Rating Charts of R-22 Alternatives Flow through Adiabatic Capillary Tubes

E. Elgendy and J. Schmidt

Abstract—Drop-in of R-22 alternatives in refrigeration and air conditioning systems requires a redesign of system components to improve system performance and reliability with the alternative refrigerants. The present paper aims at design adiabatic capillary tubes for R-22 alternatives such as R-417A, R-422D and R-438A. A theoretical model has been developed and validated with the available experimental data from literature for R-22 over a wide range of both operating and geometrical parameters. Predicted lengths of adiabatic capillary tube are compared with the lengths of the capillary tube needed under similar experimental conditions and majority of predictions are found to be within 4.4% of the experimental data. Hence, the model has been applied for R-417A, R-422D and R-438A and capillary tube selection charts and correlations have been computed. Finally a comparison between the selected refrigerants and R-22 has been introduced and the results showed that R-438A is the closest one to R-22.

Keywords—Adiabatic flow, Capillary tube, R-22 alternatives, Rating charts, Modelling.

I. INTRODUCTION

SINCE its initial recognition in 1928 and commercialization in 1936, R-22 has been used in refrigeration and air conditioning systems ranging from the smallest window air conditioning to the largest chillers and heat pumps due to its excellent properties and compatibility with various materials [1]. Unfortunately, R-22 has the ozone depleting chlorine atom. Therefore it has to be phased out eventually according to the Montreal Protocol amendments adopted by the participating countries [2]. Natural fluids are the safest option from an environmental point of view as they are present in the environment and their long term effects are known. Hydrocarbons represent one of the natural fluids that having excellent thermodynamic and the rmophysical properties which can be used as R-22 alternatives. However, they have the inconvenience of being highly flammable (ASHRAE classification, A3 security) and are only used today where a small amount of refrigerant is necessary (e.g. domestic refrigerators) which even in the case of a leakage would not create a safety problem. Among hydrocarbon mixtures, R-417A, R-438A and R-422D are characterized by zero ozone depletion potential and in the existing installations they can

easily replace R-22 [3], [4].

In order to apply these alternative refrigerants to air-conditioning and refrigeration systems, each component of the system must be redesigned to achieve a higher reliability and performance. Expansion device is one of the important components in the refrigeration systems as it reduces the pressure of the liquid refrigerant coming from condenser and regulates its flow to the evaporator. A capillary tube is a segment of tube with constant cross section area served as the expansion device which had been widely used in small vapor compression refrigeration and air-conditioning systems with simple, reliable and low costing merits. It connects the outlet of the condenser and the inlet of the evaporator, and produces a pressure drop between them. Capillary tubes can be classified into main categories according to their connection with suction line, namely; adiabatic and non-adiabatic capillary tubes. In some applications, the capillary tube is soldered to the suction line and the combination is called a capillary-tube/suction line heat exchanger or non-adiabatic capillary tube. However, refrigeration systems that use a capillary tube without the heat exchanger relationship are often referred to as adiabatic capillary tube systems.

The flow inside the capillary tube becomes two phase after flashing process. Two-phase phenomenon can be explained by different models namely homogeneous flow model, separated flow model and the drift flux model. In homogeneous two-phase flow there is no slip between the two phases, i.e., the two phases move with the same velocity. While in separated flow model, there exists slip between the two phases and when the conservation equations are applied to the mixture a variable called void fraction is introduced. In the drift flux model, the conservation equation is formulated by considering the entire mixture. The formulation is expressed in terms of four field equations; three for the mixture (mass, momentum and energy) and the fourth is the drift velocity for one of the phases.

