Numerical Study of Refrigerant Flow in Capillary Tube Using Refrigerant (R134a)

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Abstract

The present research aims at building of a mathematical model using (Engineering Equation Solver) (EES) software to analyze the flow of the refrigerant R134a into two configurations of adiabatic capillary tubes (straight and helical coiled tubes) which are used widely in a household refrigerator, freezer and water cooler. The governing equations which depend on the conservation of mass, momentum, and energy have been coded in order to calculate the length of the single-phase flow region. Moreover, the two-phase region has been divided numerically depending on homogenous flow assumption after determining the boundary conditions and calculating the physical variables at each point of the numerical divisions by using successive substitution method using iteration loops. Finite difference scheme, the length of the two-phase region and the total length for the capillary tube have also been determined. The results show that the behavior of the flow is similar for both forms with regard to the distribution of variables (pressure and temperature drop, dryness fraction, velocity and entropy) over the whole length , while the helical coiled tube length for all cases is shorter than the straight tube at the same conditions. The main parameters that affect the size of capillary tube and the behavior of refrigerant through it have been shown. The tube length increased with the increase of (condenser temperature, degree of sub-cooling and tube diameter) and decreased with the increase of mass flow rate and roughness. The helical tube length increased by increasing coil diameter and coil pitch . The results of the present study agree with the experimental data of previous works with error not exceeding ±10% .

Keywords : Adiabatic capillary tube, Household refrigerators, R134a, EES, Helical, Two-phase flow.

دراسة عددية لجريان مائع التثليج في الأنبوبة الشعرية باستخدام مائع التثليج R134a

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الخلاصة

يتضمن البحث الحالي بناء أنموذج رياضياتي لتحليل جريان مائع التثليج R134a في شكلين من الأنابيب الشعرية الأديباتية (المستقيمة والملفوفة بشكل لولبي) المستخدمة بشكل واسع في الثلاجات والمجمدات وبرادات الماء المنزلية باستخدام برنامج حل المعادلات الهندسية (EES) . لقد تمت برمجة المعادلات الحاكمة المعتمدة على معادلات حفظ الكتلة والطاقة والزخم, لحساب طول المجال أحادي الطور ومن ثم تقسيم المجال ثنائي الطور عددياً بالاعتماد على فرضية الجريان المتجانس بعد حساب الظروف المحددة وحساب المتغيرات الفيزياوية في كل نقطة من التقسيم العددي بطريقة التعويض المتعاقب وذلك بعمل حلقات تكرارية, وحساب طول المجال الثنائي الطور بطريقة الفرق المحدد وبعدها تم حساب الطول الكلي للأنبوب . تبين من النتائج أن سلوك الجريان متشابه في كلا الشكلين من حيث توزيع المتغيرات (انحدار الضغط ودرجة الحرارة ونوعية البخار والسرعة والإنتروبي) على طول الأنبوب . لكن طول الأنبوب الملفوف يكون في جميع الحالات أقصر من طول الأنبوب المستقيم عندما تكون المعالم الأخرى ثابتة . وتم توضيح المعالم الرئيسية التي تؤثر على اختيار الحجم المناسب للأنبوب وعلى سلوك جريان المائع خلاله . وبينت النتائج أن طول الأنبوب يزداد بزيادة كل من (درجة حرارة المكثف ودرجة التبريد الدوني وقطر الأنبوب) ويقل بزيادة (معدل جريان الكتلة وخشونة السطح الداخلي للأنبوب) وكذلك يزداد بزيادة (قطر اللفة وخطوة اللفة) . وأظهرت نتائج الدراسة الحالية توافقاً جيداً بالمقارنة مع البيانات التجريبية في الدراسات السابقة وبنسبة خطأ لاتتجاوز ±10% .

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List of abbreviations

Symbol	**Meaning**	**Unit**	**Symbol**	**Meaning**	**Unit**
A	Internal cross section area of capillary tube	m^2	V	Refrigerant velocity	m/s
ΔL	Incremental length in the two phase region	m	x	Dryness fraction (Vapor quality)	-----
ΔP	Pressure drop in two phase region	bar	**Greek Letters**		
C_P	Specific heat at constant pressure	kJ/kg.K	Δ	Difference	-----
d	Internal capillary tube diameter	mm	ε	Relative roughness of internal tube wall	-----
D_C	Coil pitch for helical capillary tube	mm	μ	Average dynamic viscosity	kg/m.s
De	Dean number	-----	ρ	Average density	kg/m^3
e	Internal wall roughness	µm	ν	Average specific volume	m^3/kg
$\Delta T_{sub.}$	Degree of subcooling	^0C	τ_w	Shear stress for flow inside tubes	N/m^2
f_s	Friction factor for straight tube	-----	**Subscripts**		
f_c	Friction factor for helical tube	-----	**Symbol**	**Meaning**	
G	Mass flux	$kg/m^2.s$	c	Coiled capillary tube	
h	Average enthalpy	kJ/kg	chock	Chock point	
K	Entrance losses coefficient	-----	cond.	Condenser	
L_{sp}	Single phase region length	m	evap.	Evaporator	
L_{tp}	Two phase region length	m	exit	Tube exit	
$L_{capillary\ tube}$	Total capillary tube length	m	f	Saturated liquid refrigerant	
$\dot{m}$	Mass flow rate	kg/s	g	Saturated Vapor refrigerant	
P	Pressure	bar	i	Number of iteration in the two phase region	
p	Coil pitch for helical tube	mm	max.	Maximum value	
R_e	Reynolds number	-----	s	Straight capillary tube	
s	Average entropy	kJ/kg.K	sp	Single phase region	
T	Temperature	^0C	tp	Two phase region	

Introduction and Literature Review:

The main functions of an expansion device used in refrigeration system are to reduce pressure from high condenser pressure to low evaporator one, and to regulate the refrigerant flow from the high-pressure liquid line into the evaporator at a rate equal to the evaporation rate in the evaporator. The important type of expansion device that are widely used in small vapor compression refrigeration systems is capillary tube; it is one of the main parts of the refrigeration system. It is commonly used as expansion and refrigerant flow rate controlling to evaporator to reduced the refrigerant pressure from high pressure (Condenser pressure) to low pressure (Evaporator pressure). It is a long hollow tube of drawn copper with a small inside diameter of between (0.2-3 mm) and a length of (1 to 6 m) that connects the outlet of the condenser to the inlet of the evaporator. In practical application, the capillary tube is coiled in many configurations to reduce space. It works on the principle that the liquid refrigerant passes through it much more readily than vapor, which it is due to condensate in the condenser. The capillary tube allows the pressure in the system to equalize during the off cycle (period of stopping the compressor), reducing the compressor starting torque requirements. It operates generally in acceptable performance over a wide range of operating conditions. It has many advantages such as simplicity, has no moving parts and inexpensive, which make it common in many small size refrigeration systems of single-compressor/single-evaporator. Its applications include refrigeration system of (10 kW) capacity such as household refrigerators, freezers, dehumidifiers, and room air conditioners. Lastly its application extended to larger units, such as unitary air conditioners up to (35.2 kW) .

Although the capillary tube physical configuration is very simple, the flow behavior of refrigerant through it is a very complex phenomenon. The refrigerant enters the capillary tube in subcooled liquid state, and with continuous flowing of this liquid the pressure will drop linearly due to friction, while the temperature remains constant . When this pressure drops to a level lower than the pressure of vaporization for the refrigerant, some of liquid will evaporate in a process called flash process, then the two-phase flow process begins proceeds to the tube exit .

