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EFFECT OF FRICTION FACTOR ON THE PERFORMANCE OF ADIABATIC CAPILLARY TUBE

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ABSTRACT

In this paper analysis the performance of capillary tube under the different condition. The following condition are : condensing temperature 40-55°C , subcooling 0-5°C , capillary tube diameter 1.2 – 2.4mm , mass flow rate 5 – 50 g/s , various friction factor given by different authors is used to predict the performance of capillary tube with R-22 , R-134a , R-410A refrigerant.

Key words: Capillary tube; Adiabatic; Alternative refrigerant; Friction factor.

1.INTRODUCTION

A typical residential air conditioner consists of a compressor , evaporator , condenser and expansion devices. Capillary tubes is one of the expansion devices has been extensively used for the past flow decades. Now a day electronic expansion valves and / or short tube orifices are considered as future expansion devices for some what large size air- conditioners heat pump.

HCFC 22 has been used superior as the working fluid in residential air conditioners due to its

excellent properties and fit for combining with various materials. HCFC 22 contains hydrogen and chlorine atoms and its ozone depletion potential is 0.05. Since HCFC 22 has the ozone depleting chlorine it is to be phased out eventually according to the montreal protocol amendments adopted by the participating countries [1]. The alternative of HCFC 22 are HFC 134a , R-407C (23% R-32/25% R-125/52% HFC-134a) and R-410A (50%R-32/50%R-125).

Nomenclature

A = cross-sectional area of inside of tube, m^2
 D = ID of tube, m
 f = friction factor, dimensionless
 h = enthalpy, KJ/Kg
 h_f = enthalpy of saturated liquid, KJ/Kg
 h_g = enthalpy of saturated vapor, KJ/Kg
 ΔL = length of increment, m
 P = pressure, Pa
 Re = Reynolds number
 v = specific volume, m^3/kg
 v_f = specific volume of saturated liquid, m^3/kg
 v_g = specific volume of saturated vapor, m^3/kg
 V = velocity of refrigerant, m/s
 W = mass rate of flow, kg/s
 x = fraction of vapor in mixture of liquid and vapor
 μ = viscosity, $Pa.s$
 μ_f = viscosity of saturated liquid, $Pa.s$
 μ_g = Viscosity of saturated vapor, $Pa.s$

2. MATHEMATICAL MODELING

Initially, a generalized mathematical model has been developed for straight capillary tube and for other geometries like helical and spiral capillary tubes the necessary modifications have been made in the developed model for the flow through a straight capillary tube.

The fundamental equations applicable to the control volume bounded by point 1 and 2 in fig. (A) are (1) conservation of mass, (2) conservation of energy, and (3) conservation of momentum.

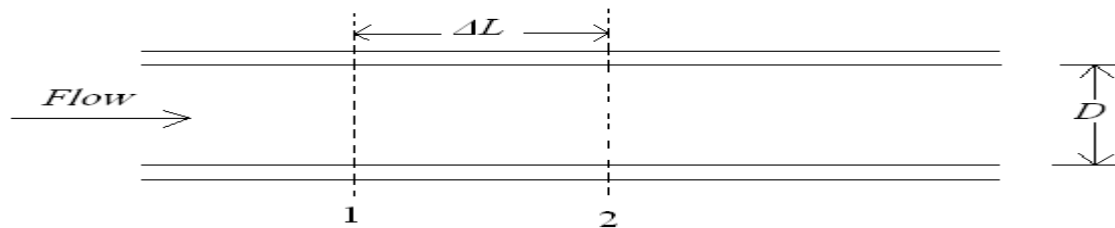


Figure 1 :- Incremental length of capillary tube.

The equation for conservation of mass states that

$$W = V_1 A / v_1 = V_2 A / v_2 \quad (1)$$

$$W/A = V_1 / v_1 = V_2 / v_2 \quad (2)$$

And W/A will be constant throughout the length of the capillary tube.

