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Micro Cryogenic Coolers for IR Imaging*

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ABSTRACT

Joule-Thomson micro cryogenic coolers (MCCs) are a preferred approach for small and low power cryocoolers. With the same heat lift, MCC's power input can be only 1/10 of a thermoelectric cooler's input, and MCC's size can be only 1/10 of a Stirling cooler's size. With futuristic planar MCC and with high frequency MEMS compressors to be developed, its size can be reduced another order of magnitude. Such "invisible" cryocoolers may revolutionize future IR imaging systems. We will review our studies on the feasibility of MCC with an emphasis on: 1) high thermal isolation levels reaching 89,000 K/W; 2) custom-designed gas mixtures with refrigeration capabilities increased by 10X and pressure ratio reduced to only 4:1; 3) compressors with low pressure ratios; and 4) excellent scalability for further size reduction.

Keywords: Cryocooler, Cryogenic, Joule-Thomson, Packaging, MEMS, Mixed refrigerants, IR Imaging, Thermoelectric, Stirling, Cooler.

1. INTRODUCTION

Due to their small volume and low power, micro cryogenic coolers (MCCs) have a great potential to create a paradigm shift in the application of sensors operating at cryogenic temperatures. MCCs are intended to cool low power-consumption sensors, thereby lowering thermal noise and enhancing bandwidth in the sensor. With its simple configuration, the Joule-Thomson (JT) cooler is the most popular approach for MCCs. Figure 1 illustrates a JT-based cryogenic cooler. Refrigerant flows continuously through the compressor, heat exchanger (high pressure channels), JT expansion valve, evaporator, heat exchanger (low pressure channels), and then compressor again to form a closed loop cycle. The gas mixture (refrigerant) is pressurized using a compressor (Figure 1, a→b), and then it flows through a cooler to be pre-cooled (Figure 1, b→b'). After precooling, the gas mixture flows through a counter flow heat exchanger where it exchanges heat with the gas flowing in the

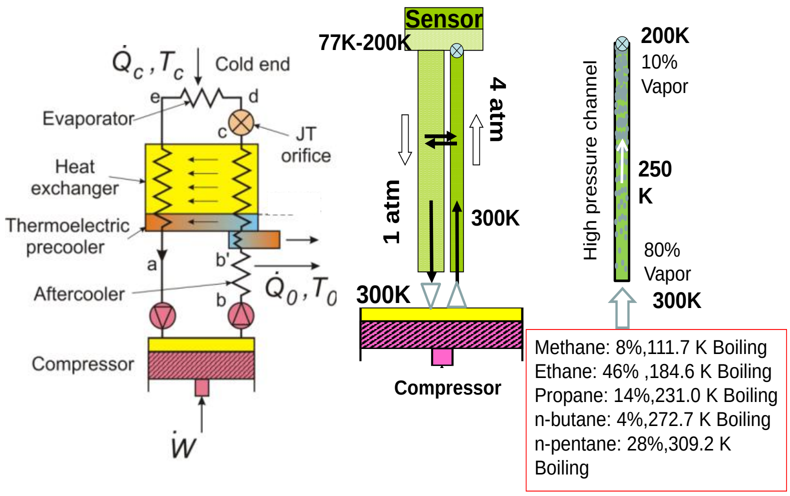

Figure 1: Joule-Thomson micro cryogenic cooler (MCC)

opposite direction inside the low-pressure line (Figure 1, b'→c). While the gas mixture meets a flow restriction (Figure 1, JT orifice), it undergoes isenthalpic expansion and the pressure drops from high to low, e.g. 16 atm or 4 atm to 1 atm for our MCCs studied. During the expansion process, the gas mixture cools and partially vaporizes (Figure 1, c →d). The liquid evaporates or boils while absorbing heat from the device and from the environment (Figure 1, d→e). From the cold head, the low pressure two-phase fluid flows back into the heat exchanger (Figure 1, e→a) to cool the incoming high pressure warm fluid for efficiency enhancement. The gas eventually goes back to the compressor system to complete a close-loop Joule-Thomson cooling cycle. Previous studies on such Joule-Thomson (JT) MCCs are reviewed as follows.

