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Adiabatic and Separated Flow of R-22 and R-407C in Capillary Tube

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Abstract

In this paper adiabatic flow in capillary tube is analyzed and modeled for R-22 and alternative R-407C. The equations of continuity, energy and pressure drop through a capillary tube are presented. A mathematical model of sub–cooled flow region and two–phase flow region is developed. The results of the calculation compared with experimental data presented in the technical literature will be shown in the present article in order to validate the model developed. This numerical model is capable of providing an effective means to analyze capillary tube performance to optimize and control an R – 22 and R – 407C in an air – conditioning systems.

Keywords: Capillary tube; alternative refrigerant; adiabatic flow.

الجريان الادياباتي المنفصل لمائعي التثليج R-22 وR-407C داخل الانبوب الشعري

الخلاصة

تم في هذا البحث بناء انموذج رياضي للجريان الادياباتي داخل الانبوب الشعري ولمائعي التثليج (R-22 و R-407C), باستخدام معادلات الاستمرارية والطاقة وتمثيل هبوط الضغط خلال الانبوب الشعري . الانموذج الرياضي يتضمن بناء انموذج لمنطقة الجريان الاحادي الطور السائل وانموذج اخر لمنطقة الجريان الثنائية الطور . تم مقارنة النتائج الحسابية مع النتائج العملية المتوفرة في البحوث السابقة لغرض التحقق من صحة الانموذج وامكانية استخدامه لتحليل اداء منظومات التثليج الانضغاطية للوصول للامثلية.

Introduction

The capillary tube serves almost all refrigeration system, and its application extends up to refrigerating capacities of the order 10 kw. Such as household refrigerator, freezers and room air conditioners. Recently its application has been extended to include larger units such as unitary air-conditioners in size up to 35.2 kw. Though a capillary tube it self is a very simple piece of hardware, the flow within the capillary tube is very complex. Liquid refrigerant enters the capillary tube, and as it flows through the tube, the pressure drops because of friction and acceleration of the refrigerant. Some of the liquid flashes into vapor as the refrigerant flows through the tube. Many refrigeration unit use a capillary tube because have advantages of being simple, have no moving parts, and are inexpensive. They also allow the pressures in the system to equalize during the off cycle and as a result the starting torque of compressor is reduced with respect to the practical use of capillary tube. The main concern of the manufactures of these systems is the determination of the length of the tube when a refrigeration capacity and the parameters at the inlet and exit of the tube are given.

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Fig. (1) shows a typical pressure distributions of a refrigerant flowing through a capillary tube, where the flow is divided into four regions: sub-cooled region (zone I: when $p \geq p_{sat}$, l, $x_g = 0$), meta-stable liquid region (zone II: when $p_{sat,l} > p \geq p_v$, $x_g = 0$), meta-stable two-phase region (zone III: when $p_v > p \geq p_{sat,g}$, $0< x_g \leq x_{g\ equil}$, $0 \leq y \leq 1$), and thermodynamic equilibrium two-phase region (zone IV: when $p_v > p \geq p_{sat,g}$, $x_{g\ equil} <x_g \leq 1$). At point a, the pressure is equal to the saturated pressure but the vaporization does not take place. A meta-stable (superheat) liquid flow occurs for a short distance until the onset of vaporization. In this region the pressure drop is almost linear. At point b, the vapor bubbles appear and pressure suddenly drops due to the beginning of the two-phase region. However, this is a meta-stable region because of the existence of superheat liquid together with saturated liquid and vapor fluid [3]. After point c, the local thermodynamic equilibrium state is reached.

