



Performance evaluation of heat exchanger for mixed refrigerant J–T cryocooler



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ABSTRACT

In mixed refrigerant Joule–Thomson cryocooler, a multi-component mixture of nitrogen–hydrocarbons undergoes evaporation and condensation process in a helical coiled heat exchanger simultaneously at different pressures. Experimental data and empirical correlations for predicting heat transfer coefficients of evaporating and condensing streams of multi-component mixtures at cryogenic temperatures are unavailable. As a result, design of these heat exchangers is a challenging task.

The present work aims to address this challenge. It assesses the existing condensation correlations against the calculated data obtained during experimentation. Experiments are conducted to determine overall heat transfer coefficients along the length of the heat exchanger for various mixtures. The paper studies the applicability of these correlations to the multi-component mixtures at cryogenic temperatures.

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1. Introduction

Mixed refrigerant Joule–Thomson (MR J–T) cryocooler consists of compressor, an after-cooler, recuperative heat exchanger, an evaporator and capillary tube as an expansion device. The mixture of refrigerants such as nitrogen, methane, ethane, propane and iso-butane is used as a working fluid. The mixed refrigerant at high pressure from the compressor is cooled in the after-cooler and then enters the counter flow heat exchanger. It is pre-cooled in the heat exchanger, prior to the J–T expansion in the capillary tube. The low pressure refrigerant after expansion is capable of producing refrigerating effect in the evaporator. The refrigerant returns to the compressor through the heat exchanger acting as a cold stream. The high pressure stream gets condensed while the return line, low pressure stream from the evaporator gets evaporated simultaneously in the heat exchanger which contributes to the high efficiency of these cryocoolers.

Many experimental and numerical studies have been carried out on the MR J–T cryocooler. These are mainly related to the optimization of mixtures used and the thermodynamic performance of the overall refrigeration system [1–6]. However, the accurate design of the recuperative heat exchanger is currently limited due to lack of general heat transfer model that allows realistic

prediction of the flow boiling and condensation heat transfer coefficients for a multi-component fluid at cryogenic temperatures.

Little work has been published about the performance of the heat exchanger for MR J–T cryocooler. Gong et al. [7] reported experimental results in terms of pressure drop and temperature distribution for different operating conditions of tubes-in-tube heat exchangers with different mixtures. Ardhapurkar et al. [8] presented a study on the performance of the multi tubes-in-tube type helical coil heat exchanger for MR J–T cryocooler. The work analyzed the effect of mixture composition on the performance of the heat exchanger, in terms of variation in overall heat transfer coefficients along the length. Alexeev et al. [9] numerically simulated multi tubes-in-tube heat exchanger for different mixture compositions. He used a modified Chen correlation to calculate the heat transfer coefficients for forced convection, as well as for condensation of mixtures. However, the calculated results were not compared with the experimental data, except for the pressure drop on the shell side. Nellis et al. [10] obtained experimental data for heat transfer coefficients for mixed refrigerants used in the cryocooler at various operating conditions. This is probably the only reported experimental study on the flow boiling of nitrogen–hydrocarbons multi-component mixtures at cryogenic temperature.

Substantial work related to boiling and condensation of mixtures is available in the literature which is compiled in the review articles by Celata et al. [11], Cheng and Mewes [12] and Cavallini et al. [13]. However, these empirical or semi-empirical correlations

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Nomenclature

A	surface area, m^2	Z_g	parameter in Eq. (11)
ALMTD	apparent logarithmic mean temperature difference, K	<i>Subscripts</i>	
C_p	specific heat, J/kg K	<i>avg</i>	average
C_T	constant in Eq. (7)	<i>bub</i>	bubble point
D	diameter, m	<i>c</i>	cold
g	acceleration due to gravity, m/s^2	<i>cond</i>	condensation
G	mass flux, $kg/m^2 s$	<i>crit</i>	critical
G_{eq}	equivalent mass flux in Eq. (6), $kg/m^2 s$	<i>dew</i>	dew point
h	heat transfer coefficient, $W/m^2 K$	<i>exp</i>	experimental
h_{lg}	latent heat of vapourization, J/kg	g	gas
i	enthalpy, J/kg	h	hot
ID	inside diameter, m	i	section of heat exchanger
J_G	dimensionless gas velocity, $J_G = xG/[gD\rho_g(\rho_l - \rho_g)]^{0.5}$	<i>in</i>	inlet, inside
J_G^*	transition dimensionless gas velocity	l	liquid
k	thermal conductivity, W/m K	<i>lo</i>	liquid only
LMTD	logarithmic mean temperature difference, K	m	mixture
\dot{m}	mass flow rate, kg/s	<i>out</i>	outlet, outside
Nu	Nusselt number, dimensionless	<i>red</i>	reduced
OD	outside diameter, m	s	saturation
p	pressure, Pa	<i>strat</i>	stratified
Pr	Prandtl number, dimensionless	<i>th</i>	theoretical
q	heat transfer in segment, J/kg	<i>total</i>	total
Q	heat transfer rate, W	<i>Greek symbols</i>	
r_1	inside radius, m	μ	dynamic viscosity, Ns/m^2
r_2	outside radius, m	ρ	mass density, kg/m^3
Re	Reynolds number, dimensionless	Δi	enthalpy difference, J/kg
Re_{eq}	equivalent Reynolds number in Eq. (5), dimensionless	ΔT	temperature difference, K
T	temperature, K	ΔT_g	temperature glide, K
U	overall heat transfer coefficient, $W/m^2 K$		
x	quality, dimensionless		
X_{tt}	Martinelli parameter for turbulent-liquid and turbulent-vapour flow		

are mainly developed for the binary mixtures of CFC, HCFC refrigerants and are useful for temperature close to ambient. Additionally, two-phase heat transfer of mixtures encountered in MR J–T cryocooler experience temperature glide more than 100 K, whereas it is only up to 10 K for the refrigerant mixtures used in heat pump and refrigeration systems. Therefore, variation in the thermo-physical properties of nitrogen–hydrocarbon mixtures at cryogenic conditions is significant. However, these studies on the mixtures, reported in the literature, give basic understanding of the heat transfer phenomenon in the mixtures. Hence, it is of significance to study flow boiling and condensation phenomenon of mixtures suitable to MR J–T cryocooler.

