The Effects of Friction Factors on Capillary Tube Length

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ABSTRACT

Capillary tubes are one of the major components of Vapor Compression Refrigeration Systems. Research efforts have shown that some parameters such as friction factor, dryness fraction, and Reynolds number affect the required length and diameter of a capillary tube for a given refrigeration capacity. Consequently, many friction factor models were developed by researchers based on these parameters. Using these developed friction factor models in the estimation of capillary tube lengths yielded different results. It was discovered that those friction factors that are based on the dryness fraction yielded better results as compared to those in the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Handbook.

The discrepancy observed from those friction factors that are based on the dryness fraction ranges between 0.05% and 0.85% while others range between 0.5% and 1.9% above that of ASHRAE under the same conditions. Furthermore, both McAdams' and Duckler's equations for two-phase viscosity were employed so that the deviation in the estimated lengths could be compared. The tube lengths generated by combining various friction factor models with McAdams' equation are much closer to that of ASHRAE standard than those of Duckler's equation. On the average, the estimated lengths using McAdams' and Duckler's equations exceeded ASHRAE standard by 1.65% and 4.13% respectively.

(Keywords: capillary, refrigeration, friction factor, tube length, mathematical model, thermodynamic properties, pressure drop)

INTRODUCTION

The most widely used cycle in the field of refrigeration and air–conditioning is the vapor-compression cycle. In this cycle, vapor is compressed then condensed to liquid form; following this, the pressure is dropped so that fluid can evaporate at low pressure. This cycle is performed by four major components, namely: the compressor, condenser, evaporator, and expansion device. Each of these components has its own peculiar behavior, and each component is influenced by conditions imposed by the other members of the quartet (Stoecker and Jones, 1982).

The influence of one component on the other results in changes in the thermal properties of the working fluid. As a result, balanced points must be reached between these components for effective performance. Stoecker and Jones (1982) reported on balanced points for the first components, while Akintunde (2004) extends the work to cover the last component; the capillary tube. A mathematical model for the generation of capillary tube length for both R12 and R22 was developed based on the thermodynamic properties of these refrigerants this model was used to develop a computer program (REF-2004), (Akintunde, 2004).

One of the major components of the vapor compression refrigeration systems is the expansion device. This serves the following purposes: reduces the pressure of liquid refrigerant; regulates the flow of refrigerant to the evaporator; and maintains the minimum possible pressure in the condenser at which all the refrigerant can condense without choke flow (Stoecker and Jones, 1982). The most common examples of expansion devices are the capillary tube and superheat controlled expansion valves. Other examples include the float valve and the constant pressure expansion valve.

Liquid refrigerant enters the expansion device and as it flows through, the pressure drops because of friction and acceleration of the refrigerant. As a result, some refrigerants flash into vapor. The amount of refrigerant that flashes into vapor depends largely on the friction factor, tube diameter, and flow rate (Jung *et al*, 1999). It was later discovered that the appropriate length of capillary tube depends on these three factors, in order to prevent choke flow and at the same time enhance system performance, (Chen *et al*, 2000).

The capillary tube, although physically simple, is behaviorally complex. Some items of complex mathematical analysis include the friction factor for the two phase flow, the thermal contact with the suction line, the metastable flow, and the oil circulation with the refrigerant, to mention a few.

Many researchers have investigated the flow through capillary tubes and a review on the subject can be found in the works of Melo *et al.*, (1988); Schultz, (1985); Stoecker and Jones, (1982); Jung *et al.* (1999), and Akintunde (2004).

In order to model a uniform length of capillary tube for a given refrigeration capacity under a stated working condition, many researchers had presented a series of friction factors. A review of these friction factors can be found in the works of Jung *et al.* (1999) and Akintunde (2004). Most of these friction factors when tested were found to generate different capillary tube lengths compared with those in American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) handbook (2002).

In this article, another mathematical model for the estimation of capillary tube length under a predetermined condition of operation was developed. The model was a modification of Jung *et al.* (1999) and Stoecker and Jones (1982) models. The new model was based on thermodynamic properties of R12 and R22, while those of Jung *et al.* and Stoecker and Jones were based on R12, R22, and their alternatives respectively.