Many experimental and numerical studies on of the fluid flow within adiabatic capillary tubes have been performed. Many investigators, most notably Cooper et al. [1], Mikol [2] and Li et al. [3] have studied capillary tubes of various diameters and length. In the experiment, the pressure and temperature before the capillary inlet and a long it, and after the tube outlet were controlled. The results showed that the flow within the capillary tube could be divided into two regions, a liquid region in which only liquid flows, and a two – phase region in which liquid and vapor flow simultaneously. The transition point is called the flash point. In the theoretical modeling, the description of an adiabatic, homogeneous equilibrium two-phase flow through the capillary tube was presented by Goldstein[4], Ahmed [5], Kim et al. [8] and Pakamat [6] although in these works, the meta-stable flow phenomenon was calculated , the predicted results compared well with the experimental data available in the literature . To take into account of the slip exists between the phases, drift flux model by Li et al. [3] and Liang & Wong [7] were presented. These models include the effects of the thermodynamic non-equilibrium vaporization and relative velocity between the two-phases. Kim et al. [8] developed a dimensionless correlation on the basis of experimental data of adiabatic capillary results for R-22, R-407C (R-32/R-125/R-134a, 23/25/52 wt.%) and R-410A (R-32/R125, 50/50 wt %). The results for straight capillary tubes were compared with capillary tubes with coiled diameters of 40, 120 and 200 mm. The mass flow rates for coiled capillary tubes are quite reduced when compared with those for straight capillary tubes especially for the cases where the coiled diameter is reduced.

Mathematical model

The flow of refrigerant in a capillary tub used as an expansion device in refrigerating system is divided into four regions ; single-phase liquid Sub-cooled region, single phase liquid Meta-stable region, liquid-vapor two-phase Meta-stable region and liquid-vapor two-phase Thermodynamic equilibrium region as shown in (fig.1) .

The following assumptions are made in formulating the separated model:

- The capillary tube is straight, horizontal, constant inner diameter and uniform roughness surface.
- Flow through the capillary tube is one dimensional, adiabatic, pure and mixed refrigerant (no oil is entrained).
- Thermodynamic equilibrium between the phases and consider the meta-stable two-phase flow region.

Sub-cooled single-phase flow region (zone I)

The single-phase sub-cooled liquid region is the region from the capillary tube inlet to the position where the saturation pressure corresponds to the temperature at the capillary inlet. The conservation of momentum gives:

$$\left(\frac{dp}{dz}\right) = -\frac{f * G^2}{2 D_i \ r_f} \qquad ......(1)$$

Where (f) is friction factor in single-phase flow region can be calculate from equation (2) [10], and thermodynamic and thermophysical properties for R-22 and R-407C given as [11].

$$\left.\begin{aligned}
f &= \left[\left(\frac{8}{\mathrm{Re}}\right)^{12} + \frac{1}{(A+B)^{1.5}}\right]^{\frac{1}{12}} \\
A &= \left\{2.457 \ln\left(\frac{1}{\left(7/\mathrm{Re}\right)^{0.9} + 0.27\, e/D_i}\right)\right\}^{16} \\
e &= 3.27 \times 10^{-4} \\
B &= \left(\frac{37530}{\mathrm{Re}}\right)^{16} \\
\mathrm{Re} &= \frac{G D_i}{m_f}
\end{aligned}\right\} .(2)$$

The length of sub-cooled region can be calculated from equation (1).

Meta-stable liquid region (zone II)

The refrigerant pressure at point (a) (fig.1) the flow becomes two-phase, but remains in a liquid state after saturation line. This phenomenon is called “delay of vaporization “. There are many researchers [1, 2, 4, 5, 12] verify it and some other see it by using a glass capillary tube in experimental rig. The delay of vaporization pressure can be calculated as [12].

$$\frac{(P_{sat} - P_{vap})\sqrt{K T_{sat}}}{s^{1.5}} = 0.679 \left(\frac{v_g}{v_g - v_f}\right) \mathrm{Re}^{0.914} \left(\frac{\Delta T_{sc}}{T_c}\right)^{-0.208} \left(\frac{D_i}{\sqrt{\left(K T_{sat}/s\right)} \times 10^4}\right)^{-3.18} \qquad ....(3)$$

Where, K is the Boltzmann constant (1.380662×10^{-23} J/K).

The meta-stable liquid region length can be determined in the same way as the sub-cooled region.