Recently, Ardhapurkar et al. [14] assessed the existing flow boiling correlations for mixtures of nitrogen–hydrocarbons using experimental data reported by Nellis et al. [10]. It is found that the existing pure component correlations are not suitable for the mixed refrigerants. They modified the existing flow boiling correlations to apply them for multi-component mixtures at cryogenic temperatures. Three different approaches: Gungor–Winteron correlation [15] along with the Thome and Shakir correction factor for the mixture effect, a modified form of Gungor and Winterton [15] correlation in Silver [16], Bell and Ghaly [17] method and modified Granryd correlation [18] are used to predict flow boiling heat transfer coefficients. These modified correlations are found to be suitable for the mixed refrigerants and are more accurate for the mixtures with relatively low temperature glides. Ardhapurkar et al. [14] further recommended modified Granryd correlation for flow boiling heat transfer coefficients of mixed refrigerants.

In the present work, experiments are carried out to deduce overall heat transfer coefficients of the heat exchanger for different mixtures. Three different mixtures are used to investigate the performance of the heat exchanger. The existing condensation correlations are assessed to study their applicability for the mixed refrigerants. For this purpose, the overall heat transfer coefficients are calculated using different condensation correlations and using modified Granryd correlation [14] for flow boiling of the mixtures. The computed values of the overall heat transfer coefficients, along the length of the heat exchanger, are compared with the experimental values.

2. Existing condensation heat transfer correlations

There are many condensation heat transfer correlations available in the literature, which are mainly developed for CFC, HCFC and HFC refrigerants. These correlations are primarily tested for the condensation of pure components. In the present work, only those correlations, which are widely used and are applicable in the present case, are presented here. These include Shah [19], Dobson and Chato [20], Cavallini and Zecchin [21] and Cavallini et al. [22] correlations.

Shah correlation [19] is often used for condenser design because of its relative simplicity. It is developed for condensation using a two-phase multiplier approach and is valid for annular flow regime. It can also be used to find the local condensation heat transfer coefficient. The condensation heat transfer coefficient is expressed in Eq. (1).

$$h_{cond} = h_{lo} \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{p_{red}^{0.38}} \right] \quad (1)$$

In Eq. (1), h_{lo} is heat transfer coefficient assuming all mass flowing as liquid which is calculated by the following equation.

$$h_{lo} = 0.023 \left(\frac{k_l}{D} \right) \text{Re}_{lo}^{0.8} \text{Pr}_l^{0.4} \quad (2)$$

where $\text{Re}_{lo} = \frac{GD}{\mu_l}$ and $p_{red} = \frac{p_s}{p_{crit}}$.

Dobson and Chato [20] investigated the condensation of pure refrigerants of R22 and R134a and mixed refrigerant of R32/R125 in horizontal tubes of 3.14–7.04 mm inside diameter. They developed two correlations based on the observed flow regimes for wavy flow and annular flow, respectively, and utilized Froude number as the criterion to distinguish the transition of flow regimes. It is suited for flow in both horizontal and vertical tubes. This correlation is commonly used in the literature for mixtures. The heat transfer coefficient during annular flow is calculated from Eq. (3).

$$\text{Nu} = 0.023 \text{Re}_l^{0.8} \text{Pr}_l^{0.4} \left[1 + \frac{2.22}{X_{tt}^{0.89}} \right] \quad (3)$$

where $\text{Re}_l = \frac{G(1-x)D}{\mu_l}$ and X_{tt} is Martinelli parameter for turbulent-liquid and turbulent-vapour flow as given in Eq. (4).

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_g} \right)^{0.1} \quad (4)$$

Cavallini and Zecchin [21] developed theoretical analysis based on the analogy between momentum and heat transfer. Basically, their correlation is modification of the Dittus–Boelter single phase forced convection correlation. The condensation heat transfer coefficient for annular flow regime is given in Eq. (5).

$$h_{cond} = 0.05 \left(\frac{k_l}{D} \right) \text{Re}_{eq}^{0.8} \text{Pr}_l^{1/3} \quad (5)$$

where the equivalent Reynolds number, Re_{eq} for the two-phase flow is calculated for an equivalent mass flux, G_{eq} , defined as

$$\text{Re}_{eq} = G_{eq}D/\mu_l \quad (6)$$

where $G_{eq} = G \left[(1-x) + x \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \right]$.

