A Modular Architecture for Helium Compressors Larger than 2.5 kW

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ABSTRACT

In oil cooled helium scroll compressors only about 10% of the heat generated is in the helium, but that helium should be cooled to <20 °C for best performance (especially with pulse tubes). The big heat load is from cooling the oil, but the oil does not need to be cooled below around 50 °C if the flow rate stays high. Thus, there are rather different cooling requirements for the helium and for the oil. To our knowledge, this distinction has not been appreciated in commercially available water-cooled or air-cooled helium compressors.

We describe two compressor configurations, with and without a chilled water loop available. For the more familiar configuration in which chilled water is available, the only changes to our compressor are to the water side of the cooling loop, which employs two separate brazed-plate heat exchangers (BPHX) in series. This modification would not be possible with traditional water-cooled compressors that use a single three-fluid heat exchanger.

When the water is not available, we break the link between the two BPHX's previously in series and provide for independent cooling of the helium and oil circuits. The oil is cooled via a commercial copper tube-in-plate heat exchanger, fan, and small water pump. The helium is cooled with a thermal expansion valve plus refrigeration connectors on the helium BPHX. The indoor unit is comprised of a Copeland condenser and a refrigerant receiver.

INTRODUCTION

Beginning in early 2010 Quantum Design has installed a number of water cooled helium compressors1 with its cryocooler based instrumentation. In these systems, the individual speed control of the compressor's capsule and cold head motor provide for intelligent oversight and budgeting of the cold head cooling power based upon the refrigeration needs required by the user experiments2. An important design consideration for these compressors is to achieve operational performance even in those facilities where chilled water was not plentiful or altogether not available.

With this in mind we worked on making our water-cooled compressors more flexible. This is especially important for customers who do not have the luxury of a chilled water loop in their lab. We call this approach a "Modular Architecture", and it can be adapted to our standard water-cooled units with fairly minimal modifications3. The only changes are to the water side of the two brazed-plate heat exchangers (BPHX). This modification would not be possible with traditional water-cooled compressors that use a single three-fluid heat-exchanger.

Cryocoolers 17, edited by S.D. Miller and R.G. Ross, Jr.

