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A NUMERICAL ANALYSIS OF ADIABATIC CAPILLARY TUBE PERFORMANCE IN VAPOR COMPRESSION REFRIGERATION SYSTEMS

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ABSTRACT
This paper represents a numerical analysis intended for the selection and geometry design of the capillary tubes used in the vapor compression refrigeration units. This expansion component is usually selected for small home air conditioning units and domestic refrigerators and freezers for their ease integration with the cooling units and cheap prices. The numerical step by step model suggested in this work depends mainly on the idea of the control volume theory for the conservation of mass, energy and linear momentum of fluid refrigerants flowing inside the capillary tube. The flow regime is divided into a number of increments for which the conservation expressions were solved simultaneously for the physical characteristics parameters. These are temperature, pressure, and vapor quality and mixture velocity along the capillary tube from the entrance at the high pressure side to the exit at the low pressure side of the refrigeration unit. The theoretical prediction for the profile of the characteristics parameters showed a good agreement with available experimental data in the literature and the ASHRAE published rating charts of the capillary tubes.

Key Words: Refrigeration, Capillary Tubes, Adiabatic, Numerical, Alternatives

تحليل عددي لأداء الأنابيب الشعرية الأديباتيه في منظومات التبريد الأنضغاطيه

.الخلاصة:

يمثل البحث الحالي تحليل عددي مزمع استخدامه لاختيار وتصميم الأنابيب الشعرية المستخدمة في منظومات التبريد الأنضغاطية. وغالباً ما تستخدم هذه الأنابيب لأغراض تخفيض الضغط في منظومات تكييف الهواء المنزلية بالإضافة للثلاجات والمجمدات لسهولة تركيبها و توافقها للعمل مع وحدات التبريد ولرخص سعرها.
النموذج المقترح يمثل تحليل عددي بطريقة الخطوة – خطوة يعتمد على مبادئ حفظ الكتلة، الطاقة والزخم الخطي لمائع التثليج أثناء مروره عبر ألأنابيب ألشعرية وتطبيقها لحجم متناهي للمائع. تم تقسيم نظام الجريان للمائع لعدد من الأحجام الصغيرة واتي تم عندها حل معدلات التمثيل لحفظ المتغيرات أعلاه لتحديد المتغيرات ألفيزياوية والتي تحدد ملامح أداء الأنابيب الشعرية. هذه المتغيرات تمثل توزيع درجات الحرارة، الضغط، النسبة المئوية للبخار وسرعة الخليط (البخار والسائل) لمائع التبريد على طول الأنبوب الشعري من نقطة الدخول لجانب الضغط العالي إلى نقطة الخروج لجانب الضغط الواطئ لمنظومة التبريد.
التنبؤ النظري لتوزيع المتغيرات أعلاه على طول الأنبوب بينت تطابق جيد مع القيم التجريبية المتوفرة في الأدبيات وكذلك عند مقارنتها مع مخططات الأداء المنشورة في أدبيات منظمة (ASHRAE) للأنابيب الشعرية.

INTRODUCTION

The capillary tube or restrictor tube needs no formal introduction to refrigeration service or design engineers. All of the pressure reducing devices used in refrigeration equipment work the same way, in principle, in one industrial application to the other. These devices are used to meter the flow refrigerant from the high pressure side to the low pressure side of the refrigerating system according to the load demand. In small refrigeration air conditioning units and domestic refrigerator, the most commonly used expansion devices is the capillary tube. It is used for the refrigeration load up to (10) kW. However, its use is also extended to larger units such as unitary air conditioners in sizes up to (35) kW capacity.

The early work of Mikol (1963) [1] showed that the vapor generation in the capillary tube not necessary to occur when the liquid approaches its bubble point. But rather the refrigerant remains in the liquid phase for further length of the capillary tube where the pressure is below the saturation value. The same trend of behavior for this situation was later observed by Li et al. (1990) [2]. The experimental work of Wei et al. (2001) [3] conducted for small refrigeration units has reported the length required for these units. The length was ranged between (0. 4) to (2.5) mm and as a result, the capillary tubes are normally folded so as to reduce the required space. An extensive data for the adiabatic capillary tubes and reliable diagram are available in the work of Bittle et al. (1998) [4], ASHRAE (1994) [5], Melo et al. (1999) [6], Sami and Maltais (2000) [7]. A qualitative data is also available from the work of Sami and co-workers in [8, 9].

A number of theoretical models, numerical and rating charts for the characteristics of the capillary tube design and selection are published. The early work conducted by Hopkins (1950) [10] and Whitesel (1957) [11] resulted in developing a rating charts for (R-12) and (R-22). ASHRAE (1979) [12] established graphical method representation for the capillary tube rating for specified entering conditions. More detailed rating charts are also included in ASHRAE (1998) [13] for pure (R-134a), (R-22), and (R-410A). The object of these curves is to establish the required capillary refrigerant flow rate for specified geometry and entering condition. This condition represents the entering pressure and the sub-cooling available for the refrigerant at inlet to the capillary tube. A generalized prediction equation based on tests with (R-134a), (R-22), and (R-410A), Wolf et al. (1995) [14] was developed for the prediction of the refrigerant mass flow rate through the capillary tube. It is based mainly on the Buckingham-pi theorem incorporating the physical factors and fluid properties that affect the capillary flow rate. Their correlation showed a good agreement with their own and other investigators experimental data for the considered refrigerants. Akintunde et al. (2006) [15] examined the performance of the helical-coiled capillary tube. They concluded that the pitch variation has no significant effect on the system performance. More recently, Akintunde (2007) [16] studied the effect of various geometries of capillary tube. He also studied the effect of the pitches of both helical and serpentine coiled capillary tubes on the performance of a vapor compression refrigeration system. He developed a correlation to describe the relationships between straight and coiled capillary tube and between helical coiled and serpentine coiled capillary tubes.

In the present work a step by step numerical analysis is developed for the prediction of the performance of the capillary tubes. The results of the present work provide a powerful tool for the rough estimate of the capillary tube geometry and flow operating conditions.