Cavallini et al. [22] proposed a correlation for heat transfer coefficient for film condensation inside horizontal tubes. The model includes two different flow categories: ΔT -dependent and ΔT -independent flow regime. Here, ΔT is the difference between the saturation temperature and the wall temperature. They suggested the Bell and Ghaly [17] correction in the calculation of the heat transfer coefficient during the condensation of non-azeotropic mixtures of HFC and HC fluids. The correction method is applied to condensation of R407C, R125/R236ea and R290/R600a mixtures. The temperature glide for these mixtures is in the range of 5–10 K. The transition dimensionless gas velocity (or modified Froude number) as expressed in Eq. (7) is used as a criterion to subdivide the flow regime.

$$J_G^T = \left\{ \left[\frac{7.5}{4.3X_{tt}^{1.111} + 1} \right]^{-3} + C_T^{-3} \right\}^{-1/3} \quad (7)$$

where $C_T = 1.6$ for hydrocarbons and 2.6 for other refrigerants. The Martinelli parameter, X_{tt} is given in Eq. (4). The heat transfer coefficient for the ΔT -independent flow regime, when $J_G > J_G^T$ is given in Eq. (8).

$$h_{cond,A} = h_{lo} \left[1 + 1.128x^{0.8170} \left(\frac{\rho_l}{\rho_g} \right)^{0.3685} \left(\frac{\mu_l}{\mu_g} \right)^{0.2363} \left(1 - \frac{\mu_g}{\mu_l} \right)^{2.144} \text{Pr}_l^{-0.1} \right] \quad (8)$$

where the liquid-phase heat transfer coefficient, h_{lo} , is calculated by Eq. (2).

For the ΔT -dependent flow regime, i.e. when $J_G \leq J_G^T$, the heat transfer coefficient is given in Eq. (9). It is related to the heat transfer coefficient, $h_{cond,A}$ i.e. Eq. (8) and to a fully-stratified flow heat transfer coefficient, h_{strat} , as expressed in Eq. (10).

$$h_{cond,D} = \left[h_{cond,A} \left(\frac{J_G^T}{J_G} \right)^{0.8} - h_{strat} \right] \left(\frac{J_G}{J_G^T} \right) + h_{strat} \quad (9)$$

$$h_{strat} = 0.725 \{ 1 + 0.741 [(1-x)/x]^{0.3321} \}^{-1} \left[\frac{k_l^3 \rho_l (\rho_l - \rho_g) g h_{lg}}{\mu_l D \Delta T} \right]^{0.25} + (1-x)^{0.087} h_{lo} \quad (10)$$

The heat transfer mechanism for condensation of non-azeotropic mixtures is same as that occurs during condensation of a pure fluid [23]. Therefore, all the above correlations, developed for condensation of pure vapours, in principle can be applied to condensation of mixtures. This is with the assumption that complete mixing in the liquid and the vapour phases occurs and overall equilibrium is maintained [13]. In case of mixtures, the condensation process is non-isothermal. Hence, there exists sensible heat effects for both liquid and vapour phases. This can be predicted by using the Silver [16], Bell and Ghaly [17] correction to account for additional mass transfer resistance due to mixture effect. Application of this well known Silver–Bell–Ghaly method for condensation of several CFC, HCFC refrigerant mixtures is shown by many researchers [22–24]. The corrected heat transfer coefficient for the condensation of refrigerant mixture, h_m , is given in Eq. (11).

$$\frac{1}{h_m} = \frac{1}{h_{cond}} + \frac{Z_g}{h_g} \quad (11)$$

where h_{cond} is the condensate film heat transfer coefficient for mixture calculated by using one of the above condensation correlations for pure component with properties of the mixture and h_g is heat transfer coefficient of the vapour phase flowing alone, which is calculated by Dittus–Boelter equation as given in Eq. (12).

$$h_g = 0.023 \left(\frac{k_g}{D} \right) \text{Re}_g^{0.8} \text{Pr}_g^{0.4} \quad (12)$$

The parameter Z_g , in Eq. (11), is the ratio of the sensible cooling of the vapour to the total cooling rate, which can be written as

$$Z_g = x \cdot C_{p_g} \cdot \frac{dT_{dew}}{di} \cong x \cdot C_{p_g} \cdot \frac{\Delta T_g}{\Delta i_m} \quad (13)$$

where C_{p_g} is the specific heat of the gas phase, $\frac{dT_{dew}}{di}$ is the slope of the dew point temperature curve with respect to the enthalpy of the mixture as it condenses, ΔT_g is the temperature glide and Δi_m is the enthalpy of isobaric condensation of the mixture.

3. Experimental set-up

Fig. 1 shows the schematic of the experimental set-up. It is described in detail elsewhere by the authors [25]. It mainly consists of a compressor, an after-cooler, oil filters, a heat exchanger, an expansion device, and an evaporator. A simple helically coiled tube-in-tube heat exchanger is used in the present work. The plain tubes are helically wound using cylindrical mandrel. It is mounted on the lathe machine and rotated manually at very low speed without any jerks to have uniform coiling of the tubes. Due care is taken

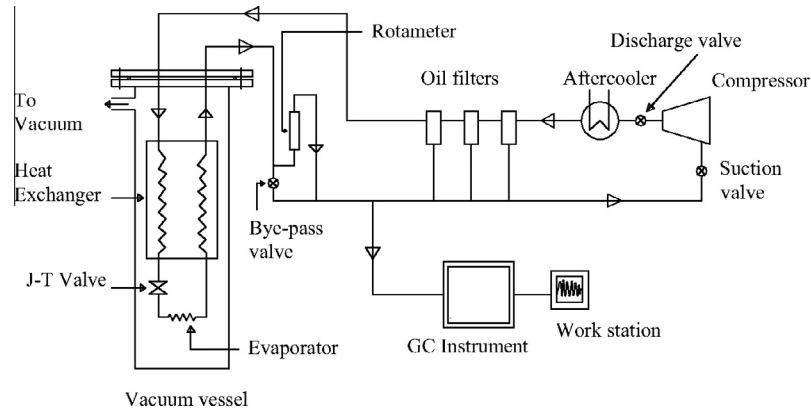


Fig. 1. Experimental set-up.

Table 1
Specifications of the heat exchanger.

Parameter	Value
Inner tube, ID (mm)	4.83
Inner tube, OD (mm)	6.35
Outer tube, ID (mm)	7.89
Outer tube, OD (mm)	9.52
Length of heat exchanger (m)	15
Coil diameter (mm)	200
Coil pitch (mm)	14.5
Number of turns of tubes	23

to avoid any flattening/pinching of the outer and the inner tubes due to excess pressure during bending process.

