Experimental investigation of low-pressure refrigerant mixtures for micro cryogenic coolers ☆

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ABSTRACT

Micro Cryogenic Coolers (MCCs) can achieve very small sizes and high efficiencies when operating with a refrigerant mixture, but micro-scale compressors have a limited pressure output. Four refrigerant mixtures were designed to operate between 0.4 MPa and 0.1 MPa, and tested in a MCC system both with and without pre-cooling. For comparison, two pure refrigerants were tested as well. Without pre-cooling, each mixture exhibited considerably lower cooling power than the design value. With pre-cooling, the mixtures exhibited unsteady cooling temperatures accompanied by flow pulsations after a period of time. The low cooling power, unsteady temperatures, and time required for the pulsations to occur are analyzed in terms of composition change due to liquid hold-up in the annular and intermittent flow regimes.

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1. Introduction

MCCs are gaining attention because their small volume, low noise, fast response, and portability make them attractive alternatives to conventional cryocoolers [1]. In the field of MCCs, those operating with a Joule–Thomson (J–T) cycle have the potential to be 10× smaller than Stirling MCCs, and 10× more efficient than thermoelectric coolers [2]. Refrigerant mixtures enable both benefits. Mixtures are designed to have a large glide between the bubble point and the dew point, such that the refrigerant in the heat exchanger of the J–T cooler is 2-phase flow. The 2-phase flow enhances heat transfer between the high-pressure and low-pressure streams, allowing the size of heat exchangers to be reduced. Mixtures are also designed to have a high specific cooling power, resulting in high cooling system efficiency [3]. Previous refrigerant mixtures typically use driving pressures of around 2 MPa [4–6], with some as low as 1.6 MPa [7]. However, to achieve a micro-system, one must use a micro-compressor, which may be limited to 0.4 MPa. Previous work from our group has shown that mixtures driven by 0.6 MPa can be effective at cooling to 177 K [8].

A number of methods have been proposed for choosing the components of a refrigerant mixture. These methods have been reviewed in Venkatarathnam's monograph on cryogenic mixed refrigerant processes [9]. The original patents of Alfeev et al. [10]

and Bioarski et al. [11] chose compositions based on experimental observations. In a patented by Little [12], refrigerants compositions are chosen to ensure the constant-pressure lines corresponding to the high- and low-side pressures of the cooler be parallel on a temperature-enthalpy diagram. Radebaugh showed that the specific cooling power of a refrigerant mixture in a J–T cooler with an ideal heat exchanger is given by the isothermal enthalpy difference between the high pressure and low pressure streams $(\Delta h|_T)_{\min}$ which represents a function that can be optimized by varying the component compositions [13]. Another method, patented by Alexeev and Quack [14], finds compositions that maximize the specific refrigeration effect $(\Delta h_{\min})$, in a manner similar to Radebaugh [13]. A third method, reported by Gong et al. [15], maximizes the Carnot efficiency (η) of the refrigerator, which is given as:

$$\eta = \frac{\Delta h_{\min}}{\text{specific compressor work}}\left(\frac{T_0}{T} - 1\right) \tag{1}$$

This method is conceptually similar to optimizing the coefficient of performance (COP), as discussed in [13]. Mixtures designed to maximize the specific cooling effect will have some components entering the warm end of the heat exchanger in a liquid phase. Boiarski referred to these as liquid refrigerant supply (LRS) systems, and differentiated them from gas refrigerant supply (GRS) systems in which all components of the mixture are in a single gas phase as they enter the cryostat [16]. GRS systems are simpler to build, but have lower specific cooling power.

One issue that arises when using refrigerant mixtures is the consistency of the composition. It has been shown that the composition of the mixture changes at different locations within the

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cryocooler, with the cold-end favoring lighter (low boiling-point) components [17]. When the refrigerant is in 2-phase flow, the liquid phase is a different composition than the vapor phase. Depending on the 2-phase flow regime, this can separate the components in a cooler. In microchannel 2-phase flow, the regimes most often encountered are annular and intermittent [18]. These can prove problematic: in annular flow the high liquid hold-up causes drastic composition changes, and in intermittent flow the varying flow-rates can cause oscillations in temperatures. We face a dilemma between the desire of reaching high cooling capacity at low pressure, on the one hand, and the desire for compactness (MEMS construction). The compactness restricts the thermodynamic feasibility to reach high cooling capacity at low compression ratios, due to the fluid flow regimes encountered at that size scale.