Experimental Studies:

Many experimental and theoretical studies and researches have been done to study the refrigerant flow through capillary tubes. Keeping in view the importance of capillary tube in low capacity refrigeration systems, it is increasingly important for the design engineers, in general, and the researchers, in particular, to be conversant with the flow behavior of various refrigerants through the capillary tubes of different sizes and geometries under adiabatic and diabatic flow conditions . Lin et al.(1991) [1] have done an experimental investigation about local pressure drop during vaporization process of (R12) through adiabatic capillary tubes . They presented a theoretical model to predict local pressure drop using the theoretical equation for pressure drop. Another experimental research presented by Melo et al. (1999) [2] includes the effect of the main various parameters on mass flow rates through the capillaries. It hinges upon using three different types of refrigerants, namely (CFC-12, HFC-134a and HC-600a) at different condenser pressure and levels of subcooling . The theoretical model uses a conventional dimensional analysis to derive correlations to predict the mass flow rates for different refrigerants. The predictions from the developed correlations were found to be in good agreement with the measured data in other studies in the literature. Kim et al. (2002) [3] have done an experimental investigation to study the performance of refrigerant flow (R22, R407C and R410A) in several capillary tubes for air- conditioners. (28) Capillary tube with different lengths and inner diameters was selected as test sections. Mass flow rate through the capillary tube was measured for different condensing temperatures and degrees of subcooling

. Dimensionless correlation was developed, and its results were compared with experimental data and (ASHRAE) design correlation and proved to be in good agreement. Other studies for coiled capillary tube have been done recently. Khan et al. (2008) [4] for example have done an experimental study for helical coiled capillary tube using refrigerant (R134a) to investigate the effect of coil pitch on the mass flow rate through the tube when coiled diameter was constant (140 mm). The same researchers Khan et al. (2008) [5] have also done an experimental investigation to study the flow of refrigerant (R134a) inside an adiabatic spirally coiled capillary tube. The effect of various geometric parameters on the mass flow rate has been investigated. It has been concluded that the effect of coiling of capillary tube reduces the mass flow rate by (5-15%) compared to those of the straight capillary tube operating under similar conditions.

Theoretical Studies:

In addition to the experimental researches presented above, theoretical studies have also been done using developed numerical models to study the flow of various types of refrigerants through the adiabatic capillary tubes. Wong and Ooi (1995) [6] have presented a model that includes a comparison of the homogenous flow and separated flow in adiabatic capillary tube for refrigerant (R12). Comparisons of the predicted results between the two-models are presented, together with experimental results from previous works. The results show that the separated flow model gives a better prediction compared to the homogeneous flow model. Bansal and Rupasinghe (1997) [7] presented a model to study the performance of capillary tube using the homogeneous adiabatic two-phase flow assumption called it (CAPIL), The effect of various design parameters results with various tube length presented to show the real flow behavior in comparison with previous works . They were validated with the previous results over a range of operating conditions within ±10%. Ibrahim and Abdul-Wahed (2008) [8] have carried out a research to study and analyses the refrigerant flow behavior in the capillary tube, and the distribution of some variables along the tube by using (FLUENT) program for the simulation, and using the separated two-phase flow assumption . Imran (2008) [9] has presented a research for building a mathematical model for separated flow through adiabatic capillary tube using refrigerant (R22) and its alternative (R407C) . The results of the calculation were compared with the experimental data in the technical literature in order to validate the developed model, and its capability of using it to analyze the capillary tube performance to optimize and control the refrigerant in the refrigeration equipment. A few theoretical studies however, have also discussed the flow in adiabatic coiled capillary tubes . Mittal et al. (2009) [10] have presented a modern study about the metastable liquid region for the adiabatic flow of refrigerant (R22) and its alternatives (i.e., R407C and R410A) through a spiral capillary tube . The effect of the pitch of spiral on the mass flow rate of refrigerant and capillary tube length has been investigated. A comparison of the flow characteristics of this refrigerant has been made at different operating conditions. The results of the present research have also been compared with the experimental results of (Khan et al. [5]). Another modern study is developed to tackle the two-phase flow of refrigerant (R22) and its alternatives (R407C and R410A) through helically coiled capillary tubes by Chingulpitak and Wongwises (2010) [11] . The results obtained from this model show reasonable agreement with the experimental data.

Mathematical analysis for refrigerant flow through capillary tube:

Assumptions used to analyze the flow through adiabatic capillary tube:-

1. The capillary tube inner diameter and surface roughness are constant.
2. The capillary tube is perfectly insulated.
3. Pure refrigerant is flowing through the capillary tube (i.e. refrigerant is free of oil).
4. One dimensional and steady state flow.
5. The flow is homogeneous in the two-phase region.
6. Flash point lies on saturated liquid line (i.e. the metastable flow region phenomena is not considered).
7. The capillary tube is horizontal (i.e. the influence of gravity is ignored).
8. The capillary tube is adiabatic (i.e. there is no heat transfer process from the tube or vice versa (Q=0))

Governing Equations:-

The governing equations used to describe the physical flow behavior through all capillary tube forms depend principally on the basic equations of the conservation of mass, momentum and energy, and the basic fundamentals of fluid flow through the tubes, which are explained by considering an infinitesimal control volume as shown in Fig.(1), and applying the conservation equations as follows :

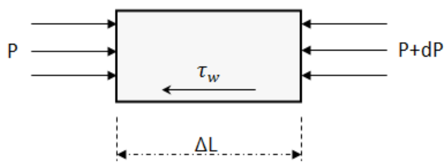


Fig.(1) the control volume (C.V) for refrigerant flow in capillary tube .

Mass balance :

$$\dot{m} = \rho . A . V \qquad \ldots\ldots\ldots\ldots\ldots (1)$$

By applying the continuity equation above, with constant cross section, the mass flux is constant (dG=0) along the flow through tube therefore :

$$\therefore \quad \frac{d\rho}{\rho} = -\frac{dV}{V} \qquad \ldots\ldots\ldots\ldots\ldots (2)$$

Momentum balance :

By applying the principle of momentum conservation on the control volume Fig.(1) the following equation is obtained :

$$P.A - (P + dP).A - \tau_{\mathrm{w}}.(\pi.d).dL = \dot{m}.dV \qquad \ldots\ldots\ldots\ldots\ldots (3)$$

As the shear stress equation for the flow inside tubes is defined by Darcy as follows [12] :

$$\tau_{\mathrm{w}} = \frac{f.\rho.V^2}{8} \qquad \ldots\ldots\ldots\ldots\ldots (4)$$

We get the following equation :

$$dL = \frac{2.d}{f}\left[\frac{d\rho}{\rho} - \frac{\rho.dP}{G^2}\right] \qquad \ldots\ldots\ldots\ldots\ldots (5)$$

Energy balance :

By applying the first law of thermodynamics (steady flow energy equation (S.F.E.E)) for open system on the control volume:

$$Q - W = g.dZ + V.dV + dh \qquad \ldots\ldots\ldots\ldots\ldots (6)$$

The mathematical analysis includes building a model for single-phase flow region and another model for two-phase flow region, and the governing equation above is applied as follows:

1. Liquid single-phase flow region:-

As the liquid single phase flow is practically incompressible, the density is almost constant, with constant tube cross section area, the liquid velocity is constant. By taking integration for both sides of equ. (5) between point (2-3) Fig.(3) therefore :

$$L_{sp} = \frac{2.d.\rho}{f_{sp}}(P_2 - P_3) \qquad \ldots\ldots\ldots\ldots\ldots (7)$$

The pressure losses due to entrance effects have been presented in equ.(8) [12] :

$$P_1 - P_2 = K.\frac{\rho.V^2}{2} \qquad \ldots\ldots\ldots\ldots\ldots (8)$$

As (K) is the entrance loss coefficient, with (0.5) value in this study assuming square edge at tube inlet, the following equation for calculating the length of single phase region is obtained :-

$$L_{sp} = \frac{d}{f_{sp}}\left[\frac{2.\rho}{G^2}(P_1 - P_3) - K\right] \qquad \ldots\ldots\ldots\ldots\ldots (9)$$

(f_{sp}) is the friction factor for single phase region. There are many correlations for friction factor of straight capillary tube, and also for helical coiled capillary tube as shown in table (1).