The dimensions of the heat exchanger are as given in Table 1. A capillary tube is used as an expansion device. The length and the inside diameter of the capillary tube is 2.0 m and 1.52 mm respectively. The suction and the discharge pressures of the compressor are measured by two pressure gauges located at the inlet and the outlet of the compressor respectively. Pressures of the low and the high pressure stream are measured both at the inlet and the outlet to the heat exchanger with the help of pressure gauges (Make: WIKA, Germany) with an accuracy of 0.1% full scale.

A rotameter is installed in the suction line near the compressor to measure the volume flow rate of the refrigerant. The mass flow rate of the refrigerant mixture is calculated using the density of the mixture in circulation, at the inlet conditions. The composition of the mixture in circulation is measured at steady state operation of the cryocooler. For every experiment, the composition of the mixture in circulation is obtained using a gas chromatograph (Make: Perkin Elmer-Clarus500GC). The gas chromatograph instrument is calibrated with components of known purity and mixtures of known composition. All the thermodynamic properties of the mixture are calculated using aspenONE [26] considering Peng–Robinson equation of state [27].

3.1. Temperature measurement

The insertion of temperature sensors into the inner tube of the heat exchanger for measuring hot fluid temperature is quite

challenging. In the present work, to measure temperatures of hot fluid, a sensor belt is made by perfectly binding the sensors on one thin supporting wire using Teflon tape. This ensured consistent distance between any two sensors.

A total of eleven temperature sensors (PT100) are used to measure temperature of the hot fluid at an interval of 1.5 m along the length of the heat exchanger. Fig. 2 shows the locations of the sensors on the heat exchanger for both the streams. The lead wires of three wire sensors are taken out from both the ends of the tube through a T-junction so as to have a bundle of lead wires of uniform thickness, passing through the inner tube. The outlets of both the T-connectors, from where lead wires of sensors are taken out from the inner tube, are filled with the low temperature epoxy (Stycast 2850) material. An equal number (11) of temperature sensors are installed on the outside surface of the outer tube to measure return fluid temperature as shown in Fig. 2. Refrigeration temperature is also measured with the sensor installed at the outlet of capillary tube.

All the temperature sensors are calibrated up to liquid nitrogen temperature (77 K). Temperature data at various locations is recorded using the data logging system, Data Taker-800. The temperatures of the hot and the cold fluid, recorded at the steady state are averaged over the period of minimum 10 min. In earlier work [25], authors carried out experiments to investigate the effect of physical existence of the temperature sensors on the hot fluid temperature measurement inside the inner tube. It is found that the obstruction caused to the flow by the temperature sensors is not significant and can be neglected. The paper [25] also presents the uncertainties propagated in all the parameters for the experiments.

4. Data reduction

The temperatures of the hot and the cold fluid are measured at different locations along the length of the heat exchanger. The temperature profiles and their dependence on the mixture composition are presented in the separate paper [25]. In order to study the performance of the heat exchanger with respect to mixture composition, the local values of the overall heat transfer coefficients for each segment are obtained using the measured temperature profiles of the fluids.

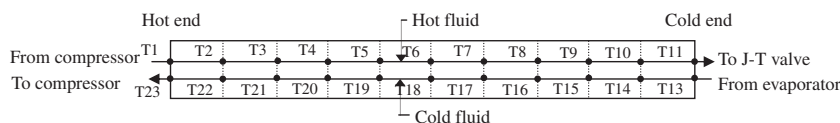


Fig. 2. Temperature sensors on heat exchanger.

The apparent logarithmic mean temperature difference (ALMTD) and heat transferred, q , are calculated in each section of the heat exchanger. The average logarithmic mean temperature difference (LMTD) for the heat exchanger is defined as given in Eq. (14).

$$\text{LMTD}_{avg} = \frac{\sum q_i}{\sum (q_i/\text{ALMTD}_i)} = \frac{i_{c,out} - i_{c,in}}{\sum (q_i/\text{ALMTD}_i)} \quad (14)$$

where $i_{c,out}$ and $i_{c,in}$ are enthalpies of the cold fluid at the outlet and the inlet to the heat exchanger respectively. The average experimental overall heat transfer coefficient, U_{exp} , based on inside surface area A , is calculated using Eq. (15).

$$U_{exp} = \frac{\dot{Q}_{total}}{A(\text{LMTD})_{avg}} = \frac{\dot{m}(i_{c,out} - i_{c,in})}{A(\text{LMTD})_{avg}} \quad (15)$$

The theoretical overall heat transfer coefficient based on the inside surface of the tube, U_{th} , is given in Eq. (16).

$$U_{th} = \left[\frac{1}{h_{in}} + \frac{r_1}{k} \ln \left(\frac{r_2}{r_1} \right) + \frac{r_1}{r_2} \frac{1}{h_{out}} \right]^{-1} \quad (16)$$

where r_1 and r_2 are inside and outside radius of the inner tube respectively, k is thermal conductivity of copper tube, h_{in} is heat transfer coefficient of the hot fluid condensing through the inner tube and h_{out} is heat transfer coefficient of the cold fluid evaporating through the annulus area. The conduction resistance of the tube wall in Eq. (16) is less compared to convective resistance due to high thermal conductivity and thin copper tube and hence, it is neglected, resulting in Eq. (17).

$$U_{th} = \left[\frac{1}{h_{in}} + \frac{r_1}{r_2} \frac{1}{h_{out}} \right]^{-1} \quad (17)$$

The flow boiling heat transfer coefficient, h_{out} , for the mixed refrigerants in Eq. (17) is calculated using modified Granryd correlation [14] in the present work. The empirical correlations such as Shah [19], Dobson and Chato [20], Cavallini and Zecchin [21] and Cavallini et al. [22] are used for condensation of the mixtures. The Silver–Bell–Ghaly correction is applied to condensation heat transfer coefficients calculated using above pure component correlations to take into account the effect of mixture. The single-phase heat transfer coefficients for both the cold and the hot fluid are predicted by the Dittus–Boelter correlation. The predicted values of the overall heat transfer coefficients, U_{th} , are compared with those obtained from the experimental data.