In the past few decades, much effort has been placed on capillary tubes modelling. Many models have been developed on the flow characteristics of refrigerants through a capillary tube. Among them, the typical ones are the homogeneous model [5]-[9] two-phase separated model [8], [10] and drift flux model [11], [12]. Accounting for the simplicity of the homogeneous model and its precision, this paper adopts a homogeneous model for adiabatic capillary tube. Capillary tubes sizing for various applications associated with common refrigerants including R-22 has been recommended primarily in the ASHRAE handbook diagrams [13]. However, there is a need to provide capillary tube rating charts for the new R-22

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alternatives including refrigerant mixtures. The objective of this paper is to perform a capillary tube rating charts for R-22 alternatives, namely R-417A, R-422D and R-438A. Moreover, a comparison between these working fluids and R-22 has been performed.

II. CAPILLARY TUBE DESIGN

Fig. 1 shows a schematic diagram of a capillary tube connecting the condenser and evaporator. In order to develop the model, the following assumptions are made:

- steady flow,
- one dimensional,
- adiabatic and homogeneous flow through the capillary tube,
- metastable effect is neglected,
- the capillary tube is straight and horizontal with constant inner diameter and uniform surface roughness.

Capillary tube length (L_{tot}) can be determined by assuming step decrements of the pressure through a particular inner diameter (d) as shown in Fig. 1, then calculating the corresponding required increments of the length (ΔL). These increments can be accumulated to give the capillary tube length required for the specified pressure drop to reach the downstream pressure, i.e.

$$L_{tot} = \Delta L_1 + \Delta L_2 + ...\Delta L_n \qquad (1)$$

In the next section, governing equations are developed to describe the flow through each element of the capillary tube.

A. Governing Equations

According to the above suppositions, the control equations of the flow in capillary tubes applying continuity, momentum conservation and energy conservation equations as;

$$\frac{d(\rho u)}{dx} = 0 \qquad (2)$$

$$\frac{d(\rho u^2)}{dx} + \frac{dp}{dx} = -\frac{1}{2}\frac{f}{d}\rho u^2 \qquad (3)$$

$$\rho u \frac{dh}{dx} + \frac{1}{2}\rho u \frac{du^2}{dx} = 0 \qquad (4)$$

A control element is selected as shown in Fig. 1; the corresponding discrete equations are;

$$\rho_1 u_1 = \rho_2 u_2 \qquad (5)$$

$$A\left((p_1 - p_2) - f\frac{\Delta l u_2}{2dv}\right) = G(u_2 - u_1) \qquad (6)$$

$$(h_2 - h_1) + \frac{u_2^2 - u_1^2}{2} = 0 \qquad (7)$$

At any increment, the mean friction factor and velocity in the control element can be determined as;

$$f_m = \frac{f_1 + f_2}{2} \qquad (8)$$

$$u_m = \frac{u_1 + u_2}{2} \qquad (9)$$

It should be noted that, the best equations yielding the results similar to the measured ones in the ASHRAE handbook are McAdams equation for the two phase viscosity and Stoecker's and Hopkins' equations for the friction factor as reported by Jung et al. [6]. These equations are as follow;

McAdams et al. [6];

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

Stoecker et al., [6];

$$f = \frac{0.33}{\text{Re}^{0.25}} \qquad (11)$$

As the refrigerant passes through the elements (increments) of the capillary tube, its pressure and temperature drop, thereby the vapor quality (x) continuously increases. Hence, in the present simulation model, the flow within the capillary tube is divided into two flow regions: single phase and two-phase flow regions as shown in Fig. 1. Two phase flow region starts when the temperature of the subcooled liquid reaches the saturated liquid temperature.

B. Solution Method

The downstream pressure at any element in the capillary tube can be calculated from;

$$p_{ds} = p_{us} - i\Delta p, \qquad (12)$$

where the script (us) refers to up-stream while (ds) refers to down-stream. For single-phase, upstream properties (h_{us}, u_{us}, ρ_{us}, and μ_{us}) of the first element are calculated based on inlet pressure and subcooled temperature. Assuming the down-stream pressure (p_{ds}) and down-stream specific enthalpy (h_{ds}) is initially equal to the upstream, the state point of down-stream point can be estimated a single or two phases. In case of a single phase, solving (5) and (7) for ρ_{ds}, u_{ds} and h_{ds} by using an iteration process for the new subcooling degree and taking into account the following equation for the resulting enthalpy (h_{ds});

$$h_{ds} = f(\rho_{ds}, p_{ds}) \qquad (13)$$

For two phase, the same equations (5, 7 and 13) are solved together but the iteration process target is to calculate the quality of the down-stream fluid. As a result the mean density

