

Vibration-Free Joule-Thomson Cryocoolers for Distributed Microcooling

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ABSTRACT

This paper reports on an innovative concept for a space-borne Joule-Thomson (J-T) cryocooler that utilizes a continuous-flow compressor to provide cooling to multiple miniature cold heads. The heat transport to each cooling site is accomplished at ambient temperature, allowing large separation distances between cryocooler components and cooling sites with minimal performance impact. The compressor uses non-contacting, gas-lubricated bearings and is a derivative of TRL 9 technology that has demonstrated long life and high reliability. The concept addresses the limitations on life and reliability normally associated with J-T cryocoolers. The key technical challenge is the development of a Low-Specific-Speed (LSS) compressor to match the operating conditions of the J-T cycle. Cycle analysis was carried out to identify the optimum operating conditions as well as the optimum composition of the cycle gas. An LSS compressor and a compact cold head were designed, and their performance, mass, and size are estimated. The cryocooler is designed to provide 10 mW of cooling at 150 K to each of the multiple cold heads. The performance of an LSS impeller is measured and the results are used to correlate a compressor design model. These studies demonstrate that a space-borne J-T cryocooler can be produced to provide efficient cooling at extremely low capacities. The cryocooler is lightweight, compact, extremely reliable, and emits negligible vibration. The scope of this paper is the design of the cryocooler, the optimization of the cycle gas composition, and the preliminary test results for an LSS compressor impeller.

INTRODUCTION

Future space applications will require low capacity cryocoolers for distributed cooling of small arrays of infrared detectors, high-temperature superconducting electronics, or payload thermal management. Passive cryogenic radiators are often impractical to integrate with the objects to be cooled and are costly to ground test. Current low-capacity cryocoolers have low thermal efficiency because of relatively large parasitic losses. Joule-Thomson cryocoolers are ideal for applications that require distributed, low-capacity cooling because: (1) the cold heads in the cooler are very compact and thus minimize the parasitic heat leak; (2) multiple remotely located cold heads can share one compressor assembly; and (3) the fluid transport to each cooling site is accomplished at ambient temperature, allowing large separation distances between the compressor assembly and cold heads with minimal performance impact.

Current J-T cryocoolers utilizing linear compressors have limitations for space applications due to high exported vibrations from the compressor.

To address these limitations, Creare began the development of a vibration-free, long-life centrifugal compressor for a J-T cryocooler. The centrifugal compressor utilizes self-acting gas bearings for vibration-free and long-life operation and an LSS impeller with unique clearance shaft seals to achieve a relatively high compression ratio and efficiency at low flow rates. A mixed-gas refrigerant was also used in the J-T cryocooler to further improve performance. Figure 1 shows the main system elements in a J-T cryocooler utilizing centrifugal compressors for distributed heat loads. The compressor assembly consists of two compressors connected in series to achieve a high compression ratio.

CRYOCOOLER DESIGN

The J-T cryocooler system is designed for a hypothetical space application that has ten remotely distributed heat loads. Each heat load requires 10 mW of cooling at 150 K. The cryocooler design effort focused on determining the optimum refrigerant and the corresponding cycle pressure and pressure ratio to maximize the cooler efficiency. This process involves iterative steps between system design and component designs. The iterative process is necessary particularly because the optimum composition of the gas mixture and the actual axial temperature distribution in the recuperator strongly depend on one another.

Cycle Gas Optimization

In general, in order for a J-T cooler to achieve a high thermodynamic efficiency, the isothermal enthalpy differences at the warm and cold ends of the recuperator must be closely matched. That is, the isothermal enthalpy difference between the high-pressure and low-pressure streams near the warm end should be about the same as the isothermal enthalpy difference near the cold end. In this particular case, however, the isothermal enthalpy difference near the cold end needs to be larger than that at the warm end to provide a larger cooling potential; this is necessary in order to accommodate a relatively large axial conduction heat leak near the cold end. The large heat leak is due to a very steep temperature gradient near the cold end of the recuperator, where the heat transfer rate between fluid streams is much higher than that at the warm end. Achieving a larger enthalpy difference at the cold end is accomplished by increasing the concentration of the more volatile component in the mixed gas.