We designed four hydrocarbon mixtures to operate between a high pressure of 0.4 MPa and a low pressure of 0.1 MPa, designed for the temperature range 140–300 K (Mix 1), 140–275 K (Mix 2), 160–300 K (Mix 3), and 200–300 K (Mix 4). Each of the mixtures is of the LRS type. These mixtures were used in a MCC system with a fiber-base cooler and a miniature compressor, described in [8]. For comparison, the system is also run with two pure hydrocarbons: propane and isobutane. The pure hydrocarbons perform as expected. The cold-tip temperature of the isobutane went as the boiling point of isobutane at the low-side pressures. The low temperature of the cold-tip with propane was limited to 242 K, which is the temperature that balances the small cooling power of the vapor with the background heat loads of the MCC. When the propane lines were cooled to 275 K (below propane's boiling point at 0.6 MPa of 281.1 K), the MCC temperature dropped to the temperature associated with the low-pressure boiling point of propane. In this situation, as with that of isobutane, the MCC acts as one would expect for a vapor-compression refrigerator.

However, the mixtures do not cool according to their design. Without pre-cooling, the MCC running the mixed refrigerant experiences steady flow and cooling powers that are much lower than the designed values. With pre-cooling, after a period of time, the flow experiences pulsations. The low cooling power and time required for the transition to pulsating flow are analyzed in terms of composition change due to liquid hold-up in annular and intermittent flow regimes, as discussed in Section 5 of this article.

2. Materials and methods

This study considered two pure hydrocarbon refrigerants (propane and isobutane) and four custom mixtures. The single-component hydrocarbons had a purity rated by the vendor to be >99.9%. The compositions of the four mixtures are given in Table 1. These compositions were chosen to maximize $(\Delta h|T)_{min}$ over the temperature range of interest, and the optimization was carried out by the software NIST4 [19]. The refrigerants were provided by a commercial gas supplier and analyzed for composition, with actual composition listed in Table 1.

Fig. 1. (a) The MCC test loop, which includes measurements of low-side pressure (P_L), high-side pressure (P_H), flowrate ($\dot{V}$), MCC cold-end temperature (T_c), heat applied to the MCC cold-end (Q_c), and base temperature (T_b). (b) Photograph of the MCC, and (c) cross-sectional drawing of the J–T valve, showing path of high-pressure refrigerant (solid line) as it expands (dashed line) to its low-pressure state (dotted line).

The test setup is shown in Fig. 1. This test method has been described elsewhere [8], but is briefly repeated here. The key components are a miniature oil-free compressor and micro cryogenic cooler. The compressor is formed by coupling a commercial piston oscillator with a pair of custom micro-fabricated check-valves. The MCC consists of a warm-end micro-coupler, a tubes-in-tube counter-flow heat exchanger (CFHX) formed by hollow-core glass fibers, and a microfabricated J–T valve with a restriction of 1.7 μm, as described in [7]. One gram of 0.3 nm molecular sieve is included between the compressor and the MCC to remove any trace moisture contaminant and thereby reduce the likelihood of icing. Refrigerant is compressed to high pressure by the miniature compressor, passes through the molecular sieve and a 7 μm particulates filter, and into the MCC, where it expands to low pressure, and through a mass flowmeter. With the mixtures, the low-side pressure was kept open to a supply tank regulated at 0.100 MPa. The high-side pressure is determined by the performance of the compressor, which is a function of the flow-rate and the compressibility of the refrigerant in question, but it is typically between 0.4 and 0.7 MPa.

With the pure hydrocarbons, the system was fed with an initial charge, and all of the valves were closed during operation. With the isobutane, the system was fed with refrigerant regulated at 0.35 MPa into the high side while the low side was still under vacuum. The propane was fed through the high-pressure side, with an initial equilibrium pressure of 0.47 MPa, which was sufficient to develop an operating pressure of 0.63 MPa on the high side with a pressure of 0.10 MPa on the low side.

Table 1
The design composition and manufactured composition of each of the four refrigerant mixtures studied.

