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Experimental and numerical investigation on the flow of mixed refrigerants through capillary tubes at cryogenic temperatures

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HIGHLIGHTS

- Pressure drop through capillary for cryogenic refrigerant mixtures is studied.
- Two phase viscosity correlations are studied at cryogenic temperatures.
- Experiments have been carried out for cryogenic refrigerant mixtures.

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ABSTRACT

Capillary tubes are the simplest form of expansion devices used in J-T refrigerators. Numerous researchers have worked on the flow of refrigerant through capillary tubes; however, only a few researchers have reported on flow through capillary tubes at cryogenic temperatures. This study focusses on the flow of refrigerant mixtures at cryogenic temperature through the capillary tube of a Mixed Refrigerant Joule-Thompson (MRJT) cryocooler. In order to design a capillary tube for such mixtures, a numerical model, which predicts pressure drop across the capillary tube, is developed in the present work. As the flow through the capillary is in two-phase region, the homogeneous flow model is employed in this analysis and various two-phase viscosity models are investigated. An algorithm to predict the design length of the capillary tube using the homogeneous flow model is developed. Experimentation is carried out to compare the same with the theoretical predictions. The model predicts capillary tube length with an accuracy of 20%.

1. Introduction

A capillary tube is widely used as an expansion device for the throttling process in Joule-Thompson (J-T) refrigerators. It offers various advantages over other throttling devices.

According to Brodyanskii et al. [1], the main advantages of using capillary tubes are simplicity and reliability as there are no moving parts and no requirement for sealing arrangements. These characteristics of capillary tubes make them suitable for cryogenic applications. These refrigerators are called as J-T cryocoolers. A mixture of refrigerants is used as the working fluid to reach lower temperatures, and that is why these cryocoolers are referred to as Mixed Refrigerant Joule-Thompson (MRJT) cryocoolers.

1.1. MRJT cryocooler

Most of the research on MRJT cryocooler is focussed on the composition of the refrigerant mixture. To study the performance of MRJT cryocooler for different refrigerant mixtures, Walimbe et al. [2] performed experimental studies on MRJT cryocooler for the temperature range of 75-–120 K using Nitrogen-Hydrocarbon and Neon-Nitrogen-Hydrocarbon mixtures. The lowest temperature of 65 K has been reported while 6.1 W of cooling effect has been obtained at 80 K using a 1 T Air-Conditioning compressor. Ardhapurkar et al. [3] have studied two-phase heat transfer inside the recuperative heat exchanger of an MRJT cryocooler numerically and experimentally. R. Satya Mehera and G. Venkatarathnam [4] have performed experimental as well as numerical investigation on the performance of MRJT cryocoolers for different heat exchangers. They have concluded that for the numerical analysis of the heat exchanger, the homogeneous flow model for the flow of mixed refrigerant in the heat exchanger [5] gives good agreement with experimental values.

1.2. Adiabatic flow through capillary

Capillary tubes are an essential part of any J-T cryocooler; they may be adiabatic or diabatic depending on the cooler design. However, due to their simplicity, adiabatic capillary tubes are preferred in J-T coolers. Melo et al. [6] have performed an extensive experimental investigation on flow through capillary tubes using R12, R134a, and R600a refrigerants by varying capillary tube size and inlet subcooling temperature. In their investigation, using dimensional analysis and experimental data, an empirical

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Nomenclature

$\dot{m}$	mass flow rate
ρ	density
V	velocity
A	area
P	pressure
g	gravitational acceleration
z	elevation
H	head loss
h	enthalpy
x	quality
G	mass flux
ν	specific volume
k	entrance loss coefficient
f	friction factor
L	length
d	diameter
Re	Reynolds number
μ	viscosity
α	void fraction

Subscripts

L,e	entrance loss
L,f	skin friction loss
sp	single phase
tp	two-phase
i	i^{th} element
f	liquid
g	vapour

correlation has been generated which is found to estimate the mass flow rates within ± 15% of experimental values. Such dimensional analysis provides a good initial approximation for flow rate through a capillary tube. Sanzovo and Mattos [7] have analysed refrigerant flow through capillary tubes using homogeneous and separated flow models. Their model predicts the mass flow rates within ± 5.8% for the homogeneous flow model and ± 7.4% for the separated flow model when compared to experimental results. They have evaluated the two-phase multiplier (ratio of two-phase to liquid-only pressure drop) for separated flow using Lin's model [8]. They have concluded that the close prediction by both homogeneous and separated flow model is due to the fact that the high refrigerant mass flux in the capillary tubes nullifies the effect of slip ratio (ratio of liquid and vapour velocities). Thus, it would be a reasonable assumption to analyse the flow in capillary tubes using the homogeneous flow model.

Dubba and Kumar [9] have performed an experimental study on flow characteristics of refrigerant R600a flow through a capillary tube. Straight and helically coiled capillary tubes with different coil diameters have been used for the analysis. It has been reported that the mass flow rate for a straight capillary tube is 1.5 – –16% higher than that for the coiled ones. Based on the dimensional analysis and experimental data, an empirical correlation has been developed which predicts mass flow rate within ± 20% of the experimental value. Rasti and Jeong [10] have developed empirical relation for refrigerant mass flow rate through straight and helical adiabatic capillary tubes. Dimensional analysis has been used to derive an empirical correlation. The empirical correlation is verified using experimental data for various refrigerants. It has been found to predict the refrigerant mass flow rate with an average absolute deviation of 6.85%.

Bansal and Rupasinghe [11] have developed a numerical solver for flow through capillary tubes using the homogeneous flow model for R134a keeping the evaporator temperature at 250 K. In their work, Churchill's correlation [12] has been used to calculate single-phase friction factor while Lin's correlation [8] has been used to evaluate two-phase friction factor. However, the information required for the evaluation of two-phase viscosity has not been specified. In their work, entropy maximisation has been used to determine the point when choking occurred. Deodhar et al. [13] have performed experimental as well as numerical studies on flow through capillary tubes for R134a using the homogeneous flow model. They evaluated capillary tube length as well as the mass flow rate using the homogeneous flow model. Colebrook's equation [14] has been used for the evaluation of single-phase and two-phase friction factor. They have studied various two-phase viscosity models reported by McAdams et al. [15], Cicchitti et al. [16], Dukler et al. [17], Beattie and Whalley [18], and Lin et al. [8] for calculating two-phase viscosity of the refrigerant. They have reported that the model given by Cicchitti et al. [16] predicts the capillary tube length and mass flow rate through the capillary tube better than other models. For determining the occurrence of choking, the concept of unrealistic length has been used. The use of unrealistic length makes the design of the solver much simpler compared to the use of maximisation of entropy [11].

1.3. Adiabatic flow through capillary tubes at cryogenic temperatures

A few research articles are available on the flow of mixed refrigerants through capillary tubes at cryogenic temperatures. Kruthiventi and Venkatarathnam [19] have reported that for a conventional refrigerator, the flow through the capillary tube is usually in choked condition, while for Mixed Refrigerant J-T(MRJT) cryocooler, it is unchoked. They showed analytically that at the exit of the capillary tube, the flow is close to being sonic and if an appropriate factor of safety is not considered during design, the flow might transition to choked condition. Ardhapurkar et al. [20] have studied pressure drop in the capillary tube for Mixed Refrigerant Joule-Thompson cryocooler using homogeneous and various separated flow models [21–23]. They have reported that predictions related to pressure drop for flow through capillary tubes were close to the experimental values using the homogeneous flow model. They have used a mixture of Nitrogen, Methane, Ethane, Propane, and Isobutane as the working fluid/refrigerant. For a capillary tube of MRJT cryocooler, as the inlet to the capillary is in two-phase, the occurrence of metastable state (which occurs during the transition from single phase to two-phase state) inside of the capillary is absent. As the cross-sectional area of the capillary tube is small, the mass flux through it will be high. High mass flux ensures that the liquid and vapour have the same velocities. Therefore, it is appropriate to use the homogeneous flow model for two-phase flow inside a capillary tube without much loss of accuracy [20].