Theoretical Model:

The general expression for the control volume theory applied to the region shown in figure (1) has the form of:

$$\frac{dN}{dt}=\frac{\partial}{\partial t}\int_{C.V}\varepsilon\rho d\forall+\int_{C.S}\varepsilon\rho\vec{V}\,dA \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots (1)$$

Where (N) represents an total general property of the control volume and (ε) is the amount of property per unit mass. Applying the above theory of the control volume with the conservation of mass, energy and momentum on a finite volume produces:

Continuity:

Its derivation is obtained by replacing (N) with the mass of the control volume (m) and (ε=1) in equation (1):

$$\frac{dm}{dt}=\frac{\partial}{\partial t}\int_{C.V}\rho d\forall+\int_{C.S}\rho\vec{V}\,dA \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

For conservation of mass $\frac{dm}{dt}=0$ (mass remains constant within the system) and since the flow is at steady state condition, then:

$$\frac{\partial}{\partial t}\int_{C.V}\rho d\forall=\left(\frac{\partial m}{\partial t}\right)_{C.V}=0$$

Therefore

$$\int_{C.S}\rho\,\vec{V}\,dA=0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots. \quad (3)$$

Since there is one stream inlet and outlet and the fluid flow is considered to be one-dimensional in the flow direction and for constant cross sectional area, (A), then:

$$\frac{\dot{m}}{A}=\frac{V_{in}}{v_{in}}=\frac{V_{out}}{v_{out}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. (4)$$

Equation (4) represents the continuity equation for compressible and incompressible flow of the refrigerant through the capillary tube.

Linear Momentum:

The 2nd law of motion for a system states that:

The rate of change of linear momentum of the system

= Sum of the external forces acting on the system

In mathematical representation is:

$$\frac{D}{Dt}=\int_{system}\vec{V}\,\rho\,d\forall=\sum F_{system} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots \quad (5)$$

Let $N=m\,\vec{V}$ in equation (1), then:

$$\varepsilon=\frac{N}{m}=\frac{m\,\vec{V}}{m}=\vec{V} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.. \quad (6)$$

Substituting for (ε) in the general form of the control volume theory and equation (5) yields:

$$\sum F_{system}=\frac{\partial}{\partial t}\int_{C.V}\vec{V}\rho\;d\forall+\int_{out}\vec{V}d\dot{m}-\int_{in}\vec{V}d\dot{m} \quad \ldots\ldots\ldots. \quad (7)$$

The net of forces acting on the control volume is due to the drag and pressure difference on opposite sides of the element in the flow direction in the form:

$$\sum F_{system}=\left[(P_{in}-P_{out})-f\,\frac{\Delta L}{d}\,\frac{V^2}{2\,v}\right]A \quad \ldots\ldots\ldots\ldots \quad (8)$$

The drag force is due to the drag of the wall on the fluid at the horizontal boundary of the system.

For the case considered in this work, where a steady state is present for the flow through the capillary tube, equation (7) and equation (8) lead to:

$$\left[(P_{in}-P_{out})-f\,\frac{\Delta L}{d}\,\frac{V^2}{2\,v}\right]A=\dot{m}\,(V_{out}-V_{in}) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots. \quad (9)$$

The above expression is applied for the one-dimensional fluid flow inside the capillary tube.

Energy:
The 1st law of thermodynamics for the system implies that:

$$\textit{Time rate of increase of the total stored energy of the system}$$
$$= \textit{Net time rate of energy addition by heat transfer into the system}$$
$$+ \textit{Net time rate of energy addition by work transfer into the system}$$

In mathematical representation of the above statement, it becomes:

$$\frac{D}{Dt}\int_{system} e\,\rho\, d\forall = \left(\dot{Q}_{net-in} + \dot{W}_{net-in}\right)_{system} \quad \ldots\ldots\ldots\ (10)$$

In which

$$\left(\dot{Q}_{net-in} + \dot{W}_{net-in}\right)_{system} = \left(\dot{Q}_{net-in} + \dot{W}_{net-in}\right)_{coincident\ C.V}$$

Let (ε=e) be the energy per unit mass of the fluid defined by:

$$e = u + \frac{V^2}{2} + g\,z \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ (11)$$

Then equations (1) and (10) would have the form:

$$\frac{\partial}{\partial t}\int_{C.V} e\,\rho\, d\forall + \int_{C.S}\left[u + \frac{P}{\rho} + \frac{V^2}{2} + g\,z\right]\rho\vec{V}\,dA = \dot{Q}_{net-in} + \dot{W}_{shaft-net}$$

The above formula when applied for unsteady state one-dimensional flow where only one stream entering and laving the control volume then:

$$\dot{m}\left[u_{out} - u_{in} + \left(\frac{P}{\rho}\right)_{out} - \left(\frac{P}{\rho}\right)_{in} + \frac{V^2{}_{out} - V^2{}_{in}}{2} + g(z_{out} - z_{in})\right]$$
$$= \dot{Q}_{net-in} + \dot{W}_{shaft-net-in} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ (12)$$

This equation is called as the "One –dimensional energy equation for steady state in the mean flow". It is valid for both compressible and incompressible flows. The specific enthalpy, (h), is related to the internal energy by:

$$h = u + \frac{P}{\rho} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ (13)$$

With substituting in eq. (12) yields:

$$\dot{m}\left[h_{out} - h_{in} + \frac{V^2{}_{out} - V^2{}_{in}}{2} + g\,(z_{out} - z_{in})\right]$$
$$= \dot{Q}_{net-in} + \dot{W}_{shaft-net-in} \quad \ldots\ldots\ldots\ldots\ (14)$$

Since the flow through the capillary tube is considered to be one-dimensional at steady state condition, the "shaft work" is zero and for adiabatic process where there is no heat lost or gained through the wall of the capillary tube, $\dot{Q}_{net} = 0$,then equation (14) for negligible potential energy gives:

$$h_{out} + \frac{V^2{}_{out}}{2} = h_{in} + \frac{V^2{}_{in}}{2} \quad \dots\dots\dots\dots\dots\dots \quad (15)$$

Relations Criteria:

The control volume theory equations for conservation of mass, eq. (4), linear momentum, equation (9) and energy, equation (15), derived in this work has the same forms as those presented by Stoecker and Jones (1982) [17].

The final expression of the linear momentum equation as expressed by equation (9) includes the most important parameters which describe the flow regime in the capillary tube. These are the specific volume, (υ), fluid velocity, (V), and the friction factor, (f).