	Design composition (mol fractions)	Actual composition (mol fractions)		Design composition (mol fractions)	Actual composition (mol fractions)
Mix1	34% Methane	34.2% Methane	Mix 3	24% Methane	24.1% Methane
	20% Ethane	19.8% Ethane		36% Ethylene	36.0% Ethylene
	18% Ethylene	15.2% Ethylene		14% Propane	14.0% Propane
	16% Isobutane	14.9% Isobutane		10% Isopentane	10.0% Isopentane
	12% Isohexane	15.9% Isohexane		16% Pentane	15.9% Pentane
Mix 2	34% Methane	33.9% Methane	Mix4	8% Methane	7.93% Methane
	22% Ethane	22.0% Ethane		46% Ethane	45.88% Ethane
	20% Ethylene	20.0% Ethylene		14% Propane	14.0% Propane
	12% Isobutane	12.0% Isobutane		4% Butane	3.99% Butane
	12% Isopentane	12.1% Isopentane		28% Pentane	28.2% Pentane

Fig. 2. Isothermal enthalpies as a function of temperature for: (a) the pure hydrocarbons, and (b) the mixtures, calculated with REFPROP with high pressures of 0.4 MPa and low pressures of 0.1 MPa. The designed operating temperature of each mixture is shown as the bolded region of the line.

Before charging the MCC with refrigerant, all lines were evacuated to <0.01 Pa (<10^{-4} mbar). The MCC was held in a vacuum during the tests. Temperature at the base and cold-tip of the MCC were monitored with platinum resistance thermometers (PRTs). The PRT on the cold-tip of the MCC could also function as a small heater by applying a small voltage; in this manner the heat applied to the MCC was also monitored. Flow-rate, high pressure, and low pressure were monitored as well. Although all four refrigerant mixtures were designed to cool from 300 K, it was found that pre-cooling was necessary to achieve low temperatures. To facilitate pre-cooling, the MCC and 20 cm of coupling tubing were held in an ice bath at 273 K. Heat loads experienced by the MCC from the environment were determined as a function of temperature by a background heat leak test.
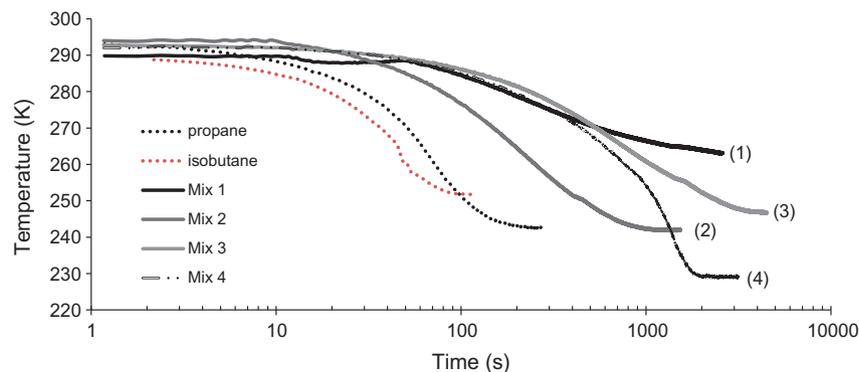
3. Theory and calculation

The gross cooling power of a Joule–Thomson cryocooler is given by the product of the minimum isothermal enthalpy change in the refrigerant between the high-pressure and low-pressure streams, with the molar flow rate: $\dot{n}(\Delta h|T)_{min}$ [13]. Here, the minimum is to be taken over the temperature range experienced by the heat exchanger. The $\Delta h|_T$ curves for the four mixtures and two pure hydrocarbons are shown in Fig. 2, with values calculated by the NIST standard reference software REFPROP [20]. For pure refrigerants, the minimum enthalpy difference will always occur at the inlet to the warm-end of the heat exchanger, whereas with mixtures it can occur at one or more pinch points.

4. Results

4.1. Cooling from ambient

The cooling profile for the six refrigerants is shown in Fig. 3. With isobutane, the minimum temperature was governed by the boiling point of isobutane at the low pressure. In each of the mixtures, and with propane, the minimum temperature was determined by balancing the cooling power with the parasitic heat loads on the MCC, including heat conducted through the heat exchanger, through the DC leads on the temperature sensor, radiation from the vacuum package, and conduction through the rarified gas in the vacuum. Such heat loads were characterized as a function of temperature by performing a background heat leak test, described in [21]. The minimum temperatures, corresponding heat loads, and cooling power are all listed in Table 2. These specific cooling powers of several hundred J/mol are values typically found in J–T cooling in which the refrigerant enters the heat exchanger as a single-phase vapor. These gas-refrigerant systems typically have $(\Delta h|T)_{min}$ values that are an order of magnitude lower than those of liquid-refrigerant systems, in which the refrigerant entering the heat exchanger contains liquid components. The test running pure propane is expected to be a gas-refrigerant system when cooling from ambient temperature, but the mixtures are designed to have some liquid at ambient temperature under high pressure.

