/h. In case 2 the values are p_{cond} = 0.64 MPa, p_{evap} = 0.36 MPa, subcooling = 1.1 °C, mass flow rate = 10.71 kg/h. Distribution predicted by the model is very close to the actual pressure distribution.

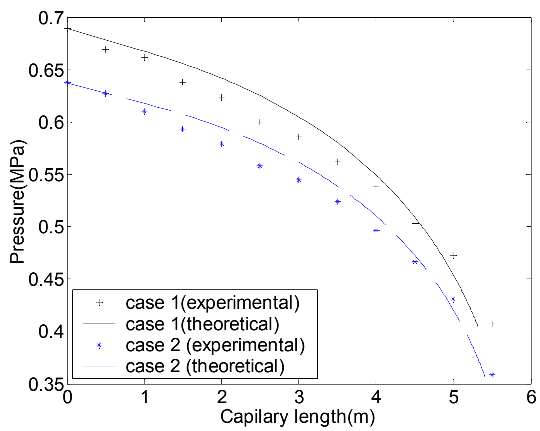

Figure 9: Actual and theoretically predicted pressure distributions for capillary

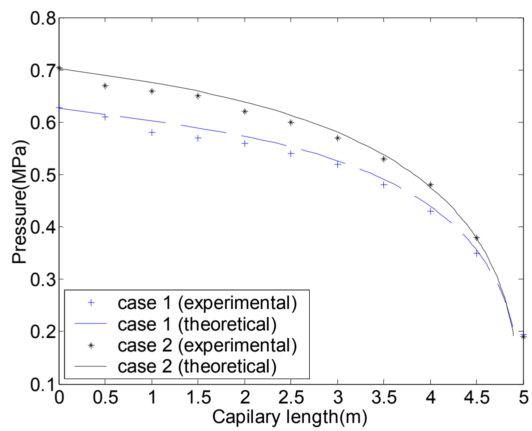

Figure 10: Actual and theoretically predicted pressure distributions for capillary with smaller helix diameter.

Figure 10 also shows the actual and theoretically predicted pressure distributions for capillary of length 5m under two operating conditions. Other geometric parameters are diameter = 1.01 m with a typical surface roughness of 1.2 micron, pitch = 6mm and helix diameter = 16mm. The values of other parameters in case 1 the values are p_{cond} = 0.70 MPa, p_{evap} = 0.19 MPa, degree of subcooling = 0.9 °C, mass flow rate = 3.52 kg/hr. In case 2 the values are p_{cond} = 0.63 MPa, p_{evap} = 0.195 MPa, degree of subcooling = 1.6 °C, mass flow rate = 3.38 kg/h.

Table 1 compares the capillary length predicted by the model with the actual length of capillary tubes used in the experiment. In the model a typical value of inside surface roughness of 1.2 micron is used throughout.
From the table it can be observed that the length of the capillary tube predicted by the model in majority of the cases is within the ± 10% limit of the actual value, which is quite reasonable.

Table 1: Comparison of experimental and theoretically predicted values of capillary length for R-134a.

Se. No.	d mm	Pitch mm	Helix Diameter mm	P_{cond} MPa	P_{evap} MPa	Sob-Cooling °C	Mass Flow Rate Kg/h	Length Predicted by model m	Actual Length m	Error in prediction (%)
1	1.62	10	66	0.91	0.21	1.0	15.12	5.14	6.0	-14.33
2	1.62	10	66	0.83	0.19	0.8	13.37	6.03	6.0	0.50
3	1.62	10	66	0.69	0.41	1.0	11.19	5.32	5.5	-3.27
4	1.62	10	66	0.63	0.36	1.1	10.71	5.41	5.5	-1.64
5	1.01	6	16	0.98	0.16	1.2	3.89	6.51	6.0	8.50
6	1.01	6	16	1.01	0.10	2.1	4.30	6.05	5.5	10.00
7	1.01	6	16	0.7	0.19	0.9	3.52	4.89	5.0	-2.20
8	1.01	6	16	0.63	0.195	1.6	3.38	4.91	5.0	-1.80
9	1.01	6	16	0.82	0.13	1.4	4.10	4.76	4.5	5.80
10	1.01	6	16	0.81	0.32	1.3	4.01	4.68	4.5	4.00

Table 2 compares the length of helical capillary tube predicted by model for two different values of *D/d* ratios of 20 and 40 with the experimental value of straight capillary length for R-134a. The *P/d* ratio is 5 in all the cases. It can be observed that for a *D/d* ratio of 20 the length required is 15-30% smaller as compared to straight. For *D/d* of 40 the length required is approximately 8-25% smaller in comparison to straight and 7% larger as compared to that with *D/d* of 20 due to smaller curvature.

Table 2. Comparison of length of helical coil predicted by model with experimental values for straight capillary for R-134a.

Se. No.	d mm	ε μm	p_{cond} MPa	p_{evap} MPa	Sub-cooling ^{0}C	Mass Flow Rate kg/h	$L_{straigh}$ actual by Melo [1999]	$L_{helical}$ by model for D/d=20	$L_{helical}$ by model for D/d =40	Percentage change for helical with D/d = 20	Percentage change for helical with D/d = 40
1	1.05	0.72	1.12	0.13	4.0	11.18	2.03	1.55	1.66	-23.65	-18.23
2	1.05	0.72	1.13	0.14	8.2	12.14	2.03	1.72	1.85	-15.27	-8.87
3	1.05	0.72	1.43	0.16	6.2	13.08	2.03	1.73	1.85	-14.78	-8.87
4	0.87	0.78	1.40	0.11	4.4	6.87	2.97	2.08	2.23	-29.97	-24.92
5	0.87	0.78	1.51	0.12	7.2	7.45	2.97	2.25	2.41	-24.24	-18.86

CONCLUSIONS

It is observed that the effect of torsion (the axial pitch) is very small as compared to the helix diameter, especially at higher helix diameter; the effect of variation of pitch is negligible. Pressure distribution given by model is compared with the pressure distribution obtained for an actual capillary and they are found to be in agreement.
The model is also validated by comparing the theoretical length predicted by model with the actual capillary length. The majority of predicted values are found to be within ± 10% limit.

The length predicted by the model for helical capillary tube with a D/d ratio of 20, is found to be 15-30% less as compared to actual length of straight tube.

NOMENCLATURE

A	cross sectional area of the capillary tube	(m^2)	ε	roughness of the inside tube surface	(m)
D	Diameter of the helical coil	(m)	μ	viscosity of the refrigerant	(kg/m.s)
G	mass flux, kg/s	(m^2)	ρ	radius of curvature of the helical coil	(m)
L	length of the capillary tube	(m)	ϕ_{fo}^{2}	two phase flow friction factor multiplier	(-)
P	pitch	(m)		**Subscripts**	
Re	Reynolds number	(-)	cond	condenser	
V	velocity	(m/s)	evap	evaporator	
d	internal diameter of the capillary tube	(m)	f	saturated liquid state	
f	Darcy's friction factor	(-)	fg	difference between gas and liquid	
h	specific enthalpy of the refrigerant	(J/kg)	fo	total flow assumed in liquid phase	
k	contraction factor	(-)	g	saturated gas state	
p	refrigerant pressure	(Pa)	s	straight	
v	specific volume	(m^3/kg)	tp	two phase	
x	quality (dryness fraction) of the refrigerant	(-)	trans	transition	

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