(ρm), fluid velocity (um) and friction factor (fm) can be obtained and then substituted into (6) to evaluate ΔL. It may be stated that the same procedure has to be repeated for the next element. Considering that the downstream conditions of the previous element are the entering (upstream) conditions to the next element, and then all incremental lengths are accumulated to obtain the total length of the capillary tube. The calculation is terminated when the fluid reaches chocked flow conditions. The chocked flow can be reached as the specific entropy is reduced or the calculated incremental

length ΔL is negative or velocity of the two phase fluid equals the sound velocity (Mach number =1). Once the flow is choked, the mass flow rate does not change even though the pressure drops further. Jung et al. [6] determined the two-phase Mach number by the following equation;

$$M_{tp} = \left\{ -G^2 \left[x \frac{dv_{sv}}{dp} + (1-x)\frac{dv_{sl}}{dp} + \Delta v \left(\frac{dx}{dp}\right)_h \right] \psi \right\}^{1/2} \qquad (14)$$
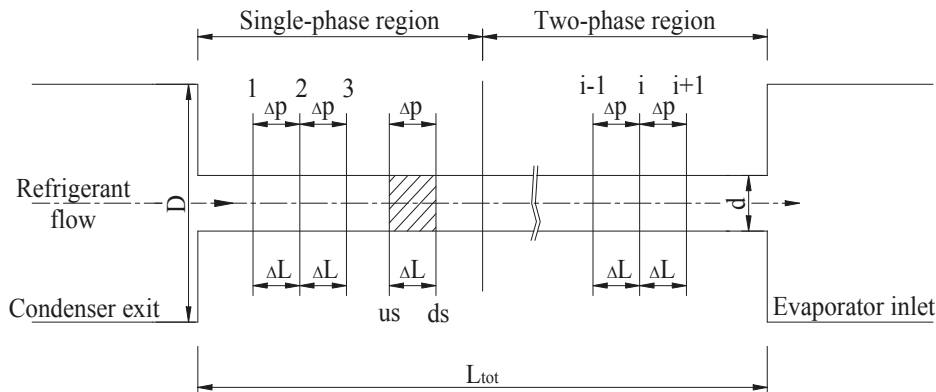

Fig. 1 Schematic diagram of an adiabatic capillary tube

where

$$\psi = \left[1 + \frac{G^2 \, \Delta v \, (v_{sl} + x \, \Delta v)}{\Delta h} \right]^{-1} \qquad (15)$$

A simulation program based on equations from (1) to (15) with necessary thermodynamic and thermo-physical properties of the selected refrigerants is developed for sizing the capillary tube. The input data for simulation are working fluid type, inlet pressure, inlet subcooling degree, inlet vapor quality and capillary tube inner diameter. Thermodynamic and transport properties of the considered working fluids are calculated using REFPROP [14]. The program calculates flow characteristics along the capillary tube, capillary tube length, refrigerant mass flow rate, and flow factors over the entire range of the considered operating conditions.

III. RESULTS AND DISCUSSION

The model results are presented in four sections; (i) detailed point analysis, (ii) model validation, (iii) rating charts for R-417A, R-422D and R-438A, (iv) comparison between selected refrigerants and R-22.