4.2. Effect of pre-cooling

When the system operating with propane was pre-cooled with an ice bath, the refrigerant going into the system was in the liquid phase. As it subsequently evaporates across the J–T valve, its low temperature is limited by the boiling point at the low pressure, shown in Fig. 4.

Fig. 3. Cool-down curves for the six refrigerants in question, plotted as a semi-log plot. Note that the pure refrigerants cool quickest, but Mix 4 reaches the lowest temperature.

Table 2
Minimum temperature, heat load required to keep the MCC at ambient temperature, and associated refrigerant enthalpy change, for propane and the four mixtures. The enthalpy change calculation is performed with REFPROP between 0.6 MPa and 0.1 MPa.

	T_{min} (K)	$\dot{Q}$ (mW) at 295 K	Measured Δh at 295 K (J/mol)	Calculated Δh at 295 K (J/mol)
Isobutane	251.7	>200	>10.000	19.800
Propane	242.2	15.8	603	723
Mix1	261.7	4.67	378.6	4.845
Mix 2	241.7	6.63	328.3	1.932
Mix 3	245.6	5.53	329.7	5.222
Mix 4	228.4	7.89	415.8	7.574
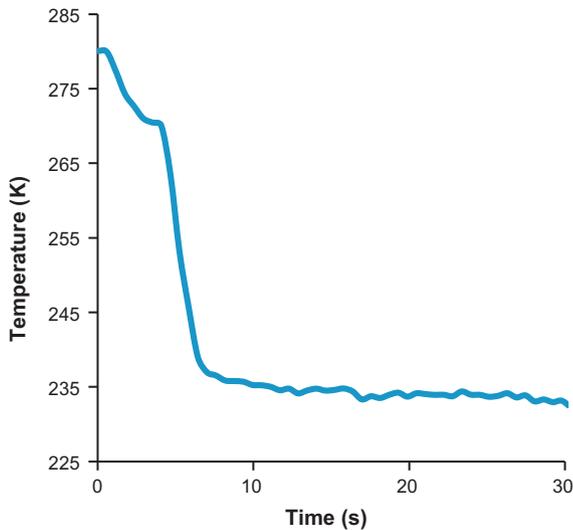
Fig. 4. Cool-down curve of propane with ice pre-cooling.

With the refrigerant mixtures, the cool-down curves are shown in Fig. 5. The test of each mixture was conducted with the same refrigerant charge used for the non-pre-cooled tests. With each mixture, the system experiences steady flow rates for a period of time, during which the MCC reaches a steady temperature. Eventually, the system experiences pulses in the flow-rate, corresponding to departures from the steady temperature. In the case of Mixtures 1 and 4, the pulses correspond to increased cooling power, and the temperature decreases. But for Mixtures 2 and 3, the overall cooling power does not increase, and the average temperature remains constant or increases.
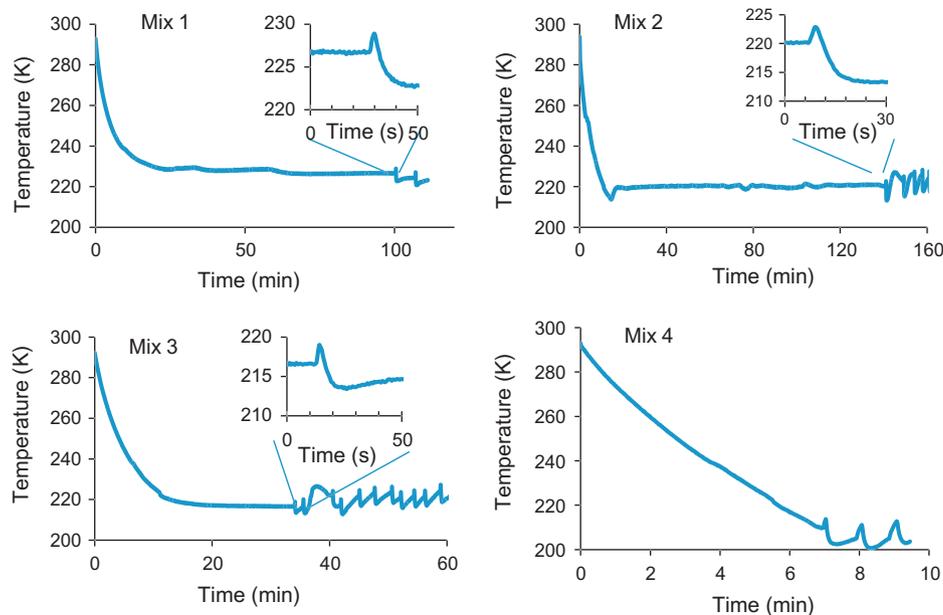
After the flow transitioned to pulsing flow, heat was applied to the PRT to lift the temperature up to the base temperature of 273 K. Measurement of the average heat applied and the flow-rate yields the average isothermal enthalpy difference of the mixture at 273 K, which is listed in Table 3.