Comparing the results of the present study with the experimental results in the literature shows that the (Charchill[16] correlation) gives better results for straight capillary tube, and (Schmidt [20] correlation) gives better results for helical coiled capillary tube .
Reynolds number (R_e) and Dean number (D_e) in these equations are obtained from the following equation:

$$R_e = \frac{\rho.V.d}{\mu} \qquad \ldots\ldots\ldots\ldots\ldots (10)$$

$$D_e = R_e\sqrt{(d/D_c)} \qquad \ldots\ldots\ldots\ldots\ldots (11)$$

2. Mixed (Liquid-Vapor) two-phase flow region :-

By applying the continuity equation between points (3-4) in Fig.(2), and also the steady flow energy equation with no external work, heat transfer or potential energy, equ.(6) turns as follows :-

$$V.dV + dh = 0 \qquad \ldots\ldots\ldots\ldots\ldots (12)$$

The flash point is due to evaporate some of the liquid, therefore the velocity will be varied during the two-phase region, (i.e. the kinetic energy can't be equal to zero), therefore:

Table (1) Friction factor correlations for straight and helical coiled capillary tubes for some researchers.

Researchers	Straight capillary tube friction factor correlation	Researchers	Helical coiled capillary tube friction factor correlation
Blassius [13]	$f_s = \frac{0.316}{Re^{0.25}}$	Dean [18]	$f_c = f_s\left(1.03058\left(D_e{}^2/288\right)^2 + 0.01195\left(D_e{}^2/288\right)^4\right)$
Moody [14]	$f_s = \frac{1.325}{\left[\ln\left(\frac{e/d}{3.7} + \frac{5.74}{Re^2}\right)\right]^2}$	Mori and Nakayama [19]	$f_c = \frac{C_1\,(d/D_c)^{0.5}}{[R_e(d/D_c)^{2.5}]^{1/6}}\left\{1 + \frac{C_2}{[R_e(d/D_c)^{2.5}]^{1/6}}\right\}$ $C_1 = 1.88411177 \times 10^{-1} + 85.2472168(\varepsilon/d) - 4.63030629 \times 10^4(\varepsilon/d)^2 + 1.31570014 \times 10^7(\varepsilon/d)^3$ $C_2 = 6.79778633 \times 10^{-2} + 25.3880380(\varepsilon/d) - 1.06133140 \times 10^4(\varepsilon/d)^2 + 2.54555343 \times 10^6(\varepsilon/d)^3$
Colebrook [15]	$\frac{1}{\sqrt{f_s}} = 1.14 - 2.\log\left(\frac{e}{d} + \frac{9.3}{Re.\sqrt{f}}\right)$	Schmidt [20]	$f_c = f_s(1 + 0.14R_e{}^x)$ $x = [1 - 0.0644/(D_c/d)^{0.312}]/(D_c/d)$
Churchill [16]	$f_s = \left[\left(\frac{8}{Re}\right)^{12} + \left(\frac{1}{(A+B)^{1.5}}\right)\right]^{\frac{1}{12}}$ $A = 2.457.\ln\left[\frac{1}{(7/Re)^{0.9} + 0.27(e/d)}\right]$ $B = \left(\frac{37530}{Re}\right)^{16}$	Mishra and Gupta [21]	$f_c = f_s(1 + [log_{10}H_e]^4)$ $H_e = R_e[(d/D_c)/\{1 + (p/\pi D_c)^2\}]^{1/2}$
Chen[17]	$\frac{1}{\sqrt{f_s}} = 2.\log\left(\frac{e/d}{3.7065} + \frac{5.042}{Re}.\log\left[\left(\frac{(e/d)^{1.1098}}{2.8257} + \frac{5.8506}{Re^{0.898}}\right)\right]\right)$	Manlapaz and Churchill [22]	$f_c = f_s[(1 - 0.18/\{1 + (35/H_e)^2\}^{0.5})^m + (1 + d/\{3D_c\})^2(H_e/88.33)]^{0.5}$ $m = 2$ for $D_e < 20$, $m = 1$ for $20 < D_e < 40$, $m = 0$ for $D_e > 40$

$$h_3 + \frac{V_3^2}{2} = h_4 + \frac{V_4^2}{2} \qquad \dots\dots\dots\dots\dots (13)$$

As the specific volume and enthalpy at each point of the two-phase mixture represent the mean, according to the homogenous flow assumption, substituting it with the continuity equation results in the equation (13), and the derived equation is:

$$\frac{G^2}{2}.v_{fg}^2.x^2 + \left(G^2.v_{fg}.v_f + h_{fg}\right).x + \left(h_f + \frac{G^2}{2}.v_f^2 - h_3 - \frac{V_3^2}{2}\right) = 0 \qquad \dots\dots\dots (14)$$

This is a quadratic equation solved by variable (x) (dryness fraction), which is expressed as:

$$x = \frac{-h_{fg} - G^2.v_{fg}.v_f \mp \sqrt{\left(G^2.v_{fg}.v_f\right)^2 - 2.G^2.v_{fg}\left(h_f + \frac{G^2}{2}.v_f^2 - h_3 - \frac{V_3^2}{2}\right)}}{G^2.v_{fg}^2} \quad \dots (15)$$

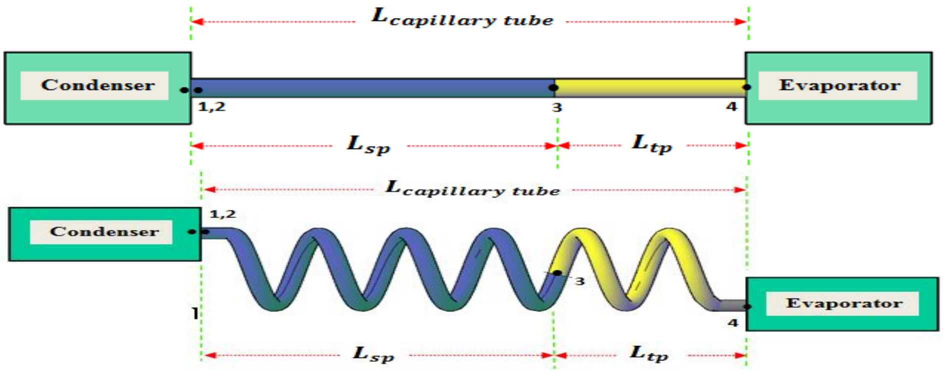


Fig.(2) The flow deatials of striaght and helical coiled tube

Method of Solution:

The solution method for the present study depends on the homogenous flow assumption using the successive substitution method to calculate the physical properties, and then the finite difference method to calculate the total length of capillary tube for the two configurations (straight and helical coiled), as follows :

- Specifying the used refrigerant and the values of main parameters depending on previous experimental research. These include (the pressure or temperature of the condenser and evaporator, the degree of subcooling, the refrigerant mass flow rate through tube, the tube diameter and roughness, the coil diameter and pitch for helical coiled tube) .
- Calculating the temperature at the tube inlet through the equation :

$$T_1 = T_{cond.} - \Delta T_{sub.} \qquad \dots\dots\dots\dots\dots (16)$$

and the pressure (P_1=$P_{cond.}$); then calculating all properties at this point, including (density, specific volume, velocity, viscosity, enthalpy and entropy), and pressure at tube inlet point (2) in Fig.(2) from equ.(8), Reynolds number from equ.(10), Dean number for coiled tube from equ.(11), friction factor for single phase region using Charchill[16] correlation for straight tube and using Schmidt[20] correlation for coiled tube, and specification of properties at point (3) and the length of single phase region from equation (9).