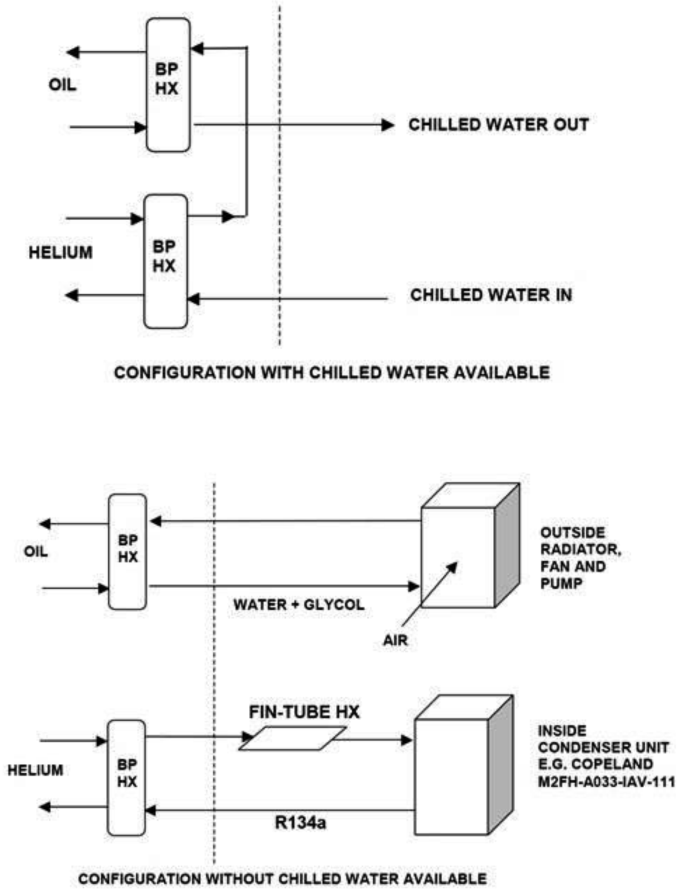

Figure 1. Possible configurations for a water cooled compressor

RADIATOR DESIGN

Figure 1 shows a schematic of the compressor configurations with and without a chilled water loop available. When the water is not available (lower diagram), we provide two additional boxes: one goes inside near the compressor and one goes outside or in a well ventilated location. We also break the link between the two BPHX's and provide an automatic expansion valve plus refrigeration connectors on the helium BPHX.

The outdoor unit is comprised of a commercial copper plate HX, fan, and small water pump. Since pressures are not large (a few PSIG) there is no concern about the pressure rating of components or lines. If freezing temperatures are possible, a mix of water and ethylene glycol would be needed. In any case, the water needs to be "copper friendly" so some additive may be helpful.

The indoor unit is comprised of a Copeland condenser and refrigerant receiver to provide an adequate amount of refrigerant for the various cooling requirements for our variable frequency driven compressors.

In practice, the outdoor and indoor units can be “packaged” in a single radiator cabinet if so desired by the user. This is in fact the preferred configuration when the radiator is housed in a warehouse or a laboratory “chase”. Figure 2 depicts the industrial design of the radiator comprising of both the water- glycol mixture and R134a cooling loops.

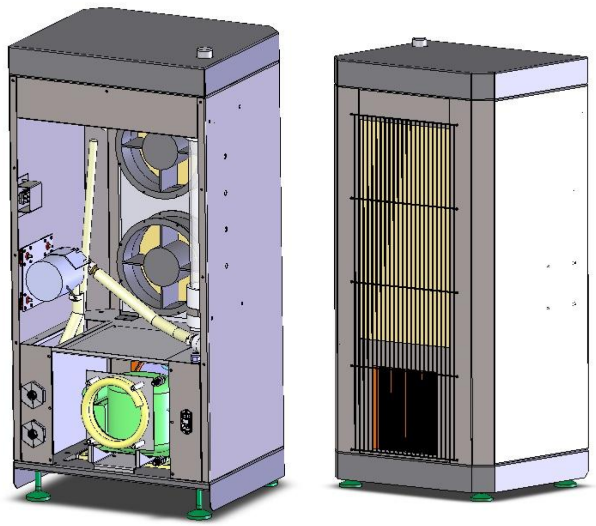

Figure 2. The industrial design of an integral radiator: lower bay houses the Copeland condenser unit while the upper bay houses the water loop radiator.

EXPERIMENTAL RESULTS

Figure 3 shows stabilized Capsule, Cool Helium and Cool Oil temperatures of the Quantum Design HLC 4500 series compressor for operation at 60 and 30 Hz using the two cooling configurations shown in Figure 1.

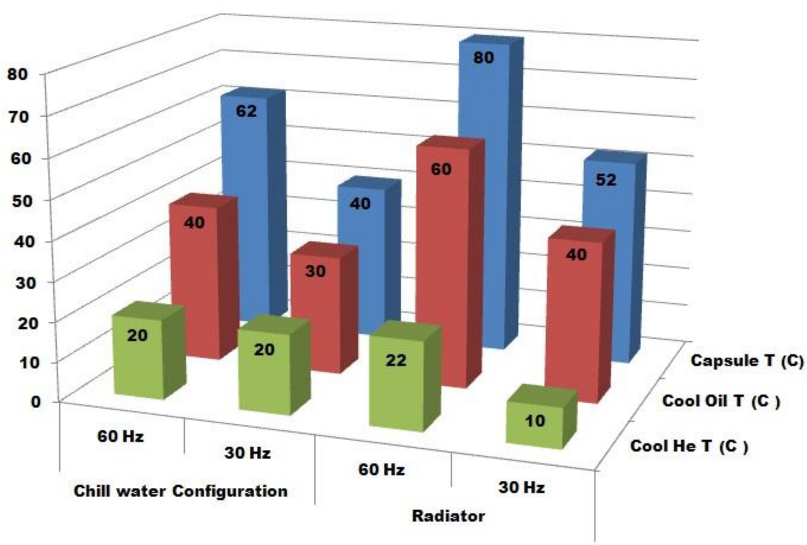

Figure 3. Cooling performance comparison using chilled water or a radiator configuration (see Figure 1).