The mathematical model was used to develop a computer program which was used to generate data for the friction factor. The program was then used to estimate capillary tube lengths using the newly developed friction factor and the friction factors of other researchers. The estimated lengths were then compared with those in ASHREA handbook under a stated working condition.

CAPILLARY TUBE MODEL

Figure 1 shows the representation of the capillary tube used for this modeling. For the analysis the following assumptions were made: the mass flow rate ($\dot{m}$) is constant and adiabatic conditions prevailed in the capillary tube (Kim, *et al.*, 2002; Wijaya, 1992; Chang and Ro, 1996; ASHRAE handbook (2002)).

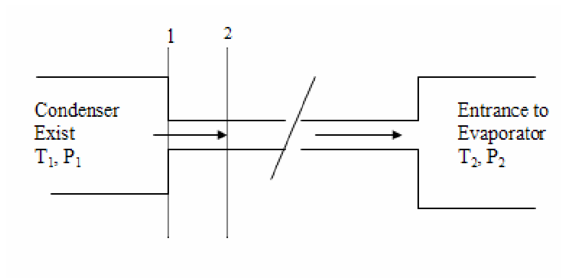

Figure 1: Schematic Diagram of a Capillary Tube.

At steady state, since the mass flow rate ($\dot{m}$) was assumed to be constant therefore the mass flow equation can be written as shown in Equation (1) (the cross-sectional area was assumed to be constant).

$$\dot{m} = \frac{V_1 A}{v_1} = \frac{V_2 A}{v_2} \qquad \textbf{(1a)}$$

or

$$\dot{m}/A = G = \frac{V_1}{v_1} = \frac{V_2}{v_2} \qquad \textbf{(1b)}$$

Generally Equation (1b) can be modified to give Equation (2).

$$V = Gv \qquad \textbf{(2)}$$

Since an adiabatic condition was assumed in the capillary tube, then an energy equation can be written as shown in Equation (3), for a flow process (Jung et al., 1999), while the momentum equation is given by Equation (4) (Akintunde, 2004)

$$500h_1 + \frac{V_1^2}{4} = 500h_2 + \frac{V_2^2}{4} \tag{3}$$

$$G(V_2 - V_1) = \left\{ (P_1 - P_2) - f\, \frac{\Delta L.G^2}{2D} v_m \right\} \tag{4}$$

The required length L can be calculated from Equation (4) in steps as given in Equation (5) and Figure 1.

$$L = \sum_i^n \Delta L \tag{5}$$

As could be seen in Equation (4), the length (L) depends on pressure variation P, friction factor (*f*), flow velocity (V), and humid volume (*v*). This is summarized in Equation (6).

$$L = \phi(\Delta P, f, V, v) \tag{6}$$

Though, the enthalpy remains constant as a result of continuous flow of refrigerant (adiabatic situation) but there will be a progressive decrease in pressure. This results in refrigerant becoming two-phase and hence the quality increases with capillary tube length. The humid volume and dynamic viscosity depend on the dryness fraction. The required equations are given in Equations (7) and (8) respectively.

$$v = (1-x)v_f + xv_g \tag{7}$$

$$\mu = (1-x)\mu_f + x\mu_g \tag{8}$$

The friction factor equations developed by various researchers are given in Table 1, representing Equations (9) to (17). Since the velocity, friction factor and humid volume changes as the refrigerant flows from point (1) to point (2), (Figure 1), the mean values of these parameters were used as given respectively in Equation (18).

$$v_m = \frac{v_1 + v_2}{2};$$

$$f_m = \frac{f_1 + f_2}{2};$$

$$V_m = \frac{V_1 + V_2}{2} \tag{18}$$

To determine dryness fraction (κ) Equations (2), (7), and (8) were combined with Equation (3), these resulted in Equation (19).

$$1000h_{f2} + 1000h_{fg2}\chi_2 + G^2 \frac{(v_{f2} + v_{fg2}\chi_2)^2}{2} = 1000h_2 + \frac{V_1^2}{2} \tag{19}$$