- The numerical method for dealing with the two-phase region is divided into infinitesimal division with uniform pressure drop (ΔP) over each section as shown in Fig.(3). The pressure at each division (i) is calculated by the equation :

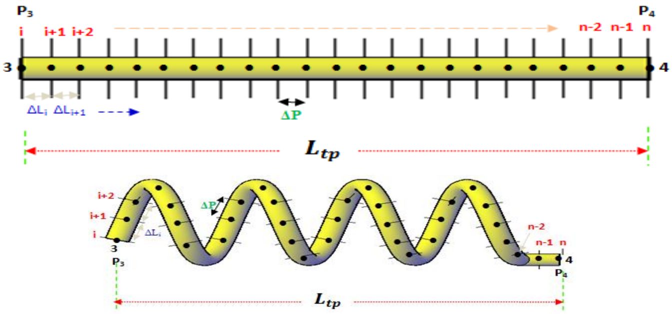


Fig.(3) Numerical divisions of the two-phase flow for straight and helical coiled tube

$$P_i = P_3 - i.\Delta P \qquad \text{… … … … … (17)}$$

- Calculating the value of dryness fraction (x_i) for each point from equ.(15), to calculate all properties at each point from two-phase region, such as density for homogenous flow which is calculated by using the following equation :

$$\rho_i = \frac{1}{\dfrac{x_i}{\rho_{gi}} + \dfrac{1-x_i}{\rho_{fi}}} \qquad \text{… … … … … (18)}$$

And entropy through the following equation :

$$S_i = S_{fi} + x_i.S_{fgi} \qquad \text{… … … … … (19)}$$

- Calculating the friction factor for each point of the numerical division in the two-phase flow region $\left(f_{tp\,i}\right)$ (using [16] and [20] correlations for straight and helical respectively table (1))after calculating Reynolds number $\left(R_{e_{tp\,i}}\right)$ at each point from the following equation :

$$R_{e_{tp\,i}} = \frac{\rho_i.V_i.d}{\mu_{tp\,i}} \qquad \text{… … … … … (20)}$$

(μ_{tpi}) is the two-phase dynamic viscosity. There are many correlations to calculate the dynamic viscosity for the two-phase flow used in previous researches. In comparison with other experimental studies it is shown that McAdams et al.[23] correlation gives better results. The velocity at each point is calculated from the equation :

$$V_i = \frac{\dot{m}}{\rho_i.A} = \frac{G}{\rho_i} = G.v_i \qquad \text{… … … … … (21)}$$

Dean number for coiled tube in the two-phase region is obtained from the following equation :

$$D_{e_{tp\,i}} = R_{e_{tp\,i}}\sqrt{(d/D_c)} \qquad \text{… … … … … (22)}$$

- At this stage, pressure (P_i), temperature (T_i), vapor quality (x_i), friction factor $\left(f_{tp\,i}\right)$, entropy (S_i) and all other properties at each limited point from numerical division in two-phase region are calculated by using the successive substitution method. Then the incremental length (ΔL_i) for each section is calculated by using the finite difference method as follows:

from the integration of equ.(5) between points (3-4) in Fig (2), the incremental length for each section will be :

$$\Delta L_i = \frac{2.d}{f_{tp\ i}}\left[\frac{\Delta\rho_i}{\rho_i} - \frac{\rho_i.\Delta P_i}{G^2}\right] \qquad \dots\dots\dots\dots\dots (23)$$

- We notice from the entropy calculation that it is in continuous increase, and when it reaches a certain value its value will begin decreased. This is why the calculations are completed at a point when the entropy begins deceasing, i.e. the entropy value at this point will be maximum ($S_{max.}$) through the flow in capillary tube, and the pressure at this value will be $\left(P_{i,S_{max.}}\right)$, compared with the limited pressure for evaporator $\left(P_{evap.}\right)$:

If:

$P_{i,S_{max.}} \le P_{evap.} \quad \Rightarrow \quad P_4 = P_{evap.}$

Or If :

$P_{i,S_{max.}} > P_{evap.} \quad \Rightarrow \quad P_4 = P_{\mathrm{i},S_{max.}} = P_{chock}$

the refrigerant velocity has reached maximum value (sound velocity), i.e. the flow reached to chocking condition (or critical condition) . Moreover the fluid flow velocity cannot exceed the sound velocity in the tubes with constant diameter blocking the continuity of calculation after this point as it will result in negative value for incremental length .

- Then the length of two-phase flow region is calculated from the following equation :

$$L_{tp} = \sum_{i=1}^{n} \Delta L_i \qquad \dots\dots\dots\dots\dots (24)$$

the total capillary tube length will be the summation of single and two-phase regions :

$$L_{\text{capillary tube}} = L_{sp} + L_{tp} \qquad \dots\dots\dots\dots\dots (25)$$

The above governing equations have been coded in the numerical formula by using developed copy of Engineering Equation Solver (EES software, Academic professional V8.156), and then inputting the data for the main parameters, and implementing them to get all the required results and analyze them and drawing their curves as explained in the results and discussion ..

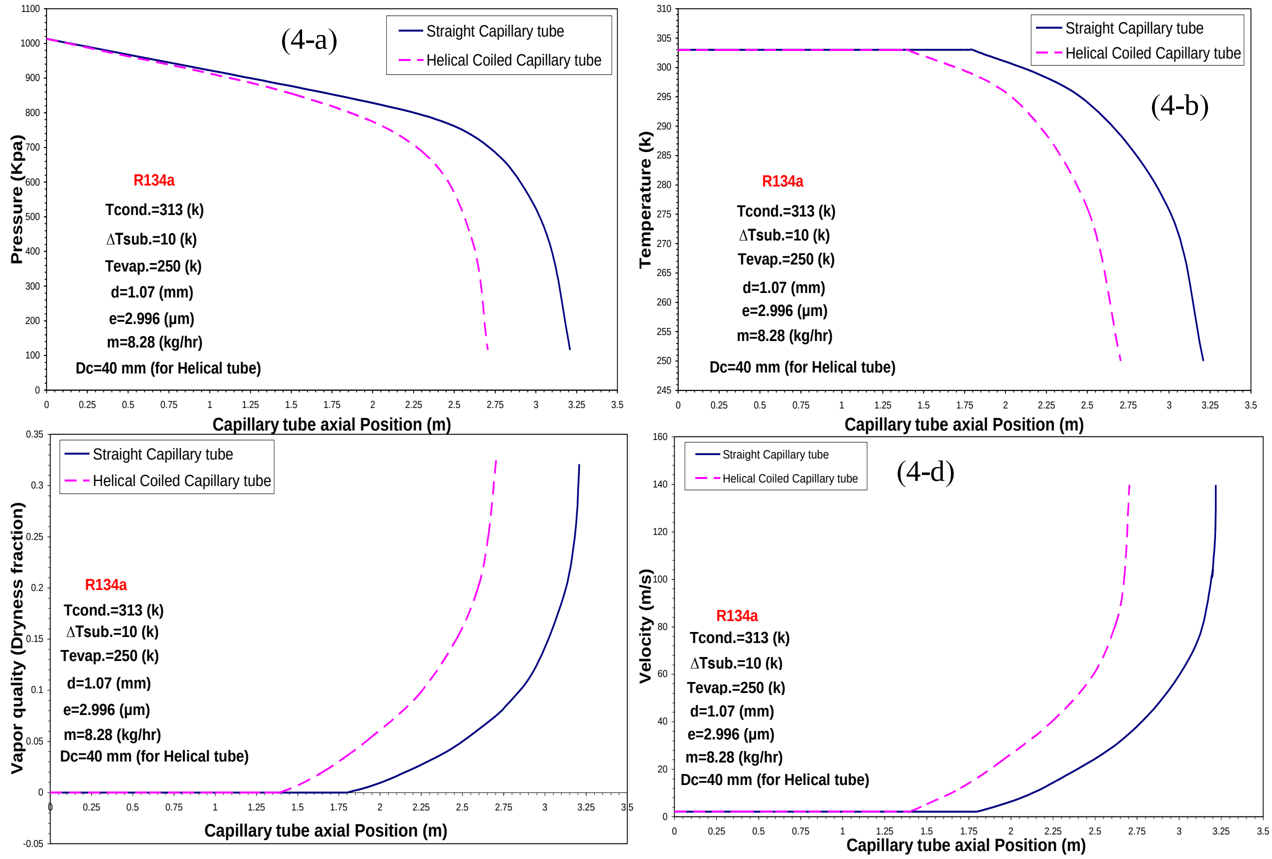
Results and Discussion:

The analyzing of results in this study depend on the experimental data range by Melo et.al[2] and Kim et.al[24] for refrigerant R134a for straight capillary tube, and the theoretical data range by Chingulpitak and Wongwises [11] for helical coiled tube.

A large number of graphs can be drawn from the output of this analysis . However due to space limitation, only typical results are shown.

1- Comparison of the main physical variables distribution for straight and helical coiled capillary tube:-

Fig.(4) shows the comparison of distribution of pressure, temperature, dryness fraction, velocity and entropy in the straight and helical coiled capillary tube length, in certain conditions of the condenser and evaporator temperature and pressure, mass flow rate, tube diameter, internal wall roughness and coil diameter . The results show for all cases that the flow behavior is similar for both configurations, while the helical coiled tube length for all cases is shorter than the straight tube at the same conditions. This is because the flow in coiled tube meets the additional resistance due to centrifugal force, which causes appearance of secondary flow, called the Dean effect, which increases the friction losses with a great rate compared with the straight tube . Therefore we note that the pressure drop in Fig.(4-a) is linear in the subcooled liquid single phase region due to friction with internal wall of tube, while it starts when it reaches the saturated condition (i.e. flash point) to drop rapidly nonlinearly due to friction and acceleration. This drop increases rapidly as the flow reaches the chocked or critical flow at the tube end . As the temperature in Fig (4-b) is constant in the single phase region which depends on the adiabatic flow assumption, the enthalpy and temperature is also constant, while in the two-phase flow region the temperature suddenly drops nonlinearly due to the pressure drop which results from the drop of that part of the enthalpy converts to kinetic energy, increasing the fluid velocity and decreasing the temperature until the flow reaches the tube end .



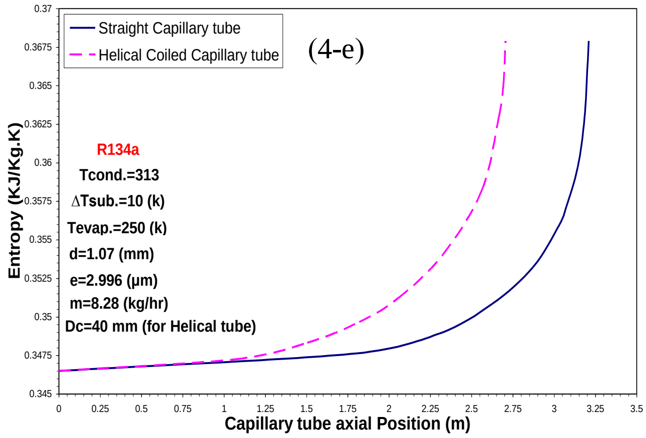


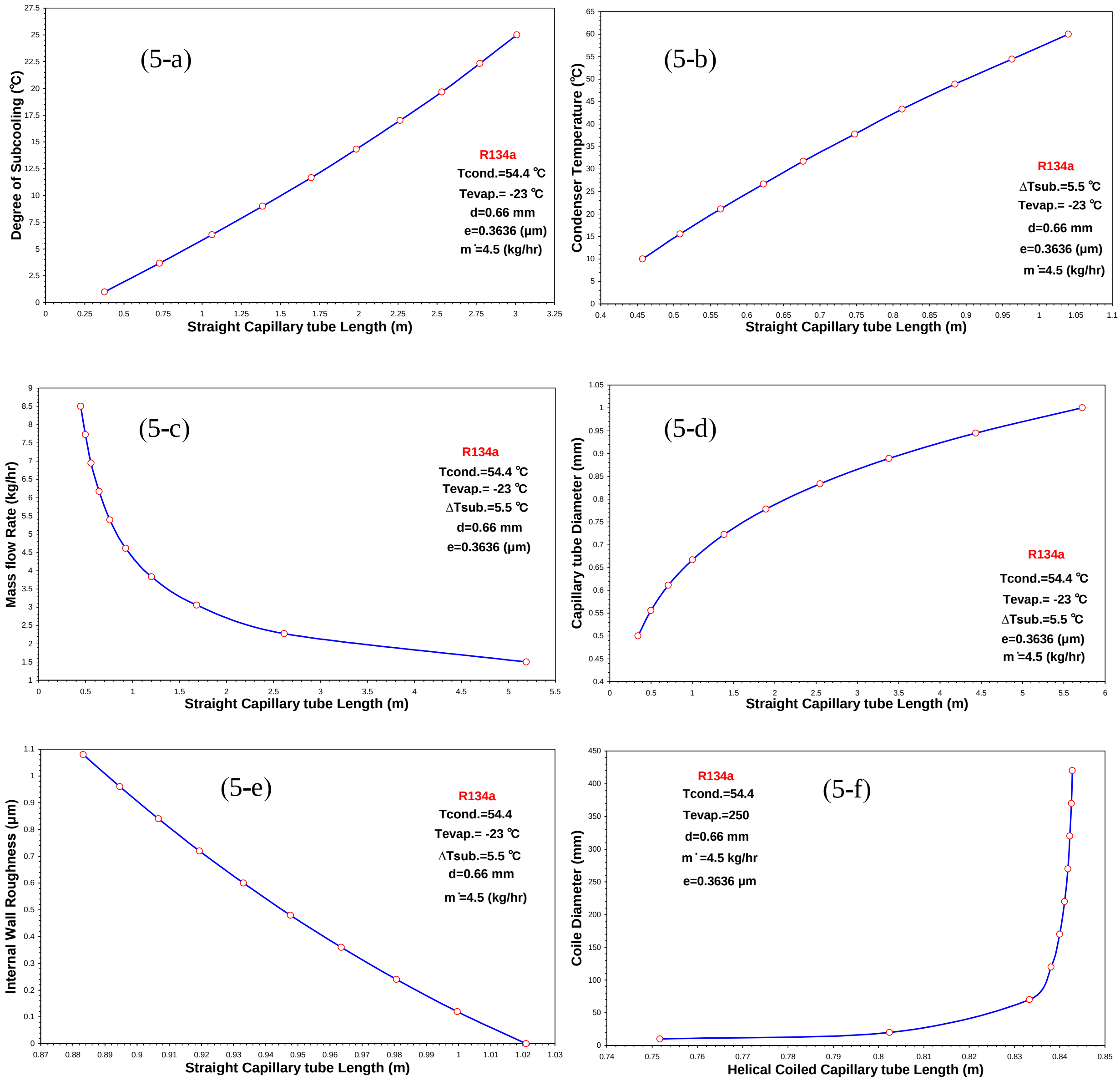
Fig.(4) Comparsion of the results of distributing the physical varibles along striaght and helical capillary tubes . (4-a) Pressure (4-b) Temperature (4-c) Vapor quality (Dryness fraction) (4-d) Velocity (4-e) Entropy .