5. Results and discussion

Experiments are conducted on MR J–T cryocooler to determine overall heat transfer coefficients along the length of the heat exchanger for various mixtures. Three specific compositions of the mixture of gases viz. nitrogen, methane, ethane, propane and iso-butane are used as a refrigerant in the system. Table 2 gives composition of the mixture charged, and of that in circulation, corresponding to each range of refrigeration temperature. These mixtures are designed to produce different refrigeration temperatures

in the MR J–T cryocooler. The heat exchanger essentially operates in two phase region at such low temperatures. The details regarding operating conditions like mass flux, pressures and temperatures for both the fluids in the heat exchanger are given in Table 3. The test conducted on each mixture is repeated at least three times to ensure repeatability of the results obtained.

During experimentation it is observed that, the inlet temperature of the hot fluid in the heat exchanger is greater than its dew point temperature for all the mixtures. Similarly, the cold fluid leaves the heat exchanger at a temperature greater than its dew point. Therefore, at the hot end of the heat exchanger, both the condensing and the evaporating streams are in single phase state for all the cases of mixtures. Hence, the experimental values of the overall heat transfer coefficients at the hot end correspond to the single phase heat transfer between the hot and the cold fluid. Here, the overall heat transfer coefficient for single phase heat transfer is calculated using the heat transfer coefficients based on Dittus–Boelter equation on either side, since, the flow is turbulent for both the streams. Table 4 compares the calculated values of single phase overall heat transfer coefficients with that of the experimental results for three different mixtures. It is noted that the experimentally obtained overall heat transfer coefficients are within 10% of those calculated using the Dittus–Boelter equation. This also verifies the accuracy of heat transfer measurements during experiments. The same methodology is applied for the rest of the heat exchanger length where the flow is two phase.

The flow boiling heat transfer coefficients calculated using modified Granryd correlation are studied for different mixtures. The variation in thermo-physical properties of the cold fluid is considered during evaluation of heat transfer coefficients. Fig. 3 shows the variation in flow boiling heat transfer coefficients against vapour quality, x for various mixtures. The cold fluid enters the heat exchanger in two-phase state ($x > 0$) for all the mixtures. The heat transfer coefficient increases as evaporation takes place due to thinning of the liquid film in the annular space and due to increase in vapour velocity in the core. It can be noticed from Fig. 3 that the heat transfer coefficients suddenly drop at high vapour quality ($x > 0.95$) in the dry-out region due to low thermal conductivity of the vapour.

It is observed from Fig. 3 that the boiling heat transfer coefficients depend on the mixture composition and operating conditions in the heat exchanger. For Mix#1, predicted values of the flow boiling heat transfer coefficients are relatively higher as compared to the other mixtures. The molar percentage of the middle boiling point components such as methane and ethane is significantly more in Mix#1 than that for Mix#2 and Mix#3. Also, the mass flux of the evaporating stream i.e. the cold fluid as shown in Table 3, is greater for Mix#1 than other mixtures. The mass flux is 221 kg/m²s for Mix#1, whereas, it is 153 kg/m²s for Mix#3, leading to a higher Reynolds number and hence higher heat transfer coefficients in the case of Mix#1. Fig. 3 also reveals that heat transfer coefficients are lower for Mix#3 than that for Mix#2 in the region of vapour quality more than 0.5. This is mainly due to lower mass flux for Mix#3. The molar percentage of the higher boiling point components such as propane and iso-butane is nearly same for Mix#2 and Mix#3. Thus, the mass flux and the mixture

Table 2
Mixture specifications.

Mixture	Mixture composition, N ₂ /CH ₄ /C ₂ H ₆ /C ₃ H ₈ /iC ₄ H ₁₀ (% mol)		Temperature range (K)
	Charged	Circulation	
Mix#1	5.5/42.5/36.0/5.0/11.0	6.99/46.335/33.533/3.996/9.146	140–150
Mix#2	36.0/15.0/13.0/19.0/17.0	39.86/16.865/12.845/17.38/13.045	<100
Mix#3	15.5/31.0/16.5/21.0/16.0	18.455/32.785/16.05/20.14/12.57	110–120

Table 3
Experimental conditions.

Mixture	Mass flux, kg/m ² s		Mean pressure, kPa		Cold fluid temperature, K		Hot fluid temperature, K	
	Cold fluid	Hot fluid	Cold fluid	Hot fluid	Inlet	Outlet	Inlet	Outlet
Mix#1	221	208	466	1190	144.9	296.9	303.2	149.3
Mix#2	215	202	411	1435	100.2	293.5	301.5	110.5
Mix#3	153	144	394	1135	114.8	296.2	302.7	116.3

Table 4
Single phase heat transfer coefficients.