For the configuration where chilled water is available, the water inlet temperature was set to 20 °C with a flow of 5.7 L/min. The data show that even at these relatively small water circulation rates the two separate BPHX in series are cooling the capsule very efficiently with capsule temperatures reaching 62 °C at capsule speeds of 60 Hz and 40 °C at capsule speeds of 30 Hz. This is because the chilled water at the inlet of the compressor is allowed to cool the helium gas independently of the oil using the first BPHX. Only after the helium has been cooled, the slightly warmer water is then used to cool the oil through a separate BPHX connected in series. The capsule temperatures recorded in this experiment are satisfactory and well below the maximum "discharge temperature" of 85 °C specified by the scroll helium capsule manufacturer. While this is good for the Helium gas, which in both cases is maintained close to 20 °C, it is not so for the oil temperature which should be maintained between 45 and 55 °C. The data show that during slow speed operation of the capsule at 30 Hz, the oil temperature cools to 30 °C, which will cause the oil to become more viscous and might make compressor start up more difficult. For optimal operation of this variable speed compressor, the chilled water flow should then be decreased even further and/or the inlet water chilled temperature increased.

For the radiator configuration, the data were gathered in an environmental temperature of 40 °C during 60 Hz operation, and 28 °C during 30 Hz operation. The capsule temperature once again is below 85 °C, even at 60 Hz. Likewise, adequate helium temperature is maintained by the Copeland condenser (i.e. the R134a refrigerant loop), with the Cool Helium temperature reaching 22 °C at 60 Hz, and the very nice temperature of 10 °C at 30 Hz. The Cool Oil temperature, cooled by the water cooling loop radiator equipped with a small water circulation pump (max flow of 7.5 L/min), is now 60 °C at 60 Hz and 40 °C at 30 Hz. The data show that with this modular design, the different cooling requirements for the helium gas and the oil can be met independently by varying the thermal expansion valve on the R134a refrigerant loop and the fan speed of the water loop radiator.

CONCLUSIONS

Three-fluid heat-exchangers, although widely used in the cryo-refrigeration industry, are not ideal for helium compressors. We have demonstrated a novel modular architecture that makes use of two separate BPHX which allow for: 1) ideal cooling conditions for both helium and oil and, 2) reduced water cooling requirements. We are confident that the radiator design described in this paper will be employed by many users and especially in those locations where there might be a lack of chilled water.

REFERENCES

1. Model HLC 4500: a 7.5 KW class liquid cooled compressor manufactured by Quantum Design Inc.
2. Sayles T., Martien D., Diederichs J., Sloan J., Bird D., Spagna S., "Enhanced Helium Compressor Operation for sensitive Measuring Instrumentation," *Cryocoolers 16*, ICC Press, Boulder, CO (2011), pp. 371-373.
3. 2012 Quantum Design Inc. Patent Pending.