Extensive work has been conducted on flow through adiabatic as well as diabatic capillary tubes with various pure and mixtures of refrigerants. Besides, the inlet condition to these capillary tubes has been kept in subcooled or two-phase state. However, less work has been reported for the flow through capillary tubes at cryogenic temperatures. As for MRJT cryocoolers, the selection of the capillary tubes is mostly done by trial and error method. This is mainly due to scarce experimental and theoretical information available on capillary tubes at cryogenic temperature. Also, there is a lack of experimental data for validation of any mathematical model that may be developed for the flow of refrigerants through capillary tubes at cryogenic temperature.

To address these issues, the present work focusses on determining the pressure drop through capillary tubes for MRJT cryocoolers for different inlet flow conditions. A numerical solver is developed using the homogeneous flow model to determine the length of a capillary tube for a given pressure drop and mass flow rate. An extensive experimental investigation is carried out on flow through capillary tubes using different refrigerant mixtures at cryogenic temperatures for comparison with numerical predictions.

2. Mathematical model for flow through a capillary tube

The homogeneous flow model is widely used to simulate the flow of refrigerants through capillary tubes. This is considered as the most accurate model for high mass flux flow as is the case for flow through capillary tubes [5]. In the present study, a numerical model is developed to simulate the flow of refrigerant mixtures through a capillary tube of an MRJT cryocooler to produce low temperatures. In the capillary tube of a conventional refrigerator, there are two regions: initially, a single-phase region followed by a two-phase region as shown in the schematic of a capillary tube given in Fig. 1. For capillary tubes of MRJT cryocoolers, only two-phase flow exists throughout the entire length of the capillary tube. However, to develop a robust numerical solver, both single-phase and two-phase regions are considered in the theoretical analysis.

To determine the pressure drop for flow through capillary tubes, mass, momentum, and energy equations are solved. In the single-phase region, the properties of the fluid are not affected by pressure, and therefore, an average value of properties can be used to calculate the pressure drop in the single-phase region. However, for the two-phase region, the changes in fluid properties are quite significant, and a global average may result in large quantities of error.

2.1. Assumptions

To simplify the analysis of the flow through capillary tubes, the following assumptions are made:

(i) Flow through the capillary tube is steady.
(ii) Flow through the capillary tube is homogeneous and turbulent.
(iii) Metastable state inside the capillary tube (if present) is neglected.
(iv) Flow through the capillary tube is assumed to be one dimensional.
(v) For single-phase length, the fluid properties are assumed to be constant.

2.2. Governing equations

The governing equations for single-phase as well as two-phase flows are given below. These equations are solved to determine the pressure drop across the capillary tubes. Mass and momentum equations are sufficient to evaluate pressure drop for a single phase, while the energy equation is required to evaluate pressure drop for the two-phase region. Eqs. (1), (2), and (3), as given below, represent mass, momentum, and energy conservation equations respectively.

$$\dot{m} = \rho_1 V_1 A = \rho_2 V_2 A \tag{1}$$

$$\frac{P_1}{\rho g} + \frac{V_1^2}{2g} + z_1 = \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + z_2 + H_L \tag{2}$$

$$h_1 + \frac{V_1^2}{2} = h_f + x(h_g - h_f) + \frac{G^2}{2}(v_f(1-x) + v_g x)^2 \tag{3}$$

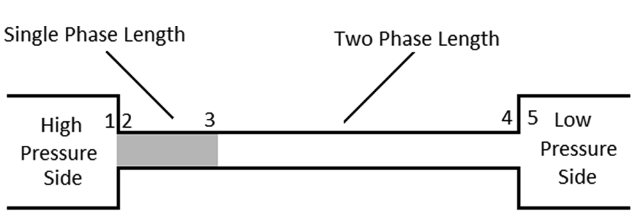


Fig. 1. Schematic of capillary tube.

States 1 and 2 correspond to any arbitrary points across which conservation equations are applied. In the energy equation, the first two terms on the right hand side represent enthalpy at that given point and are written in terms of quality and liquid and vapour enthalpy.

2.3. Single-phase region

For single-phase flow region, the total head losses h_L is evaluated taking into account the entrance losses (h_{Le}) and the skin frictional loss (h_{Lf}) in the single-phase length of the capillary tube. Entrance head loss h_{Le}, is calculated as:

$$H_{L,e} = \frac{kV_3^2}{2g} \tag{4}$$

where k is the entrance loss coefficient.

Similarly, head loss due to wall shear h_{Lf} is calculated as:

$$H_{L,f} = f_{sp}\frac{L_{sp}}{d}\frac{V^2}{2g} \tag{5}$$

Single-phase friction factor f_{sp} is evaluated using Colebrook's equation [14] or Blasius formula [14] as given in Eqs. (6) and (7) respectively.

$$\frac{1}{\sqrt{f_{sp}}} = 1.14 - 2\log\left(\frac{e}{d} + \frac{9.3}{\mathrm{Re}\sqrt{f_{sp}}}\right) \tag{6}$$

$$f_{sp} = 0.316Re^{-0.25} \tag{7}$$

where

$$Re = \frac{\rho V d}{\mu} \tag{8}$$

As for single-phase flow, the property variations are neglected and the average velocity V is taken throughout the region. Substituting the values of each head loss in the momentum equation (Eq. (2)) applied across points 1 and 3 as in Fig. 1 and neglecting V_1, single-phase length is calculated as:

$$L_{sp} = \left[(P_1 - P_3)\frac{2\rho}{G^2} - (k+1)\right]\frac{d}{f_{sp}} \tag{9}$$

2.4. Two-phase region

Most of the pressure drop and cooling happens in the two-phase region of a capillary tube. Due to the large change in properties of the fluid with respect to the pressure, the two-phase region is divided into many small elements.

By applying energy balance to the control volume, the length of the element for a given infinitesimal pressure drop (dP) can be obtained as given below [13].

$$dL_i = \frac{2d}{f_{itp}}\left[\frac{-\rho_i dP}{G^2} + \frac{d\rho_i}{\rho_i}\right] \tag{10}$$

where f_{itp} is the average two-phase friction factor for the elemental control volume and is calculated using Eqs. (6) or (7). The two-phase density required to evaluate two-phase Reynolds number is calculated using Eq. (11) as:

$$\frac{1}{\rho_{itp}} = \frac{1-x}{\rho_f} + \frac{x}{\rho_g} \tag{11}$$

Viscosity is the cause of frictional pressure drop in any flow. In two-phase flow, the determination of viscosity is difficult due to two phases co-existing with each other. To simplify the evaluation of two-phase viscosity, various researchers have developed empirical relations to calculate two-phase viscosity taking into account different assumptions.

In the present work, a few viscosity models are chosen to evaluate two-phase viscosity. The two-phase viscosity can be calculated using any of such viscosity models as given in Eqs. (12)–(16).

McAdams model [15]:

$$\frac{1}{\mu_{itp}} = \frac{x_i}{\mu_{ig}} + \frac{1 - x_i}{\mu_{if}} \tag{12}$$

Cicchitti's Model [16]:

$$\mu_{itp} = x_i\mu_{ig} + (1 - x_i)\mu_{if} \tag{13}$$

Dukler's Model [17]:

$$\mu_{itp} = \frac{x_i v_{ig}\mu_{ig} + (1 - x_i)v_{if}\mu_{if}}{x_i v_{ig} + (1 - x_i)v_{if}} \tag{14}$$

Beattie and Whalley's model [18]:

$$\mu_{itp} = \alpha_{tp}\mu_{ig} + (1 - \alpha_{tp})\mu_{if}(1 + 2.5\alpha_{tp}),\ \alpha_{tp} = \frac{x_i v_{ig}}{v_{if} + x_i v_{ifg}} \tag{15}$$

Lin's Model [8]:

$$\mu_{itp} = \frac{\mu_{ig}\mu_{if}}{\mu_{ig} + x_i^{1.4}(\mu_{if} - \mu_{ig})} \tag{16}$$

Quality (x) is determined using energy equation (Eq. (3)) which is applied from the start of the two-phase length to any point in the two-phase region where the quality is to be evaluated.