A. Detailed Point Analysis

Fig. 2 shows a sample of results for R-22 detailed point to illustrate the chocked flow conditions and consequently the calculations termination. The selected point condenser pressure is 2MPa, subcooling degree 10K, refrigerant mass flow rate 70kgh-1, capillary tube diameter is 1.68mm. The estimated capillary tube length is 1.702m at chocked

conditions. The thermodynamic conditions are represented by Fanno line shown on the enthalpy-entropy diagram of Fig. 2.A, where the enthalpy decreases while the entropy increases as the fluid flows through the tube. At the chocked flow conditions the Fanno line demands a decrease in entropy, which is forbidden by the second law of thermodynamic for this adiabatic process. A relation between specific enthalpy and exit pressure is also presented in Fig. 2.A just to find chocked exit pressure corresponding to the point of entropy decreasing. Hence, the calculations procedure is terminated and the accumulated length increments are the capillary tube length.

The same chocked flow conditions can be also reached as the calculated capillary tube accumulated length starts to decrease which means that the calculated length increment (ΔL) is negative as seen in Fig. 2.B. Also, this confirms the ending of calculation process. The same results can be reached by Mach number as illustrated in Fig. 2.C. As the Mach number reaches one, the fluid velocity is sonic velocity and the flow is chocked.

B. Model Validation

Model validation is performed on the available experimental data for R-22. Experimental data for pressure distribution along an adiabatic capillary tube up to chocked flow conditions are presented by both Jung et al. [6] and Kim et al. [15]. Predicted mass flow rate using the present model is compared with the experimental data of Jung et al. and Kim et al. in Fig. 3. The presented validation data has been done over a wide operating range of condensation temperature (40-

55°C), subcooling degree (0-10K) and inner diameters (1.25-2mm). It can be stated that, the present model predictions are close to the experimental data with an average error of 4.6% and 4.3% as compared with Jung et al. and Kim et al., respectively. Hence, the present model can be used to size capillary tubes and predict flow characteristics of new refrigerants along them.

C. Rating Charts for R-417A, R-422D and R-438A

The capillary tube size (inner diameter and length) must be selected for drop-in replacements for ozone depleting substances to obtain the balance point between the compressor and the capillary tube. In order to facilitate a selection of appropriate capillary tube inner diameter and length for domestic refrigerators/freezers, rating charts for R-417A, R-422D and R-438A are developed based on the obtained results from the present model for steady adiabatic flow through capillary tube. Charts similar to those charts reported by ASHRAE [13] for R-22 are shown in Figs. 4-6.