Two Stage Pure Gas and Single Stage Mixture Gas Microcooler Developments at University of Twente

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ABSTRACT

The development of micromachined Joule-Thomson (JT) coolers has been an on-going and successful research project at the University of Twente for many years. In this research, we develop MEMS-based cryocooling systems for cooling small electronic devices, such as amplifiers and infrared sensors, to improve their performance. After successfully fabricating and testing single-stage JT microcoolers with cooling capacities of around 10 mW at 100 K, we now present the development of two-stage pure gas microcooler and of single-stage microcooler that operates with gas mixtures. The first stage of the two stage cooler operates with nitrogen gas between 85.0 and 1.1 bar and the second stage with hydrogen gas between 70.0 and 1.0 bar. The mass flow in the first and second stage is 16.7 mg s^{-1} and 0.93 mg s^{-1}, respectively, result in a cooling power of 21 mW at 95 K and 30 mW at 32 K. The single-stage microcooler is designed and produced for operating with hydrocarbon gas mixtures. This microcooler is powered with a linear compressor to obtain a closed-cycle cooler. Operating with a ternary mixture of methane, ethane and isobutane between 9.4 and 1.3 bar, a net cooling power of 46 mW at 150 K is obtained at a mass flow of 1.35 mg s^{-1}.

INTRODUCTION

Earlier we have reported on the single stage microcoolers manufactured using standard micromachining technologies. These microcoolers are operated with pure fluids to attain temperature of 103 K with nitrogen gas and 138 K with methane gas[1]. Typical operating pressures are 80 bar Nitrogen and 50 bar methane. The high pressure gas source is from a gas bottle. In this paper, we describe two new developments on microcooler research at University of Twente namely, two-stage pure gas microcooler to attain temperature of 30 K from 295 K and a closed cycle mixed gas microcooler powered by a linear compressor.

TWO-STAGE PURE-GAS JT MICROCOOLER

In our design, the two-stage microcooler is made in a stack of three glass wafers (see Fig. 1a). For each stage, the high and low-pressure lines are isotropically etched in the middle and bottom wafers. The channels contain pillars to limit the maximum mechanical stress caused by the high pressure. In each stage, the high-pressure line ends in a flow restriction, which is extended to the evaporator volume and connected to the low-pressure line. Thus, a counter-flow heat exchanger (CFHX) is formed by the high and low-pressure channels and the thin intermediate glass wafer. In order to increase the heat transfer between pre-cooler and the first evaporator, they are integrated into a compact design with a relatively large heat-transfer area. In the evaporator I, the low-pressure fluid of the first stage, after expansion, needs to take up heat from the high-pressure fluid of the second stage.

The working fluids of the two-stage JT microcooler operating at 30 K were optimized on the basis of COP, and nitrogen and hydrogen were chosen as the first stage and the second stage working fluid, respectively[2]. The pre-cooler, which is a heat exchanger thermally connecting the nitrogen and the hydrogen stages is an important component and careful design is crucial because the size and required mass-flow rate of the nitrogen stage is determined by the pre-cooler thermal performance. Heat conduction from the ambient temperature parts to the cold parts, thermal radiation on the cold surface, and pressure drop due to gas flow in the channels further add to the design complexity of a heat exchanger. We have optimized the dimensions of the CFHXs and restrictions considering all these losses and the results are described in elsewhere[2].

Figure 1b shows a picture of the produced two-stage microcooler. A thin gold layer is deposited on the cooler's surface to reduce radiation losses. Nitrogen and hydrogen gas is supplied from pressurized gas bottles. The microcooler is mounted in a vacuum chamber and a vacuum pressure of less than 10^{-4} mbar is maintained. Temperature is measured with resistance thermal sensor at the precooler and with a diode thermal sensor at the evaporator of the hydrogen stage. Standard surface-mounted-device (SMD) resistors are used as heaters for supplying heat to the microcooler to measure the cooling power. The temperature sensors and heaters are glued to the microcooler with conducting silver paint and are connected to the printed circuit board (PCB) with bond wires (made from 1% silicon and 99% aluminum) of diameter of 25 μm.