2.5. Solution methodology

An algorithm is developed in the present work to evaluate capillary tube length for given variable parameters like mass flow rate, pressure drop, inlet and outlet pressures, capillary tube diameter, and surface roughness of the capillary tube. These parameters are given as input to the numerical solver for which the capillary tube length is evaluated. Also, depending on the inlet conditions, inlet subcooling or inlet quality are provided as inputs to the model. Fig. 2 gives the flow chart for this algorithm.

The capillary tube length for a given pressure drop is calculated using the required inputs. The inlet to the capillary tube may be in a subcooled or two-phase state. For the subcooled inlet, using inlet subcooling, entrance losses, and skin friction losses for single-phase, the capillary length is evaluated for the single phase. The two-phase length is divided into small elements across which a small pressure drop is decided. The incremental length for additional pressure drop is evaluated using the mathematical model explained in Section 2.4. The exit condition for the previous node serves as the inlet condition for the next node in the two-phase region.

Properties are averaged out over one elemental control volume over which Eq. (10) is applied. Computation is stopped when choking happens or when the pressure value reaches evaporator pressure. To determine the occurrence of choking, the concept of unrealistic length [13] is used.

In the present mathematical model, the fluid properties are assumed to be linearly varying along the control volume. Thus, the error induced due to discretisation is of the order $(\Delta P)^2$.

3. Experimental analysis of cryogenic flow through capillary

Several experiments are conducted to compare the predictions obtained using the mathematical model for capillary tube length for different refrigerant mixtures at cryogenic temperatures. The experiments are also carried out to study flow through capillary tubes with respect to different inlet temperatures to the capillary tube.

3.1. Experimental setup and Instrumentation

Fig. 3 shows the schematic of the experimental setup used. The setup consists of two compressors with intercooler and aftercooler to remove the heat of compression. The refrigerant mixture after compression is precooled using Liquid Nitrogen (LN2) bath. The precooling heat exchanger consists of a helically coiled tube dipped in LN2 bath. The refrigerant mixture can be precooled up to 100 K using the LN2 bath. The cold refrigerant mixture is expanded to low pressure using a capillary tube of 0.5 m length and 1.14 mm diameter. The working fluid returns to the suction of the first stage compressor at room temperature. To reduce the heat loss from the capillary tube, it is wrapped with Multi Layer Insulation (MLI) and kept inside a vacuum chamber as shown in the figure.

Pressure, temperature, and mass flow rate are the primary measurements taken during experimentation. To aid the analysis, the composition of the mixture in circulation is also measured. It should be noted that as the precooling coil is dipped in LN2 bath, liquid holdup occurs. This results in the variation of the composition of the refrigerant

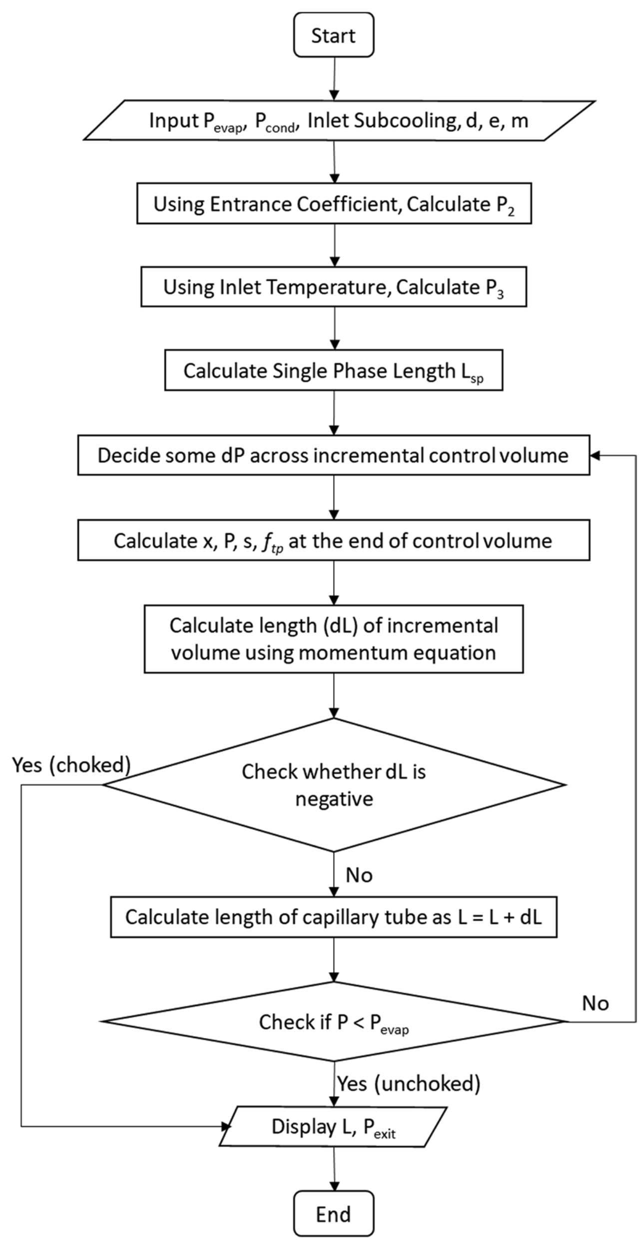


Fig. 2. Capillary tube length prediction algorithm.

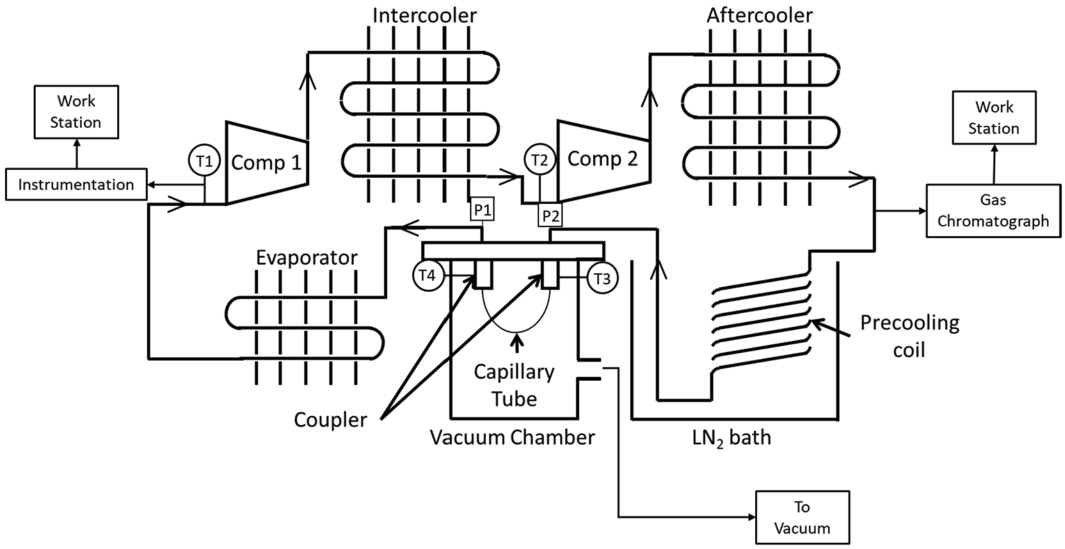


Fig. 3. Experimental setup to study the flow of refrigerant mixture through a capillary tube.

mixture in circulation from what is charged initially. Also, as the number of precooling coils dipped in LN2 increase (i.e. capillary inlet temperature reduces), the amount of composition shift increases. This helps in simulating actual MRJT systems. To ensure that mixture in circulation is the same throughout the system, the sampling of circulating mixture using gas chromatograph is done before and after the precooling coil. It is found that the composition of the mixture in circulation at both the location is approximately the same. The mixture composition in circulation for one set of experiment is provided in Table 5. Various measurement instrument details like type, range, and sensitivity of various instruments used are given in Table 1.