5. Discussion

5.1. Low cooling powers from ambient

The performance of these refrigerant mixtures can be considered in terms of composition change due to liquid holdup. At a temperature of 295 K and pressure of 0.6 MPa, each refrigerant mix will have some fraction of the mixture in the vapor phase. The liquid will form first along the side-walls of the coupling channels. A previous paper focused on Mixture 4 has shown that the ratio of vapor flow to that of liquid flow to be given by [22]:

$$\frac{\dot{n}_l}{\dot{n}_v} = \frac{\rho_l}{\rho_v} \frac{-1 - (\frac{r_i}{R})^4 + 2(\frac{r_i}{R})^2}{-2(\frac{r_i}{R})^4 - (\frac{r_i}{R})^4 \frac{\mu_l}{\mu_v} + (\frac{r_i}{R})^2}, \tag{2}$$

where ρ represents molar density, μ dynamic viscosity, R the channel radius and r_i the radius to the liquid/vapor interface. In a cylindrical geometry, r_i/R can be calculated from the vapor quality of the mixture (X) and the liquid and vapor densities as:

$$\frac{r_i}{R} = \sqrt{\frac{X/\rho_v}{X/\rho_v + (1-X)/\rho_l}} \tag{3}$$

Values of $\dot{n}_l/\dot{n}_v$ are listed in Table 4. Density, viscosity, and quality values are computed from REFPROP. Note that under such conditions, liquid flow rates contribute a maximum of 0.03% to the total molar flow rate, and thus liquid components will not contrib-

Fig. 5. Cool-down curves of the 4 mixtures with ice pre-cooling. Inserts for Mix 1, 2, and 3 show a zoomed-in profile of the first departure from steady-state. Note that there are different time scales for each plot.

Table 3
Steady-state temperature (before pulses), average temperature after pulses, isothermal enthalpy difference at 273 K measured with pulses, calculated enthalpy differences, measured time between cooling and pulses, and a calculation of that time based on a model given in Section 5.2.

	T before pulses (K)	T after pulses (K)	Measured Δh at 273 K (J/mol)	Calculated Δh at 273 K (J/mol)	Time to pulses (min)	Calculated time to pulses (min)
Mix1	226.8	222.6	384.1	3751	100.4	54
Mix 2	220.3	222.3	545.4	4737	140.5	17.3
Mix 3	216.9	220.2	747.1	7660	34.1	29.4
Mix 4	209.9	193.6	1135.8	8412	7.1	7.25

Table 4
Calculated enthalpy change of the vapor-phase of each refrigerant at 295 K between 0.6 and 0.1 MPa, ratio of liquid phase molar flow-rate to vapor phase molar flow-rate, and composition of the vapor-phase of each mixture at 295 K and 0.6 MPa, as calculated by REFPROP.

	Compositions of vapor components			Compositions vapor components
Mix1				
Δh = 240 J/mol	40.9% Methane	Mix 3	Δh = 351 J/mol	30.0% Methane
	23.3% Ethane			43.4% Ethylene
	18.0% Ethylene			15.3% Propane
$\dot{n}_L/\dot{n}_v = 6.05 \times 10^{-5}$	13.8% Isobutane		$\dot{n}_L/\dot{n}_v = 8.65 \times 10^{-5}$	4.87% Isopentane
	3.89% Isohexane			6.41% Pentane
Mix 2				
Δh = 329 J/mol	34.1% Methane	Mix 4	Δh = 421 J/mol	11.4% Methane
	22.1% Ethane			61.2% Ethane
	20.1% Ethylene			15.8% Propane
$\dot{n}_L/\dot{n}_v = 6.47 \times 10^{-8}$	11.9% Isobutane		$\dot{n}_L/\dot{n}_v = 27.1 \times 10^{-5}$	2.75% Butane
	11.6% Isopentane			8.75% Pentane

ute significantly to the flow. With such a liquid holdup value, the mixture would be enriched in the lower-boiling-point components. The compositions of the vapor phase of each mixture are listed in Table 4. These altered refrigerant mixtures will have their minimum isothermal enthalpy difference occur at the warm end, and the $\Delta h|_T$ values at 295 K are calculated and included in Table 4, showing reasonable agreement with the measured values listed in Table 2.