The overall J-T effect of a gas mixture depends on the J-T effect of each constituent and its concentration in the mixture. As system pressure increases, each constituent will have a higher J-T

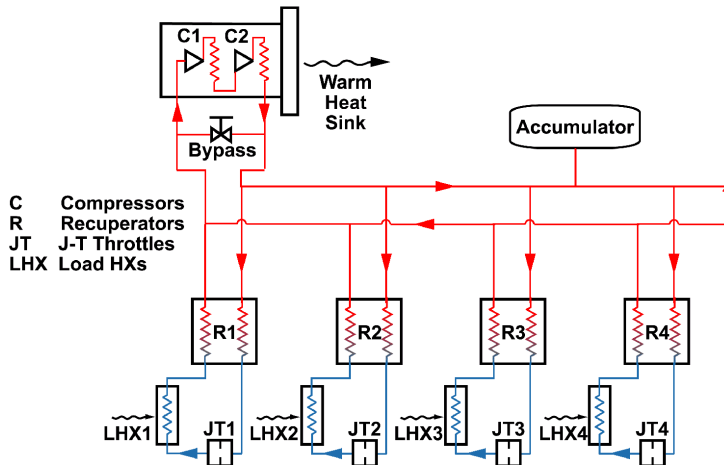


Figure 1. Continuous-flow J-T cryocooler for distributed heat loads.

effect; this tends to raise the overall J-T effect of the mixture. However, increasing the compressor inlet pressure requires the use of a gas mixture with a greater concentration of the more volatile constituent to maintain the same cooling temperature. The more volatile constituent has a lower J-T effect, and therefore tends to reduce the overall J-T effect. The result of these two competing effects is that the maximum J-T effect is at a pressure of about 4.5 bar for the selected constituents and pressure ratio, as shown in Figure 2.

These analyses are for a system with an ideal compressor having a fixed pressure ratio. The efficiency of a real centrifugal compressor will decrease with the operating pressure due to higher internal leakage and drag losses. Therefore, in a real J-T cryocooler, the optimum compressor inlet pressure will be lower than the 4.5 bar value shown before. A pressure of 2.0 bar was therefore selected in our baseline design.

Cryocooler Performance

Table 1 summarizes the system performance with an optimized gas composition and realistic performance estimates for the centrifugal compressor and cold heads. The total input power is 26 W, which corresponds to a cryocooler COP of 0.4% of a Carnot cycle. This value is consistent with the trend of existing data for coolers with much larger capacity, as illustrated in Figure 3.¹ The J-T cryocooler is characterized at two different load values corresponding to a single 10 mW coldhead as well as the entire 100 mW cryocooler. The conceptual layout design for the cryocooler is shown in Figure 4. The total mass of the mechanical cryocooler is 4.5 kg, most of which is associated with the compressors. The mass of each cold head is only 10 grams. The scaling of the cryocooler input power and mass with the number of cold heads is shown in Figure 5. The cryocooler scales very well in terms of performance and mass to higher capacities and number of cold heads.

Cold Head Design

A simple tube-in-tube configuration was selected for the recuperator. The overall O.D. of the recuperator is only 1/16 in. and the length is 16 in. The small diameter reduces the exposed surface area of the recuperator, and thus reduces the parasitic heat leak. The relatively long length reduces the axial conduction heat leak. These features are particularly important for a cold head with a very small cooling capacity, where the parasitic heat leak and axial conduction heat leak can be even higher than the net cooling power. The recuperator achieves a thermal effectiveness of about 0.998. After subtracting the estimated parasitic heat leak from ambient, each cold head has a net cooling power of about 11 mW, slightly greater than our target of 10 mW. A simple capillary tube is used as the flow throttle device to minimize the size and mass of the cold heads.