In Fig.(4-c) we notice the distribution of vapor quality (dryness fraction) along the tube length. Its value is zero in the single-phase region until the flash process occurs, and then it increases nonlinearly and rapidly due to the increase in the amount of vapor in the two-phase mixture until it reaches the tube exit. Fig.(4-d , 4-e) show the variation of refrigerant velocity and entropy. We note that the velocity is constant in the single phase, then it begins to increase in great form in the two-phase flow, because the flow energy after evaporation is converted to kinetic energy, therefore the enthalpy decreases and the velocity increases. While there is a linear increase for entropy in the single-phase flow, when generating the first vapor bubble the entropy increase readily and nonlinearly in the two-phase flow . Actually the great increase in velocity through two-phase flow contributes to the increasing of the entropy until it reaches the critical condition at the tube end.

2- The effect of the main parameters on the capillary tube length:-

Fig.(5) shows the effect of the main parameters on the capillary tube length, which include the operation parameters (degree of subcooling, condenser pressure or temperature and mass flow rate) and the geometric parameters (tube diameter and internal wall roughness). Fig.(5-a) shows that the tube length increases directly with the increase of the degree of subcooling when the other parameters are constant, because with the increase of the degree of subcooling the amount of liquid in tube increases also (i.e. increase the length of single phase). It is known that the resistance of liquid flow is less than the resistance of two-phase flow. This causes shifting the flash point to the direction of the tube exit, therefore the higher degree of subcooling the greater length of the tube required to get the required pressure drop from the condenser pressure to the evaporator pressure i.e. the length of tube increases from (0.3765 m) to (3.009 m) when the degree of subcooling increases from (1 ^{0}C) to (25 ^{0}C) . Also the tube length increases for the same reason with the increase of the condenser temperature but in less ratio, i.e. when the condenser temperature increases from (10 ^{0}C) to (60 ^{0}C) the tube length increases from (0.4573 m) to (1.04 m) when the degree of subcooling is (5.5 ^{0}C) and the other parameters are constant, as shown in Fig.(5-b) . Fig.(5-c) shows the effect of mass flow rate variation on the capillary tube length. It shows inversely relationship between them, therefore when the mass flow rate increases from (1.5 kg/hr) to (8.5 kg/hr) the length of tube decreases from (5.192 m) to (0.446 m), and the reason behind this is that increasing the mass flow rate means increasing the resistance of the flow presented in friction with the internal wall for tube and the friction between refrigerant particles, which is due to increase in the pressure drop, therefore it requires shorter length of tube. The results also show that there are directly relation between the tube length and its internal diameter, as shown in Fig.(5-d), in that the varying of tube diameter from (0.5 mm) to (1 mm) is due to increasing the tube length in a great ratio, i.e. from (0.3428 m) to (5.723 m) in that the internal tube diameter has a great effect on its length reducing the friction effects with the increase of the internal tube area which requires a large length to get the required pressure drop . There are inverse relationship between the tube length and its internal wall roughness. when the internal wall is smooth (e=0 µm) in the Fig.(5-e) the tube length is (1.021 m), and when the roughness increases to (1.08 µm), the tube length decreases to (0.8832 m) when the other parameters are constant, because the increasing of roughness means increasing the friction effects of refrigerant with tube internal wall .

In addition to the parameters presented above, two other parameters have a direct effect on the helical tube length, and on the distributions of physical properties through it; these are coil diameter and coil pitch. Fig. (5-f) shows that with increasing coil diameter from (10 mm) to (420 mm), the helical tube length increased from (0.7517 m) to (0.8428 m) when the other parameters are constant. This is due to the fact that the smaller the coil diameter is the greater



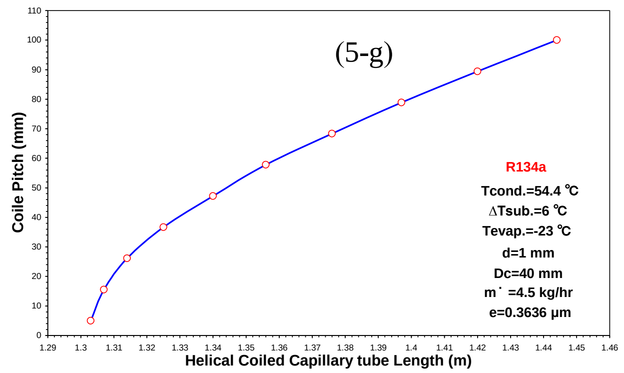
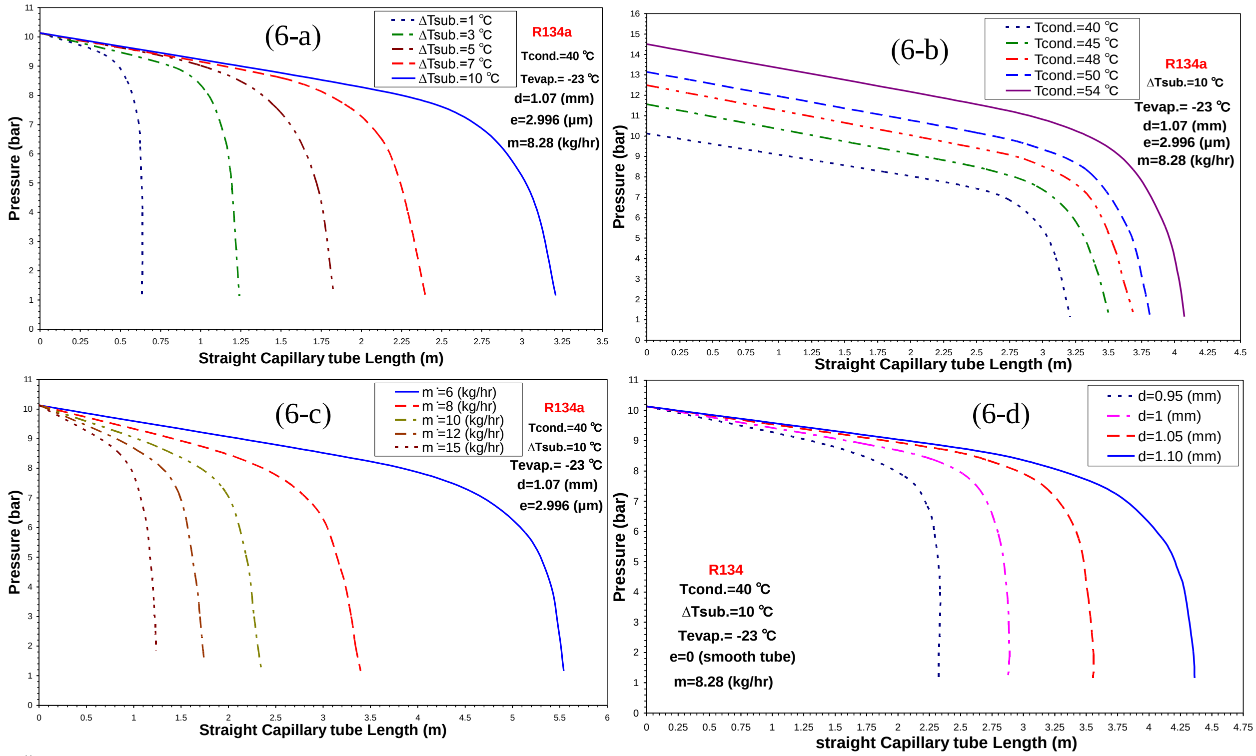


Fig.(5) Results of the effect of the variations of the main parameters on the capillary tube length .
Capillary tube length VS.
{(5-a) Degree of subcooling ,
(5-b) Condenser temperature ,
(5-c) Mass flow rate , (5-d) Tube diameter
(5-e) Tube roughness , (5-f) Coil diameter
(5-g) Coil pitch} .

the Dean number has according to equ.(11), Therefore the friction effect increases due to the secondary flow, and then the pressure drop which requires a shorter tube length increases. However, with the increase of the coil diameter to more than (100 mm) the tube length becomes nearly constant, and so does the effect of secondary flow. The effect of the coil pitch on the total length of helical tube is shown in Fig.(5-g) which uses (Mishra and Gupta correlation [21]) to calculate the helical tube friction factor . This equation contains the coil pitch parameters (p), where the increase of the coil pitch gives a clear increase in the tube length . This is due to reducing the effect of flow resistance that is represented by friction due to the secondary flow with the increase of the coil pitch. The figure shows that with increasing coil pitch from (5 mm) to (100 mm) the helical tube length increase from (1.303 m) to (1.44 m).