These rating charts can be used to determine either capillary tube size (inner diameter and length) or mass flow rate of the prescribed refrigerants.
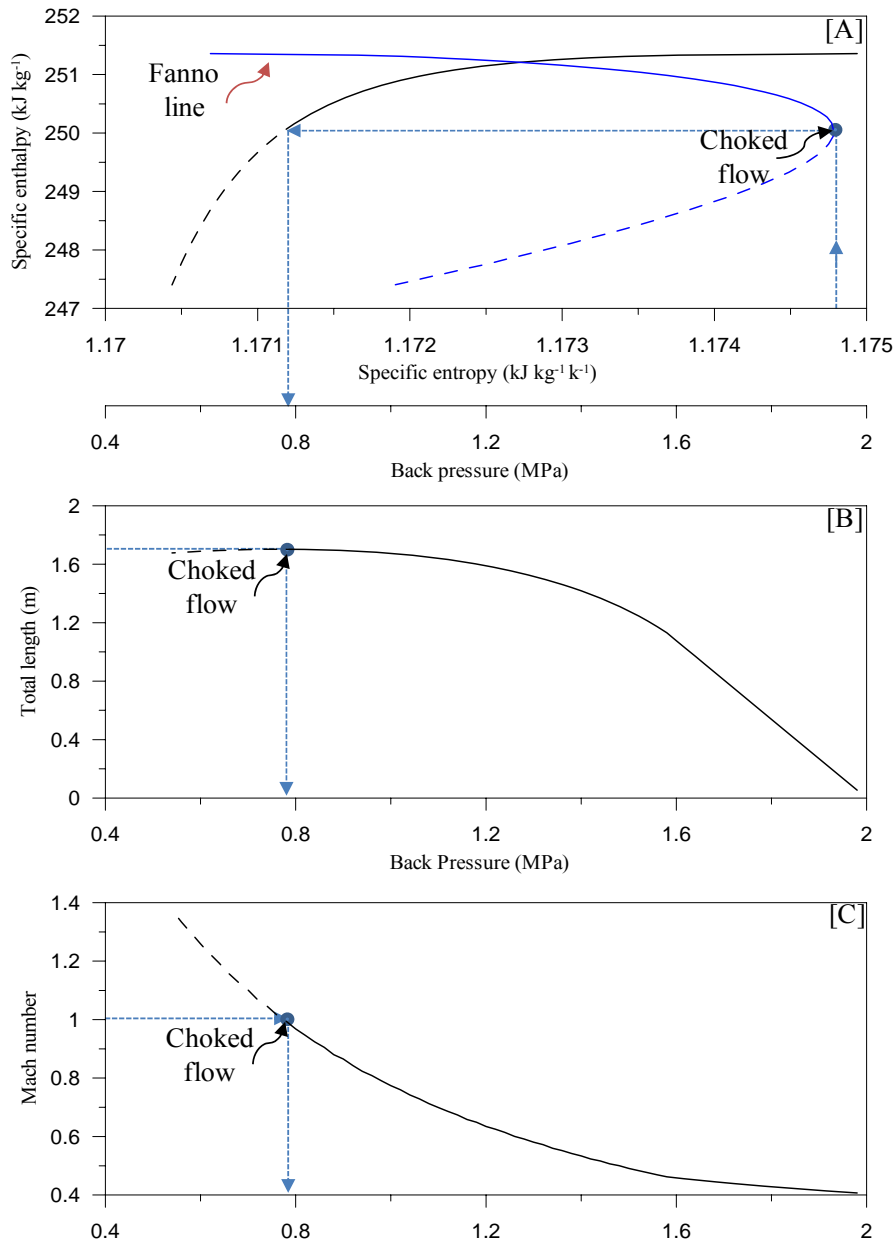

