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RAYTHEON RSP2 CRYOCOOLER LOW TEMPERATURE TESTING AND DESIGN ENHANCEMENTS

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ABSTRACT

The High Capacity Raytheon Stirling / Pulse Tube Hybrid 2-Stage cryocooler (HC-RSP2) was originally developed to provide simultaneous cooling at temperatures of 85 K and 35 K. During testing performed in 2008 it was demonstrated that this stock-configuration cryocooler is capable of providing significant amounts of heat lift at 2nd stage temperatures as low as 12 K, and modeling indicated that minor changes to the 2nd stage inertance tube / surge volume setup could yield improved performance. These changes were implemented and the cooler was successfully retested, producing >350 mW of heat lift at 12 K. A comprehensive redesign of the system has been performed, the result of which is a robust 2-stage cryocooler system that is intended to efficiently produce relatively large amounts of cooling at 2nd stage temperatures <12 K. This cryocooler, called the Low Temperature RSP2 (LT-RSP2) will be fabricated and tested over the next 12 months. This paper reports on the recently-completed test activities, as well as details relating to the system redesign. Expected performance, mass and packaging volume are addressed.

KEYWORDS: pulse tube, Stirling, aerospace, hybrid

INTRODUCTION

The development of practical low temperature (sub-15 K) closed-cycle cryogenic coolers is being actively pursued by Raytheon Space and Airborne Systems due to the beneficial detector chemistries that are enabled by such a system. The kernel of the ongoing research is the Raytheon Stirling / pulse tube two-stage (RSP2) cryocooler architecture, which in the past has been primarily optimized for operation at higher temperatures such as 110 K / 40 K and 85K / 35 K (note: xx K / yy K is shorthand for xx K Stirling first stage temperature and yy K pulse tube second stage temperature). Over the last 18 months the 85 K / 35 K High Capacity RSP2 (HC-RSP2) has been modified and tested for operation at 2nd stage temperatures below 15 K, and the lessons-learned from that activity are being applied to a new design, the Low Temperature RSP2 (LT-RSP2).

The LT-RSP2 expander module shares many mechanical design features with the existing and well-understood RSP2 cryocoolers, however the cold head portion has been fully optimized to efficiently produce refrigeration at first and second stage temperatures of ~60 K and ~12 K, respectively. The compressor module of the LT-RSP2 has also been optimized, resulting in a module that is physically smaller and architecturally simpler than the existing HC-RSP2 compressor while being significantly more powerful and efficient.

LOW TEMPERATURE HC-RSP2 TESTING

Several RSP2 cryocoolers have been designed and extensively tested, most notably the 2nd generation Medium Capacity (MC-RSP2) and High Capacity (HC-RSP2) versions. The MC-RSP2, produced using internal funding, is a 200 Watt-class machine that is optimized for 1st and 2nd stage temperatures of approximately 110K and 40K, respectively. The HC-RSP2, shown in FIGURE 1, is a product of Air Force Research Laboratory and Missile Defense Agency funding. The HC-RSP2 is a much larger 500 Watt-class machine primarily intended to produce refrigeration at 85 K and 35 K. The HC-RSP2 simultaneously produces 16.1 W at 85 K and 2.6 W at 35 K, with the ability to actively *load shift* from one stage to the other [1]. This capability, uniquely inherent to linear coolers with the Raytheon-patented Stirling / pulse tube hybrid architecture, allows for the ratio of refrigeration at the two stages to be altered via simple software commands, without a significant loss of overall efficiency. This capability is extremely valuable to systems utilizing two-stage cryocoolers in that it is unlikely that the refrigeration ratio of the two stages used to generate the thermodynamic design will be the actual ratio required of the cooler in operation. A two-stage cryocooler solution without the ability to load shift will be forced to operate in an off-nominal condition and may, as a result, suffer significantly in terms of efficiency.