To verify that the refrigerants' compositions are actually changing, refrigerant samples were collected and analyzed using gas chromatography/thermal conductivity detection (GC/TCD) by a local commercial natural gas analysis lab. To collect refrigerant samples, the test setup was modified, as shown in Fig. 6a. The technique used is similar to that published previously [19], but it differs due to the addition of the collection reservoir. Refrigerant starts at the 160 L reservoir, regulated at 0.10 MPa, gets compressed by the miniature compressor, passes through the molecular sieve and filter, and passes through the MCC before it is collected. During an initial period, valve 2 is closed and valve 3 is open, and the refrigerant is collected in a collection reservoir. The use of the collection reservoir ensures that no composition change occurs upstream of the compressor. Once the system reaches an equilibrium high pressure, valve 2 is closed and valve 3 opened, and refrigerant is collected in an evacuated 300 mL sample bottle. By controlling speed at which valves 2 and 3 are opened, we keep the low-side pressure at 0.1 MPa. The MCC temperature was kept at 295 K during these tests. For mixtures 1 through 4, the high pressures generated by the compressor during these tests were 0.50, 0.60, 0.41, and 0.38 MPa, respectively. The compositions measured are shown in Figs. 6 b–d, as well as compositions of vapor components of each mixture calculated at the high pressures, at a temperature 295 K. Note that there is excellent agreement between the measured and calculated compositions for Mixture 4, and reasonably good agreement with the other mixtures, especially among the lightest and heaviest components.

5.2. Time to pulsating flows

When pre-cooling the coupling channels, more of the heavy components of the mixtures will condense and be held up in the

form of a liquid film along the channel sidewalls. In this analysis, the majority of the liquid is not flowing out of the pre-cooled coupling tubes, yet new refrigerant is constantly being fed into this section and partially condensing. As a result, the liquid film will increase in thickness with time. Once the liquid film reaches a critical thickness, slugs can form in the pre-cooled channels, which would cause pulses in the flow-rate. According to an analysis by Coleman and Garimella, slugs will form in a cylindrical mini-channel if the vapor quality is given by [23]:

$$X_c < \frac{a}{b+G} \quad \begin{aligned} a &= 69.5673 + 22.595 \exp(0.2586 d_H) \\ b &= 59.9899 + 176.8137 \exp(0.3826 d_H) \end{aligned} \tag{4}$$

where G is flowrate in kg/(m² s) and d_H is hydraulic diameter in mm. In 1/8″ Cu tubing with a flowrate of 30 standard cubic cm per minute (sccm), the critical vapor fraction is 0.39. Based on this analysis, one can calculate the time required for liquid slugs to first appear as

$$T = \frac{\rho_l A}{\dot{n}(1-X)} \sqrt{\frac{X_c \rho_l}{X_c(\rho_l - \rho_v) + \rho_l}} \tag{5}$$

where A is the surface area experiencing cooling, X the molar vapor quality of the fluid entering the pre-cooled channels, and X_c is 0.39, as calculated previously. Results are listed in Table 3. Note that the calculated times agree well for Mixtures 3 and 4, and show agreement to within a factor of 2 for Mixture 1, but are off by nearly an order of magnitude for Mixture 2. The difference between Mixture 2 and the other three mixtures is that Mixture 2 was designed to have no liquid in 0.4 MPa channels at ambient temperature, whereas the other three mixtures were designed to have up to 22% of the refrigerant liquefy in the high-pressure ambient temperature channels.