Specific Volume (υ):

The specific volume of the flowing refrigerant, (υ), represents the volumetric flow rate of the fluid as it passes through the tube accompanied by state variation. If the refrigerant state is considered to be saturated liquid at the entrance, then the quality of the vapor continue to increase progressively toward the exit end. This is due to reduction in the fluid pressure and temperature. This variation will produce a mean velocity distribution profile along the tube section. Hence, the friction coefficient will not be a constant value all over the capillary tube. The specific volume of the fluid mixture may be estimated from:

$$\upsilon = \upsilon_f(1-x) + \upsilon_g x \quad \dots\dots\dots\dots\dots\dots\dots\dots \quad (16)$$

Fluid Velocity (V):

The fluid velocity, (V), represents the mean velocity between the entering and leaving sections of the control volume. Further, these velocities are also mean values for the liquid and vapor in the increment length, (ΔL). Using the result of the continuity equation presented by equation (4) in the 2nd term of the left hand side of equation (9) yields:

$$f\,\frac{\Delta L}{d}\,\frac{V^2}{2\,\upsilon} = f\,\frac{\Delta L}{d}\,\frac{V}{2}\,\frac{\dot{m}}{A} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots \quad (17.a)$$

In this expression the mean velocity is suggested to be used by assuming a linear velocity distribution along the increment defined as:

$$V_m = a_1 + a_2 X \quad \dots\dots\dots\dots\dots\dots\dots\dots.. \quad (17.b)$$

Where (a1) and (a2) are constants. The longitudinal direction of the fluid flow through the capillary tube is presented by (X). This formula states that:

$$V_m = \frac{V_{in} + V_{out}}{2} \quad \dots\dots\dots\dots\dots\dots\dots\dots. \quad (17.c)$$

In addition, the mean velocity at any position in the longitudinal direction is represented by V(X) indicates the mutual effect of liquid and vapor phases which is satisfied by the continuity equation.

Friction Factor (f):

The friction factor, (f), of the two phase flow is difficult to be predicted. A suggested method for the prediction of the mean friction factor is outlined in this work. The length of the control volume, (ΔL), which is considered as the increment length will be further refined to a number of sub-increment depending on the quality of vapor contained by the control volume. In this category, the temperature variation from the inlet to exit sides of the increment is assumed to have a linear variation with the length of control volume. The increment temperature is divided into equal sub-increment values in the form:

$$\Delta T_{ni} = \frac{(T_{out} - T_{in})}{n} \quad \ldots\ldots\ldots\ldots\ldots (18.a)$$

Where (n) represents the number of sub-increments considered in the control volume with a length of (ΔL). This doesn't mean that the sub-increments contain the same amount of vapor contents, since it is accumulative property numerical value. This of course will produce a (Re) number distribution which in turns give a friction factor profile for the refrigerant of the control volume.

This will reduce the possible error may arise from using the single phase equations in the prediction o the friction factor. It may be estimated from the mean value defined by:

$$f_m = \frac{f_1 + f_2 + \cdots + f_n}{n} \quad \ldots\ldots\ldots\ldots\ldots (18.b)$$

In which, (fi) represents the friction factor for the (ith) sub-increment. Noting that these sub-increments are not necessarily equal in length, since the length of the sub-increment depends on the ability of producing the specified quality of vapor. The value of (fi) for the refrigerant is estimated using the Blasius (1911) [18] relation expressed by:

$$f_i = \frac{0.33}{(Re_d)^{0.25}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (19)$$

It may be also predicted by the Haaland (1983) [18] correlation presented in the form:

$$\frac{1}{f^{0.5}} = -1.8 \log\left[\frac{6.9}{Re_d} + \left(\frac{\epsilon/d}{3.7}\right)^{1.11}\right] \quad \ldots\ldots\ldots\ldots\ldots (20)$$

Colebrook (1939) [19] correlation for the friction factor can also be used in the following expression:

$$\frac{1}{f^{0.5}} = -2 \log\left[\frac{\epsilon/d}{3.7} + \frac{2.51}{Re_d f^{0.5}}\right] \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (21)$$

Where in the above correlations

$$Re_d = \frac{V\, d}{\upsilon\, \mu} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (22.a)$$

The Reynolds number presented in the above equation will incorporate the effect produced by the presence of the two-phase flow with using the fluid viscosity, (μ), and the specific volume,(υ), as those of the mixture depending on the vapor quality at that section where:

$$\mu = \mu_f (1 - x) + \mu_g x \quad \ldots\ldots\ldots\ldots\ldots (22.b)$$

The solution of equation (21) requires an iterative procedure for the friction factor, (f). This procedure was incorporated in the computer program built for this purpose.

The roughness (ϵ) of the capillary copper tube is assumed to be (0.002) mm as recommended for these tubes by Frank (2001) [20].

Computational Scheme:

General Description:

The object of the capillary tube design is to predict the length and diameter required for specified operation conditions of the vapor compression refrigeration system.

These conditions are related to the mass flow rate of refrigerant which matches that required for stable operation of the compressor. Moreover, it should be capable of providing the proper pressure ratio for smooth running of the whole unit and avoiding the problems of starving or overfed of the evaporator. The later situation occurs when too much refrigerant for the amount of cooling needed, resulting in slugging of the compressor, liquid drops enter the compressor.
The numerical analysis suggested in this work explores the idea of using the step by step technique. This model depends on dividing the length of the capillary tube into a number of increments of temperature drop, (ΔT). This in turn represents the available pressure drop may occur for the increment length, (ΔL). The output condition from the control volume is considered to be the entering conditions to the next step and so on. The calculations proceed until the final exit pressure, at the evaporator inlet, is obtained, for which the capillary tube accumulative length then estimated.
Combining the conservation of mass expression, equation (4) and the energy conservation criteria, equation (15), results:

$$h_{out} + \left(\frac{\dot{m}}{A}\right)^2 \frac{(\upsilon_{out})^2}{2} = h_{in} + \frac{V^2{}_{in}}{2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots. \quad (23)$$

Replacing the thermodynamics relations into the above expression, for the specific volume of the refrigerant, (υ), presented by equation (16), and the following relation for the enthalpy, (h)

$$h = h_f(1 - x) + h_g x \quad \ldots\ldots\ldots\ldots\ldots.. \quad (24)$$

This will produce a general formula for the simulated energy criteria including the mass conservation formula in the form:

$$h_{f_{out}} + (h_{fg_{out}})x + \frac{1}{2}\left\{\upsilon_{f_{out}} + \left(\upsilon_{g_{out}} - \upsilon_{f_{out}}\right)x\right\}^2 \left(\frac{\dot{m}}{A}\right)^2 = h_{in} + \frac{V^2{}_{in}}{2} \quad .. (25)$$

The above equation is solved for the quality, (x), with a given entering operation conditions to the incremental control volume together with the refrigerant mass flow rate. When the quality is obtained then the exit fluid condition and leaving velocity are estimated. With the aid of the linear momentum equation, equation (9), and the friction factor estimation discussed in sec. (3-2 & 3-3), the increment length is calculated. This length is explicitly satisfies the specified exit required conditions from that increment.