In the 1980s, W. Little *et al.* [1,2] made a series of matchbox size JT type MCCs based on an etched glass plate heat exchanger, with the lowest temperature ranging from 88 K to 70 K. In 2001, J. Burger *et al.* [3] inserted a glass capillary tube (ID/OD=0.25mm/0.36 mm) into a larger glass capillary tube (ID/OD=0.53mm/0.67 mm), forming a coaxial heat exchanger for a 77 mm x 9 mm MCC. In 2006, Lerou *et al.*[4] fabricated a 30 mm x 2.2 mm glass plate based JT cooler which achieved a temperature of 100 K. These heat exchangers achieved reasonable performance for the MCCs demonstrated. However, the pressures required were very high (e.g. 80 atm), and refrigerants used were based on single gas components with low refrigeration capabilities. We have improved the MCCs substantially by using custom-designed gas mixtures and using MEMS technologies for high thermal isolation.

The use of gas mixtures is also illustrated by Figure 1. The mixture, consisting of gas components with different boiling temperatures, is optimum-designed for a specific temperature range (300 to 200K as shown in the figure) and a pressure ratio (4:1 as shown in the figure). The mixture is delivered into the high pressure channel, i.e. 6 hollow fibers (ID/OD=75μm/125μm), of the heat exchanger by a compressor. At the entrance of the channel, most of the mixture would be in a vapor form. Moving toward the coldhead, most of the mixture would be liquefied at cryogenic temperatures, e.g. 200K as shown. The feasibility of such MCCs has been demonstrated, as shown by Figure 2[5,6,7]. This figure presents the temperatures of the cold head while the system was being pumped by the compressor. We reached stable 140-150 K for over 2 hours. We employed a mixture consisting of five components: Propane (231 K/85 K), Ethane (184.6 K/90.4 K); Methane (111.6 K/90.7 K); Neon (27 K/24.5 K); and Nitrogen (77 K/63 K) listed as gas (boiling point/triple point). In addition, when the low pressure level was changed, the cold head temperature dropped even lower down to 76K. Unfortunately, it was not a stable operation. Nevertheless, the stable cryogenic operation at 140-150K demonstrated the feasibility of the MCC with novel features such as gas mixtures and fiber-based MEMS technologies.

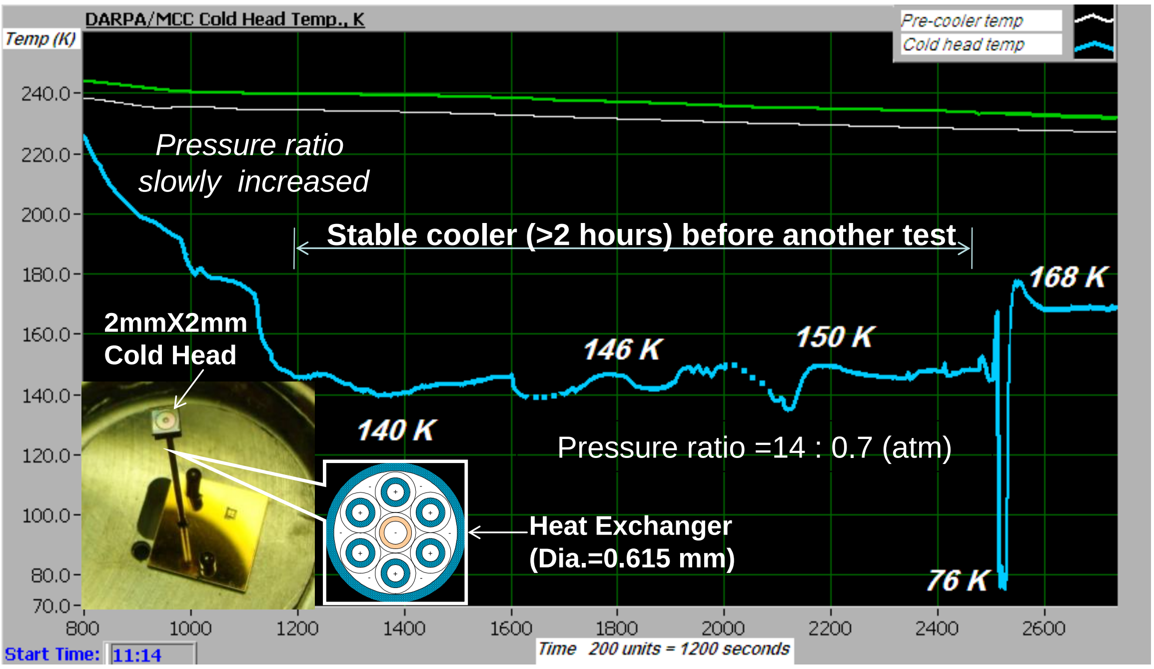

Figure 2: J-T micro cryogenic cooler demonstrated with a stable operation at 140-150K