Meta-stable two-phase region (zone III)

After vapor appears, there exists a meta-stable two-phase region before the fluid reaches thermodynamic equilibrium. Feburie et al. [13] suggested that the flow can be separated into three states: superheated liquid (subscript m), saturated liquid (subscript f) and saturated vapor (subscript g). In this zone, the governing equations are used to evaluate the mass flow rate, the pressure and the mean enthalpy respectively. A variable y, defined as mass ratio of total saturated phase to total phase,

$$y = (m_f + m_g)/(m_f + m_g + m_m) \qquad ….(4)$$

Is used to evaluate the superheat liquid mass flow. This variable is evaluated by correlation proposed by Feburie et al. [13]:

$$\frac{dy}{dz} = 0.02 \left(\frac{p D_i}{A}\right)(1-y)\left[\frac{P_{sat} - P}{P_c - P_{sat}}\right]^{0.25} \qquad ....(5)$$

The mean enthalpy (obtained from the energy equation) is related to the individual enthalpies through their mass fractions, that is:

$$\left.\begin{aligned}
\bar{h} &= x_m h_m + x_f h_f + x_g h_g \\
&= (1-y) h_m + (y - x_g) h_f + x_g h_g
\end{aligned}\right\} ..(6)$$

This expression allows the evaluation of the vapor mass fraction (x_g). Following Feburie et al. [13], the temperature of the superheat liquid has been assumed constant in this region, the averaged temperature distribution is defined as:

$$T = (1 - y)T_m + yT_{sat} \qquad \text{..........(7)}$$

When y approaches unity, the superheated liquid vanishes and the flow process enters to equilibrium two-phase flow.

Equilibrium two-phase region (zone IV)

Separated flow model can be consider in this region, is more appropriate for use in such region in which one phase is completely separated from the other within the flow channel. The model as developed by Collier [14] gives the expression for the two-phase pressure gradient in separated flow as:

$$-\frac{dP}{dZ} = \frac{A + B + C}{D} \qquad \text{.......... .(8)}$$

Where

$$A = \frac{2 f_{fo} G^2 v_f \Phi_{fo}^2}{D_i} \qquad \text{.......... (9)}$$

$$B = G^2 \frac{dx}{dz}\left[\left\{\frac{2xv_g}{a} - \frac{2(1-x)}{(1-a)}v_f\right\} + \frac{da}{dx}\left\{\frac{(1-x)^2}{(1-a)^2}v_f - \frac{x^2}{a^2}v_g\right\}\right] \text{..(10)}$$

$$C = g\sin q\left[r_g a + r_f(1-a)\right] \qquad \text{....(11)}$$

$$D = 1 + G^2\left[\frac{x^2}{a}\left(\frac{dv_g}{dp}\right) + \frac{da}{dp}\left\{\frac{(1-x)^2}{(1-a)^2}v_f - \frac{x^2}{a^2}v_g\right\}\right] \text{ .(12)}$$

Eq. (8) above assumes negligible liquid compressibility. The separated flow model takes into account the fact that the two-phase can have different velocities; it may be developed with various degree of complexity. Using the separated flow model will take into account mostly interfacial slip with other data and assumption characteristic for the flow structure therefore as a first approximation, the pressure drop in the annular flow region is obtained using eq.(8) in an integrated form

$$\Delta p_{tp} = \left[\left(\frac{2 f_{fo} G^2 v_f (\Delta z)}{D_i (x_2 - x_1)}\right)\int_{x_1}^{x_2}\Phi_{fo}^2 dx + G^2 v_f\left(\frac{x^2 v_g}{a v_f} + \frac{(1-x)^2}{(1-a)} - 1\right) + \frac{g(\Delta z)\sin q}{(x_2 - x_1)}\int_{x_1}^{x_2}\left[r_g a + r_f(1-a)\right]dx\right] \text{.......(13)}$$

x_1 and x_2 in eq.(13) are the quality values at the beginning and end of test section length in separated flow eq. (12) assumes that flashing and vapor compressibility are negligible and that the specific volume and friction factor remain constant. Two-phase friction factor multiplier which can be determined by using collier equation [14] as:

$$\Phi_{fo}^2 = \Phi_f^2 (1 - x)^2 \frac{f_f}{f_{fo}} \qquad \text{..........(14)}$$

By using Blusies equation

$$f_f = 0.046\left[\frac{1}{\left(GD_i / m_f\right)^{0.2}}\right] \qquad \text{.......... .(15)}$$

And from same reference [14]

$$\Phi_f^2 = \frac{1}{(1 - a)^2} \qquad \text{.........(16)}$$

Where α is the void fraction can be calculated from [14] as:

$$a = \frac{x r_f}{x r_f + (1 - x) r_g S} \qquad \text{..........(17)}$$

The expression for the slip ration defining the vapor to liquid velocity ration as $S = V_g/V_f$, according to Schulz correlation [15] as:

$$S = 1 + (A \times TH_o^B x^C) \qquad \text{............(18)}$$

Where,

$$TH_o = G^2 \frac{v}{p} \qquad \text{.........(19)}$$

A, B and C are constant and equal (2.475), (0.915) and (- 0.25) respectively.