Solution Procedure:

To simplify the scheme of the numerical analysis, a computer program was built to establish the vapor quality, velocity, pressure, and temperature distribution of the refrigerant flowing through the capillary tube. A flow chart of the present work computer program is presented in figure (2). The following procedure was used for the solution object of the above parameters:
Specify the operating conditions of the capillary tube including, refrigerant mass flow rate, $\dot{m}$, entering pressure, Pin, leaving pressure, Pout, temperature, Tin, leaving temperature, Tout, and the inlet quality for the mixture condition or sub-cooled temperature if it was present.
Select a proper capillary tube size including outside diameter and internal diameter for the above operating conditions.
Divide the whole temperature range between the entering and leaving condition of the capillary tube to equal temperature increments, ΔT. This in effect as assuming the exit

temperature out of the capillary tube increment, ΔL. Hence, the pressure ratio across the capillary tube increment may be determined.
For the selected circulated refrigerant through the cooling unit, calculate the pressure, Pout, liquid enthalpy, hf out, latent heat of vaporization, hfg out, vapor specific volume, υg out and liquid specific volume, υf out at the exit of the increment.
Solve equation (25) for the exit quality, x, of the refrigerant mixture leaving the increment. This will ease the estimation of the leaving refrigerant condition out of the increment including hout, υout, and the leaving stream velocity, Vout.
Compute the refrigerant mean velocity as described in section (3-2), and the mean friction factor from section (3-3), using one of the correlations presented for prediction.
Estimate the required increment length, ΔL, to satisfy the energy and continuity equation by applying equation (9).
Store the exit conditions from the increment and to be considered as the entering condition to the next length increment. The calculation was allowed to continue until the exit condition from the last tube increment is within acceptable accuracy limit of the required conditions.

Results and Discussion:

Model Verification:

The predicted capillary tube geometry for a given operating conditions may be obtained for any specified conditions at the entering and leaving sides of the capillary tube. It is more convenient if the results of the present work was compared directly with any available experimental data in the literature or available known rating charts such as those presented by ASHRAE. The data obtained from Sami et al. [21] represents a well published detailed operation conditions for the capillary tube. They used three different refrigerants as alternatives for the systems circulating (R-22) as a refrigerant. They applied (R-407C), (R-410A) and (R-410B) in addition to the (R-22) for a good range of pressure and mass flow rate for three different sizes (1.778, 1.905, and 2.159) mm of capillary tube diameter. Their tests data obtained for the operating conditions of saturated (R-22) flowing in a capillary tube diameter of (2.159) mm at the operating pressure of (11.1) bar and refrigerant flow rate of (54) kg/h is used for verification. The data obtained for the same mass flow rate, capillary tube diameter and pressure of (10.3) bar with (R-407C) flow was also used for comparison.

The predicted capillary tube performance with the present analysis was also compared with the charts presented by ASHRAE [12] for selected similar conditions of (R-22) system.

The thermal properties of the refrigerants used for comparison were obtained for the ASHRAE [22]. A polynomial fitting was used to correlate the properties in terms of temperature for the saturation conditions with acceptable accuracy limits.

Comparison with Experimental Data:

The predicted performance for the adiabatic capillary tube may be classified for a constant mass flow rate of refrigerant according to the following parameters:

Pressure and Temperature Distribution:

The analysis of the present work for the conditions mentioned above for the Sami et al. [21] data is presented. The predicted performance for (R-22) with these conditions is shown in figure (3). It is obvious that the numerical analysis for the pressure distribution shows the same behavior as that of the experimental data. Further it shows the acceptable agreement with the experimental data. The maximum discrepancy between the experimental and predicted values of the tube length fell within (13%). In figure (4), the pressure drop distribution along the capillary tube is outlined. Here, the pressure drop increases with the capillary tube length due to the increase in the friction effect with the flow direction. The pressure drop for the two-phase flow is usually lower than that of the single liquid at the same mass flow rate and pressure condition. In fact the predicted pressure drop for each increment showed a decrease as the flow proceeds toward the exit end. Again the predicted behavior has the same trend as that of experimental data of Sami et al. [21] and as noticed by other investigators [7, 8, 9 & 10].

As a result of the expansion and pressure variation, the temperature drops with the direction of the flow toward the exit from the capillary tube, figure (5). The comparison for the temperature distribution is also exhibited acceptable agreement with that of the experimental data. The maximum discrepancy from the experimental data was within (12%). Here, the percentage is calculated from the error in the tube length to produce the same temperature. This percentage falls to lower values when using Haaland [18] or Colebrook [19] correlations.

The mean friction factor estimated from the above equations was (0.022), (0.0233) and (0.0234) for Blasius, Haaland and Colebrook respectively. The latter correlations predicted higher friction factor with maximum discrepancy of (6%).

Velocity and Quality Distribution:

Figure (6) shows the predicted variation of the velocity along the capillary tube for the refrigerant (R-22) considered for comparison in this category. Here, the predictions when using the different correlations for the friction factor are shown for comparison. All of these performance predictions showed the same behavior although they have different numerical values. The velocity of the refrigerant exhibited quite a smooth behavior and revealed a high increase in the tail portion of the tube. This is due to the increase in the vapor quality which causes an increase of the specific volume of the mixture which in turn shows a rise for the velocity. The refrigerant vapor quality variation along the stream path is shown in figure (7). It is worthwhile mentioning here that when using Blasius correlation, the predicted fluid quality and velocity were higher than those of the other correlations. Further, the trend of the velocity variation has the same profile as that of the quality curve.

Evaluation of Friction Factor (f):

It has been found that the choice of the friction factor correlation affects the final results of the capillary tube design for specified operating conditions. Figure (3) shows the results of the present model when using different equations for the prediction of the friction factor for the (R-22) case. It is obvious that the capillary tube performance prediction when using the correlations of Blasius, equation (19), and Haaland, equation (20), and the Colebrook, equation (21), is overestimated. Although they predicted longer tube path length, but produced different extra length percentage.

These correlations showed an increase in the tube length by (12%), (7.8%) and (7.5%) respectively. In fact the last two correlations predicted almost the same refrigerant flow path lengths to achieve the required evaporator pressure for the flow of (R-22) case. Here the predictions of the capillary tube performance, pressure and temperature profiles were closer to the experimental data than that of Blasius correlation.

The experimental pressure profile of the refrigerant (R-407C) for Sami et al. [21] is compared with the different friction factor correlations are shown in figure (9). The predicted pressure profile by the present model exhibited the same trend as that of the experimental data with accuracy of less than (6%). The predicted pressure drop of the refrigerant along the capillary tube is compared with that of the refrigerant (R-407C) experimental data. The comparison showed a good agreement with the same trend and numerical profile as that of Sami et al. [21] data.