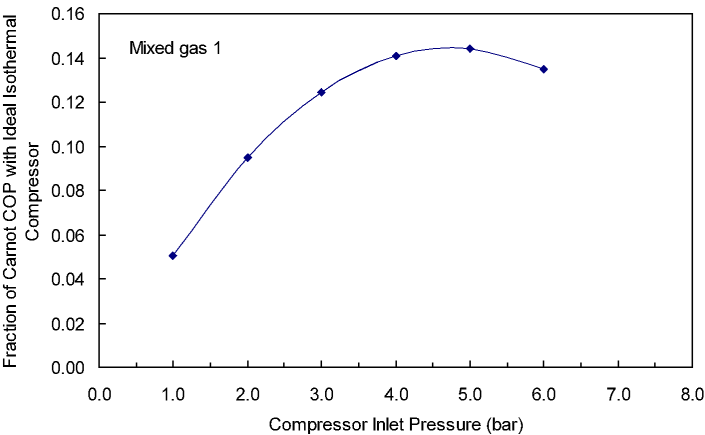


Figure 2. Effect of system pressure on J-T cycle performance.

Table 1. Predicted performance of J-T cryocooler.

Heat rejection temperature	(K)	300
Two-stage compressor pressure ratio	(-)	2.4
Load temperature	(K)	150
Total system net cooling power	(mW)	100
Compressor mass flow rate	(g/sec)	0.019
Ideal isothermal compressor input power	(W)	1.08
Refrigerant cooling capacity	Δh (J/g)	5.3
Compressor efficiency	%	5.1
DC input power	(W)	26
Cryocooler efficiency relative to Carnot cycle	(%)	0.4

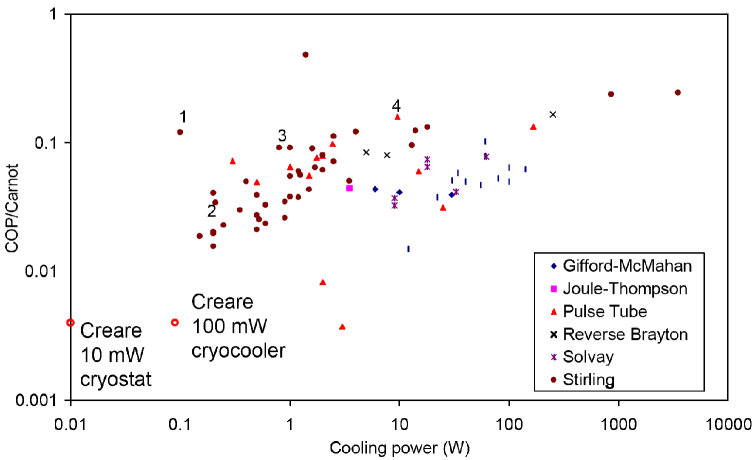


Figure 3. Cryocooler performance comparison.

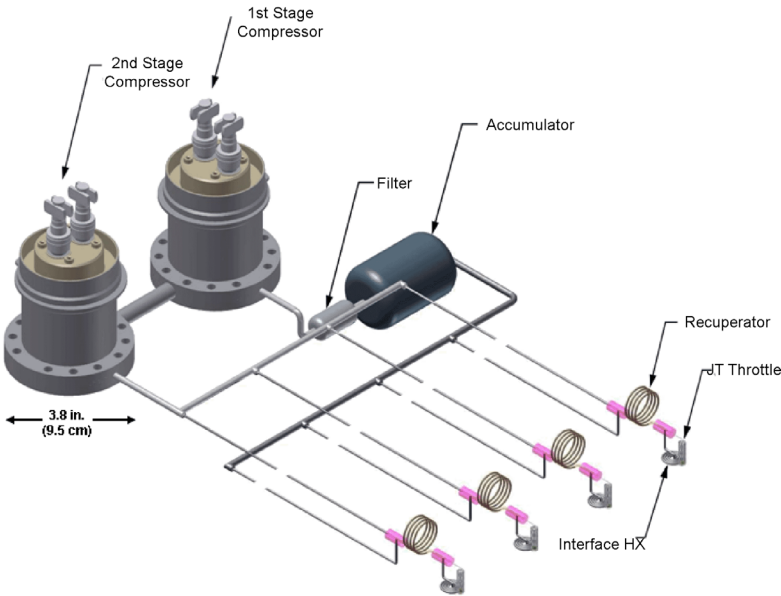


Figure 4. Conceptual design of J-T cryocooler.