One assumption of this analysis is that the refrigerant at the suction port of the compressor has the original composition. This is not necessarily the case. Although the low-pressure lines are open to a large refrigerant reservoir, they are also open to the returning channel, which should have a different composition. This would result in the refrigerant at the suction port of the compressor having a lower concentration of the heavy components. As a result, less of the refrigerant fed into the high-pressure lines would
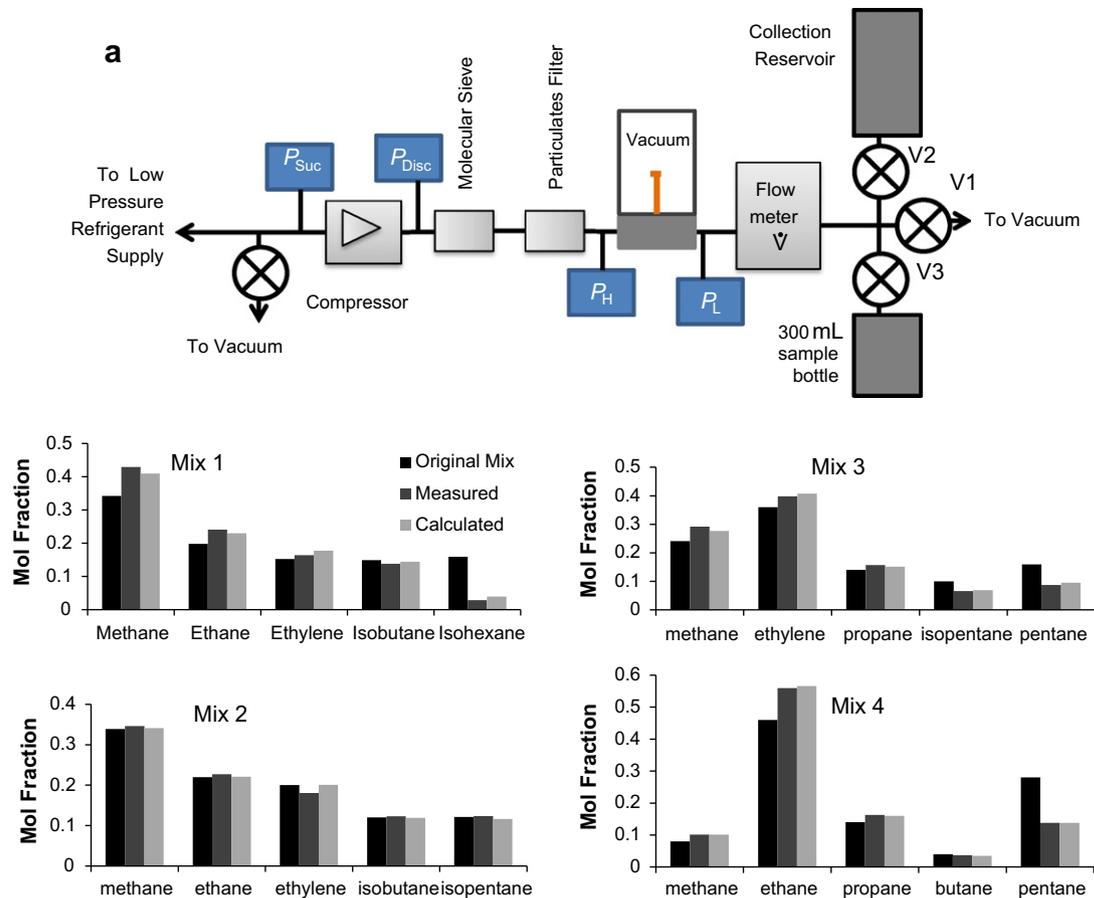
Fig. 6. (a) modified test set-up to measure the composition of the mixture that passes through the MCC, (b–d) measured compositions of 4 mixtures, as determined by GC/TCD.

condense along the walls, and the time required for the liquid film to reach its critical thickness would be longer.

5.3. Temperature fluctuations

Note from Fig. 5 that when a pulsation first occurs, the MCC temperature increases dramatically. When the heat load on the MCC changes, the instantaneous temperature response will be determined by the thermal mass of the MCC, which should not change with temperature, in the range we are considering. The temperature rise rate is plotted as a function of heat applied in Fig. 7a. With Mixture 1, the temperature rise rate is 1.74 K/s during a pulse, which corresponds to a heat load of 35.2 mW. This is significantly higher than the background heat load of 7.36 mW at 226.7 K, as determined by a heat-leak test (Fig. 7b) [21].

Consider a slug of liquid formed in the high-pressure pre-cooled coupling channel. Such a liquid slug will have a volume of roughly 4 μL, which is greater than the total volume of the fibers in the MCC (1 μL). The 4 μL liquid slug will completely fill the high-pressure micro-channels of the MCC with liquid before any vapor enters the flow. While the liquid is entering the high-pressure channels of the CFHX, vapor is passing through the low-pressure channels. The molar flow-rate of the liquid is similar to the vapor, but the liquid has a considerably higher specific heat than that of the vapor. As a result, when the liquid reaches the J–T valve, it has not been cooled to the temperature of the vapor, and the temperature at the MCC cold tip consequently increases rapidly.