Comparison with ASHRAE Charts:
The comparison of the present work for the performance prediction of capillary tubes with available charts in the ASHRAE [12] which are based on:

- i- The knowledge of the entering capillary tube pressure and flow condition, sub-cooled, saturated and two-phase, and
- ii- Fixed value of the capillary tube geometry, (d), and (L).

Then refrigerant mass flow rate, ($\dot{m}_{ref}$, for the above conditions is obtained.

Let us assume the refrigerant (R-22) is saturated at the inlet to the tube with a pressure of (20) bar. The selected geometry is (1.63) mm and (203) cm for the capillary tube diameter and length respectively. Then from the charts of ASHRAE [12], the refrigerant mass flow rate to satisfy the above operating conditions for the chocked flow condition equal to (43.5) kg/h. With this refrigerant mass flow rate, ($\dot{m}_{ref}$), entering capillary tube saturation pressure and tube diameter, the present model predicted a tube length ranged from (191) cm to (215) cm for the chocked condition, table (1). This range of tube length as predicted when using different friction factor formulae. This prediction fell within (± 6%) discrepancy when comparing with the ASHRAE [12] charts.

Table (1) also shows the prediction of the present model with the ASHRAE [12] charts for the case where two-phase flow exists at the entrance to the capillary tube. Here, the results showed that the Blasius correlation predicted very close tube length to that of the ASHRAE [12] charts within (1.6 %). Whereas, the other equations predicted the length accuracy within (15%) of that estimated by ASHRAE.

It is interesting to compare the performance of both refrigerants (R-22) and the alternative (R-407C) for the same mass flow rate and capillary tube geometry, diameter and length. The results of the experimental data and the present model showed that these refrigerants will establish pressure ratios as (1.8) and (2) for (R-22) and (R-407C) respectively. This highlights the idea of using the latter as a substitute for the former in the existing cooling units. However, it is very important to note that the operating temperature range will not be the same.

Conclusions:
The numerical model presented in the this investigation provides a technical tool for estimation of the capillary tube geometry with acceptable accuracy. This geometry will establish the proper diameter and length for stable operation of the vapor compression refrigeration unit. A good agreement was obtained with the available experimental data and those predicted by other methods. The following points can be withdrawn from the present model prediction:

- i- The pressure drop increases with the capillary tube length due to the increase in the friction effect with the flow direction. The predicted pressure drop for each increment showed a decrease as the flow proceeds toward the exit end due to the two-phase effect.

ii- The velocity of the refrigerant exhibited quite a smooth flowing path and revealed a high increase in the tail portion of the tube due to the increase in the specific volume of the mixture which in turn shows a rise for the velocity.

iii- The friction factor correlations tested with this model showed their capability of handling the different conditions expected during the stable operation.

iv- The present model is suitable for the prediction of the capillary tube geometry when using alternatives of refrigerants in the refrigeration units. Although, the sub-cooled flow at the inlet of the capillary tube was not examined, the present model is qualified to sustain such condition in the numerical step by step solution.

Table (1): The comparison of the predicted capillary tube performance with the ASHRAE [12] estimations for the refrigerant (R-22).

Reference No.	Pcond. (bar)	Tcond. (°C)	xinlet (%)	$\dot{m}_{ref}$ (kg/h)	Lcap. (cm)	dcap. (mm)
ASHRAE [12]	20	51	0	43.2	203.0	1.63
P.Work [B-18]	20	51	0	43.2	215.0	1.63
P.Work [H-18]	20	51	0	43.2	192.0	1.63
P.Work [19]	20	51	0	43.2	191.2	1.63
ASHRAE [12]	20	51	5	48.0	152.0	1.68
P.Work [B-18]	20	51	5	48.0	149.5	1.68
P.Work [H-18]	20	51	5	48.0	131.6	1.68
P.Work [19]	20	51	5	48.0	131.2	1.68

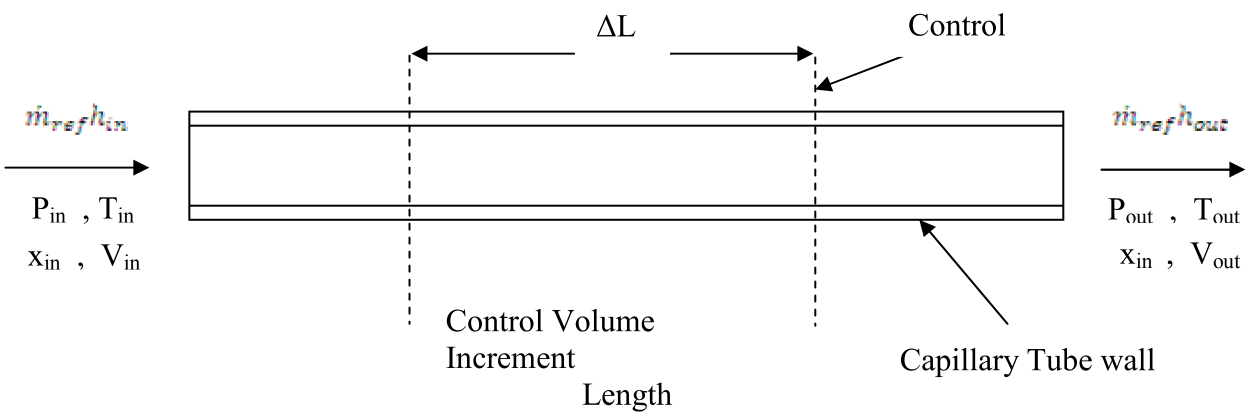

Figure (1): A Schematic diagram for the control volume of fluid flowing inside the capillary tube.