Equation (3) was solved for κ_2 [κ_1, is assumed to be zero- saturated or subcool liquid leaving the condenser] the dryness factor at the state point (2). For the purpose of the computer modeling the program will be terminated at the point where the Mach number becomes unity. Hence for the dynamic states, the Mach number (M_ϕ) as expressed by Jung *et al.* (1999) (Equation 20a and b) was modified by Akintunde (2004) for steady state as shown in Equation (21).

$$M_\phi = \left\{ G^2 \left[x\frac{dv_g}{dp} + (1-x)\frac{dv_f}{dp} + v_{fg}\left(\frac{dx}{dp}\right)_h \right] \phi \right\}^{1/2} \tag{20a}$$

where:

$$\phi = \left[1 + G^2 \frac{v_{fg}(v_f + xv_{fg})}{h_{fg}} \right] \tag{20b}$$

$$M_\phi = \left\{ G^2 \left[x\frac{v_{g1} - v_{g2}}{P_1 - P_2} + (1-x)\frac{v_{f1} - v_{f2}}{P_1 - P_2} + v_{fg}\left(\frac{x_1 - x_2}{P_1 - P_2}\right)_h \right] \phi \right\}^{1/2} \tag{21}$$

The property equations for R12 and R22 can be obtained from the works of Jung et al. (1999); Stoecker and Jones (1982); and Akintunde (2004). The flow chart for the computer program is shown in Figure 2.

METHODOLOGY

The friction factor equations were tested under varied conditions as follows, using 10 kW capacity refrigerator:

(i) Saturated condition of 40 C condensing temperature, 0.01 kg/s flow rate; 1.62 mm; capillary tube internal diameter and –5 C evaporation temperatures. The ambient temperature was assumed constant at 35 C.

Table 1: Friction Factor Equations (Source: Jung et al., 1999 and Akintunde, 2004).

Code	Author	Friction Factor Equations
F1	Stoecker	$f = \dfrac{0.33}{\mathrm{Re}^{0.25}}$ (9)
F2	Blausius	$f = \dfrac{0.3}{\mathrm{Re}^{0.25}}$ (10)
F3	Goldstein	f = 0.02 (11)
F4	Erth	$f = \dfrac{3.1}{\mathrm{Re}^{0.5}} Exp\left(\dfrac{1.0 - x^{0.25}}{2.4}\right)$ (12)
F5	Sami	$f = \dfrac{3.1}{[\mathrm{Re}(1-x)]^{0.5}} Exp\left(\dfrac{1.0 - x^{0.25}}{2.4}\right)$ (13)
F6	Pate *et al*	$f = \dfrac{3.49}{\mathrm{Re}^{0.47}}$ (14)
F7	Hopkins	$f = \dfrac{0.217}{\mathrm{Re}^{0.2}}$ (15)
F8	Jung *et al*	$f = \dfrac{0.25}{\mathrm{Re}^{0.27}}$ (16)
F9	Present work	$f = \dfrac{0.25}{[\mathrm{Re}(1-x)]^{0.28}}$ (17)

(ii) Varied subcooling temperatures. The subcooling temperature ranges between 2 C and 6 C in step of 0.5 C.

The generated tube lengths were compared with that of ASHRAE (which is 2.03 m) under the stated condition in (i) above. This was used as the benchmark for the comparison of the generated lengths when the various friction factors were combined with either McAdams' or Duckler's equations for two-phase viscosity. These two equations were judged the best for two phase flow by Jung *et al.* (1999). Each of the friction factors combined with either McAdams' or Duckler's equations for two-phase viscosity were used in turn to generate data under various conditions using the developed computer program. A new friction factor equation (F9) was then developed by fitting data into Equation (22) using dimensional analysis.

$$f = \phi(\mathrm{Re}, x, d, v) \qquad \textbf{(22)}$$

RESULTS AND DISCUSSIONS

The generated lengths when each of the friction factor equation was combined with the McAdams' equation for two-phase viscosity was denoted by L_M while the combination of the friction factor equations with Duckler's equation was tagged L_D. L_A stands for the corresponding length given by ASHRAE handbook under the same conditions.