Mixture	U_{exp} (W/m ² K)	Single-phase local heat transfer coefficient, h (W/m ² K)		U_{th} (W/m ² K)	% Difference in U
		Inside	Outside		
Mix#1	610.3	1083.5	1386.6	679.6	10.2
Mix#2	486.4	853.2	1108.5	538.2	9.6
Mix#3	412.1	729.7	937.8	458.4	10.1

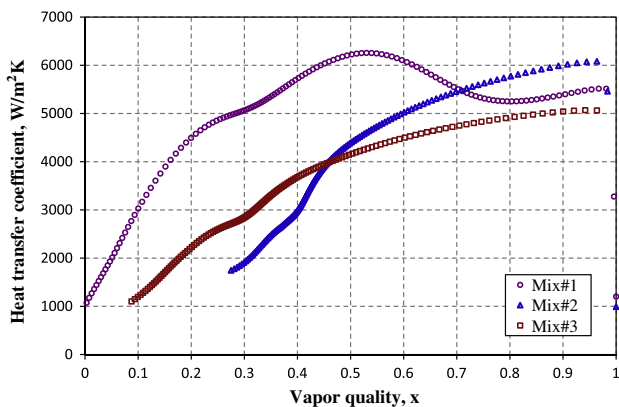


Fig. 3. Predicted flow boiling heat transfer coefficients for Mix#1, Mix#2 and Mix#3.

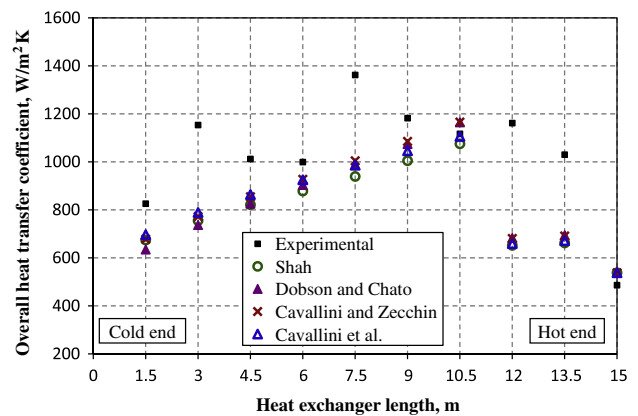


Fig. 5. Local overall heat transfer coefficients along the length of the heat exchanger for Mix#2.

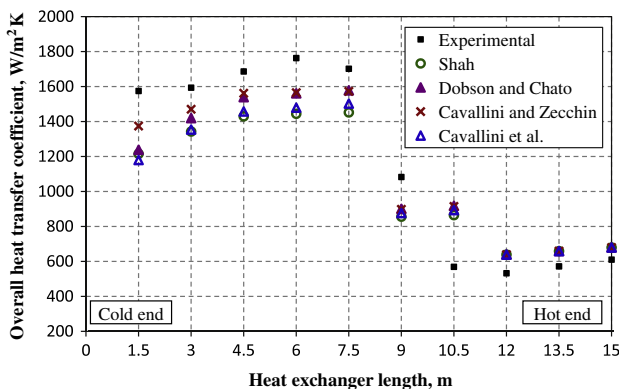


Fig. 4. Local overall heat transfer coefficients along the length of the heat exchanger for Mix#1.

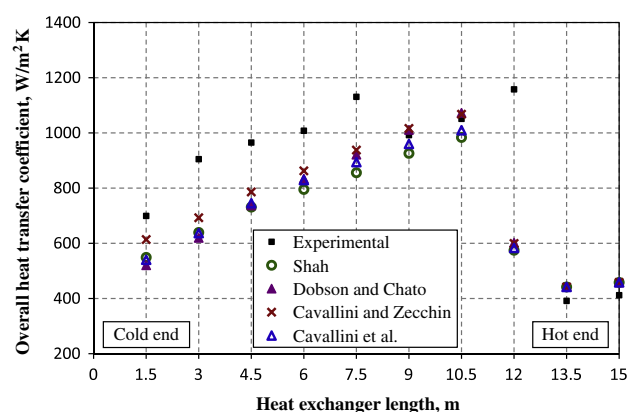


Fig. 6. Local overall heat transfer coefficients along the length of the heat exchanger for Mix#3.

composition have significant influence on the local heat transfer coefficient for the flow boiling.

Figs. 4–6 show the comparison of the theoretical values of the overall heat transfer coefficients obtained using different correlations with those obtained based on the experimental data for Mix#1, Mix#2, and Mix#3, respectively. The predicted variation in overall heat transfer coefficients is quite similar to those obtained experimentally for all the mixtures, while the scatter for experimental values is relatively less for Mix#1. The scatter is significant in the region of phase change, particularly for Mix#2

and Mix#3. It is clear from the figures that the overall heat transfer coefficients predicted using Dobson and Chato correlation and Cavallini and Zecchin correlation are close to the experimental values. On the other hand the overall heat transfer coefficients obtained using Shah and Cavallini et al. correlation are relatively less than the experimental values.

As pointed out earlier, the mixed refrigerant at low temperature and low pressure enters the counter flow heat exchanger in two-phase state ($x > 0$) and leaves the heat exchanger in vapour state

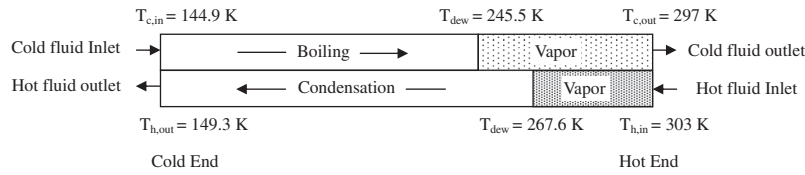


Fig. 7. Phase change process in the heat exchanger for Mix#1.

Table 5
Average values of overall heat transfer coefficients.

Mixture	Average U_{exp} , W/m ² K	Average U_{th} , W/m ² K			
		Shah [19]	Dobson and Chato [20]	Cavallini and Zecchin [21]	Cavallini et al. [22]
Mix#1	1168.65	1058.23	1113.18	1133.50	1071.98
Mix#2	1033.02	800.25	824.27	839.87	828.35
Mix#3	871.34	695.61	721.83	747.72	709.89

Table 6
Deviation in predicted values of overall heat transfer coefficients.