Details of the gas mixtures are given in Table 2. Mixture 1 contains Methane, Ethane, Propane, and Isobutane in equal percentage by volume. While in Mixture 2, in addition to the four hydrocarbons, nitrogen is added keeping the concentration of each component equal. For the gas mixtures, the charging pressures used are 1 and 1.6 MPa.

The uncertainty associated with the measuring instruments is provided in Table 1. Using the uncertainty of individual instruments, the total uncertainty is evaluated using the methodology provided by Moffat [24]. According to him, if a quantity (R) is dependent on m parameters as given in Eq. (17), then the uncertainty in the measurement of the quantity can be calculated as given in Eq. (18).

$$R = X_1^a X_2^b X_3^c \cdots\cdots\cdots .. X_M^m \tag{17}$$

$$\frac{\delta R}{R} = \left\{ \left(a\frac{\delta X_1}{X_1} \right)^2 + \left(b\frac{\delta X_2}{2} \right)^2 + \cdots + \left(m\frac{\delta X_M}{X_M} \right)^2 \right\}^{0.5} \tag{18}$$

Using Eq. (18), the uncertainty for the experimental results is found to be 7%.

As shown in Fig. 3, the capillary tube used here is a straight copper capillary tube of 0.5 m length, 1.14 mm inner diameter, and 75 µm surface roughness (as provided by the manufacturer). Due to the size of the vacuum chamber which is used for insulation, the capillary tube is bent in a U-type configuration. To avoid heat inleaks from the surrounding, the capillary tube is wrapped with MLI and kept inside a vacuum chamber during experimentation.

3.2. Methodology

As mentioned in the previous section, the set of experiments is performed for the two refrigerant mixtures at different charging pressures i.e. 1 and 1.6 MPa. Before starting the experiments, the vacuum chamber, housing the capillary tube, is evacuated to minimise the heat in leaks. The sampling tube is evacuated to get sample of refrigerant mixture in circulation. The experiments begin by turning the aftercooler and intercooler fan on. After waiting for a few minutes, both the compressors are switched on. Liquid nitrogen is poured into the precooling coil until the desired precooling temperature is achieved. The desired precooling temperature is obtained based on the number of coils dipped in liquid nitrogen. Due to continuous boil-off of liquid nitrogen, the liquid level in LN2 bath decreases and is closely monitored. However, this results in a slight variation of precooling temperature which is kept to less than ± 1 °C to maintain accuracy. Temperature and pressure data are recorded once the desired precooling temperature is achieved and when the steady state reaches. The volume flow rate is measured using a rotameter which is calibrated for nitrogen gas. The mass flow rate is calculated using the volume flow rate and density of the mixture.

The mixture of gases is charged in the system according to the composition given in Table 2 at 1 MPa and 1.6 MPa. However, due to the separated two-phase flow in the precooling coil which is dipped in LN2, the phenomena of liquid holdup is observed. As a result, the mixture composition in circulation is not the same as initially charged composition. Also, the amount of liquid holdup increases as the amount of precooling provided in the precooling coil increases. In the MRJT cryocooler also, as the temperature drop increases, the composition shift increases. Thus, the precooling coil replicates the composition shift of an actual MRJT cryocooler. To determine the composition of the mixture in circulation, the sampling of working fluid in circulation is carried out to at the exit of the precooling coils.

3.3. Isenthalpic curves on T-S chart

In order to ascertain the validity of the experimental results for flow through capillary tubes, the experimental results are compared with those obtained using theoretical charts. The comparison is done with respect to temperature after expansion. The theoretical isenthalpic line

Table 1
Instrumentation details.

Instrument	Type	Range	Uncertainty
Flowmeter	Rotameter	0–20 m^3/h	0.5 m^3/h
RTD	PT100	−200 to 400 °C	0.1 °C
Pressure Transducers	Piezoelectric transducers	0–3.5 MPa	1%
Gas Chromatograph	Adsorption column type	–	5%

Table 2
Composition of Refrigerants mixture charged.

Mix. No.	$\%N_2$	$\%CH_4$	$\%C_2H_6$	$\%C_3H_8$	$\%iC_4H_{10}$
Mix 1	–	25	25	25	25
Mix 2	20	20	20	20	20

is drawn on the T-s chart using the capillary inlet and outlet conditions. The theoretical property data for the mixture in circulation is calculated using REFPROP [25]. The outlet condition thus obtained is compared with the experimental results. Such a comparison for two experiments is shown in Fig. 4. From the figure, it can be seen that the theoretical and actual isenthalpic lines are in good agreement with each other, considering the uncertainties in the experiments and the error associated in Equation of State used for theoretical calculations. At the outlet of the capillary tube, the difference between theoretically and experimentally obtained temperature is as shown in Fig. 4 for Mix-2 at a capillary inlet temperature of 150 and 200 K and at a charging pressure of 1 MPa. As the temperature drop across the capillary tube (ΔT) obtained experimentally is in good agreement with ΔT obtained using theoretical means (as obtained using REFPROP [25]), we can conclude here that the experiments conducted in the present case are accurate enough for further validation.

4. Results and discussion

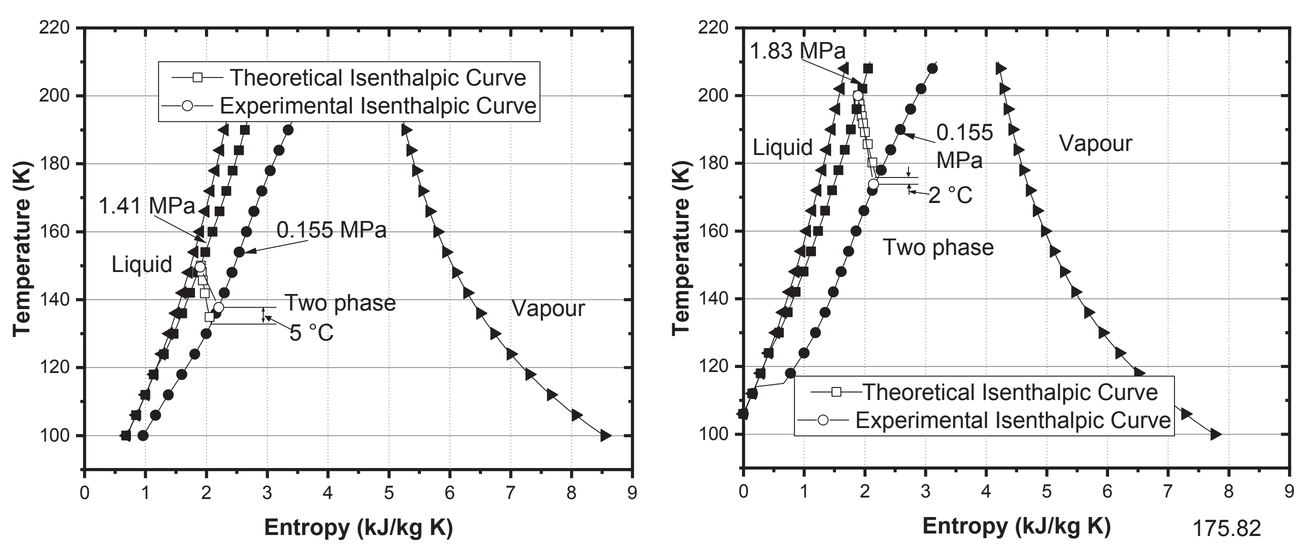
In order to judge the validity of the theoretical prediction, for a given capillary tube, several experiments are carried out using different refrigerant gas mixtures at different charging pressures. Various parameters like capillary outlet temperature (T_o), capillary inlet pressure (P_i), capillary outlet pressure (P_o), the pressure ratio across the capillary tube (P_r), and mass flow rate through the capillary tube ($\dot{m}$) are studied with respect to different Capillary inlet temperatures. The capillary tube is kept the same throughout the experimental analysis. The model also evaluates various two-phase viscosity models for refrigerant mixtures and considers the applicability of different friction factor equations at cryogenic temperatures.