The statement of conservation of energy is

$$1000h_1 + V_1^2/2 = 1000h_2 + V_2^2/2 \quad (3)$$

Which assumes negligible heat transfer in and out of the tube?
The momentum equation in words states that the difference in forces applied to the element

because of drag and pressure difference on opposite ends of the element equals that needed to accelerate the fluid

$$[(p_1 - p_2) - f(\Delta L/D)(V^2/2\nu)]A = W(V_2 - V_1) \quad (4)$$

As the refrigerant flows through the capillary tube, its pressure and saturation temperature

progressively drop and the fraction of vapor x continuously increases. At any point

$$h = h_f(1-x) + h_g x \quad (5)$$

and

$$v = v_f(1-x) + v_g x \quad (6)$$

In Eq. (4) V , ν , and f all change as the refrigerant flows point 1 to point 2 but some simplification results from Eq. (2) which shows that V/ν is constant so that

$$f(\Delta L/D)(V^2/2\nu) = f(\Delta L/D)(V/2)(W/A) \quad (7)$$

In the calculation the V used in Eq. (7) will be the mean velocity

$$V_m = (V_1 + V_2)/2 \quad (8)$$

Since expressing the friction factor f for the two phase flow is complex, we shall use different friction factor to predict the performance of capillary tube given by Stoecker [2], Pate [3], Hopkins [4] respectively

$$\begin{aligned} f &= .33/\text{Re}^{.25} \\ f &= 3.4/\text{Re}^{.47} \\ f &= .217/\text{Re}^{0.2} \end{aligned} \quad (9)$$

The viscosity of the two-phase refrigerant at a given position in the tube is a function of the vapor fraction x .

viscosity of the two-phase can be calculated by Mcadams et al. [2] or Duckler et al. [5]

$$\mu = \mu_f(1-x) + \mu_g x \quad (10)$$

The mean friction factor f_m applicable to the increment of length 1-2 is

$$f_m = (f_1 + f_2)/2 \quad (11)$$

2.1 Calculating the length of an increment

The essence of analytical calculation method is to determine the length of the increment 1-2 in Fig. (A) for a given reduction in saturation temperature of the refrigerant. The flow rate and

all the conditions at point 1 are known, and for an arbitrarily selected temperature at point 2 the remaining conditions at point 2 and the ΔL will be computed in the following specific steps.

1. Select t_2 .
2. Compute p_2 , h_{f2} , h_{g2} , v_{f2} , and v_{g2} , all of which are function of t_2 .
3. Combine the continuity equation (2) and the energy equation (3)

$$1000h_2 + (v_2^2/2)(W/A)^2 = 1000h_1 + V_1^2/2 \quad (12)$$

Substitute Eqs. (5) and (6) into Eq. (12)

$$1000h_{f2} + 1000(h_{g2} - h_{f2})x + [v_{f2} + (v_{g2} - v_{f2})x]^2/2(W/A)^2 = 1000h_1 + V_1^2/2 \quad (13)$$

Everything in Eq.(13) is known except x , which can be solved by the quadratic equation

$$x = [-b \pm \sqrt{b^2 - 4ac}]/(2a) \quad (14)$$

where

$$a = (v_{g2} - v_{f2})^2 (W/A)^2 (1/2)$$

$$b = 1000(h_{g2} - h_{f2}) + v_{f2}(v_{g2} - v_{f2})(W/A)^2$$

$$c = 1000(h_{f2} - h_1) + (W/A)^2 (1/2) v_{f2}^2 - V_1^2/2$$

4. With the value of x known, h_2 , v_2 , and V_2 can be computed.
5. Compute the Reynolds number at point 2 using the viscosity from Eq. (10), the friction factor at point 2 from Eq. (9), and the mean friction factor for the increment from Eq. (11).
6. Finally, substitute Eq. (7) and Eq. (8) into Eq. (4) to solve for ΔL .