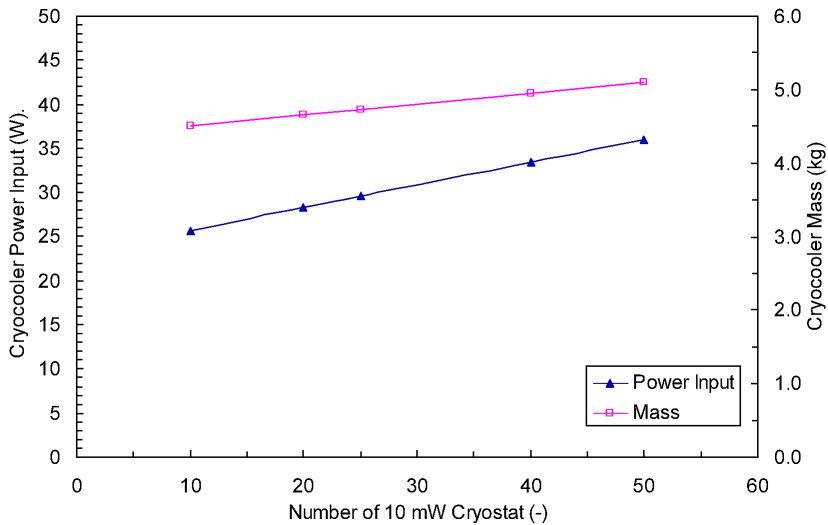


Figure 5. Impact of number of cold heads on cryocooler input power and mass. The cryocooler scales well to higher capacities.

Compressor Design

The volumetric flow rate through the J-T compressor assembly is very small because of its relatively high operating pressure and the low mass flow rate associated with the low cooling loads. These operating conditions call for an LSS centrifugal compressor. The compressor employs a small LSS impeller and operates at a very high rotating speed to achieve a high pressure rise and a low flow rate.

In order to achieve high compressor efficiency, the internal leakage in the compressor must be reduced to a level much lower than the flow rate through the compressor. The typical internal leakage in a centrifugal compressor for a reverse-Brayton cryocooler is typically about 1-2% of the through flow rate. The volumetric flow rate through a low-capacity J-T cryocooler, however, is about two orders of magnitude lower than the flow through a typical reverse-Brayton cryocooler. Therefore, the internal leakage in a centrifugal J-T compressor could be a significant fraction of the through flow if existing rotary seal technology is used here.

To overcome this problem, we developed a floating labyrinth clearance seal for a J-T compressor. A unique feature of this seal is that the inner diameters of labyrinth teeth will self-align with the center axis of a rotating shaft during normal operation. This allows us to use a clearance between the I.D. of the labyrinth seals and the O.D. of the shaft that is much smaller than the clearance in existing labyrinth seals, thereby significantly reducing internal leakage.

The J-T compressor assembly consists of two centrifugal LSS compressors connected in series and operating at 8000 rev/s to provide an overall pressure ratio of about 2.4. In a flight configuration, the compressors are bolted to the spacecraft heat rejection interface. Heat generated inside the compressor is conducted through the compressor housing, and then to the spacecraft heat rejection interface. Because the flow rate through the compressor is very small and the aerodynamic power input to the mixed gas is only about 1.2 W, the outlet gas will rapidly cool down to the housing temperature as it flows past the internal structure, eliminating the need for a separate aftercooler.

COMPRESSOR TESTS

Tests were performed to quantify the performance of an LSS centrifugal compressor. The goal of this activity was to prove that an LSS centrifugal compressor can achieve the flow rate, pressure ratio, and efficiency needed for a mixed-gas J-T cryocooler. The key measurements taken were (1) measurements of the static seal leakage and (2) head-flow curves for the LSS compressor impeller.