Figure 3 compares J-T MCC, thermoelectric (T/E) cooler and Stirling cooler for a temperature range from 300 to 200K. As indicated, for the same heat lift, MCC input power is about 10% of the T/E cooler's. MCC's size is about 10% of the Stirling cooler's. The T/E cooler is well known for its poor efficiency at temperatures below 240 K. The Stirling cooler's size is limited by it's frequency (<120 Hz) for the oscillating flow. With the simple configuration, MCC's compressor can be operated at high frequencies, e.g. 1 KHz. We have also derived other comparisons corresponding to different temperature ranges with conclusions similar to those illustrated in Figure 3.

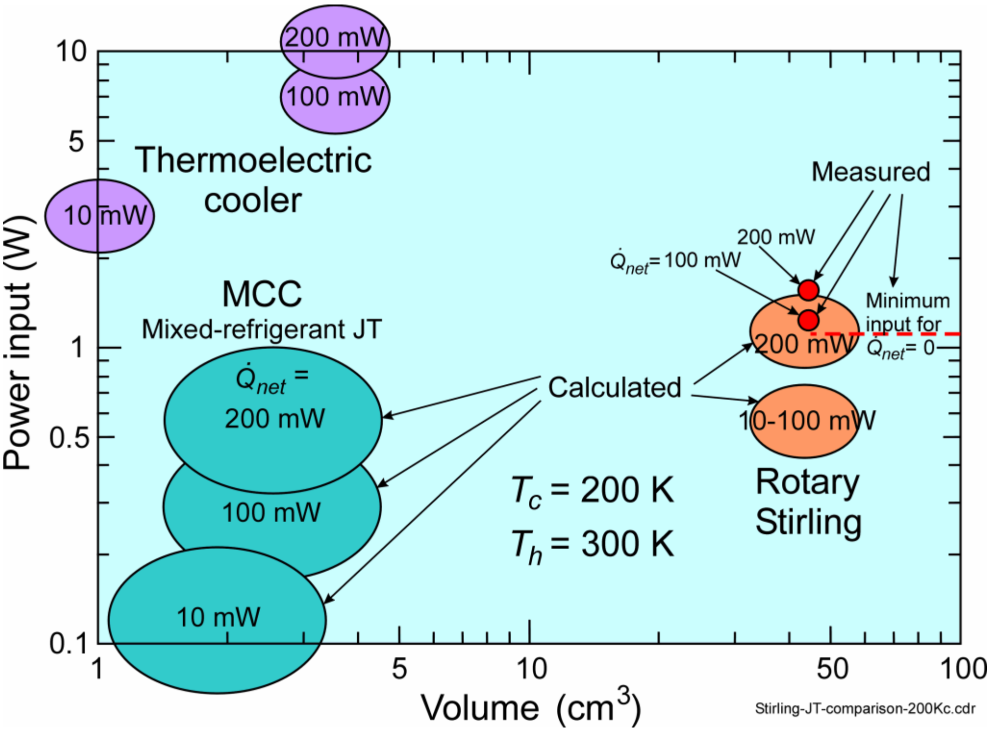

Figure 3: MCC compared with thermoelectric and Stirling coolers for a 300-to-200 K temperature range

In addition to the above-mentioned advantages, Figure 3 indicates a great potential for MCC-enabled future IR imaging systems. MCC's power and size are reduced significantly when the heat lift is reduced from 100-200 mW to 10 mW. According to our understanding, IR imaging module's power dissipation can be substantially reduced with novel read-out approaches. The best system with very low power and small size can be accomplished by integrating an optimum MCC cooler and an optimum IR imaging module.

MCC is expected to make a major impact to IR imaging and many other sensing applications. However, many engineering challenges have to be solved. We now review our studies on some of these challenges with an emphasis on thermal isolation, gas mixture and compressor, followed by a discussion on opportunities.