Figure 3 shows the generated tube lengths under the saturated conditions, while Figures 4 to 8 show the corresponding lengths under various conditions of subcooling. Figure 9 shows the summary of the findings.

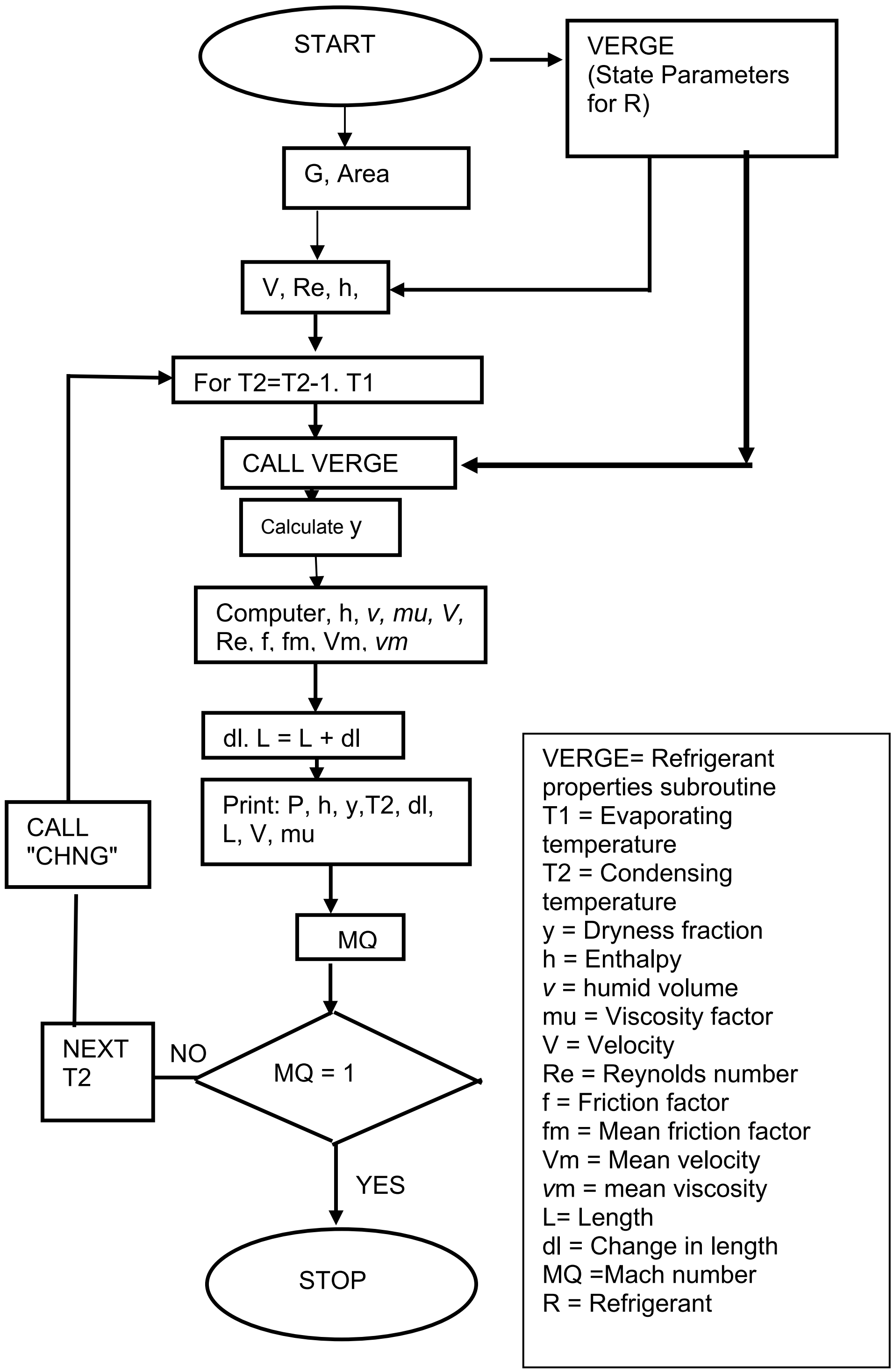

Figure 2: Flow Chart for Capillary Tube Model.