4.1. Experimental results for different refrigerant mixtures at different charging pressure

The inlet temperature to the capillary tube depends on the precooling provided by the LN2 bath, and accordingly the inlet and outlet pressures of the refrigerant mixture change after isenthalpic expansion. Fig. 5 shows the effect of capillary tube inlet temperature on inlet and outlet pressures and subsequently the pressure ratio across the capillary tube. This study provides hindsight into the effect of the component and composition of refrigerant mixtures with respect to temperature at the inlet of the capillary on various operating parameters mentioned above. The temperature of the refrigerant mixture entering the capillary tube is kept in the range of 250–100 K depending on the composition of the mixture. The inlet temperature is varied in steps of 25 K. The capillary tube for all the experiments is kept the same at a length of 0.5 m and a diameter of 1.14 mm.

For Mix 1 ($CH_4/C_2H_6/C_3H_8/iC_4H_{10}$), where each component is 25% by volume for a total charging pressure of 1 MPa, the variation of inlet and outlet pressures and pressure ratio is shown in Fig. 5(a). It may be observed that inlet pressure (P_i) varies from 1.9 MPa to 1.04 MPa as the precooling temperature gets reduced from 250 K to 175 K. Corresponding outlet pressures (P_o) are 0.15 MPa to 0.125 MPa respectively which results in the pressure ratio (P_r) varying from 12.7 to 8.3 respectively.

The effect of the addition of nitrogen in the refrigerant mixture is studied using Mix 2 ($N_2/CH_4/C_2H_6/C_3H_8/iC_4H_{10}$), where each component in the mixture is kept 20% by volume for the same total pressure of 1 MPa. As can be seen from Fig. 5(b), inlet pressure changes from 2 MPa for an inlet temperature of 250 K to 0.75 MPa for an inlet temperature of 100 K. Corresponding outlet pressure varies from 0.15 MPa to 0.09 MPa respectively. As a result, the variation in pressure ratio is from 13.3 to 8.3. It may be noted that for this mixture, a lower capillary inlet temperature of 100 K is achieved as compared to mix 1 due to the presence of nitrogen in Mix 2. This is due to the fact that nitrogen content in the mixture ensures the circulation of the refrigerant mixture at a low precooling temperature of 100 K. At a precooling temperature of 175 K, Mix 1 and Mix 2 develop a pressure ratio



(a) Mix 2, 1 MPa charging pressure, 150 K Capillary inlet temperature

(b) Mix 2, 1 MPa charging pressure, 200 K Capillary inlet temperature

Fig. 4. Comparison between experimental and theoretical isenthalpic expansion.

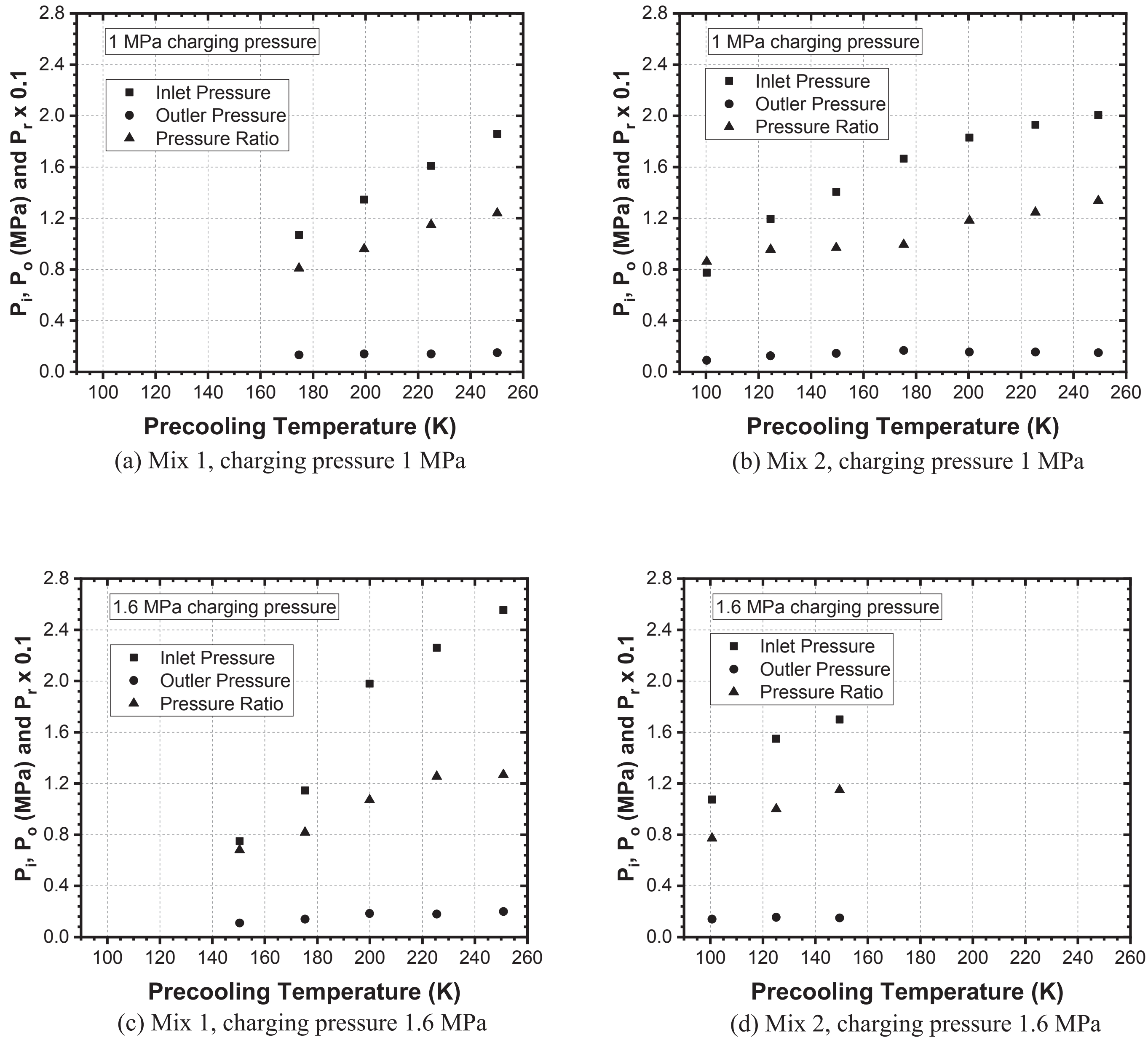


Fig. 5. Effect of charging pressure and mixture composition on Inlet Pressure (P_i), Outlet Pressure (P_o) and Pressure Ratio (P_r).

of 8 and 10 respectively across the capillary tube. As Mix 2 contains nitrogen along with hydrocarbons, the pressure ratio is higher for Mix 2 compared to Mix 1 for the same capillary tube inlet temperature. This is because of an increase in the composition of low boiling components (Nitrogen and Methane) which increases the quality of vapour entering the capillary tube. High quality vapour has less density resulting in large fluid velocities, thereby resulting in large pressure drop across the capillary tube. This also leads to high inlet pressure for refrigerant mixtures with more concentration of low boiling components. For example, for a capillary inlet temperature of 200 K, capillary inlet pressure is 1.35 and 1.83 for Mix 1 and Mix 2 respectively.

The above study is done for 1 MPa charging pressure. A similar study is carried out for higher charging pressure of 1.6 MPa to understand the effect of charging pressure on the operating parameters. The variation of inlet and outlet pressures and pressure ratio for Mix 1 ($CH_4/C_2H_6/C_3H_8/iC_4H_{10}$), for a total charging pressure 1.6 MPa is shown in Fig. 5(c). Here, the inlet and outlet pressures vary from 2.56 to 0.75 and 0.2 to 0.11 respectively, as the precooling temperature decreases from 250 K to 150 K. The corresponding pressure ratio varies from 12.7 to 6.8.