Mixture	% Deviation			
	Shah [19]	Dobson and Chato [20]	Cavallini and Zecchin [21]	Cavallini et al. [22]
Mix#1	9.45	4.75	3.00	8.27
Mix#2	22.53	20.20	18.70	19.81
Mix#3	20.17	17.16	14.19	18.52

Table 7
Temperature glide for mixtures.

Mixture	Hot fluid temperature, K			Cold fluid temperature, K		
	T_{bub}	T_{dew}	ΔT_g	T_{bub}	T_{dew}	ΔT_g
Mix#1	139.29	267.64	128.35	114.62	245.46	130.84
Mix#2	103.61	287.15	183.54	86.74	253.77	167.03
Mix#3	110.97	281.79	170.82	92.42	253.56	161.14

due to its evaporation. As an example, the change of the phases in the cold and the hot fluid is shown in Fig. 7 for Mix#1. The boiling heat transfer coefficient increases with respect to increase in quality and then suddenly drops corresponding to single phase heat transfer coefficient in the vapour region of the heat exchanger at the hot end as shown in Fig. 3. The mixed refrigerant at high temperature and high pressure enters the heat exchanger from right side, as shown in Fig. 7, which undergoes condensation process. It can be noticed from Fig. 7 that at the hot end of the heat exchanger, both the cold and the hot fluid are in vapour state leading to heat transfer from hot vapours to cold vapours. Hence, the overall heat transfer coefficients in this region are lower than those in the two phase region which is towards the cold end of the heat exchanger. The overall heat transfer coefficient is maximum at the section where both the cold and the hot fluids change their phases. Towards the cold end of the heat exchanger, the overall heat transfer coefficient decreases. This is due to lower values of boiling and condensation heat transfer coefficients. As the refrigerant condenses (as the quality decreases), the local condensation heat transfer coefficient decreases gradually along the tube. This is mainly due to the reduced flow speed of refrigerant and the increased liquid film thickness which results in a higher thermal resistance.

Table 5 gives the average values of the overall heat transfer coefficients obtained using various correlations against the experimental data. The percentage deviation in the calculated overall heat transfer coefficients is compared in Table 6 for all the mixtures. The experimental results are in good agreement with

all the correlations tested for the case of Mix#1. It is noted that the deviations in Dobson and Chato correlation and Cavallini and Zecchin correlation are 4.75% and 3% respectively for Mix#1. On the contrary, the deviations for Shah and Cavallini correlation are 9.45% and 8.27%, respectively. The deviation for Cavallini and Zecchin correlation is lowest as compared to other correlations. It is 3%, 18.7% and 14.2% for Mix#1, Mix#2, and Mix#3, respectively. Hence, Cavallini and Zecchin correlation is found to be more suitable for prediction of condensation heat transfer coefficient of the mixed refrigerants used in the cryocooler.

It is also seen from Table 6 that the deviation for Mix#2 is more as compared to Mix#1 and Mix#3. This may be due to the difference in composition of the mixtures. The temperature glide of the mixture depends on its composition. Table 7 gives the bubble point temperature, T_{bub} , the dew point temperature, T_{dew} and temperature glide, ΔT_g of the hot and the cold fluid for all the mixtures. These temperatures are evaluated at mean pressures of the fluids using the software – AspenONE. It is noted that the temperature glide for Mix#2 is more than that for Mix#1 and Mix#3. Mixture effect is more for the mixture having more temperature glide. Therefore, existing empirical correlations are more suitable for the mixed refrigerants for which temperature glide is relatively less (up to 130 K). It would be apt to highlight the fact that the present work may be the first of its kind on the condensation of multi-component mixtures at cryogenic conditions. However, it is emphasised here that the study has been conducted for three mixture compositions only. Hence, before making any definitive conclusions, extensive experimentation is necessary using other mixture compositions.

6. Conclusions

In the heat exchanger for MR J–T cryocooler, the multi-component mixture of nitrogen–hydrocarbons evaporates and condenses simultaneously at different pressures along the length, thereby improving the performance of the cryocooler. However, at present, the design of such heat exchanger is crucial due to lack of experimental heat transfer data and generalized empirical correlation for

boiling and condensation of the mixed refrigerants. There are many empirical correlations for flow boiling and condensation heat transfer which are either developed for pure components or binary mixtures. However, these correlations have not been tested for multi-component mixtures and for operating conditions of the cryocooler.

In the present work, experiments are carried out to measure the overall heat transfer coefficients for various mixture compositions. These experimental results are compared with the calculated overall heat transfer coefficients using several condensation correlations. A comparison of all the data is carried out to assess the suitability of these correlations to estimate the heat transfer coefficients. Following conclusions are drawn from the present study.

1. The existing pure component condensation correlations are applied to mixtures of nitrogen–hydrocarbons at cryogenic temperatures using the Silver–Bell–Ghaly correction to take into account the effect of mixture.
2. The predicted overall heat transfer coefficients, obtained using Dobson and Chato, Cavallini and Zecchin, and Cavallini et al. correlations for condensation heat transfer and modified Granryd correlation for flow boiling, are in good agreement with the experimental data for all the tested mixtures.
3. The percentage deviation in the calculated overall heat transfer coefficients using Cavallini and Zecchin correlation for condensation heat transfer are least, 3% for Mix#1, 18.7% for Mix#2 and 14.2% for Mix#3, only. Hence, use of this correlation is recommended for the calculation of condensation heat transfer coefficients of the mixed refrigerants at cryogenic temperatures.
4. For Mix#1, all the tested correlations predict overall heat transfer coefficients within 10% deviation from the experimental values. The temperature glide for Mix#1 is relatively less. Therefore, these condensation correlations along with the modified Granryd correlation for flow boiling can be used more accurately for the mixtures having low temperature glide.