3- The main parameters effecting the distribution of pressure through the straight capillary tube:-

The other results of the effects of the main parameters on the straight tube length and on the variable distribution of refrigerant R134a that flows through it are presented in Fig.(6). We note in Fig (6-a) the effect of the variation in the degree of subcooling from (1 ^{0}C) to (10^{0}C), we note that the tube length increases from (0.6356 m) to (3.209 m) i.e. nearly with a ratio (80%) when the other parameters are constant. We also note that the pressure drop is reduced with increasing the degree of subcooling. These results from the increasing of the liquid single phase region that has low flow resistance compared with the two-phase region . This requires increasing the total tube length to get the required pressure drop . Fig.(6-b) shows the effect of the variation of the condenser temperature on the total length of tube and on the distribution of pressure through it . Therefore with increasing the condenser temperature for five degrees from (40 ^{0}C) to (54 ^{0}C), the pressure drop will reduce and the tube length increases from (3.209 m) to (4.07 m) i.e. nearly with a ratio (27%). The reason behind this is the liquid single phase region increasing when the degree of subcooling is (10 ^{0}C) and the other parameters are constant . Fig.(6-c) shows the effect of the variation of the mass flow rate on the tube length and the distribution of pressure through it. We note that with increasing of the mass flow rate, the total length of tube is reduced, i.e. when the mass flow rate increased from (6 kg/hr) to (15/ kg/hr) the tube length is reduced from (3.209 m) to (1.237 m) i.e. nearly with a ratio (61%). The reason behind this is that with increasing the mass flow rate the friction with the internal wall increases, and also the friction between refrigerant particles, therefore the pressure drop increases to an extent that requires a shorter tube length for this increasing pressure drop . To show the effect of geometric parameters the distribution of pressure along the straight capillary tube length and its variations with tube diameter have been drawn in the Fig.(6-d). We note that with increasing the tube diameter the pressure drop is reduced because of increasing the internal wall cross section area, which reduces the friction resistance and requires increasing the tube length. With increasing the tube diameter from (0.95 mm) to (1.1 mm) i.e. with a ratio nearly (16%) the tube length increases from (2.323 m) to (4.362 m) i.e. nearly in (1.86) times or with a ratio nearly (47%) to give the same pressure drop, when the other parameters are constant . The effect of the internal wall roughness of tube on the distribution of pressure is shown in Fig.(6-e). With increasing the internal wall roughness the pressure drop increases because of increasing the friction effect. This requires a shorter tube length, i.e. when the internal wall is smooth (e=0 µm) i.e. the friction of refrigerant with internal wall is very small, but when there is friction between refrigerant particles with each other the tube length was (3.87 m), and when increasing the roughness to (7.49 µm) the pressure drop increases because of the friction increase, which requires a shorter tube length to convey the pressure to the evaporator pressure, therefore the tube length becomes (2.713 m) i.e. the length is reduced in (30%) ratio .



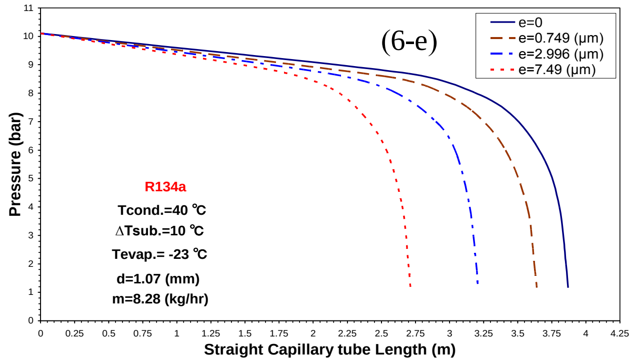


Fig.(6) Results of effect the variations of main parameters on straight tube length and on the pressure distribution through it .
(6-a) Degree of subcooling ,
(6-b) Condenser temperature ,
(6-c) Mass flow rate ,
(6-d) Tube diameter ,
(6-e) Tube roughnes

4- Pressure distribution in straight and helical capillary tube at different coil diameter and coil pitch:-

In Fig.(7) the comparison of pressure distribution for flow refrigerant R134a through straight and helical capillary tube has been done at different coil diameters to show the effect of coil diameter on the coiled tube length and on the distribution of pressure through it. We note in Fig.(7-a) that the pressure drop in the straight tube is less than the pressure drop in the helical coiled tube which is increased with reducing the coil diameter. The coiled tube length with coil diameter (10 mm) is less than the straight tube in a ratio of nearly (29%), whereas it is less in a ratio of nearly (13%) when the coil diameter (400 mm). The reason behind this is that the friction loss in the helical tube is greater compared with straight tube. In Fig.(7-b) a comparison of the pressure distribution through the straight and helical capillary tube is done at different coil pitch to show the effect of coil pitch on the helical tube length and on the distribution of pressure through it using (Mishra and Gupta[21]) equation to calculate the friction factor. We note that the pressure drop decreases with increasing the coil pitch due to reducing the friction resistance, which requires a large tube length to get the required pressure drop .

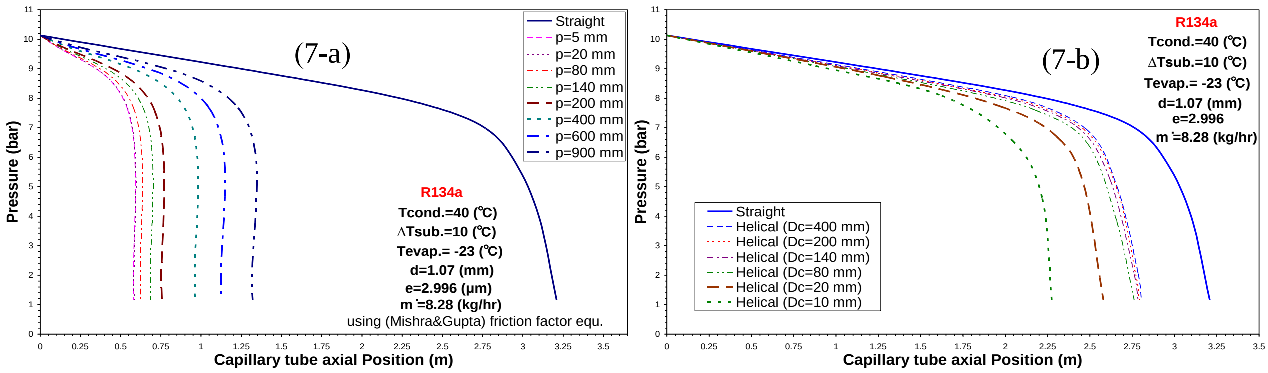


Fig.(7) shows comparison of the results of pressure distribuation through straight and helical tubes at different coil diameter and coil pitch .