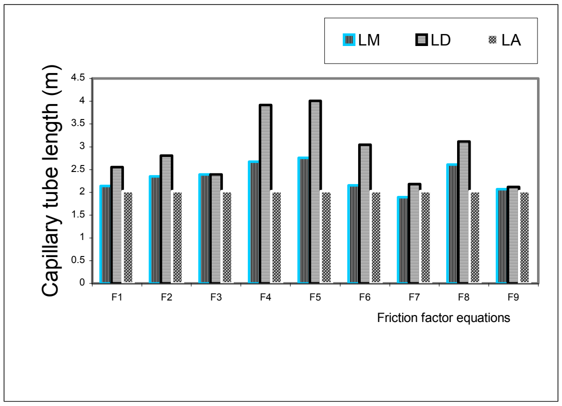

Figure 3: Capillary Tube Lengths at Saturated Points.

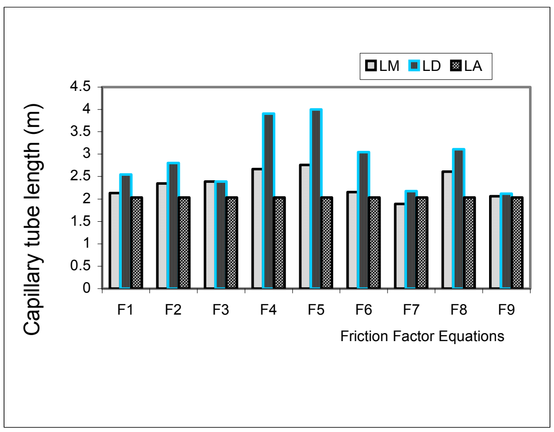

Figure 4: Capillary Tube Length at 2 C Subcooling.

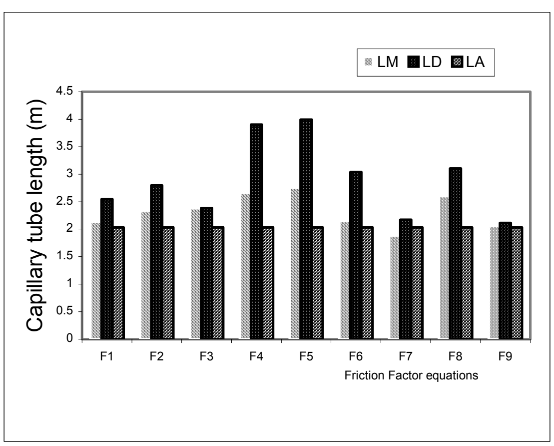

Figure 5: Capillary Tube Length at 3 C Subcooling.

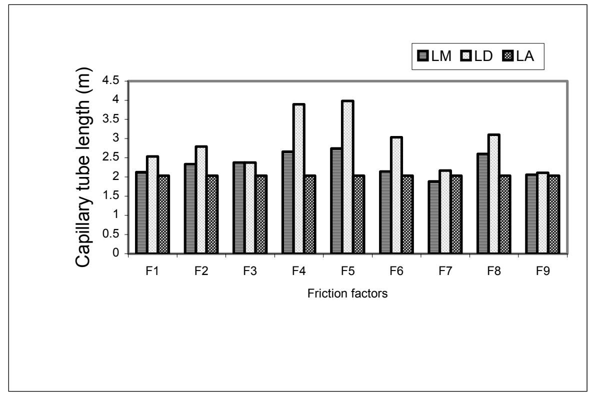

Figure 6: Capillary Tube Length at 4 C of Subcooling.

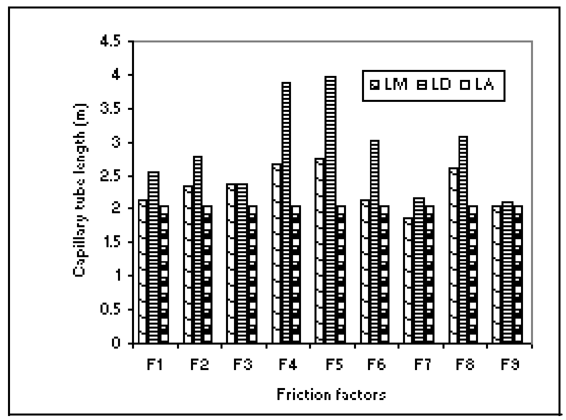

Figure 7: Capillary Tube Length at 5 C Subcooling.