The nitrogen stage cools down from 295 to 94 K in about 20 minutes and the hydrogen stage cools down to 30 K from 295 K in about 60 minutes. The high and low pressures of the nitrogen stage are 85.1 and 1.1 bar, respectively, and of the hydrogen stage are 70.0 and 1.0 bar, respectively. The cool-down curve for both of the stages is shown in Fig. 2. The mass-flow rate of the nitrogen stage increases from 6.1 to 21.7 mg s^{-1} for a corresponding decrease in temperature of the precooler from 295 to 94 K. In the steady state the mass-flow rate of the nitrogen gas fluctuates around 17.5 mg s^{-1}. The mass-flow rate of the hydrogen gas increases from 0.1 to 1.3 mg $s^{-1}$with the corresponding decrease in the cold-tip temperature of the hydrogen stage from 295 K to 30 K.

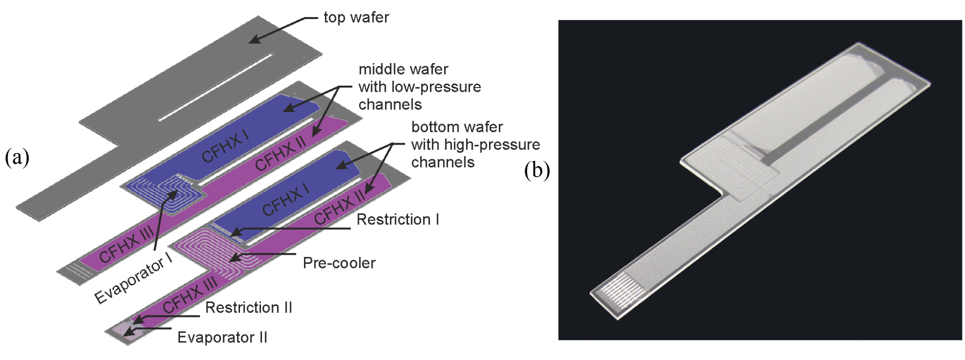

Figure 1. Two-stage microcooler geometry. a) Exploded view of the two-stage microcooler, b) Photograph of the two-stage microcooler.

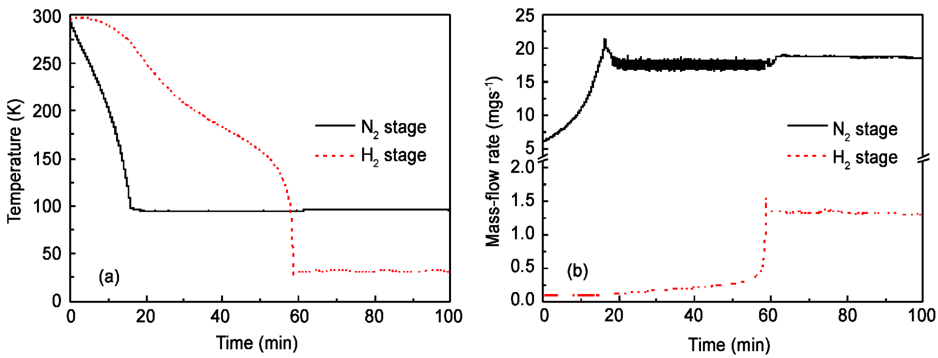

Figure 2. Measurement results of the microcooler. a) Cold-tip temperatures of the two stages, b) Mass-flow rates of the two stages. The N_2 stage is operated between 85.1 bar and 1.1 bar, and the high and low pressures of the H_2 stage are 70.0 bar and 1.0 bar respectively.