$$\frac{f_f}{f_{fo}} = \frac{\left(GD_i / m_f\right)^{0.2}}{\left(G(1 - x)D_i / m_f\right)^{0.2}} = \frac{1}{(1 - x)^{0.2}} \qquad \text{........(20)}$$

By substituting (eq.20) in (eq.14) gives:

$$\Phi^2_{fo} = (1-x)^{1.8}\,\Phi^2_f \quad .... (21)$$

For constant void fraction, (eq. 16) and (eq. 21) show that Φ^2_{fo} is also constant. In two-phase region, the viscosity is calculated by the Dukler equation [16] as:

$$m_{tp} = \frac{x v_g\, m_g + (1-x) v_f\, m_f}{x v_g + (1-x) v_f} \quad ....(22)$$

Results and discussion

Using the finite difference method solved equation (13), to validate the separated model presented above. Comparison has been made with available experimental results by Kim [8] and Koizumi [9] . Experimental results for adiabatic two phase flow of R-22 in capillary tube where presented by Koizumi [9] and for R-407C presented by Kim [8]. figures (3, 4) show that the numerical results of pressure distributions along the capillary tubes obtained from the present model, and agree well with the experimental data are presented by Kim [8] and Koizumi [9] . The good agreement between those results indicates that the present model is suitable for the prediction of refrigerant two phase flow in capillary tube. figures (5, 6, 7) show comparison performance R-22 and R-407C , from these figures conclusion it that the length of tube for R-407C is longer than that for R-22 for the same condition with (20 %), because the viscosity of R-407C is less than R-22.

Figure (8) show the effect of mass flow rate on the pressure drop along capillary tube, with increase in the mass flow rate the friction increase as a result. So the length of capillary tube will decrease.

Figure (9) show the effect of subcooling on the pressure drop along capillary tube length, when increase subcool temperature there is an increase in the length of capillary tube. Figure (10) show the effect of condensing temperature on the pressure drop along capillary tube length, when increasing condensing temperature there is an increase in the length of tube.

Conclusions

Mathematical models for flows in capillary tube have been developed using the two phase separated model. The models have been validated with the existing data available in the literature. Adiabatic flow in a capillary tube is analyzed for R-407C, which are a non-azeotropic mixture and an alternative to R-22. Numerical results for the flow of R-407C in capillary tube compared with the performance data for R-22. The results show that this numerical model is adequately precise and valuable to provide an effective means by which analyzing capillary tube performance to optimize and control an R-407C air conditioning system.

Nomenclature

A cross-section area (m^2)
D diameter (m)
f friction factor
g acceleration due to gravity (m/s^2)
G mass velocity ($kg/m^2\,s$)
h enthalpy (kJ/kg)
K Boltzmann's constant
L length (m)
m mass (kg)
m^o mass flow rate (kg/s)
p pressure (Pa)
pv pressure of vaporization (Pa)
Re Reynolds number
S slip ratio
T temperature (K)
v velocity (m /s)
υ specific volume (m^3/kg)
x vapor quality
y mass ratio of total saturated phase to total phase
z axial coordinate

Greek letters

Δp pressure drop (pa)
ΔT_{sc} degree of sub-cooling (K)
ε Roughness (m)

α void fraction
ξ two-phase frictional multiplier
μ dynamic viscosity (kg /m s)
θ tube inclination angle (rad)
ρ density (kg /m^3)
σ surface tension (N /m)

Subscripts

c critical
f saturated liquid condition
ℓ based on liquid part of total flow
g saturated vapor condition
i inner
fo based on total flow assumed to be liquid
m superheated liquid
sat saturation
sc sub-cooled
tp two-phase
vap vaporization