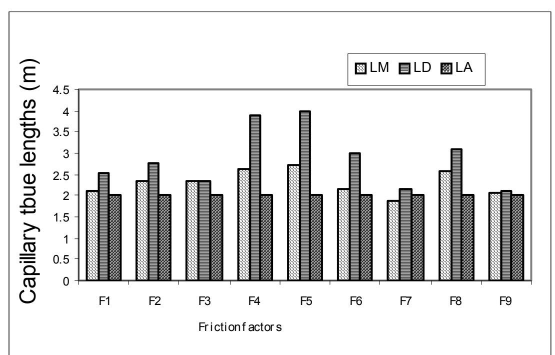

Fig. 8 Capillary tube length at 6 C subcooling

Figure 8: Capillary Tube Length at 6 C Subcooling.

Figure 3 shows that under the saturated condition, both L_M and L_D generated by the use of the friction factors are longer than that of ASHREA except for F7. A critical look shows that L_M (that is, the length generated by McAdams' equation for two-phase viscosity with the various friction factors) are much closer to that of ASHRAE than the corresponding L_D.

As could be observed in Figures 4 to 8, as the degree of subcooling increases, the various length approaches stipulated by ASHRAE change. Figure 9 also justifies this, which indicates that the tube length may likely converge at higher degree of subcooling.

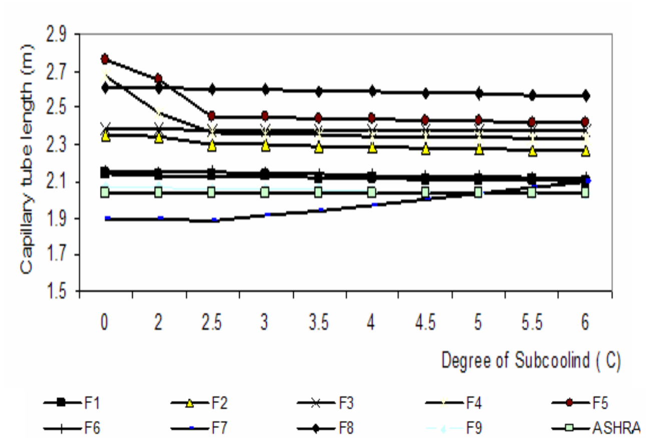

Figure 9: Variation of Generated Capillary Tube Length with Degree of Subcooling.

These friction factors were now grouped into three groups: A; B and C.

(A) Those friction factors that dependent on the Reynolds number and the dryness fraction. These are F4, F5 and F9.

(B) Those friction factors that depend on the Reynolds number alone. These are: F1; F2; F6; F7 and F8.

(C) Those friction factors that neither depends on the Reynolds number nor dryness fraction (F3).

Figures 3 to 9 show very clearly that group A, (that is, Figure 4 and 5), gives outrageous lengths as compared with the corresponding ASHRAE standard length, while F9 gives lengths that are very closed to that of the ASHRAE (that is L_A). It should be noted that both Figure 4 and 5 are given as exponential function of dryness fraction, while Figure 9 was not. From the same figures and Figure 1, 2, 6, 7 and 8 (group B) generated lengths that are much closer to that of ASHRAE as compare to Group A.

From those figures also, it could be observed that F3 generated constant length (both L_D and L_M), which could be expected since F3 is a constant. More so, it is observed that both L_M and L_D are longer that L_A. Judging from the lengths generated by group C, it is clear that the friction factor could not be given as a constant term, while the observation from group A (without F9) predicted that the friction factor may not depends, so much, on the dryness fraction and the Reynolds number. This is because the lengths generated by group C are closer to the benchmark than those of group A.

Considering the lengths generated by Figure 9, (the newly developed friction factor equation), it shows that the required capillary tube length depends on both Reynolds number (Re) and dryness fraction (x). This shows further that the dependency of friction factor on Re and x is not exponential.