Fig. 2 Chocked flow conditions. (A) Fanno line, (B) capillary tube length and (C) Mach number
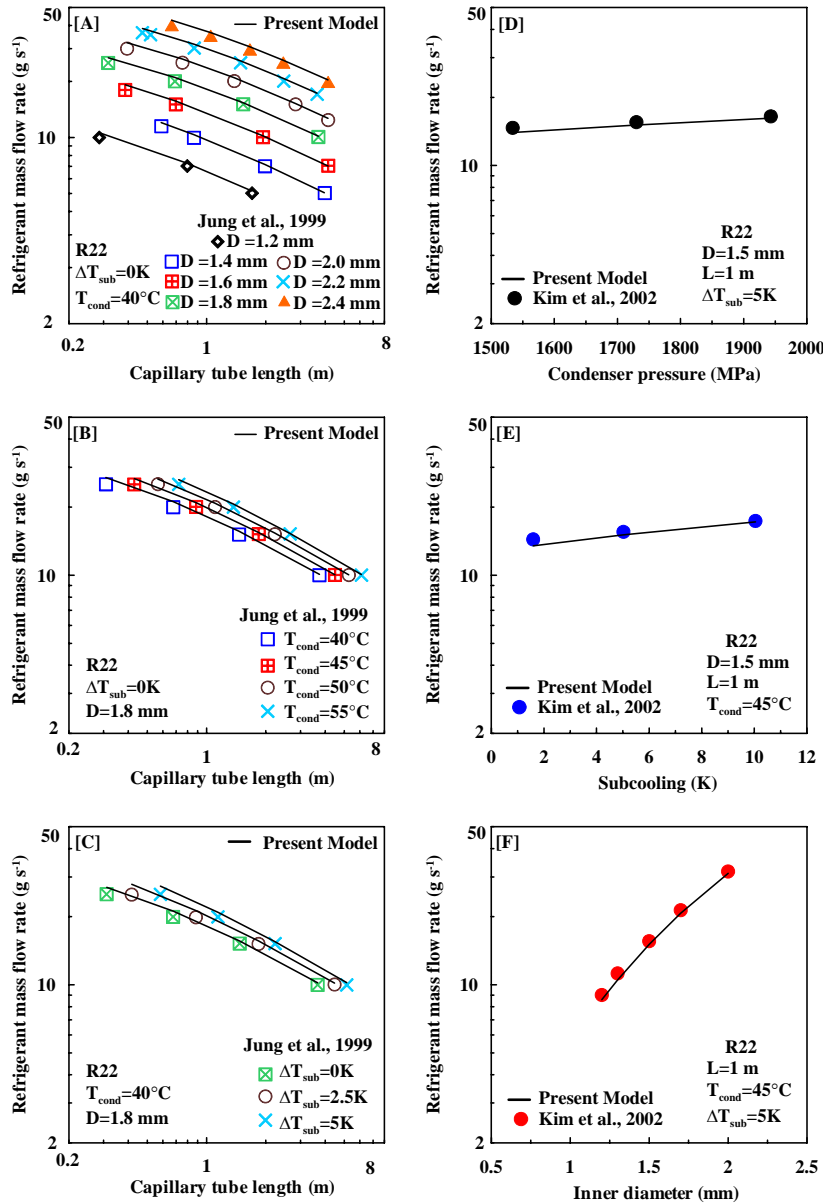
Fig. 3 Comparison between present model and experimental data of Jung et al. [15] and Kim et al. [15]