Under the operating condition that the high and low pressures of the nitrogen gas are 85.1 bar and 1.1 bar, respectively, and those of the hydrogen gas 70.1 bar and 1.0 bar, respectively. The net cooling power of the hydrogen stage is measured using a heater. When the hydrogen stage reaches about 30 K, the dissipated heater power is increased to the point where the hydrogen stage temperature begins to rise. Using this method, the measured net cooling power of the hydrogen stage is estimated to be 32 mW with a mass-flow rate of 0.95 mg s^{-1}. In this case, the temperature of the nitrogen stage is about 94 K. The net cooling power of the nitrogen stage is measured using a PID controller that maintains a fixed nitrogen stage temperature of 95 K. In this measurement, the heater power of the hydrogen stage is set to 30 mW, a little lower than the measured net cooling power of 32 mW, because the nitrogen stage is controlled at 95 K and there is almost no two-phase flow in the evaporator. The heater power of the nitrogen stage increases gradually with the decreasing mass-flow rate due to water molecule deposition inside the restriction[3]. That is because the temperature of the nitrogen gas decreases with the decreasing pressure drop in the low-pressure channel. The increased heater power is required to heat the nitrogen gas at a lower temperature to reach 95 K. The net cooling power of the nitrogen stage is about 21 mW at 95 K for a mass-flow rate of 16.7 mg s^{-1}.

CLOSED-CYCLE SINGLE-STAGE MIXED-GAS JT MICROCOOLER

In this section, a microcooler operating with mixed gas powered by a linear compressor is described. Our aim is to design a microcooler that operates at 150 K with a gross cooling power of 60 mW. The warm end is at 295 K. On the basis of maximizing the minimum isothermal enthalpy difference between the high and low-pressure flows in the CFHX ($\Delta h_{\min}$), a mixture of 39 mol% methane, 20 mol% ethane and 41 mol% isobutane is selected as the working fluids. Detailed calculations and analysis of this gas mixture is described in the PhD thesis of Derking[4]. The isothermal enthalpy difference between the high-pressure and low-pressure flows is a function of temperature, and smallest enthalpy difference is at 184 K ($\Delta h_{\min}$ is 0.58 x 10^5 J kg^{-1}). To produce a gross cooling power of 60 mW, the required mass-flow rate is 1 mg s^{-1}.

Figure 3 shows a picture of the fabricated microcooler with instrumentation. The dimensions of the cooler are 60.0 mm x 10.0 mm x 0.7 mm. One heater and two resistance type thermal sensors are glued to the cold tip of the cold stage with conducting silver paint. This microcooler is connected with gas tubes to a two stage linear motor compressor. The schematic of the measurement set-up is shown in Fig. 4. The high-pressure fluid flows through a getter filter to decrease the amount of water particles in the gas mixture to below one part-per-billion to avoid clogging of the restriction[3]. The gas then flows through the JT cold stage and at the low-pressure side of the system the mass-flow rate is measured. The tubing in the center can be used to clean the system by means of pumping and flushing.

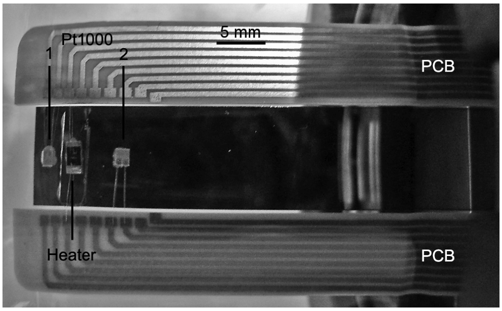

Figure 3. Photograph of the micromachined JT cold stage prepared for the experiments. The cold stage is equipped with two temperature sensors and one heater. The temperature sensors are located at a distance of 2 mm and 9 mm from the cold end.

This compressor was developed by the Rutherford Appleton Laboratory and a similar variant is used for a 4 K closed-cycle JT cooling system for the high frequency instrument on the Planck explorer that was launched in 2009[5,6]. In our experimental setup, the system is filled with the selected gas mixture to a pressure of 4.6 bar. The compressor is operated at an oscillating frequency of 42 Hz and a piston stroke of 7 mm. This results in low, medium, and high pressures of 1.3 bar, 3.9 bar and 9.4 bar, respectively. The total input power to the compressor set is about 50 W. The compressed gas is cooled to room temperature with an aftercooler.