3. Analysis of the calculation results

In order to analysis the performance of adiabatic capillary tube for refrigerant R-22, R-134a, and R-410A with different friction factor given by

authors. Performance of capillary tube with 0°C subcooling, $D=1.8\text{mm}$, $w=10\text{g/s}$, $T_{\text{cond}}=40^\circ\text{C}$ is shown by Fig. 2 to 10.

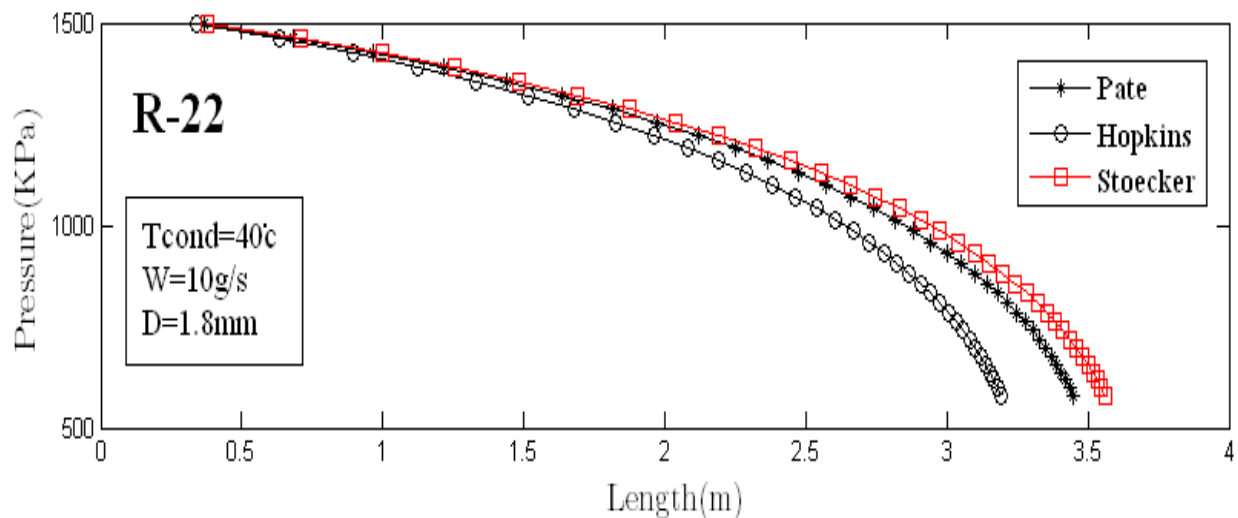


Fig. 2 Pressure variety versus the increment of capillary tube length for R-22