Similarly, the effect of precooling temperature on inlet and outlet pressures and pressure ratio for Mix 2 (N_2/ $CH_4/C_2H_6/C_3H_8/iC_4H_{10}$) for a total pressure of 1.6 MPa is shown in Fig. 5(d). Inlet and outlet pressures vary from 1.7 to 1.08 and 0.15 to 0.14 respectively for a precooling temperature of 150 K to 100 K. This results in the pressure ratio variation from 11.5 to 7.7. For this case too, compared to Mix 1, the pressure ratio is higher for Mix 2 at a particular precooling temperature. This is because the amount of low boiling component (Nitrogen and Methane) is more in Mix 2. Thus, the trend remains similar for both the mixtures irrespective of charging pressure.

The effect of charging pressure on the inlet and outlet pressure and subsequently the pressure ratio for a Mix 1 can be seen from Fig. 5(a) and (c). From the figures, it is observed that for the same mixture composition and at a given capillary inlet temperature, the pressure ratio does not change for different charging pressures. For Mix 1 at a capillary inlet temperature of 175 K, the pressure ratio is 8.1 and 8.2 for

Table 3
Experimental results for Mix 1 & 2 (1 MPa charging pressure).

Sr. No.		T_i (K)	T_o (K)	P_h (MPa)	P_l (MPa)	P_r	ṁ (kg/hr)
1	Mix 1	250.15	203.14	1.86	0.15	12.4	11.4
2	Mix 2	249.42	207.37	2.01	0.15	13.4	10.5
3	Mix 1	224.97	189.38	1.61	0.14	11.5	10.4
4	Mix 2	225.49	194.91	1.93	0.16	12.5	10.6
5	Mix 1	199.47	169.16	1.345	0.14	9.6	10.8
6	Mix 2	200.31	173.84	1.83	0.16	11.8	10.6
7	Mix 1	174.68	147.78	1.07	0.13	8.1	10.3
8	Mix 2	175.3	156.12	1.67	0.17	9.9	11.7
9	Mix 2	149.6	137.67	1.41	0.15	9.7	10.5
10	Mix 2	124.59	105.9	12	0.13	9.6	8.5
11	Mix 2	100.27	98.21	0.78	0.09	8.6	6.9

Table 4
Experimental results for Mix 1 & 2 (1.6 MPa charging pressure).

Sr. No.		T_i (K)	T_o (K)	P_h (MPa)	P_l (MPa)	P_r	ṁ (kg/hr)
1	Mix 1	250.85	201.53	2.56	0.2	12.7	13.8
2	Mix 1	225.49	190.57	2.26	0.18	12.6	11.9
3	Mix 1	199.89	170.22	1.98	0.19	10.7	12.1
4	Mix 1	175.3	147.7	1.15	0.14	8.2	9.5
5	Mix 1	150.37	125.92	0.75	0.11	6.8	7.6
6	Mix 2	149.25	123.84	1.7	0.15	11.5	10.1
7	Mix 2	125.105	100.555	1.55	0.15	10	11.8
8	Mix 2	100.65	93.35	1.075	0.14	7.7	10.7

a charging pressure of 1 MPa and 1.6 MPa respectively. However, the total pressure drop increases in the case of higher charging pressure. A similar trend is also observed in the case of Mix 2.

It may, therefore, be concluded that, as the precooling temperature reduces, the inlet pressure reduces significantly, whereas the outlet pressure reduces marginally. This is mainly because of condensed liquid hold-up. As liquid hold-up occurs in the precooling coil, the mass flow available in circulation reduces, resulting in the reduction of the inlet and outlet pressures. With the decrease in precooling temperature, the amount of liquid in the mixture increases which in turn increases the density of the fluid. The higher density fluid has less velocity for the same mass flow rate. Therefore, the resistance provided by the capillary tube is less for a less velocity, high density fluid. This is the primary reason for the reduction in pressure ratio for all the sets of experiments shown here. This is also similar to what is observed during the transient phase of an MRJT cryocooler where initially the pressure ratio is high, and eventually, it reduces to reach a steady state [3].

Tables 3 and 4 give experimental results for both the mixtures for charging pressures of 1 and 1.6 MPa respectively. Various parameters like Capillary Inlet Temperature (T_i), Capillary Outlet Temperature (T_o), Inlet and Outlet Pressures, and the corresponding pressure ratio along with the mass flow rate (ṁ) through the capillary tube is given in these tables. In order to compare the results for the same capillary inlet temperature for different refrigerant mixtures (Mix 1 and Mix 2) and charging pressures (1 MPa and 1.6 MPa), Table 3 depicts the results for 1 MPa while Table 4 gives results for 1.6 MPa. From Table 3, it may be noted that as the capillary inlet temperature reduces the magnitude of the temperature difference across the capillary tube also reduces. For Mix 1 temperature drop across the capillary tube is 49 K, and 28 K for the capillary inlet temperature of 250 K and 175 K respectively. While, it may be noted that, for the capillary inlet temperature of 175 K, the temperature drop across the capillary tube is 27 K and 19 K for Mix 1 and Mix 2 respectively. It may be seen that the mass flow rate reduces with the decrease in capillary tube inlet temperature. This is a result of the reduction of capillary exit pressure. This can be seen from Table 4; similar trends are observed for 1.6 MPa charging pressure.

Table 6
Comparison between Colebrook's and Blasius friction factor equation.

Mix No.	Charging Pressure (MPa)	Capillary Inlet Temperature (K)	Numerically Obtained Capillary Tube Length	
			Colebrook's Equation [14] (m)	Blasius Equation [14] (m)
Mix 1	1	200	0.4224	0.4431
Mix 1	1.6	250	0.4603	0.4835
Mix 2	1	150	0.4707	0.4801
Mix 2	1.6	125	0.6194	0.6318

From Tables 3 and 4 it can also be highlighted that using Mix 1 for the same capillary inlet temperature, the temperature difference across the capillary tube is nearly independent of charging pressure. However, for Mix 2 it is observed that temperature difference across the capillary tube is higher for 1.6 MPa charging pressure compared to 1 MPa charging pressure. It can also be observed that for higher charging pressure, the mass flow rate for the two refrigerant mixtures is higher. A high mass flow rate yields a more cooling effect.

The refrigerant mixture in circulation will be different from the refrigerant mixture charged in the system initially as explained in Section 3.1. The mixture in circulation for Mix 1 at 1 MPa charging pressure is as given in Table 5. From the table, it can be seen that as the capillary inlet temperature reduces, the composition of high boiling component (propane and butane) reduces, that of low boiling component (Nitrogen and Methane) increases, and that of medium boiling component (ethane) almost remains constant. This is because of the liquid being concentrated in the high boiling components and vapour being concentrated in the low boiling component. Similar trends are observed for both the refrigerant mixtures at charging pressure of 1 MPa and 1.6 MPa.

4.2. Comparison of Colebrook's and Blasius friction factor equations

In order to calculate the capillary length for a given mass flow rate and pressure drop, the friction factor is an important parameter. It depends on various flow properties and surface roughness of the walls.

Table 5
Refrigerant mixture in circulation $N_2/CH_4/C_2H_6/C_3H_8/iC_4H_{10}$ (20% each, total pressure 1 MPa).