Figure 5 shows the measured temperature, mass-flow rate and heater power versus the time. The JT cold stage cools down from 295 to 128 K in 14 minutes. The second temperature sensor reads 134 K. During the cool down phase, the mass-flow rate increases from 0.9 to 1.5 mg s^{-1} and after cool down it becomes 0.8 mg s^{-1} due to the temperature dependence of density and viscosity of the mixture. The fluctuations in the mass-flow rate indicate that two-phase fluid is formed in the evaporator. The net cooling power of the JT cold stage is measured at different temperatures by placing sensor 1 and the heater resistor in a PID control loop. At a set temperature of 140 K (at time 0.8 hour in Fig. 5), the mass-flow rate increases to 1.25 mg s^{-1}. The net cooling power in this case is 38 mW. At 150 K, cold-tip set-point temperature, the mass-flow rate further increases to 1.35 mg s^{-1} and the net cooling power to 46 mW. With further increase in cold-tip temperature to 160 K, the mass-flow rate decreases. This is caused by the deposition of ice crystals inside the restriction, reducing the height and thereby the mass-flow rate. This accumulation of ice crystals in the restriction would result in clogging of the restriction

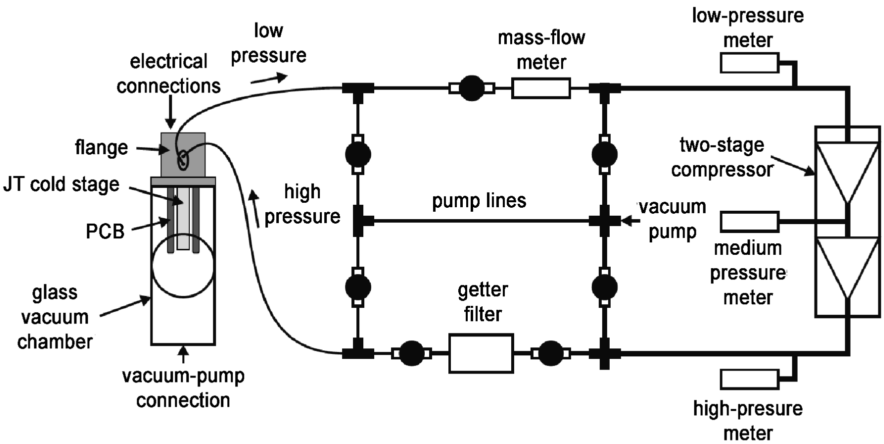

Figure 4. Schematic of the experimental set-up used to measure the performance of a closed-cycle JT microcooler

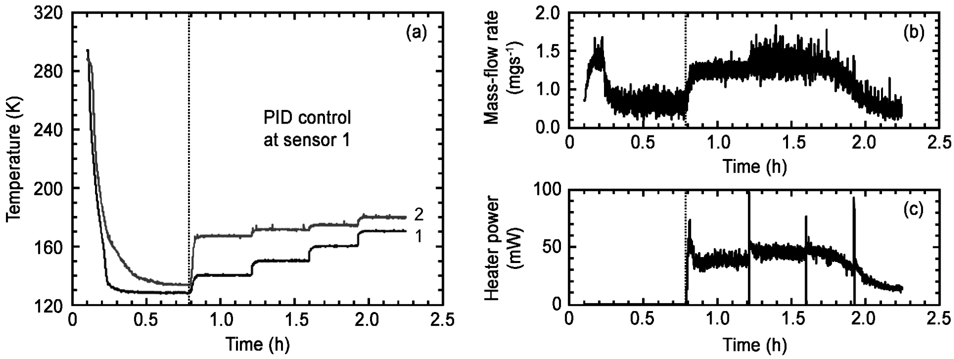

Figure 5. Measurement results of a closed-cycle JT cooler operating with a gas mixture of 39 mol% methane, 20 mol% ethane and 41 mol% isobutane between 1.3 bar and 9.4 bar. a) Temperature of two sensors, b) Mass-flow rate and c) Supplied heater power versus the time.

and heating up of the cold stage[3]. A cold stage operating at 160 K is more sensitive to clogging than one operated at a lower temperature, because the cold-tip temperature is much closer to the phase-transition temperature range of 180-220 K when ice crystals deposit corresponding to the partial pressure of water in the gas mixture.