If the capillary tube inner diameter and length are the same as those of the reference capillary tube, the refrigerant flow rate through it will be the standard refrigerant mass flow rate which can be determined using the upstream conditions (inlet pressure and sub-cooling degrees or quality). On the other side, if the capillary tube size is different from the reference capillary tube, actual refrigerant flow rate can be calculated as follows:

$$\dot{m}_{act} = \phi \dot{m}_{std} \qquad (16)$$

where $\dot{m}_{std}$ is the standard mass flow rate and ϕ is the flow factor.

Fig. 4 A illustrates standard mass flow rate ($\dot{m}_{std}$) of R-417A through a reference capillary tube of 1.68mm inner diameter and 1.524m long as a function of inlet pressure for different inlet conditions such as sub-cooled, saturated and two-phase refrigerant. Fig. 4.B presents a geometric correction factor as a function of capillary tube geometry (d and L) for sub-cooled flow conditions. In order to select the inner diameter and length of capillary tube for a given refrigerant mass flow rate, the standard mass flow rate has to be determined from Fig. 4 A by knowing the inlet pressure and the subcooling degree. Then, flow factor (ϕ) has to be calculated. As a final point, capillary tube geometry (inner diameter and length) can be selected from Fig. 4.B according to the value of flow factor.
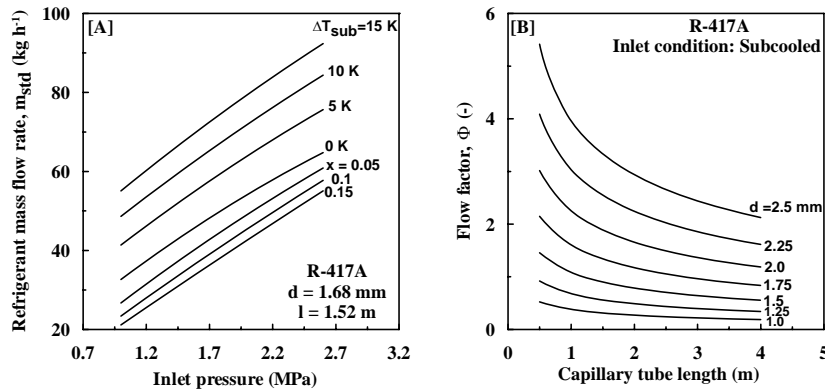
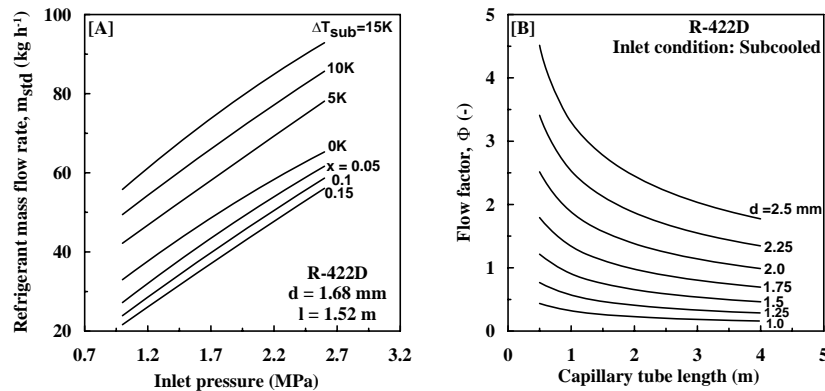Fig. 4 Capillary tube rating charts for R-417A