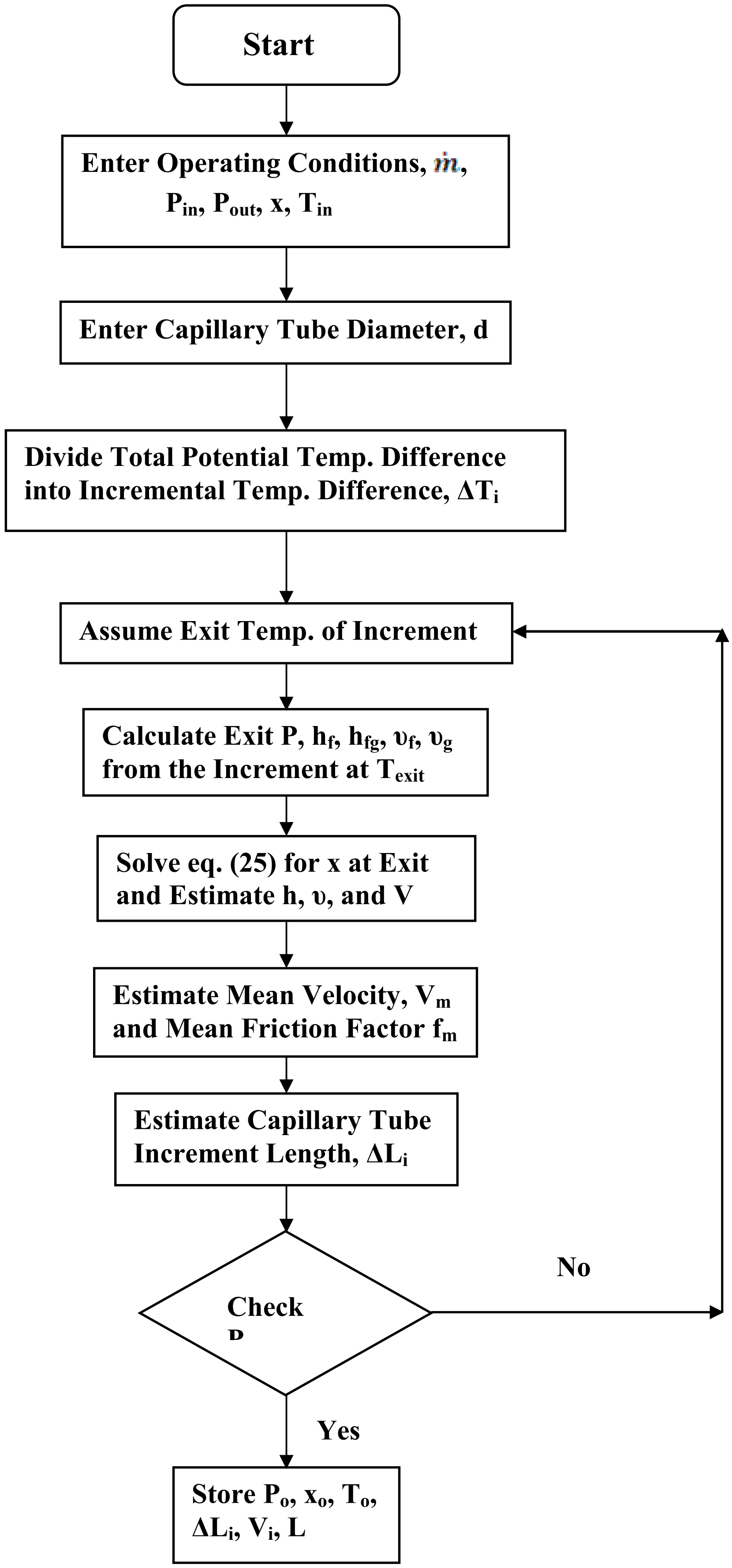

Figure (2): A Flow Chart of the Present Work Computer Program.

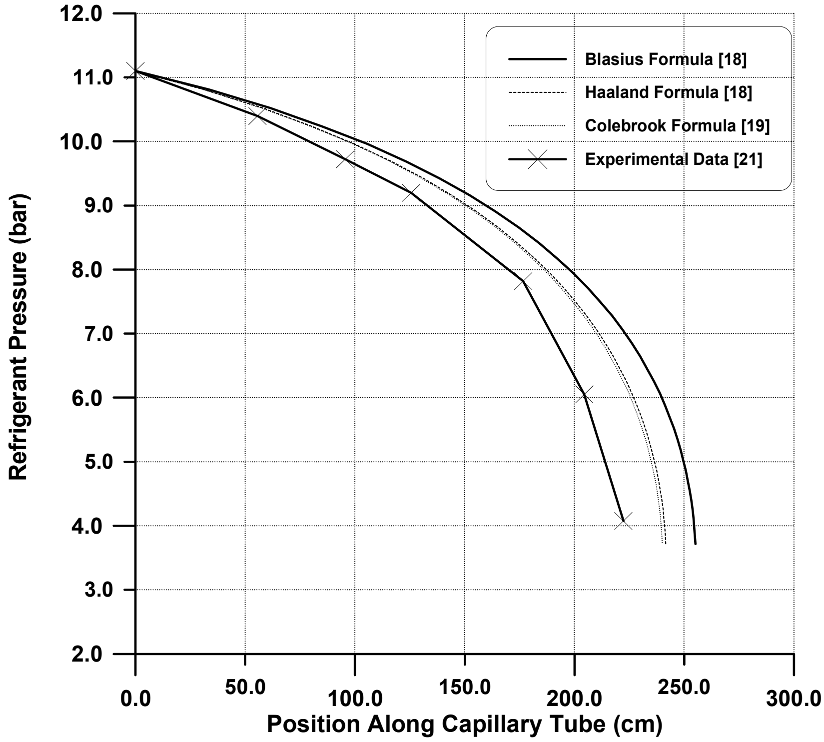

Figure (3): The predicted pressure profile of the refrigerant (R-22) compared to the data of Sami et al. [21] for different friction factor estimation methods.

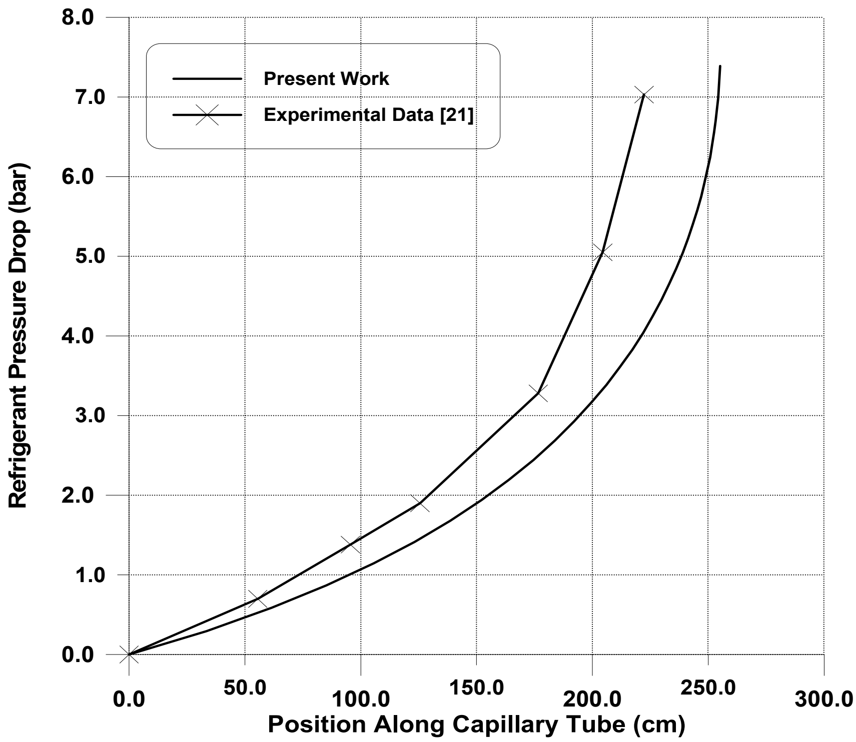

Figure (4): Comparison of the predicted refrigerant (R-22) pressure drop profile with the experimental data of Sami et al. [21].