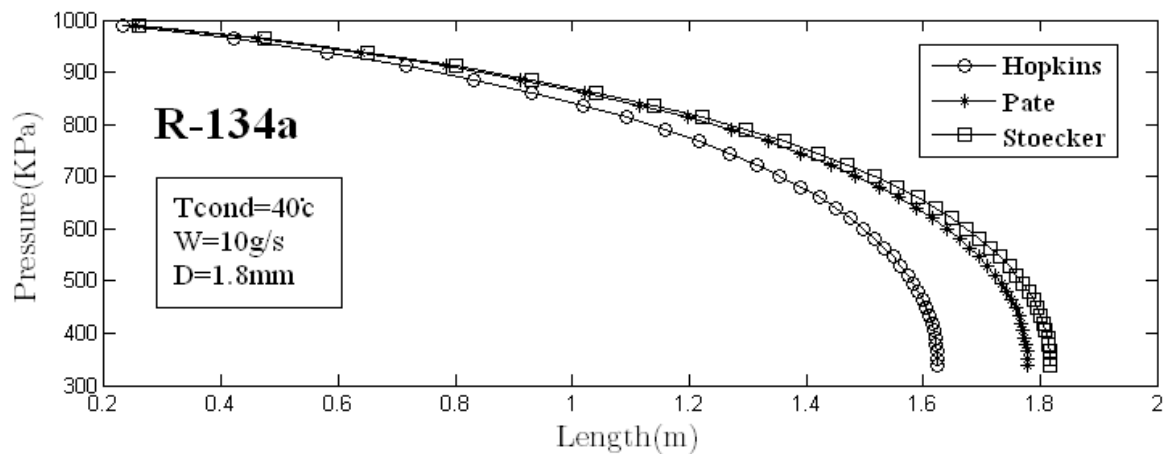


Fig. 3 Pressure variety versus the increment of capillary tube length for R-134a

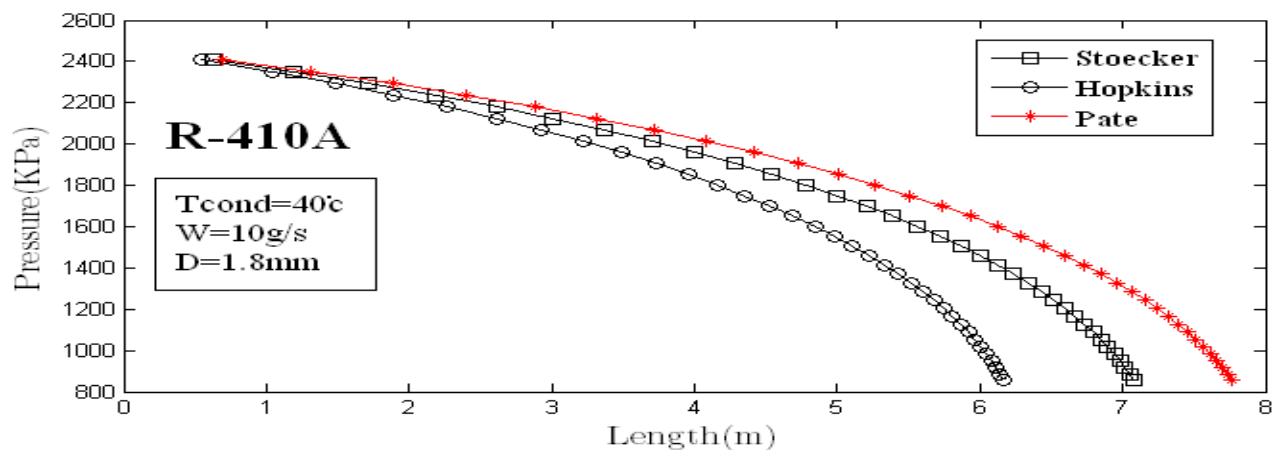


Fig. 4 Pressure variety versus the increment of capillary tube length for R-410A

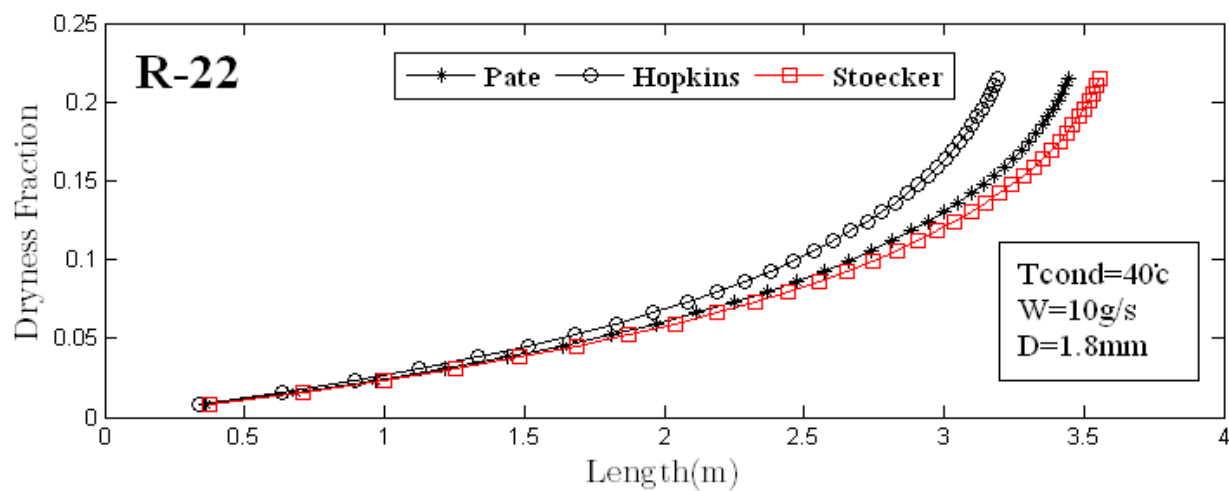


Fig. 5 Dryness Fraction versus the increment of capillary tube length for R-22

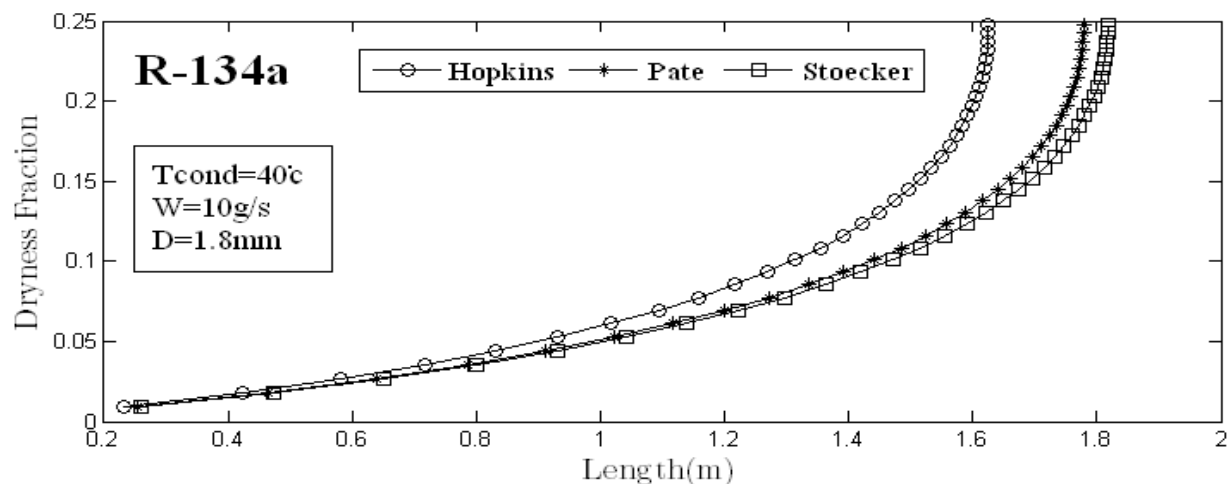