Trial No.	Capillary Inlet Temperature *(K)*	Capillary Inlet Pressure *(MPa)*	Capillary Inlet Quality/Degree of subcooling	Concentration *(% vol)*				
				N_2	CH_4	C_2H_6	C_3H_8	iC_4H_{10}
1	249.42	2.01	0.34	20.12	21.79	22.21	24.73	14.02
2	225.49	1.93	0.29	20.54	22.62	22.59	21.12	13.17
3	200.31	1.83	0.27	22.15	24.55	22.48	19.5	11.32
4	175.3	1.67	0.23	22.84	25.62	22.06	18.65	10.84
5	149.6	1.41	0.16	22.32	23.84	21.26	20.0	12.6
6	124.59	11	0.14	25.49	26.64	22.02	15.77	9.79
7	100.27	0.78	1 °C	21.17	21.32	19.98	21.7	15.83

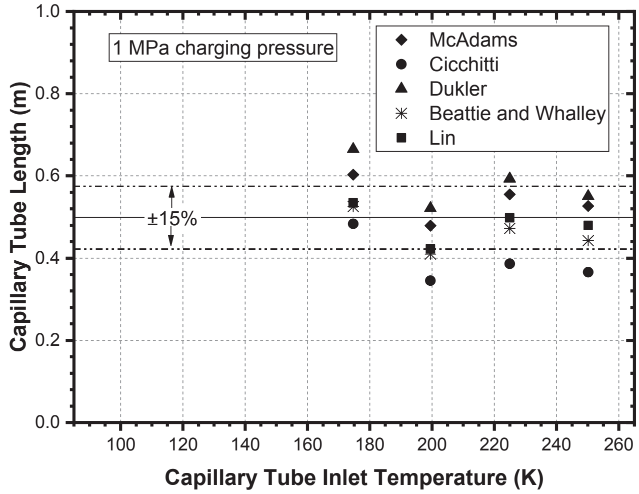


Fig. 6a. Numerical Length Prediction results (Mix 1, 1 MPa charging pressure).

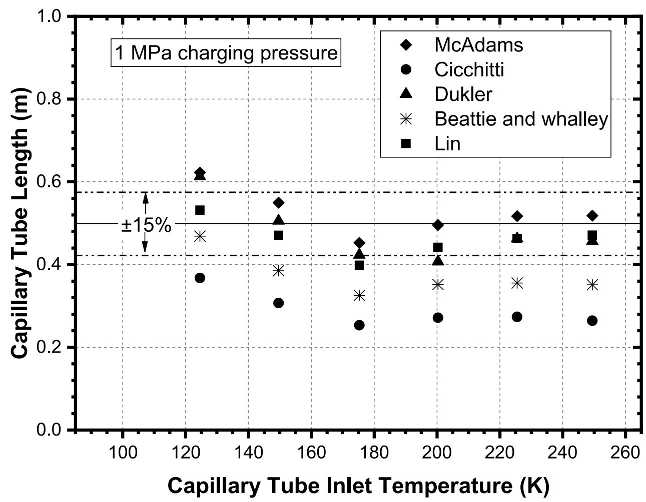


Fig. 6b. Numerical Length Prediction results (Mix 2, 1 MPa charging pressure).

To evaluate the friction factor, various correlations are available in the literature. In the present work, Colebrook's [14] and Blasius [14] friction factor correlations are used to evaluate the friction factor inside the capillary tube. Capillary tube length obtained using the two friction factor correlations is given in Table 6. The numerically obtained capillary tube length for different experimental parameters is shown for Lin's viscosity model [8]. This is done at different charging pressure for the two refrigerant mixtures used in the present work.

From the table, it is clear that the capillary tube length obtained using both the correlations is in good agreement with each other for both the refrigerant mixtures. In view of this, all the future calculations in the present work are carried out using Blasius friction factor correlation only.

4.3. Capillary length predictions for different viscosity models

The main objective of the numerical simulation in the present work is to evaluate capillary tube length based on the experimental data in terms of inlet and outlet pressures, inlet temperature, and mass flow rate for a given capillary diameter. These predictions are compared with the actual capillary length used in the experiments. Fig. 6 shows the comparison between the actual capillary tube length and the numerically predicted capillary tube length based on various viscosity models as described earlier.

Fig. 6a shows this comparison for Mix 1 at the charging pressure of 1 MPa. In the figure, the solid line shows the actual capillary tube length of 0.5 m used for all the experiments, while the dashes show the margin of ± 15% against the actual value. This is done to compare the length predictions obtained using various viscosity models.

It can be observed that the viscosity models proposed by McAdams, Beattie and Whalley, and Lin predict the capillary tube length within ± 15% for most of the experiments. This can be noticed clearly from the figure at a capillary inlet temperature of 225 K. It is also observed that Dukler's viscosity model over predicts the capillary tube length while Cicchitti's viscosity model under predicts the capillary tube length for this temperature. A similar trend with a slight variation can be observed for other capillary inlet temperatures as well. Fig. 6b shows the comparison between actual and numerically predicted lengths for Mix 2 at 1 MPa charging pressure.

The same observations are seen for Mix 2 as for Mix 1. For Mix 2, at a capillary inlet temperature of 175 K, the capillary tube length prediction using Lin's viscosity model is slightly out of bound, however, it is still close to the bound of ± 15%. In Fig. 6b, tube length prediction for an inlet temperature of 100 K is not shown as the predictions are out of the scale of the figure. This is because of the inlet to the capillary being in the subcooled state. When the inlet to the tube is in the subcooled state, the metastable state would be present during phase transition. As modelling of the metastable state is not incorporated in the

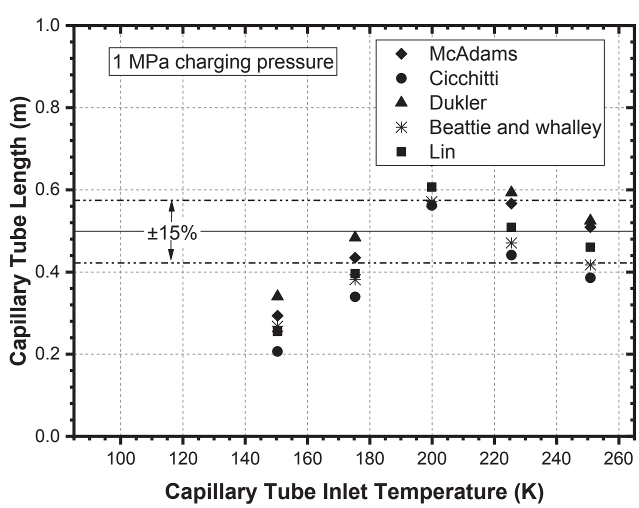


Fig. 6c. Numerical Length Prediction results (Mix 1, 1.6 MPa charging pressure).

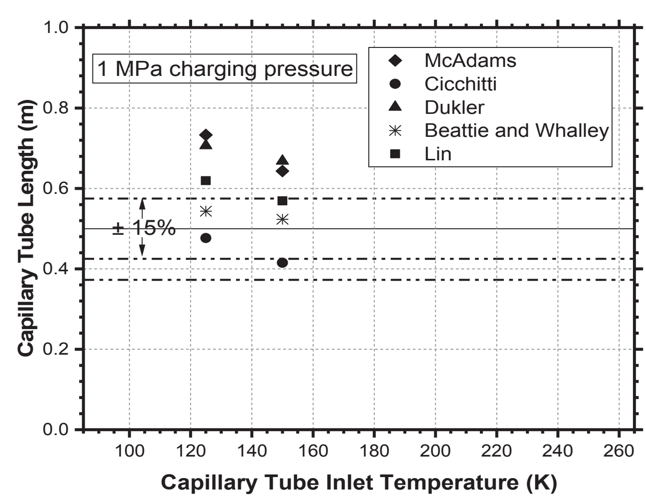


Fig. 6d. Numerical Length Prediction results (Mix 2, 1.6 MPa charging pressure).

present mathematical model, the error involved for capillary tube length prediction for such a case is high.

Figs. 6c and 6d show the capillary tube length prediction for Mix 1 and Mix 2 respectively at 1.6 MPa charging pressure. From the figure, it can be observed that Beattie and Whalley and Lin's viscosity models predict the capillary tube length within an error of ± 15% for almost all the cases. For Mix 1, for a capillary inlet temperature of 200 K, 175 K, and 150 K, the error associated with Beattie and Whalley and Lin's viscosity model is greater than ± 15%. However, for the capillary inlet temperature of 200 K and 175 K, the error is still within ± 20%. For the capillary inlet temperature of 150 K, the error in prediction is large as the inlet condition to the capillary is subcooled. For Mix 1, at 1.6 MPa charging pressure, McAdams viscosity model predicts the capillary tube length within ± 20% for all experimental conditions except for a capillary inlet temperature of 200 K and 150 K. However, it fails to give good prediction for Mix 2 at 1.6 MPa charging pressure.