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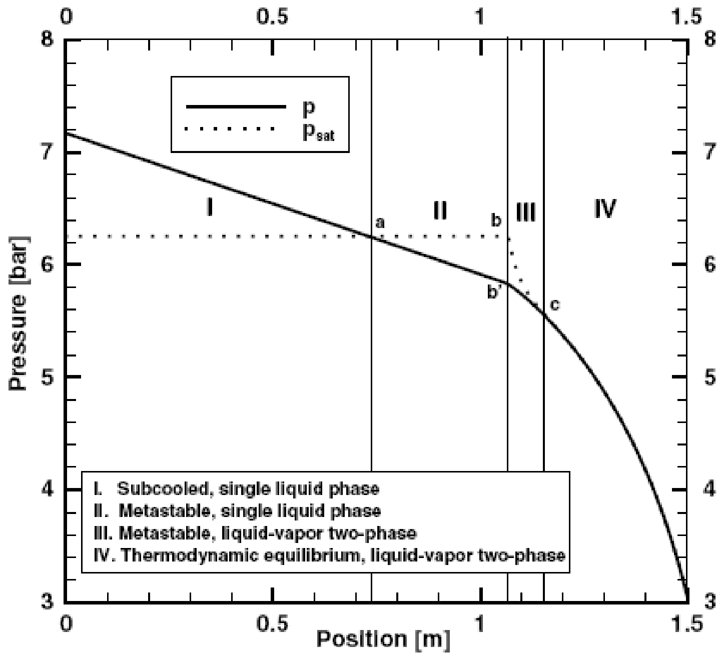

Figure (1) Typical pressure distribution along a capillary tube [12]

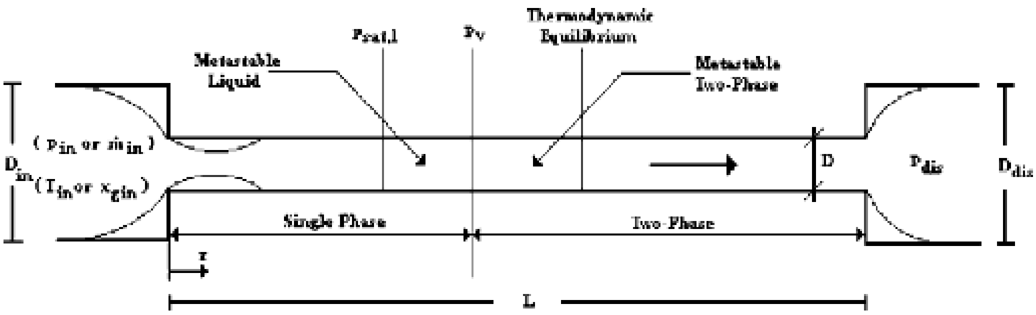

Figure (2) Schematic diagram of a capillary tube [12]