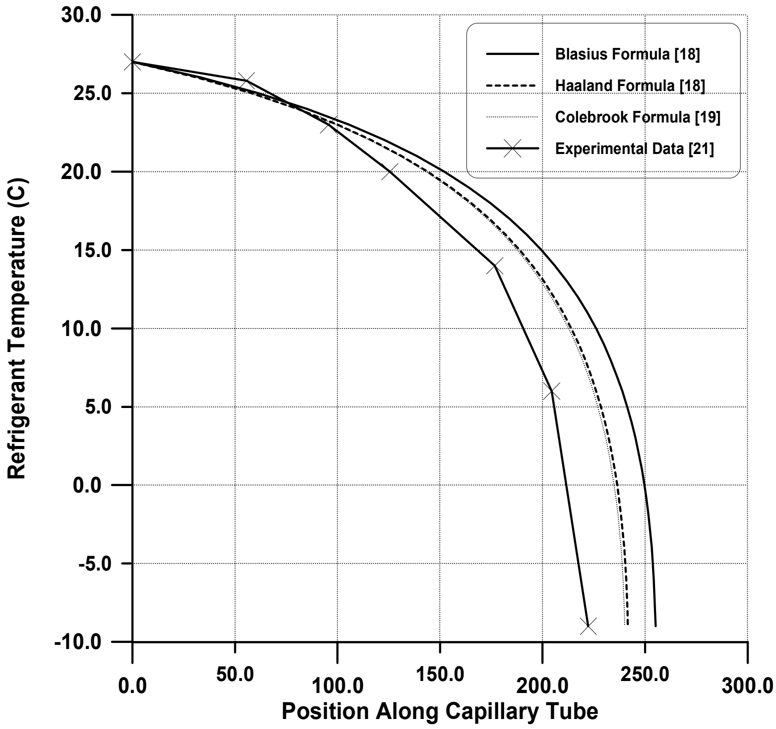

Figure (5): Comparison of the predicted temperature profile for the refrigerant R-22 with that of Sami et al. [21] experimental data.

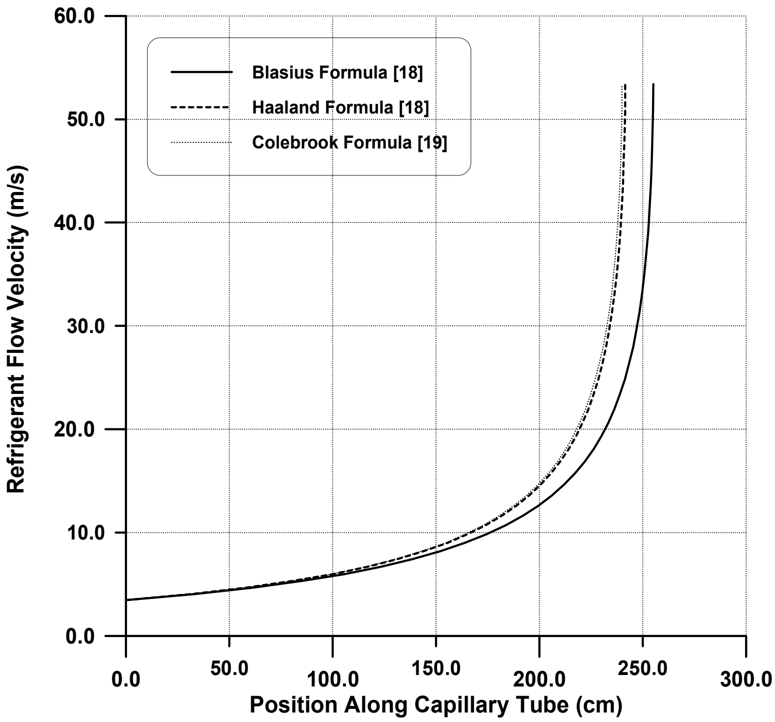

Figure (6): The present model predicted refrigerant (R-22) velocity profile for Sami et al. [21] operating conditions of the capillary tube.

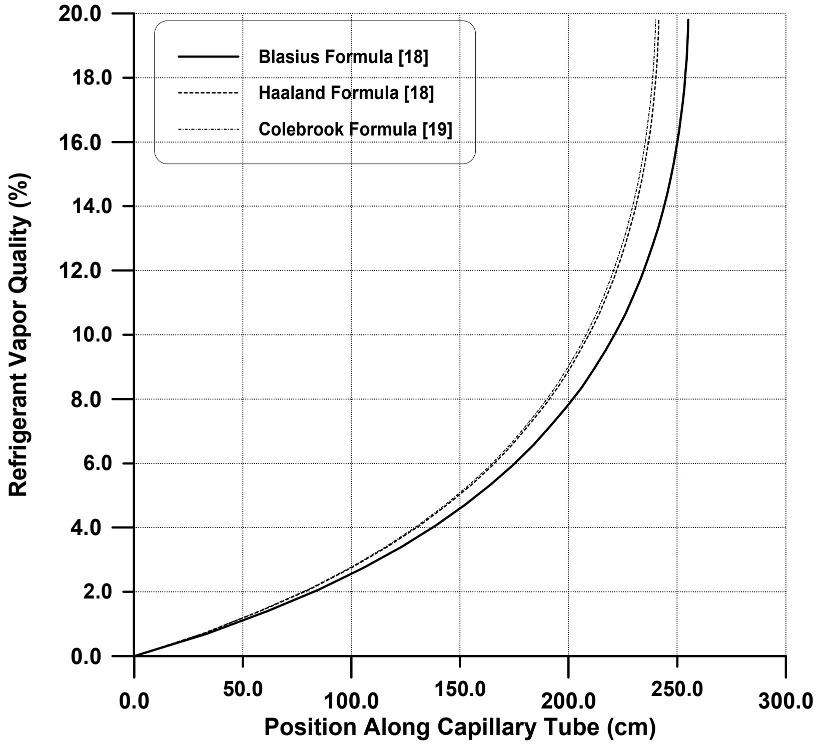

Figure (7): The present model predicted refrigerant (R-22) vapor quality distribution for Sami et al. [21] operating conditions.