2. THERMAL ISOLATION

Figure 4 illustrates the thermal isolation concept for the MCC and an illustration of a functional device. The MCC is enclosed in a vacuum chamber ($< 2 \times 10^{-5}$ torr) to avoid air conduction and convection. The major heat loads result from conduction through the heat exchanger consisting of 6 glass fibers and a capillary, and radiation onto the cold head and the heat exchanger. The hollow-core fiber-based heat exchanger is 25 mm long and cold head is 2 mm square. Three major components of MCC are the micro cold head, the micro heat exchanger, and the micro coupler. Refrigerant flows are indicated by arrows as high pressure in light/red and low pressure in bold/blue. In the micro cold head, three chips are bonded together to form an expansion valve and returning channels. The expansion valve consists of a 760-nm-deep and 500-µm-across radial gap, formed between the Pyrex glass and silicon chips. Six hollow-core fibers (ID/OD=75 µm/125 µm) are solder-bonded into an etched silicon coupling structure. Another etched silicon chip is then solder-bonded onto the cold head with the gap between these two chips forming the embedded returning channels. The 6 fibers are fed into a glass capillary (ID/OD=536 µm/617 µm), forming the heat exchanger with high pressure refrigerants in the fibers and low pressure refrigerants in the capillary.

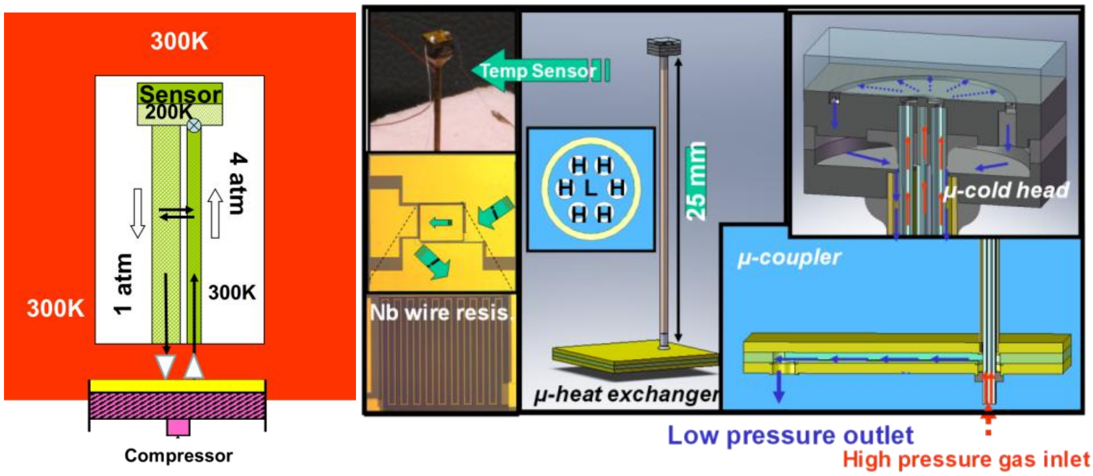

Figure 4: High thermal isolation achieved by using fiber-based MEMS technologies.

In a thermal study, we characterized the heat loads on the MCC with a 77 K cold head and different surrounding temperatures. At 240 K and 300 K surrounding temperatures, the heat loads on the MCC with segmental metal coatings were 9.62 mW and 5.09 mW, respectively. With an Aluminized Mylar surface shielding, the heat loads without metal coating were 3.81 mW and 1.83 mW, respectively. With 1.83 mW heat load at 77K and shielding at 240 K, the thermal isolation could be calculated as $(240-77)/(1.83 \times 10^{-3}) = 89{,}000$ K/W. This study is to be reported in the near future[8]. This extremely high thermal isolation achieved is very important to IR imaging and other sensor applications. In addition to device power dissipation, cryocoolers usually have to remove heat loads resulting from heat transferred from the signal and power/ground leads and conduction and radiation from environment. Without excellent thermal isolation, such heat loads could be much higher than the device power dissipation.

It should be noted that thermal isolation is strongly affected by the device and MCC configurations and temperature difference between the environment and the cold head. In addition, it is not necessary to reach the maximum possible thermal isolation. For real applications, we have to develop an optimum design considering total heat loads from the device and the environment, manufacturability, and mechanical integrity under vibration and mechanical shock tests.