5- Validation the results of the present study with experimental data in the literatures:-

The experimental studies that deal with the flow in capillary tubes have been done to evaluate the effect of main parameters on the mass flow rate through the capillary tube. They were done by using tubes with certain length, diameter and form in the test section, and then controlling the variation of operation parameters, and observing its effects on the mass flow rate and the behavior of the flow of refrigerant through it.

In Fig.(8-a) the results of the present study are compared with the experimental results of Wijaya [25], for the variation of straight capillary tube with mass flow rate in various condenser temperatures. We note that the mass flow rate reduces with reducing the condenser temperature and increasing capillary tube length. Because the friction loss in the tube increases with increasing the tube inlet temperature, the mass flow rate reduces through it (i.e. the reducing in the tube length means increasing the pressure drop through it, to get the same pressure drop when reducing the tube length requires reducing the mass flow rate) . These results show a good agreement with these experimental data with error nearly (±9%) .

Fig. (8-b) shows a comparison of the present study results with the experimental results of Khan et.al[4], of variation in the helical coiled capillary tube length with the mass flow rate at different degree of subcooling. It shows that the mass flow rate increases with increasing the degree of subcooling and reduces with increasing the tube length, when the other parameters

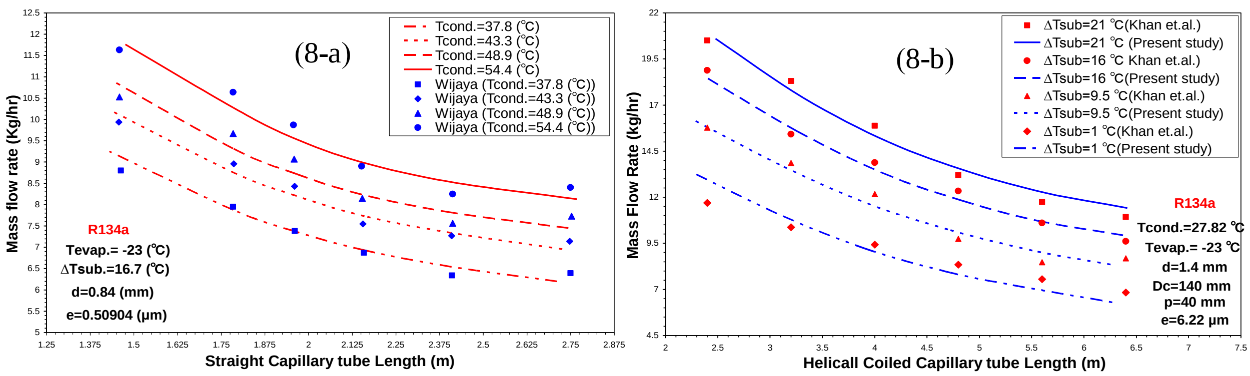


Fig.(8) Comparison between the present study results and experimental results ([25] and [4]) for variation straight and helical tube length with mass flow rate at different condenser temperatures and degrees of subcooling .

are constant, i.e. with helical tube length (4 m) when increasing the mass flow rate from (9.5 kg/hr) to (12 kg/hr) i.e. with a ratio of nearly (26%). This requires increasing the degree of subcooling from (1 ^{0}C) to (9.5 ^{0}C) in order for the capillary tube length to be constant. The results show a good agreement with the experimental results with error nearly (±7%).

6- The applicability of the present study model on other types of refrigerants:-

To show the possibility of using other types of refrigerants in this theoretical analysis, the results of distributing the pressure along the straight capillary tube have been presented in Fig.(9) using refrigerant (R12) that has bad effect on environment, and its suggested alternatives (R134a, R600a and R152a) . Fig. (9-a) shows that the pressure drop of refrigerant R12 is less than its alternatives, therefore the tube length is greater. The results show that the straight tube length using refrigerant (R134a, R600a and R152a) reduces with a ratio of (30%, 53% and 64%) respectively when the other parameters are constant . i.e. the refrigeration equipment that uses R12, when redesigned using its alternatives requires a shorter tube length . This result shows a good agreement with the experimental data of (Mikol [26]) with an error ratio of (-5%) for refrigerant R12. Fig.(9-b) shows another comparison with the experimental data of (Li et.al [27]), it gives a good agreement with an error ratio of nearly ((-3) to (+8)%) . It shows that the length of straight capillary tube using refrigerants (R134a, R600a and R152a) reduces with a ratio of nearly (11%, 20% and 50%) respectively.

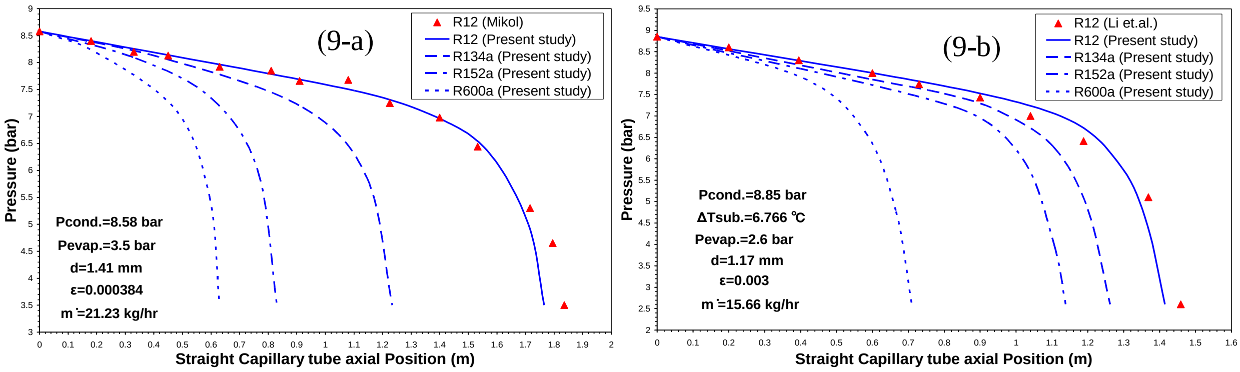


Fig.(9) Comparison of the results of the pressure distribuation for refrigerant R12 and its alternatives and with the exprimental data for ([26] and [27]) .

Conclusions :

The present research implies building two mathematical models to compare the analysis of refrigerant flow through two different configurations of adiabatic capillary tubes (straight and helical coiled tube). Each model has been analyzed with building a region for liquid single phase flow and another for mixture two phase flow, and calculating the length of each region and the total tube length, and then presented the calculation and its curve results. The following results are concluded:

1- The pressure drop is linear in the single phase region, while the temperature is constant . When the flow reaches the saturated conditions (flash point) the pressure and temperature drop rapidly in the two-phase region . Whereas the vapor quality value equals zero, fluid velocity is constant and the entropy increases linearly in the single phase region. Then they all increases rapidly nonlinearly in the two-phase region until reaching the tube exit .

2- The results show that the capillary tube length increases with increasing the degree of subcooling, condenser temperature or pressure, internal diameter of tube, coil diameter and coil pitch, and decreases with increasing the mass flow rate and internal wall roughness, with varying the input parameters in this numerical model .

3- The straight capillary tube length is greater than the helical tube at any value of coil diameter and coil pitch . Therefore the use of helical coiled tube is better for reducing the cost and also reducing the space .

4- The variation of the degree of subcooling and tube diameter have a greater effect on the tube length in comparison with other parameters . Therefore if the refrigeration system requires replacing the capillary tube with another one of shorter length to reduce space or cost, this should be done by reducing the degree of subcooling or the tube internal diameter .

5- The distribution of pressure and other physical variables is constant along the helical coiled tube when increasing the coil diameter to more than (100 mm), i.e. the tube length will be nearly constant.

6- The results also show the possibility of using any other type of refrigerant in the present study model, and that the tube length using refrigerant R12 is greater in comparison with its alternatives (R134a, R600a and R152a) .

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