Fig. 5 Capillary tube rating charts for R-422D

Figs 5 A and 5 B are rating charts for R-422D through adiabatic capillary tubes. Fig. 5.A plots the capillary tube flow rate as a function of inlet condition and inlet pressure for the same reference capillary tube geometry of 1.68mm ID and 1.524m long. Fig. 5 B is a geometric correction factor. Using the desired capillary tube geometry, ϕ may be determined and then multiplied with the flow rate from Fig. 5.A to determine the predicted capillary tube flow rate. Figs. 6.A and 6.B present the rating charts for R-438A through adiabatic capillary tubes. The method of selection from these charts is identical to that previously presented for R-417A and R-422D.

In order to generalize the predicted data of the present capillary tube model, the capillary tube length up to choked flow conditions for an inlet having subcooled conditions is correlated with the relevant parameters in the following form:

$$L = C_o(D)^{C1}(\dot{m}_{ref})^{C2}(p_{in})^{C3}(\Delta T_{sub})^{C4} \qquad (17)$$

where L is the capillary tube length up to choked flow conditions and the numerical constants (Co, C1, C2, C3 and C4) in (17) are given in Table I. Multiple correlation factors for R-417A, R-422D and R-438A are 0.963, 0.964 and 0.965 respectively.

D. Comparison between R-417A, R-422D, R-438A and R-22

Comparisons between R-417A, R-422D, R-438A and R-22 are made over wide ranges of inlet pressure (1-2.6MPa) and presented in Fig. 7. Pressure distribution along the capillary tube up to choked flow conditions for R-417A, R-422D, R-438A and R-22 is shown in Fig. 7.A, which indicates that saturated pressure decreases as the capillary tube length increases. Clearly, rate of decrease in the saturated pressure of R-417A is higher than that of R-422D, R-22 and R-438A. This is due to the fact that the specific volume of R-417A is higher than those of R-422D, R-22 and R-438A at any section of the capillary tube. Thus, R-417A velocity along the capillary tube is higher than those of R-422D, R-22 and R-438A. This leads to increase in Mach number of R-417A, thereby its choked flow occurs earlier than those of R-422D, R-22 and R-438A for given mass flow rate and inner diameter of the capillary tube. Fig. 7.A reveals that choked pressures of 0.54, 0.54, 0.52 and 0.52MPa occur at capillary tube lengths of 3.216, 3.292, 3.546 and 3.476m for R-417A, R-422-D, R-438A and R-22, respectively. Hence, pressure drop per unit length is 0.454, 0.444, 0.426 and 0.417MPa/m for R-417A, R-422D, R-22 and R-438A, respectively. Thus, the nearest refrigerant to replace R-22 is R-438A. As a result, to maintain the same pressure drop (Δp), the length of capillary tube has to be decreased for a given inner diameter when this alternative is used.
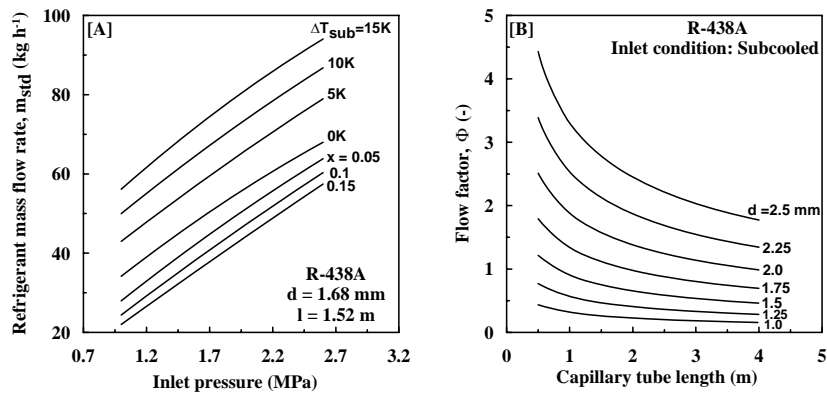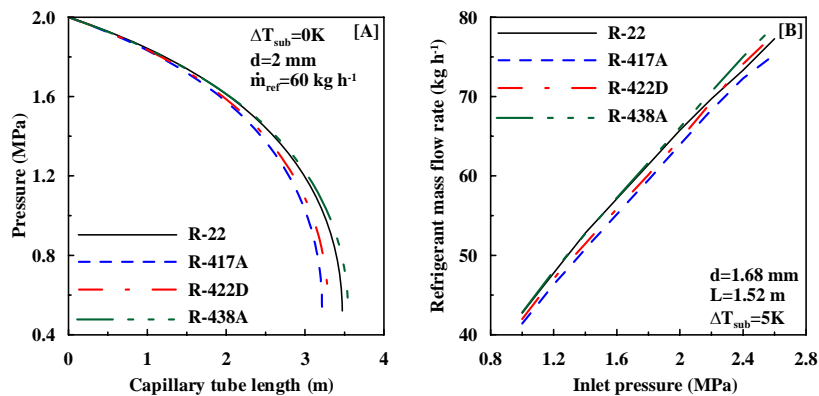
Fig. 6 Capillary tube rating charts for R-438A

Fig. 7 Comparison between R-22 and its alternatives (R-417A, R-422D and R-438A)

TABLE I
COEFFICIENTS OF CORRELATION (17)

Refrigerant	C_0	C_1	C_2	C_3	C_4
R-417A	228.3626	217.4307	0.126303	3.489663	1.071673
R-422D	233.0508	216.4476	0.12653	3.479014	1.072599
R-438A	247.5061	220.6294	0.125562	3.515935	1.06757

Fig. 7 B shows variations of standard mass flow rate ($\dot{m}_{std}$) of R-22, R-417A, R-422D and R-438A through the reference capillary tube (1.68mm inner diameter and 1.52m long) with inlet pressure under choked conditions for inlet subcooled (5K). Clearly, R-417A yields the lowest refrigerant mass flow rate over the inlet flow conditions. Also, it could be noted that R-438A has the same refrigerant mass flow rate of R-22 at lower operating range of condensation pressure (less than 1.8MPa).

IV. CONCLUSIONS

In the present work, a theoretical modelling is carried out to size a capillary tube of small scale refrigeration and air conditioning systems working with R-22 alternatives. The considered refrigerants are R-417A, R-422D and R-438A. Based on the simulation results, the following conclusions are presented:

- Capillary tube sizing procedure is terminated at chocked flow conditions which can be determined by either decreasing of specific entropy or decreasing calculated length or Mach number equals one.
- The presented model has an average error of 4.4% as compared to experimental data by different literature work.
- Rating charts for capillary tube working with R-417A, R-422D and R-438A are presented which can be used to predict both capillary tube inner diameter and length.
- Comparison between R-22 and its alternatives revealed that R-438A is the closest refrigerant to R22. Hence, R-438A can be dropped in refrigeration and air conditioning system without changing capillary tube of R-22.

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