3. GAS MIXTURES

The heat flows in and out of the MCC system can be represented by the equation below[9]:

(1)

where is the gross refrigeration delivered by the mixed refrigerant pumped by the compressor; $\dot{n}$ is the flow rate in mol/s; and $(\Delta h_T)_{min}$ is the minimum molar isothermal enthalpy difference of the refrigerants between the high pressure and low pressure enthalpies within the temperature range of interest, and is the net refrigeration power or heat lift. Other components are: (1) , the refrigeration loss resulting from heat exchanger ineffectiveness; (2) , the refrigeration loss resulting from the pressure drop on the low pressure side of the heat exchanger; (3) , conduction heat loads through the heat exchanger or DC leads used to power the device; and (4) , radiation heat loads from the environment.

In order to reduce the MCC compressor's size and power, we have custom-design a gas mixture with the highest possible minimum isothermal enthalpy difference for a given set of pressures, e.g. from 16 atm or 4 atm to 1 atm, and temperature difference, e.g. from 300 to 200 or 77K. As shown in the above equation, for a given gross refrigeration power, the flow rate can be reduced substantially if we can apply a gas mixture with a high minimum enthalpy difference. With the reduced flow rate, the size and the power of the compressor can be decreased.

In macro-scaled Joule-Thomson refrigeration systems, mixed refrigerants have been widely applied to enhance the efficiency and refrigeration power. Radebaugh[9], Missimer[10], and Boiarski[11] reviewed recent developments and history of mixed refrigerants. Fuderer and Andrija[12] first used mixed gases in a single stream without phase separators in 1969. They found that the mixtures experienced mostly two-phase flow in the heat exchanger. As a result, boiling and condensing heat transfer of two-phase flow greatly enhanced cooling efficiency. Boiarski and Longsworth[11] demonstrated mixtures for 67 K cooling for a JT system with only 2 MPa pressure applied. They also pointed out that a possible liquid-liquid separation of the nitrogen from hydrocarbons in the mixtures happened at extremely low temperature if the pressure was lower than 6 MPa. To quantify its performance, Marquardt et al. [13] further developed models and optimization approaches for mixtures. Little[14,15] also verified great enhancement in refrigeration power using mixtures compared with pure nitrogen. However, no studies have ever been conducted to apply gas mixtures for micro-scaled cryocoolers, and no studies have ever been conducted to integrate the mixture designs and the compressor for an MCC.

To determine the behavior of mixtures in MCCs, the normal boiling points of the components, mixture solubility, and refrigeration loss due to pressure drop on the low pressure side of the heat exchanger are evaluated. In fluids, the largest enthalpy difference usually occurs at or close to the temperature of the phase change from liquid to gas. For pure refrigerants, $(\Delta h_T)_{\min}$ occurs at the highest temperature of interest which is the warm end of the heat exchanger. As a result, several cooling stages and different gases have to be applied to enhance efficiency to reach cryogenic temperatures. In mixed refrigerants, components are selected with boiling points across the temperature range of interest. By controlling the amount of different components in a mixture, the enthalpy difference is made more uniform across the temperature range, and $(\Delta h_T)_{\min}$ is maximized.

Figure 5 shows the plot of the isothermal enthalpy difference of the optimized five-component mixed refrigerant. The optimized mixed refrigerant is designed using NIST software known as NIST4[16]. The mixture in terms of mole fraction consists of 14% Propane, 16% Ethane, 22% Methane, 42% Nitrogen, and 6% Neon, and their normal boiling points are 231 K, 184.6 K, 111.6 K, 77 K, and 24 K respectively. The minimum enthalpy difference between 0.1 MPa to 1.6 MPa (~1.35 kJ/mole) occurs at 140 K.

Table 1 lists $(\Delta h_T)_{\min}$ for pure nitrogen and mixtures under different pressure ratios. The five-component mixed refrigerant has the largest $(\Delta h_T)_{\min}$ ~ 1.35 kJ/mol. Before the 5-component gas mixture, we had designed a 7-component gas mixture for the same temperature and pressure ranges. It reached a higher minimum enthalpy difference (1.576 kJ/mol); however, we decided to use the 5-component mixture due to its simplicity. To deliver 15 mW of gross refrigeration power, JT MCC using mixed refrigerants only require 1.6 MPa pressure input and 11 µmol/s or 15 sccm flow, while those using pure nitrogen require about 3 times the pressure input and flow as shown in Table 1.