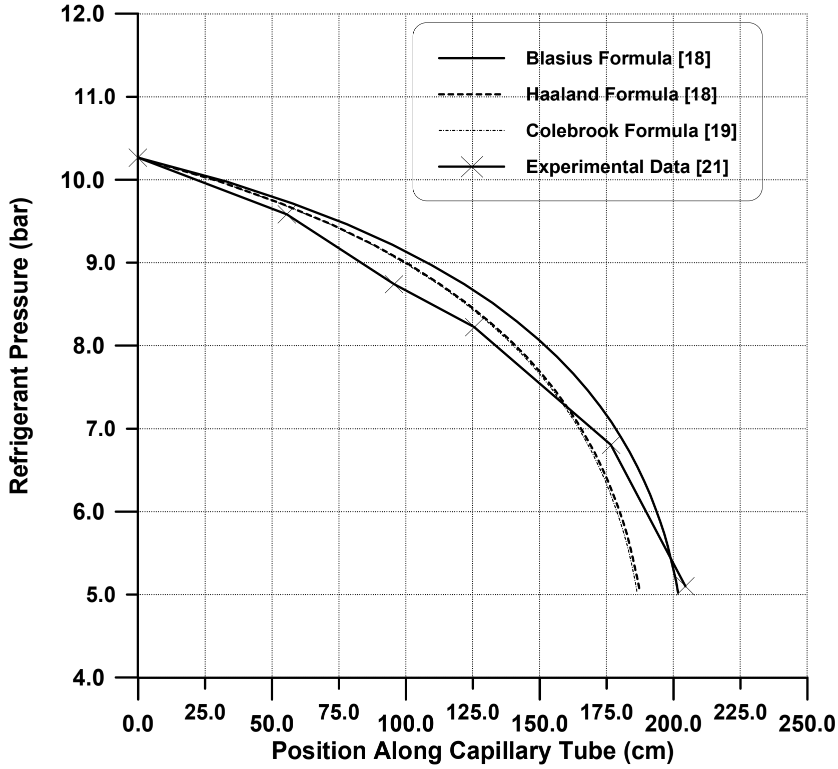

Figure (8): Comparison of predicted pressure profile for refrigerant (R-407C) of the present model with Sami et al. [21].

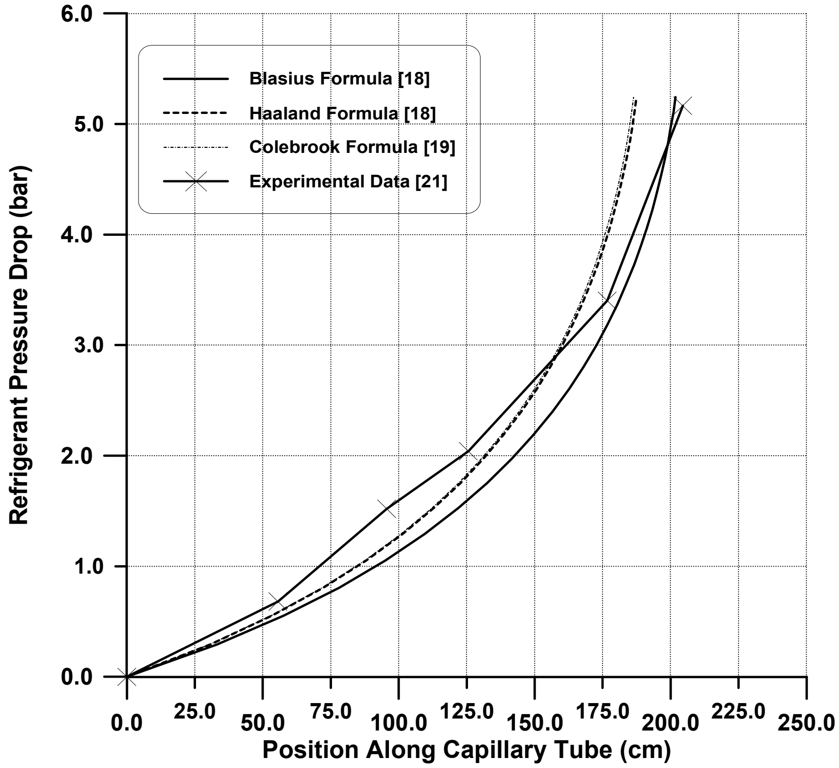

Figure (9): Comparison of predicted pressure drop for refrigerant (R-407C) of the present model with Sami et al. [21] experimental

Nomenclature:

A : Area, (m2)
d : Tube diameter, (m)
e : Specific total energy, (kJ/kg)
f : Friction factor, (Dimensionless)
F : Force, (N)
g : Gravitational acceleration, (m/s2)
h : Specific fluid enthalpy, (kJ/kg)
hfg: Latent heat of vaporization, (kJ/kg)
L : Length, (m)
$\dot{m}$: Mass flow rate, (kg/s)
ni : Increment number
N : General property of the control volume
P : Fluid Pressure
$\dot{Q}$: Heat load, (kW)
Re : Reynolds number, (Dimensionless)
t : Time, (s)
T : Temperature, (ºC)
u : Specific internal energy, (kJ/kg)
υ : Specific fluid volume, (m3/kg)
V : Fluid flow velocity, (m/s)
$\forall$: Volume, (m3)

$\dot{W}$: Work energy rate, (kW)
x : Vapor quality, (Dimensionless)
X : Axial position in the flow direction, (m)
z : Elevation, (m)

Greek Letters:

ε : Specific property
ϵ : Roughness of tube surface, (mm)
ΔL: Control volume length, (m)
ΔT : Temperature difference, (ºC)
μ : Fluid viscosity, (Pa.s)
ρ : Fluid density, (kg/m3)
Σ : Summation of variables

Subscripts:

c : Cross sectional
cap: Capillary tube
cond: Condenser value
C.V: Control volume
C.S: Control surface boundary
f : Liquid phase
g : Vapor state
in : Inlet condition
m : Mean value
net : Net of variable
out: Outlet condition
ref : Refrigerant

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