The heat load experienced by the MCC under such conditions can be calculated from the enthalpy of the high-pressure liquid and low-pressure vapor at the warm and cold temperatures. Assuming a perfect heat exchanger, the heat applied will be given by:

$$\dot{Q} = \dot{n}[(h_{liq}^{warm} - h_{vap}^{warm}) - (h_{cold}^{liq} - h_{vap}^{cold})] \qquad (6)$$

For Mixture 1 with a flow-rate of 17.5 sccm, this equation yields a heat load of 57 mW, which corresponds to a temperature rise rate of 2.06 K/s. This is within a factor of 2 of the observed temperature rise rate, and it is likely that the instrumentation integration period of 0.5 s would smooth out the sharp temperature rise that occurs instantaneously once the liquid reaches the J–T valve, so agreement to within a factor of 2 is reasonable.

Once the liquid travels through the J–T valve, it has a higher cooling power, so the temperature will subsequently drop. This behavior of liquid flow has been observed in the microchannels of a similar MCC fabricated without radiation shielding [24].

In order to eliminate the flow pulsations and corresponding temperature fluctuations, one would want the refrigerant to condense first in the microchannels of the MCC. This can be accomplished by either eliminating the coupling tubing of the MCC and extending the high-pressure microchannels such that they can act as an after-cooler, or by actively pre-cooling the microchannels.

6. Conclusions

The following conclusions can be drawn from this article:

· Four low-pressure refrigerant mixtures were designed and tested in a MCC system, each exhibiting lower-than-expected cooling powers without pre-cooling, and temperature fluctuations accompanied by pulsating flow with pre-cooling.

(a) Characterize Thermal Mass

-7.36mW

(b) Background Heat Leak
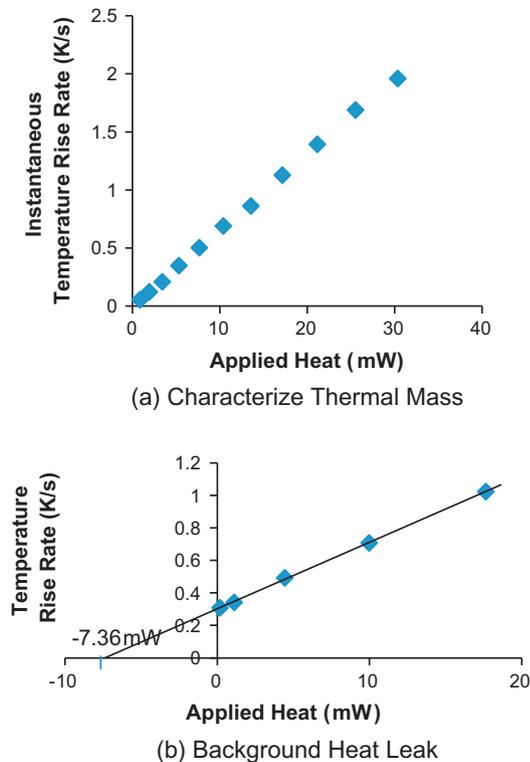
Fig. 7. Relationship between applied heat and temperature increase rate. Data in (a) are for the instantaneous temperature response to heat applied after the cooler has reached steady state. Data in (b) correspond to a background heat leak test at 226 K. For such a test, the compressor is turned off and the flowrate brought to zero, and the temperature rise rate at 226 K is measured under different applied heat loads. Note that if one were to extrapolate the relation back to a temperature rise rate of 0, the heat absorbed is 7.36 mW, corresponding to the background heat load which must be absorbed by the cooler in order to stay at a steady temperature at 226 K.

· Two pure hydrocarbon refrigerants were tested for comparison, with cooling results expected from a vapor compression refrigeration cycle when the refrigerant entered the MCC as a liquid.
· The lower-than-expected cooling powers of the four mixtures are analyzed in terms of a composition change due to the liquid-phase components being held-up in the coupling line between the compressor and cooler.
· The pulsating flow is analyzed in terms of the time required before held-up liquid reaches a critical thickness and forms a liquid slug in the coupling line. The slug of liquid can cause the observed temperature fluctuations.

Acknowledgements

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