Table 1 Minimum enthalpy difference of pure nitrogen and mixed refrigerants over the temperature range of 77 K to 240 K for the high pressure P_H and low pressure P_L are shown in the table. Also shown are the ideal COP, the efficiency, and required flow rate for each refrigerant to provide 15 mW of gross refrigeration power from the above equation.

Refrigerant	P_H (MPa)	P_L (MPa)	$(\Delta h_T)_{\min}$ (kJ/mol)	COP_{ideal}	η %Carnot	$\dot{n}$ (µmol/s)
N_2	5.0	0.1	0.468	0.0480	14.4	32.0
N_2	2.5	0.1	0.232	0.0289	8.7	64.6
N_2	1.6	0.1	0.146	0.0211	6.3	102.7
5-comp mix	1.6	0.1	1.35	0.2497	52.9	11.1

For IR imaging applications, the preferred target temperatures could be around 140K instead of 77K. As a result, we have identified another opportunity by using custom-designed gas mixtures to reduce the pressure ratio required. Figure 6 presents a breakthrough design of the gas mixtures good for 4:1 pressure ratio. For a given temperature range, i.e. 300 to 140 K in this case, the best 5-componet design could result in the minimum enthalpy difference reaching 2.01 kJ/mole, which is in fact higher than 1.35 kJ/mole of the gas mixture used in our successful demonstration a shown in Figure 2. With this conceptual breakthrough, we decided to use this new gas mixture and reduced the high pressure from 16 to 4 atm. With the 4:1 pressure ratio, we were allowed to use compact compressors. In addition, we will be able to use a polyimide-based planar cold stage instead of the fiber-based vertical one and use a MEMS compressor in the future. The experimental study on the use of such gas mixtures requiring only 4:1 pressure ratio will be presented in the next section.

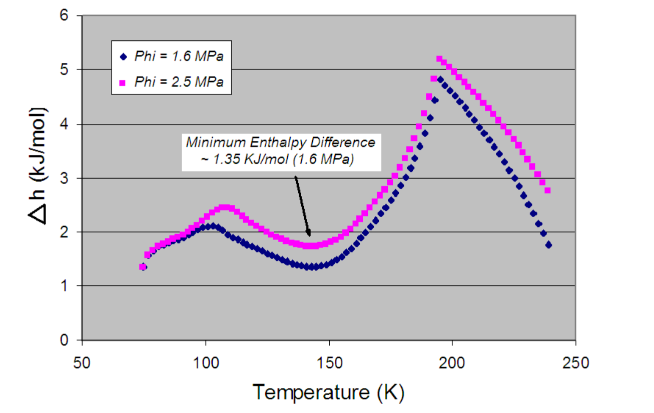

Figure 5: Enthalpy difference vs. temperature of an optimized five-component mixed refrigerant. Both enthalpy difference of 0.1 MPa (1 bar) to 1.6 MPa (16 bar) and 0.1 MPa (1 bar) to 2.5 MPa (25 bar) are plotted as a function of temperature.

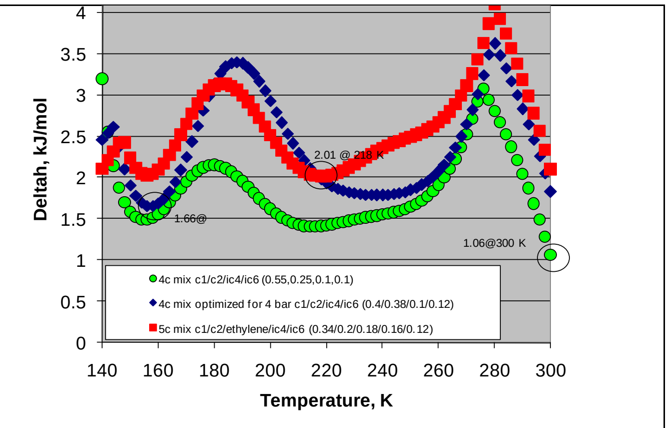

Figure 6: Gas mixtures designed for a 300-to-140 K temperature range and 4:1 pressure ratio. Gas mixture of methane/ethane/ethylene/isobutane/isohexane with mole fractions 0.34/0.20/0.18/0.16/0.12 achieved a 2.01 kJ/mol refrigeration capability.