References

- [1] Brodyanskii VM, Gromov EA, Grezin AK, Yagodin VM, Nikolaskii VA, Tashchina AG. Efficient throttling cryogenic refrigerators which operate on mixtures. *Chem Petroleum Eng* 1971;7(12):1057–61.
- [2] Longworth RC, Boiarski MJ, Klusmier LA. 80 K closed cycle throttle refrigerator. *Cryocoolers* 1995;8:537–41.
- [3] Walimbe NS, Narayankhedkar KG, Atrey MD. Experimental investigation on mixed refrigerant Joule–Thomson cryocooler with flammable and non-flammable refrigerant mixtures. *Cryogenics* 2010;50(10):653–9.
- [4] Alexeev A, Haberstroh Ch, Quack H. Further development of a mixed gas Joule Thomson refrigerator. *Adv Cryogenic Eng* 1998;43:1667–74.
- [5] Keppler F, Nellis G, Klein SA. Optimization of the composition of a gas mixture in a Joule–Thomson cycle. *HVAC&R Res* 2004;10(2):213–30.
- [6] Gong MQ, Luo EC, Zhou Y, Liang JT, Zhang L. Optimum composition calculation for multicomponent cryogenic mixture used in Joule–Thomson refrigerators. *Adv Cryogenic Eng* 2000;45:283–90.
- [7] Gong MQ, Wu JF, Luo EC, Qi YF, Hu QG, Zhou Y. Study on the overall heat transfer coefficient for the tube-in-tube heat exchanger used in mixed-gases coolers. *Adv Cryogenic Eng* 2002;47B:1483–90.
- [8] Ardhapurkar PM, Sridharan A, Atrey MD. Investigations on two-phase heat exchanger for mixed refrigerant Joule–Thomson cryocooler. *Adv Cryogenic Eng* 2012;57A:706–13.
- [9] Alexeev A, Thiel A, Haberstroh Ch, Quack H. Study of behavior in the heat exchanger of a mixed gas Joule–Thomson cooler. *Adv Cryogenic Eng* 2000;45:307–14.
- [10] Nellis G, Hughes C, Pfothner J. Heat transfer coefficient measurements for mixed gas working fluids at cryogenic temperatures. *Cryogenics* 2005;45(8):546–56.
- [11] Celata GP, Cumo M, Setaro T. A review of pool and forced convective boiling of binary mixtures. *Exp Thermal Fluid Sci* 1994;9:367–81.
- [12] Cheng L, Mewes D. Review of two-phase flow and flow boiling of mixtures in small and mini channels. *Int J Multiphase Flow* 2006;32:183–207.
- [13] Cavallini A, Censi G, Del Col D, Doretti L, Longo GA, Rossetto L, et al. Condensation inside and outside smooth and enhanced tubes – a review of recent research. *Int J Refrig* 2003;26:373–92.
- [14] Ardhapurkar PM, Sridharan A, Atrey MD. Flow boiling heat transfer coefficients at cryogenic temperatures for multi-component refrigerant mixtures of nitrogen–hydrocarbons. *Cryogenics* 2014;59:84–92.
- [15] Gungor KE, Winterton RHS. Simplified general correlation for saturated flow boiling and comparisons with data. *Chem Eng Res Des* 1987;65:148–56.
- [16] Silver L. Gas cooling with aqueous condensation. *Trans Inst Chem Eng* 1947;25:30–42.
- [17] Bell KJ, Ghaly MA. An approximate generalized design method for multicomponent/partial condenser. *AIChE Symp Ser* 1973;69:72–9.
- [18] Granryd E. Heat transfer in flow evaporation of non-azeotropic refrigerant mixtures – a theoretical approach. In: *Proc. 18th int congress of refrigeration*, vol. 3. Montreal; 1991. p. 1330–4.
- [19] Shah MM. A general correlation for heat transfer during film condensation inside pipes. *Int J Heat & Mass Transfer* 1979;22:547–56.
- [20] Dobson MK, Chato JC. Condensation in smooth horizontal tubes. *J Heat Transfer, Trans ASME* 1998;120:193–213.
- [21] Cavallini A, Zecchin R. A dimensionless correlation for heat transfer in forced convection condensation. In: *Proceedings of the 6th int heat transfer conference*, vol. 3. Tokyo; 1974. p. 309–13.
- [22] Cavallini A, Col DD, Doretti L, Matkovic M, Rossetto L, Zilio C, et al. Condensation in horizontal smooth tubes: a new heat transfer model for heat exchanger design. *Heat Transfer Eng* 2006;27(8):31–8.
- [23] Col DD, Cavallini A, Thome JR. Condensation of zeotropic mixtures in horizontal tubes: new simplified heat transfer model based on flow regimes. *J Heat Transfer* 2005;127:221–30.
- [24] Shah MM. An improved and extended general correlation for heat transfer during condensation in plain tubes. *HVAC&R Res* 2009;15(5):889–913.
- [25] Ardhapurkar PM, Sridharan A, Atrey MD. Experimental investigation on temperature profile and pressure drop in two-phase heat exchanger for mixed refrigerant Joule–Thomson cryocooler. *Appl Thermal Eng* 2014;66:94–103.
- [26] AspenONE, V 7.1, Aspen Technology Inc, Burlington, MA 01803, USA; 2009.
- [27] Peng D, Robinson DB. A new two-constant equation of state. *Ind Eng Chem Fundam* 1976;15(1):59–64.