The schematic of the test setup is shown in Figure 6. The setup consists of two main components: (1) a commercially available spindle, with a rotational speed up to 250,000 rpm (4200 rev/s) to drive the rotor; and (2) a test chamber that houses the rotor and floating seals. To reduce the complexity of sealing the test chamber at the shaft pass-through, we tested with a ambient pressure gas with a high molecular weight (SF_6), instead of the actual high-pressure gas. The pressure in the test chamber was maintained slightly above the ambient pressure to prevent air from flowing into the test rig.

Leak Tests

The leakage through the floating seals was directly measured in the test setup by supplying a high-pressure gas at the outlet and using a shaft without an impeller. The backward flow rate from the outlet to the chamber is equivalent to the leakage rate across the seals. The measured and predicted leakages are shown in Figure 7. The error bars shown represent the measurement uncertainty. There is an uncertainty in the predicted leakage because of uncertainty in the exact clearance and concentricity between the shaft and seal. This clearance uncertainty is due to the uncertainties in the physical measurement of the shaft and seal diameters that are of the same order of magnitude as the clearances. The measured leakage was slightly higher than the predicted leakage for all clearances, but consistent with predicted values when the uncertainties of the predictions (0.0001 in. clearance uncertainty) are also considered.

Using floating seals with a clearance of about 0.0002 in., no mechanical interference was found for a precision-ground shaft spinning at speeds of about 150,000 rpm. From this result, along with the leakage tests, it was concluded that a radial clearance value of 0.0002 in. to 0.0003 in. was physically feasible in that the shaft and seals could be manufactured to the required tolerance and assembled. A seal clearance of approximately 0.0002 in. was then used in the compressor design.

Compressor Pressure Rise

The performance of the impeller was measured for one seal-clearance value of 0.0005 in. The impeller test results are shown in Figure 8. Here, pressure rise is shown as a function of rotation speed, with error bars representing the measurement uncertainty. The measured pressure rise was slightly lower than the predicted pressure rise for a given rotational speed, but the trend with speed is consistent with the predictions. The lower pressure rise may be attributed to larger-than-predicted internal leakage and over-restricted flow channels in the impeller. Assuming the measured pressure rise was not improved upon by future design improvement, the impeller would need to operate at a 9% higher tip speed, which would result in an increase of electrical input power from 26 W to 31 W.