4. COMPRESSOR

The development of a compressor has been an ongoing challenge in our MCC studies. A laboratory meter-scaled compressor was used in the demonstration study as shown in Figure 2. For a fully integrated MCC, however, we need a compact compressor. We have developed a piezoelectric actuator-based compressor as shown in Figure 7; its design concept was reported in a paper published in 2009[17]. It reached 5:1 pressure ratio with 500 Hz operation. The results were very encouraging. Unfortunately, it was difficult to reach a reliable performance. We decided to use a commercially available compressor that was proven repeatable and reliable. Figure 8 shows a commercial miniature compressor chosen for its compact size. A set of two check valves was added to the compressor, which was manufactured for a Stirling cooler that did not require valves. In addition, a coupler was fabricated precisely to reduce the dead volume in order to generate pressure ratios higher than 4:1. Again, pressure ratio was a new requirement since a Stirling cooler requires a low pressure ratio <1.5:1. The modified compressor was successful and achieved pressure ratio as high as 7:1.

In an experiment using a gas mixture modified from the ones shown in Figure 6, we have successfully demonstrated cryogenic cooling with the miniature compressor running with a pressure ratio around 4:1. The temperature fluctuations were reduced to ±1 K when a micro heater was applied to control the cold head temperatures. In addition to 200K, we have also reached other much lower temperatures. More details are to be reported in a conference in the near future[18].

These results are very encouraging. Gas mixtures requiring low pressure ratios, e.g. 4:1, are proven feasible. In addition, compressors developed for Stirling coolers can be modified for JT MCCs. With these accomplishments, we have demonstrated that JT MCCs can be further developed for real IR imaging applications in the near future.

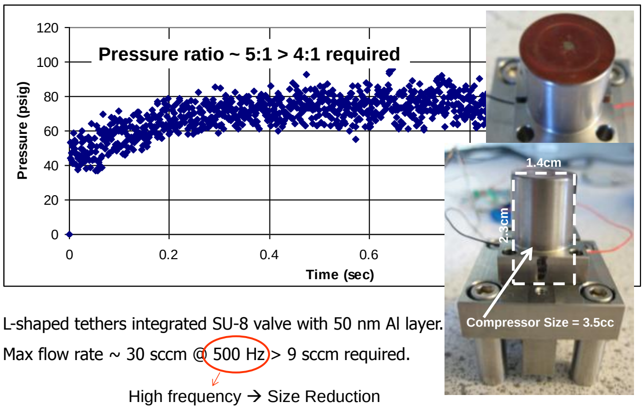	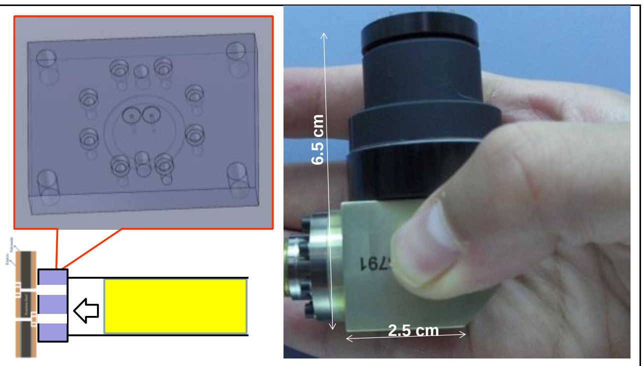
Figure 7: Piezoelectric actuator-based compressor developed for MCC with a metalized Kapton™ diaphragm driven by the actuator. Pressure ratio of 5:1 was achieved.	Figure 8: A miniature compressor used for a rotary Stirling cooler was modified with a set of check valves and a coupler to reduce dead volume for MCC.