Similarly, for Mix 2, Beattie and Whalley viscosity model predicts the capillary tube length within ± 20% of the actual value for a capillary inlet temperature of 150 K and 125 K. Lin's viscosity model predicts the capillary tube length within ± 20% of the actual value for the capillary inlet temperature of 150 K, while for the capillary inlet temperature of 125 K, the error is slightly higher than 20%. For the capillary inlet temperature of 100 K, the inlet to the capillary is in the subcooled state resulting in the numerical predictions being out of bound of the scale, and therefore it is not shown in the figure.

Thus, it can be seen from Fig. 6 that only for a few viscosity models, the numerical predictions match well with the experimental results. It is observed that the viscosity models proposed by McAdams, Beattie and Whalley, and Lin predict the capillary tube length within ± 15% of the actual value for most of the experimental conditions. However, for a few experimental data points, the numerical prediction involves large error when compared to the actual capillary tube length.

It may be noted that for the refrigerant flow through capillary tube at cryogenic temperature, average error associated with length prediction is 38.21% for McAdams model, 43.98% for Cicchitti's model, 39.33% for Dukler's model, 33.86% for Beattie and Whalley model, and 31.05% for Lin's model. However, if we exclude the cases where inlet to the capillary tube is in the subcooled state, i.e. metastable state is present, then the error goes down considerably. Therefore, considering only the cases where inlet to the capillary tube is in the two phase state, errors for capillary tube length prediction are 16.07% for McAdams model, 28.99% for Cicchitti's model, 18.65% for Dukler's model, 17.85% for Beattie and Whalley model, and 12.17% for Lin's model.

It may be noted from the figure that overall Dukler's model overpredicts the capillary tube length, while Cicchitti's model underpredicts the capillary tube length for most of the numerical simulations. From the figure, it can also be seen that as the precooling temperature reduces, the error in numerical prediction increases. This is due to the reduction of reliability of property data for refrigerant mixtures at cryogenic temperatures. It can also be highlighted that inside the capillary, there can also be the presence of Vapour-Liquid-Liquid equilibria (VLLE) instead of Vapour-Liquid equilibria (VLE). Accurate property data is not available in REFPROP [25] for VLLE. This results in inaccurate thermodynamic property data and thereby inaccurate capillary tube length prediction.

Some of the errors can also be attributed to the Equation of State that was used for calculating the thermodynamic and transport properties of refrigerant mixtures.

5. Conclusions

The present paper highlights the effect of capillary inlet temperature on various parameters, namely capillary inlet and outlet pressures, pressure ratio across the capillary tube, and mass flow rate for flow of refrigerant mixtures through capillary tubes at cryogenic temperatures. It also aims at developing a mathematical model to predict the capillary tube length for given the conditions. Several experiments are carried out for various inlet temperatures, mixture compositions, and charging pressures. Additionally, a comparison between the friction factor using Colebrook's and Blasius correlations is also carried out.

It may be concluded that numerical results based on McAdams, Beattie and Whalley or Lin's models for two-phase viscosity predict capillary tube length within ± 20% of actual value. Lin's viscosity model is found to be consistent and is recommended for the prediction of two-phase viscosity.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

References

[1] V.M. Brodyanskii, E.A. Gromov, A.K. Grezin, V.M. Yagodin, V.A. Nikolaskii, A.G. Tashchina, Efficient throttling cryogenic refrigeration, Khimicheskoe Neftyanoe Mashinostroenie 12 (1971) 16–18.
[2] N. Walimbe, Investigations on Mixed Refrigerant Joule-Thompson Cryocooler for 80 K Applications, PhD thesis Indian Institute of Technology, Bombay, 2009.
[3] P. Ardhapurkar, Investigations on Two-Phase Heat Exchanger for Mixed Refrigerant Joule-Thompson Cryocooler, PhD thesis Indian Institute of Technology, Bombay, 2015.
[4] R. Satya Maher, G. Venkatarathnam, Estimation of performance of a J-T Refrigerators Operating with Nitrogen-Hydrocarbon Mixtures and a Coiled tubes-in-tube heat exchanger, Cryogenics 92 (2018) 27–35.
[5] J.G. Collier, J.R. Thome, Convection Boiling and Condensation, Oxford University Press, New York, 1994.
[6] C. Melo, R.T.S. Ferreira, C.N. Boabaid, J.M. Goncalves, M.M. Mezavi, An experimental analysis of adiabatic capillary tubes, Appl. Therm. Eng. 19 (1999) 669–684.
[7] F. Sanzovo, O. Mattos, Refrigerant mixtures flow through capillary tubes: a comparison between homogeneous and separated-flow models, HVAC&R Res. 9 (2003) 33–53.
[8] S. Lin, C.C.K. Kwork, R.Y. Li, Z.H. Chen, Z.Y. Chen, Local frictional pressure drop during vaporization for R12 through capillary tubes, Int. J. Multiph. Flow 17 (1) (1991) 95–102.
[9] S.K. Dubba, R. Kumar, Adiabatic flow characteristics of R-600a inside a helically coiled capillary tube, Appl. Therm. Eng. 132 (2018) 500–507.
[10] M. Rasti, J.H. Jeong, A generalized continuous empirical correlation for the refrigerant mass flow rate through adiabatic straight and helically coiled capillary tubes, Appl. Therm. Eng. 143 (2018) 450–460.
[11] P.K. Bansal, A.S. Rupasinghe, A homogeneous model for adiabatic capillary tubes, Appl. Therm. Eng. 18 (1997) 207–219.
[12] S. Churchill, Frictional equation spans all fluid flow regimes, Chem. Eng. J. 84 (1997) 91–92.
[13] D. Subodh, S.D. Deodhar, B. Hardik, H.B. Kothadia, K.N. Iyer, S.V. Prabhu, Experimental and numerical studies of choked flow through adiabatic and diabatic capillary tubes, Appl. Therm. Eng. 90 (2015) 879–894.
[14] R.K. Bansal, A textbook of Fluid Mechanics and Hydraulic Machines, Laxmi Publications (2004).
[15] W. McAdams, W. Wood, R. Bryan, Vaporisation inside horizontal tubes Benzene-oil mixture, ASME 64 (1942) 193.
[16] A. Cicchitti, C. Lombardi, M. Silvestri, G. Soldaini, R. Zavattarelli, Two-phase cooling experiments-pressure drop, heat transfer and burnout measurements, Energia Nucleare 7 (1960) 407–425.
[17] A. Dukler, M. Wicks, R. Cleveland, Frictional pressure drop in two-phase flow, AIChE J. 10 (1964) 38–51.
[18] D. Beattie, P. Whalley, A simple two-phase frictional pressure drop calculation

method, Int. J. Multiph. Flow 8 (1981) 83–87.
[19] S. Kruthiventi, G. Venkatarathnam, Studies on capillary tube expansion device used in J-T refrigerators operating with nitrogen-hydrocarbon mixtures, Cryogenics 87 (2017) 76–84.
[20] P. Ardhapurkar, A. Srisharan, M. Atrey, Investigation of pressure drop in capillary tube for Mixed Refrigerant Joule-Thompson cryocooler, AIP Conf. Proc. 1573 (2014) 155.
[21] L. Friedel, Improved friction pressure drop correlations for horizontal and vertical two-phase pipe flow, The European Two-phase Flow Group Meeting, 1979.
[22] R. Gronnerud, Investigation of liquid hold-up, flow-resistance and heat transfer in circulation type evaporators, part IV: Two-phase flow resistance in boiling refrigerants. Bull. de l'Inst. du Froid, 1979, 1.
[23] H. Muller-Steinhagen, K. Heck, A simple friction pressure drop correlation for two-phase flow in pipes, Chem. Eng. Process 20 (1986) 297–308.
[24] R.J. Moffat, Describing the uncertainties in experimental results, Exp. Therm Fluid Sci. 1 (1988) 3–17.
[25] National Institute of Standards and Technology (NIST), Maryland USA, 2017, 'Reference Fluid Thermodynamic and Transport Properties Database (REFPROP)', ver. 10.