Fig. 6 Dryness Fraction versus the increment of capillary tube length for R-134a

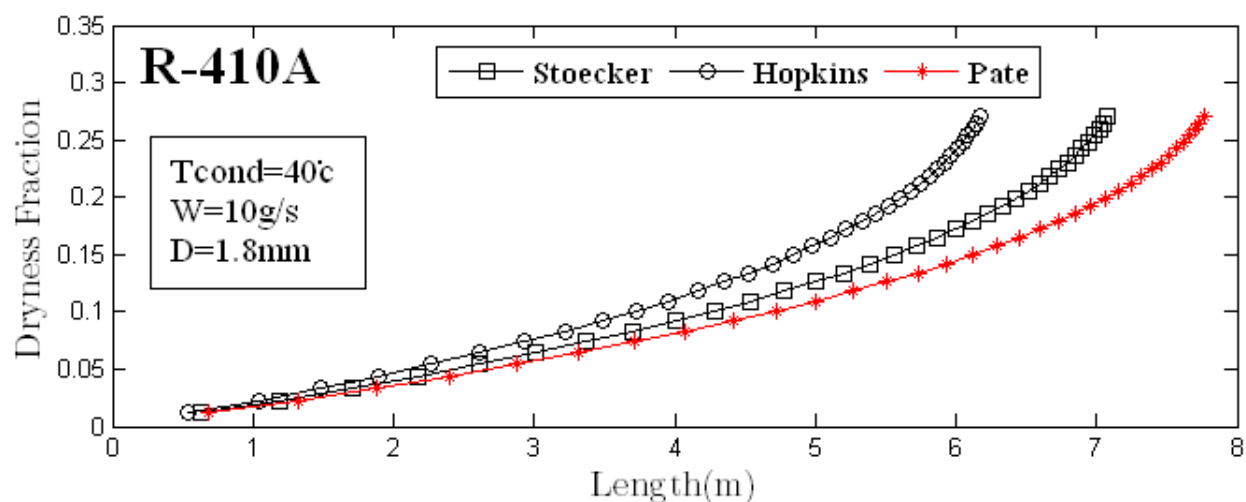


Fig. 7 Dryness Fraction versus the increment of capillary tube length for R-410A

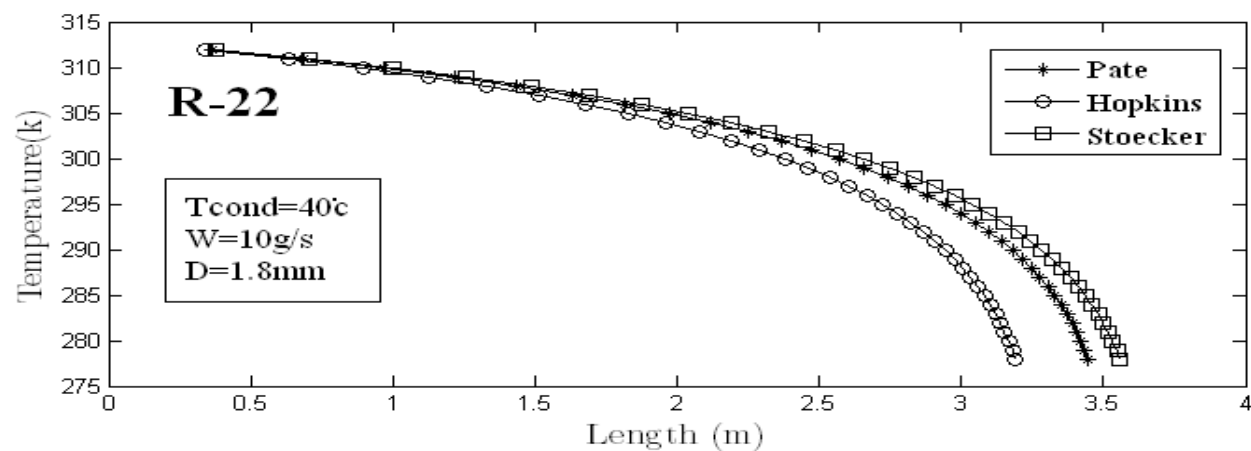


Fig. 8 Temperature versus the increment of capillary tube length for R-22

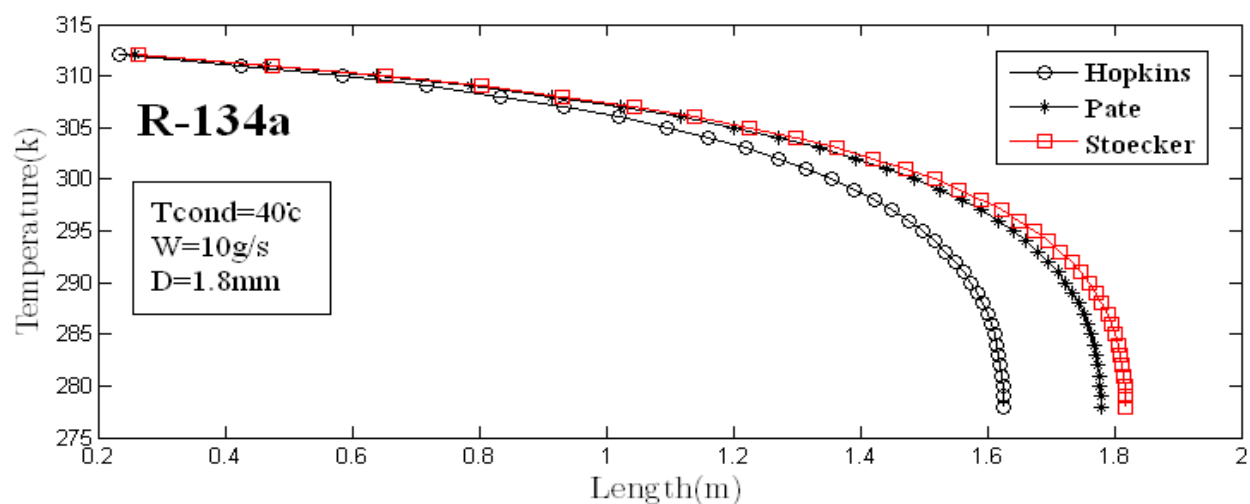


Fig. 9 Temperature versus the increment of capillary tube length for R-134a