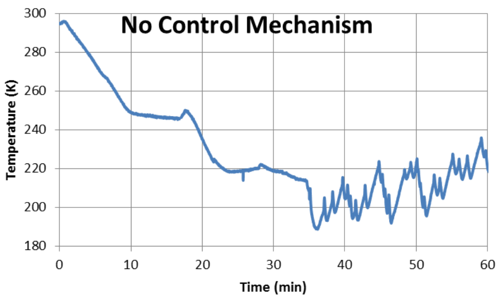	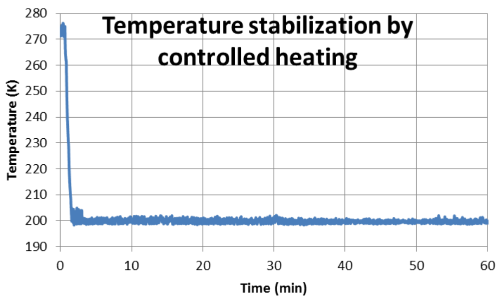
Figure 9: Cryogenic cooling demonstrated using a gas mixture modified from those shown in Figure 6.	Figure 10: Improved performance by controlled heating.

5. OPPORTUNITIES

Joule-Thomson MCCs have been demonstrated with novel features sufficient for IR imaging applications. More importantly, we have identified opportunities to develop a planar MCC that will reach the full potential as illustrated in Figure 3. The planar MCCs will be very compact, manufacturable, reliable and cost-effective. The planar heat exchanger fabricated is shown in Figure 11. These heat exchangers are realized on a wafer with an excellent potential to be fabricated, assembled and packaged through batch processes. Its details will be reported in a conference this year[19].

Another exciting potential is for additional 10 or 100X size reductions by using a MEMS compressor as shown in Figure 12. In general, for a given heat lift, there is a specific refrigerant's flow rate required. Running the compressor at a higher frequency would require a smaller volume displaced in each stroke. As a result, the compressor's volume is roughly inversely proportional to the operating frequency. When the frequency reaches 100 kHz, it is possible develop a MEMS-scaled compressor that will be much smaller than the compact compressors shown in Figures 7 and 8.

However, the highest pressure ratio achieved to date by a MEMS compressor to date is close to 2:1. The major barrier to reaching high pressure ratios is the control of dead volume and achievable maximum actuation force using MEMS devices operating at 100 kHz frequencies. When the desirable MEMS compressor is developed, the MCC size range as shown in Figure 3 can be further reduced. There is clearly an opportunity to develop nearly "invisible" MCCs for IR imaging.

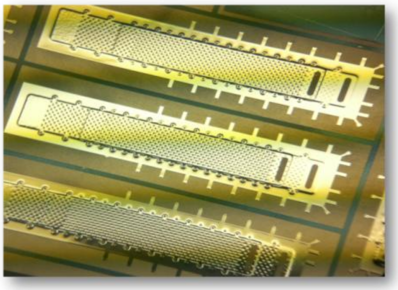

Figure 11: Planar heat exchanger for planar MCCs.

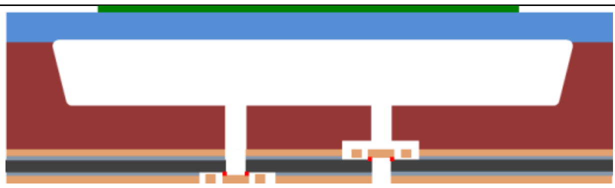

Figure 12: MEMS compressor for planar MCCs.

6. ACKNOWLEDGMENTS

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7. SUMMARY

MCCs with mixed refrigerant are very promising for cooling small electronics because they can require a tenth the input power of a corresponding TE cooler, and fill a tenth the volume of a corresponding Stirling cooler. To realize that potential, an MCC requires high thermal isolation, a refrigerant with high cooling power, and a miniature compressor. We have demonstrated high thermal isolation of 89,000 K/W using a fiber-based MCC. Optimized mixed gas refrigerants, composed of light hydrocarbons, were designed to operate at a 4:1 compression ratio with >2 kJ/mol refrigeration capacity. This low pressure allowed the use of a miniature compressor designed for a Stirling cooler and fitted with check valves, for stable cooling to 200 K. The future planar heat exchanger and high-frequency MEMS-based compressor have exciting potential for reliability, cost-effective manufacturability, and an order of magnitude smaller volumetric footprint. At such small sizes, these “invisible” MCCs may revolutionize future IR imaging systems.

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