The linear compressor used in the current test rig can deliver a much larger flow than the 1.35 mg s^{-1} level required by a single miniature JT cold stage. The penalty of decrease in compression ratio is quite small. Therefore, this compressor can be used in a distributed cooling system in which multiple miniature JT cold stages are combined with a single compressor. Further experiments showed that the linear compressor used can deliver a mass-flow rate that is adequate to drive about 19 micromachined JT cold stages in parallel. In this mode, the compressor pressure ratio is slightly less (between 2.2 and 8.9 bar compared to between 1.3 and 9.4 bar) which will result in a lower net cooling power of 23 mW per miniature JT cold stage.

CONCLUSION

A two-stage 30 K JT micocooler with overall dimensions of 20 mm x 90 mm x 0.7 mm has been developed and tested. With nitrogen and hydrogen gas as the working fluids, the nitrogen stage cools down to 94 K in about 20 minutes and the hydrogen stage cools down to 30 K in 60 minutes. The high and low pressures of the nitrogen stage are 85.1 and 1.1 bar, respectively, and that of the hydrogen stage are 70.0 and 1.0 bar, respectively. A closed-cycle JT cooler consisting of a micromachined JT cold stage and a linear compressor is investigated. The cooling system is operated with a gas mixture of 39 mol% methane, 20 mol% ethane and 41 mol% isobutane between 1.3 bar and 9.4 bar. At a cold-tip temperature of 150 K, a net cooling power of 46 mW is obtained with a mass-flow rate of 1.35 mg s^{-1}. The linear compressor used can deliver a mass-flow rate that is adequate to drive about 19 micromachined JT cold stages in parallel. In that way, a distributed cooling system can be realized.

ACKNOWLEDGMENT

This work is supported by the Dutch Technology Foundation (STW) under Contract No. 08014 and the European Space Agency under Contract No. 20768/NL/EM.

REFERENCES

1. Derking, J. H., Holland, H. J.,; Lerou, P. P. P. M., Tirolien, T., ter Brake, H. J. M., "Micromachined Joule-Thomson cold stages operating in the temperature range 80–250 K,". *Int. J. Refrig.,* vol. 35, no. 4 (2012), pp.1200-1207.

2. Cao, H. S., Mudaliar, A. V., Derking, J. H., Lerou, P. P. P. M., Holland, H. J., Zalewski, D. R., Vanapalli, S., ter Brake, H. J. M., "Design and optimization of a two-stage 28 K Joule–Thomson microcooler," *Cryogenics*, Vol. 52, Issue: 1 (January 2012), pp. 51-57.

3. Lerou, P. P. P. M., Brake, H. J. M. t., Holland, H. J., Burger, J. F., Rogalla, H., "Insight into clogging of micromachined cryogenic coolers," *Appl. Phy. Lett.*, vol. 90, no. 2 (2007), pp. 064102.

4. Derking, J. H., "Distributed Joule-Thomson microcooling for optical detectors in space". PhD thesis, University of Twente, Enschede, 2011.

5. Morgante, G., Pearson, D., Melot, F., Stassi, P., Terenzi, L., Wilson, P., Hernandez, B., Wade, L., Gregorio, A., Bersanelli, M., Butler, C., Mandolesi, N., "Cryogenic characterization of the Planck sorption cooler system flight model," *J. Instrum.*, Vol. 4 (2009), pp. T12016.

6. Bradshaw, T. W., Orlowska, A. H., Jewell, C., "Life Test and Performance Testing of a 4K Cooler for Space Applications In 10th International Cryocooler Conference," *Cryocoolers 10*, Kluwer Academic/Plenum Publishers, New York (1999), pp 521-528.