CONCLUSION

The present study examined the generated capillary tube lengths based on friction factors and viscosity equations for two-phase flow, which is prevalent in the capillary tube. The lengths generated by various friction factors under stated conditions were compared with the standard lengths given by ASHRAE. It was clearly shown that the required capillary tube length for a specified condenser condition depends on both Reynolds number (Re) and dryness fraction (x) and not on either alone. It was also noted in the study that the required relationship between these two factors (that Re and x) should not be in exponential form. It was shown that the generated lengths approach the ASHRAE requirement as the degree of subcooling is increased. A new friction factor was then developed from the generated data. This friction factor developed was tested along with the others under the same conditions and was found to yield better-required length as compared with ASHRAE standard.

NOMENCLATURE

A	Surface or cross sectional area (m^2)
ϕ	Diameter (m)
f	Friction factor
G	Mass flux (kg/m^2)
L	Length required (m)
ΔL	Length increment (m)
$\dot{m}$	Mass rate of flow (kg/s)
M	Mach number
P	Pressure (N/m^2) (or Power (W))
Re	Reynolds number
T	Temperature (K)
T_a	Ambient temperature (K)
v	Velocity (m/s)
κ	Dryness fraction
μ	Dynamic viscosity (Pa.s)
ν	Humid volume (m^3/kg)
ρ	Density (kg/m^3)

SUBSCRIPTS

f	fluid
fg	mixture
g	gas (or vapor)
1	Inlet
2	Outlet
m	mean value
A	ASHRAE
M	McAdams' combination
D	Duckler's combination

REFERENCES

1. Akintunde, M. A. 2004. *Theoretical Design Model for Vapour Compression Refrigeration Systems*. AMSE Periodicals: France.

2. ASHRAE 2002. *ASHRAE Handbook: Refrigeration*. Chapters 49.7 and 45.26. "Adiabatic Capillary Tube Selection Procedure".

3. Chang, S.D. and Ro, S.T. 1996. "Experimental and Numerical Studies on Adiabatic Flow of HF Mixtures in Capillary Tubes". *International Journal of Refrigeration*. 6(1):83 – 88.

4. Chen, S.L., Cheng, Y.R., Liu, C.H., and Jwo, C.S. 2000. "Simulation of Refrigeration Flowing Through Adiabatic Capillary Tubes". *HVAC&R Research*. 6(2): 101 – 115.

5. Hopkin, N.E. 1996. "Rating the Restrictor Tube". *Journal of Refrigeration Engineering*. 58(11):1087 – 1094.

6. Jung, D., Park, C., and Park, B. 1999. "Capillary Tube Selection for HCFC-22 Alternatives". *International Journal of Refrigeration*. 22(8):604 – 614.

7. Kim, S.G., Kim, M.S,. and Ro, S.T. 2002. "Experimental Investigations of the Performance of R22, R407 and R410A in Several Capillary Tubes for Air-Conditioning". *International Journal of Refrigeration*, 25(3):521 – 531.

8. Melo, C., DaSilva, R.T., Ferreira, R., Pereira, H., and Negrao, C.O.R. 1988. "Dynamic Behaviour of a Vapour Compression Refrigerator: A Theoretical and Experimental Analysis". *Journal of Refrigeration Science and Technology*. Commission B1, International Institute of Refrigeration. 141.

9. Schultz, G.D. 1985. "Experimental Investigation of Refrigerant Flow Through Adiabatic Capillary Tube". *International Journal Of Refrigeration*. 4(6):878 – 886.

10. Stoecker, W.F. and Jones, W.P. 1982. *Refrigeration and Air-Conditioning. Second Edition*. McGraw-Hill: New York, NY.

11. Wijaya, H. 1992. "Adiabatic Capillary Tube Test Data for HF-134a". *International Journal of Refrigeration*. 11(1):63 – 71.

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SUGGESTED CITATION

Akintunde, M.A. 2007. "The Effects of Friction Factors on Capillary Tube Length". *Pacific Journal of Science and Technology*. 8(2):238-245.

Pacific Journal of Science and Technology