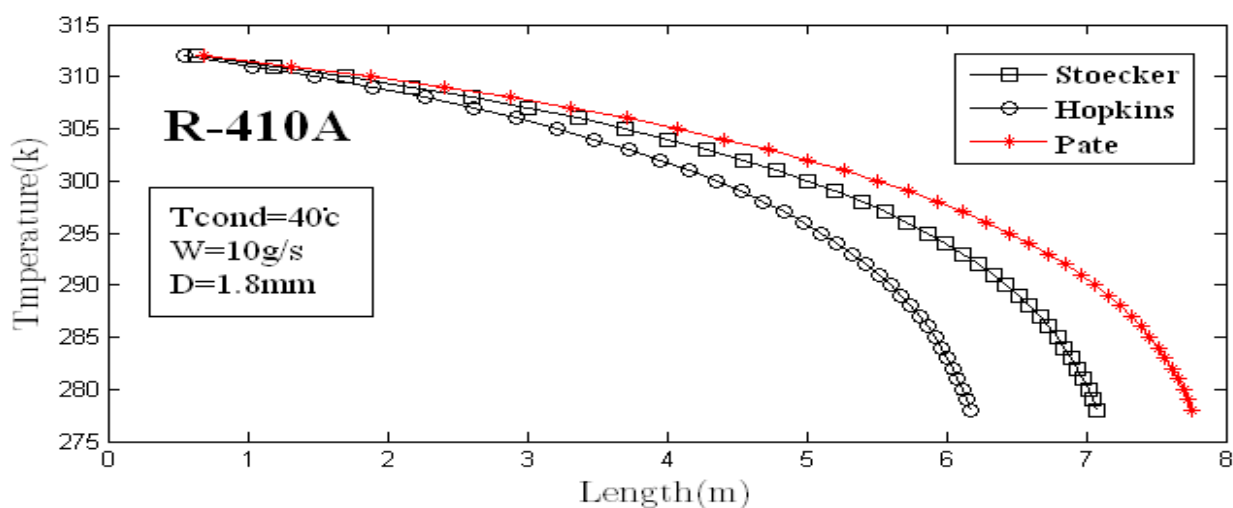


Fig. 10 Temperature versus the increment of capillary tube length for R-410A

4.CONCLUSIONS

In this paper, the performance of adiabatic capillary tubes used is predicted for R-22, R-134a, R-410A. The following conclusions can be drawn.

(1) Duckler's equation is not better than McAdams' equation for two-phase viscosity calculation.

(2) Kinetic energy term in energy conservation equations is not impacting on the final results.

(3) pressure drop due to area contraction is impacting on the model so it needs to be considered in the model.

- (4) Refrigerant R-134a show the optimum length as compare to others refrigerant for the given condition.
- (5) From the above analysis for predicting the performance of capillary tube Hopkins [5] friction factor gives optimum result under the all conditions.

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