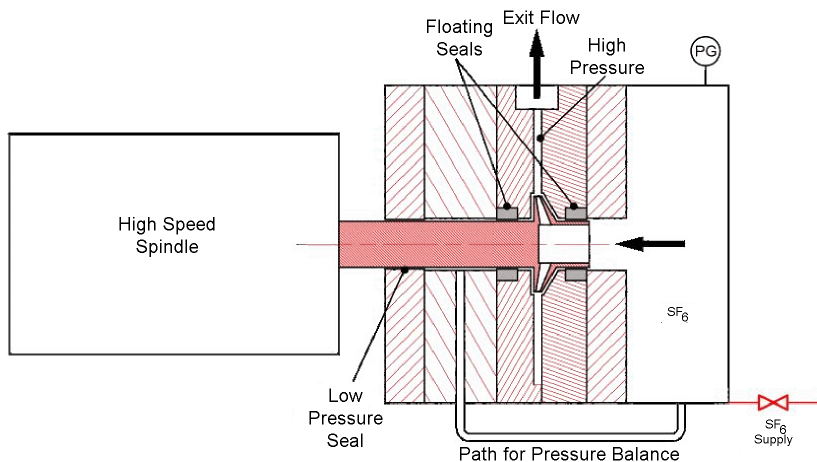


Figure 6. Schematic of compressor test setup

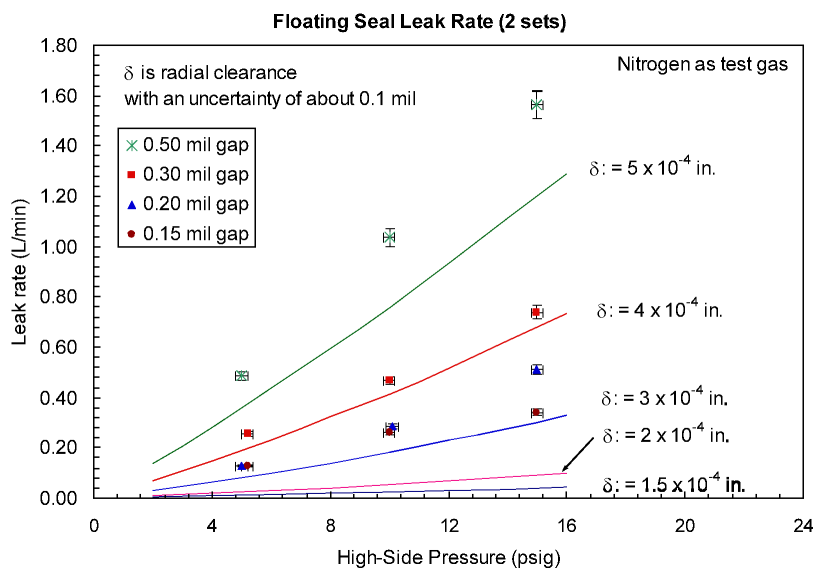


Figure 7. Seal leakage test data and predictions.

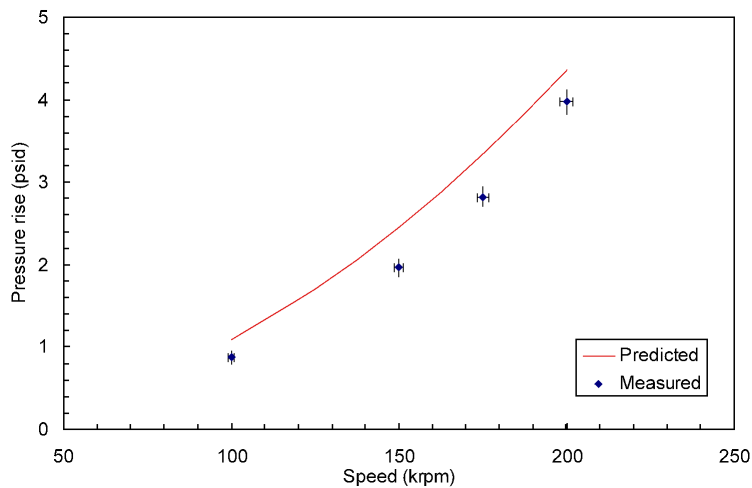


Figure 8. Measured and predicted pressure rise.

CONCLUSIONS

This study describes a new type of space cryocooler to provide extremely low capacity cooling simultaneously to multiple cold heads. The approach is a J-T cryocooler that utilizes an LSS centrifugal compressor. Cycle analysis was carried out to identify the optimum system operating conditions as well as the optimum composition of the cycle gas. An LSS compressor and a compact cold head were designed, and their performance, mass, and size were estimated. The resulting system provides 10 mW of cooling at 150 K at each of 10 cold heads for only 26 W of total dc input power. This performance corresponds to a COP of 0.4% of a Carnot cycle, a reasonable value for a very low capacity cryocooler. The compressor impeller was optimized to attain high efficiency at extremely low flow rates. The proof-of-concept tests demonstrated the feasibility and characterized the performance of the LSS centrifugal compressor. The only uncertainty is the total compres-

sor input power. However, the uncertainty has been bounded by the results of these tests. The total mechanical cryocooler mass is estimated to be 4.5 kg. Each cold head weighs only 10 grams. The system scales well to higher capacities, requiring only 0.25 W of additional input power for each 10 mW cold head.

The technology discussed in this study can be applied to low-capacity cooling at temperatures ranging from 200 K to 70 K. In addition, it can be used as the lowest stage cooler in a multistage, low-temperature cryocooler with cooling temperatures below 10 K to provide efficient, remote cooling.

ACKNOWLEDGMENT

The support and guidance provided by the Missile Defense Agency and the Air Force Research Laboratory are gratefully acknowledged.

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