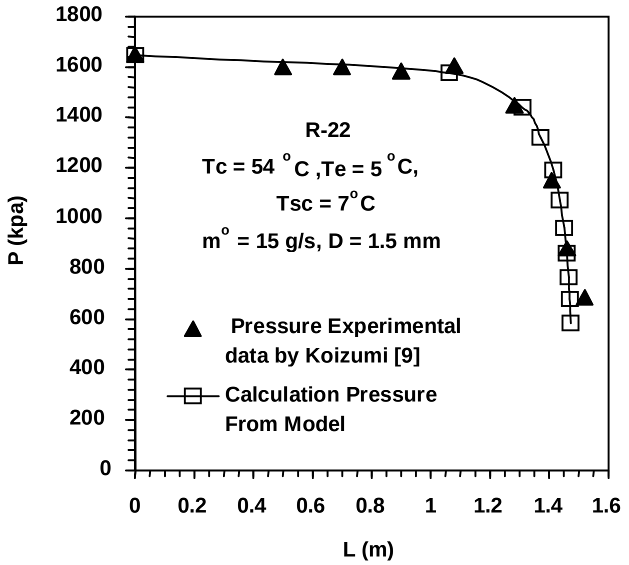

Figure (3) Comparison between measured and predicted results for R-22

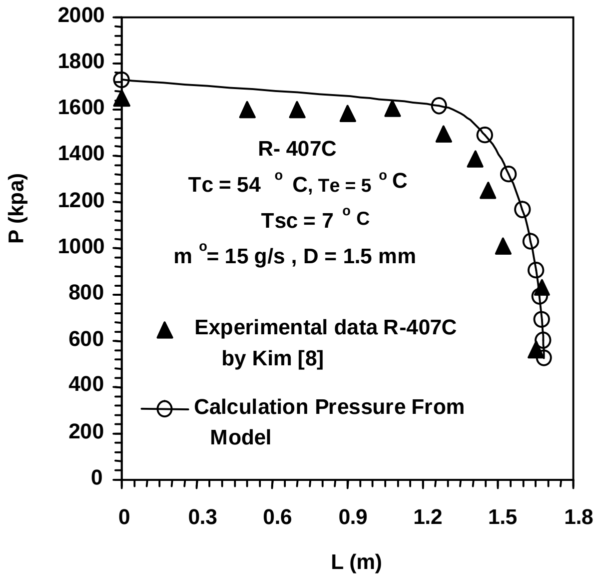

Figure (4) Comparison between measured and predicted results for R-407C

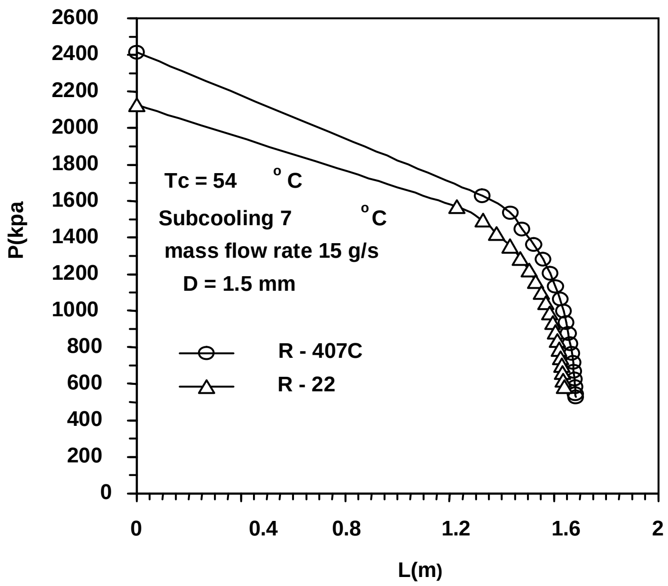

Figure (5) Comparison of pressure drop with the length of tube for R-22 and R-407C

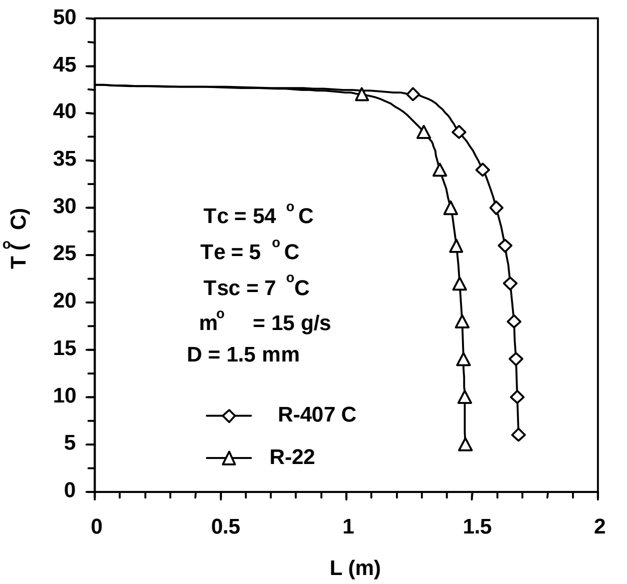

Figure (6) Comparison of temperature drop with the length of tube for R-22 and R-407C

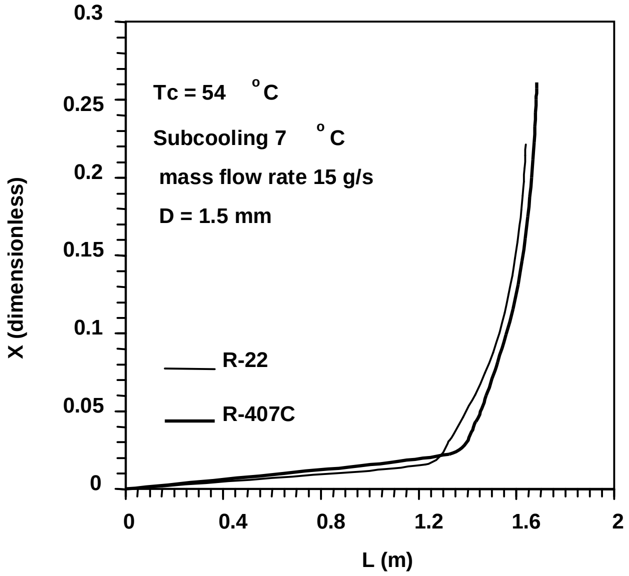

Figure (7) Comparison of quality with the length of tube for R-22 and R-407C

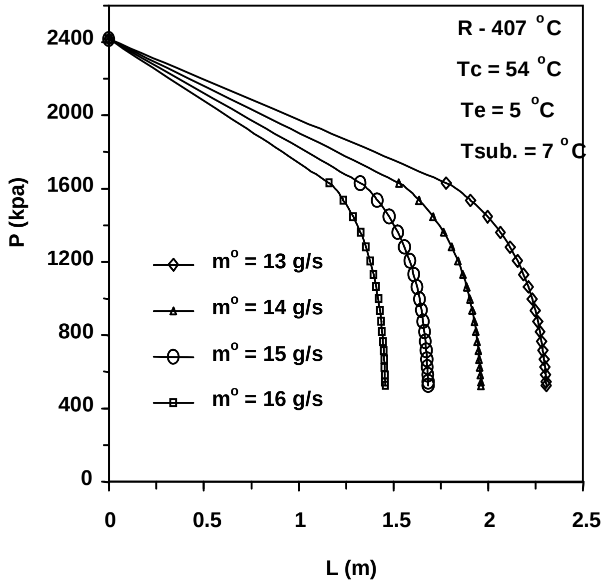

Figure (8) Effect of mass flow rate on pressure drop along the tube length

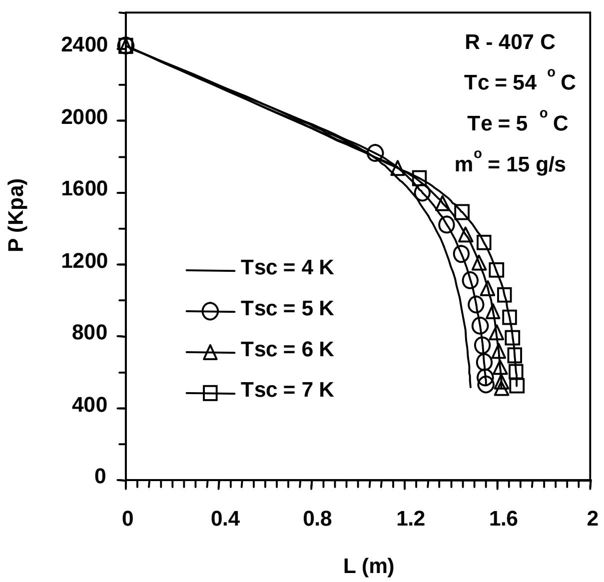

Figure (9) Effect of subcooling on pressure drop along the tube length

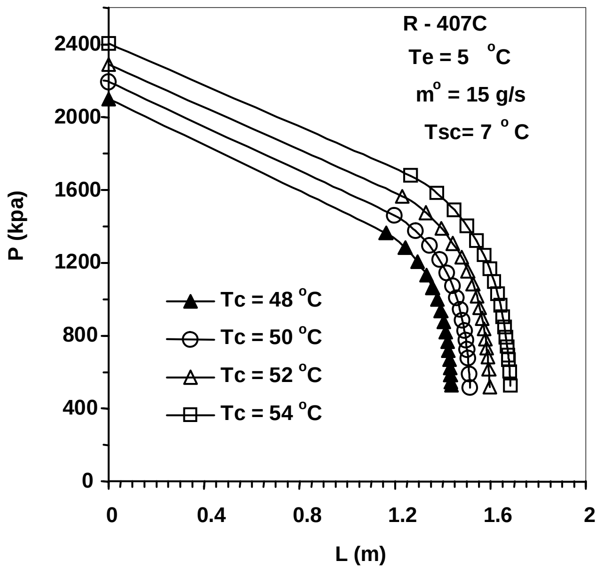

Figure (10) Effect of condensing temperature on pressure drop along the tube length