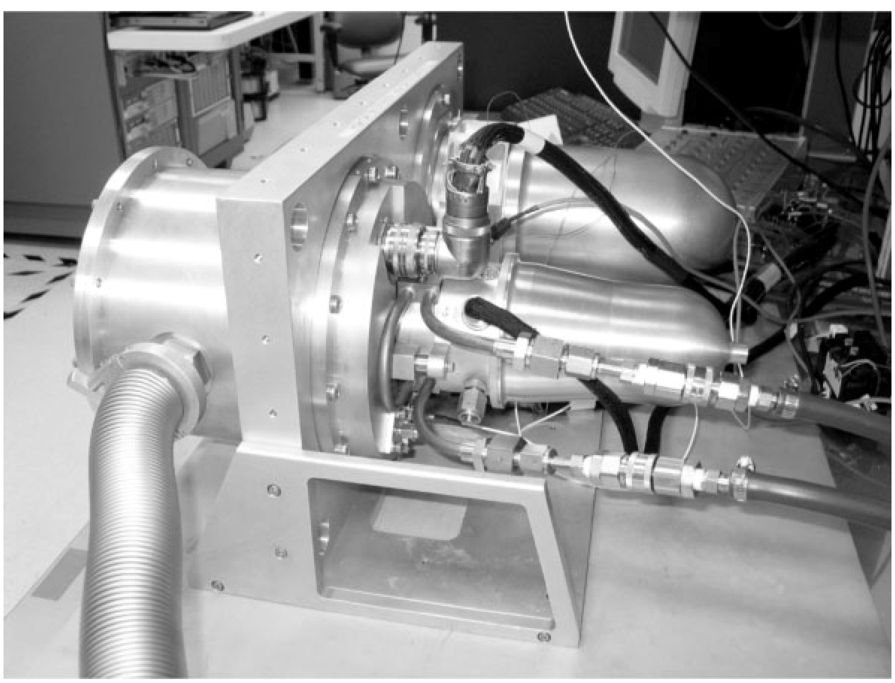

FIGURE 1. HC-RSP2 cryocooler in test stand undergoing benchtop characterization testing.

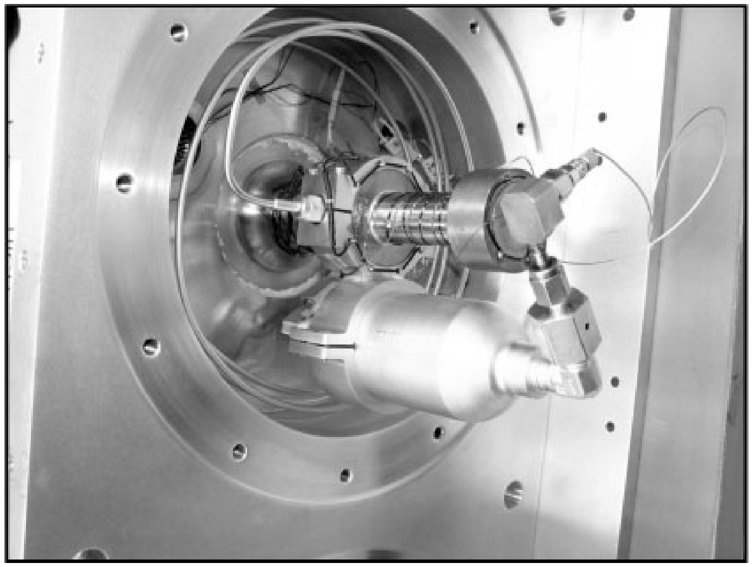

FIGURE 2. "Stock" HC-RSP2 cold head configured for 35 K / 85 K optimum performance for 2007 testing. Commercial surge volume and tubing used for convenience. Custom compact, packaged solution provided for space-flight applications.

Low temperature load line and no-load tests were performed in late 2007 during the process of correlating the HC-RSP2 to the thermodynamic system model. It was found that, despite being optimized for much higher temperatures, the HC-RSP2 was capable of reaching 2nd stage temperatures well below 15 K; in the benchtop test configuration the HC-RSP2 was capable of producing greater than 200 mW of refrigeration at a 2nd stage cold tip temperature of 12 K. The cold head configuration (i.e., the inertance tube and surge volume) was left unchanged from the 35 K / 85 K as-delivered configuration; see FIGURE 2. Though the input power required to generate the refrigeration exceeded 500 W, this was seen as an extremely positive result given that the 2nd stage had been designed expressly for operation at significantly higher temperatures. In particular, the ability of the regenerator to function below 15 K was encouraging given the rapidly decreasing heat capacity of most materials at these temperatures. Further, the HC-RSP2 moving mechanisms are designed to operate at frequencies in the range of 40-50 Hz, well above the frequency range ideal for low temperature operation. During low-temperature testing the system operational frequency was set to a system-level optimal value, i.e., the compressor and expander were operated below the structural resonant frequency in order to improve the regenerator effectiveness; the efficiencies of both modules were thus well below what would be expected of a fully-optimized system.

During 2008 the Raytheon Space Cryocooler product line received inquiries as to our abilities in the sub-15 K range, and the results of the 2007 HC-RSP2 testing were presented to the interested parties. Understanding the limitations of the as-built HC-RSP2 when operating at 2nd stage temperatures below 15 K, a proposal to lightly modify the machine for increased performance was put forth and agreed-to. The 2nd stage surge volume and inertance tubes were re-optimized specifically for operation below 15 K, the system fill pressure was modified, and a new set of multi-layer insulation (MLI) was fabricated in order to minimize the effects of parasitic radiation. The low temperature cold head test configuration is shown in FIGURE 3 (without MLI). Thermodynamic modeling indicated that these modifications would yield an increase in 2nd stage refrigeration of approximately 200 mW, from the previously-tested 220 mW to 410 mW. The modifications were performed and the unit was retested in late 2008.

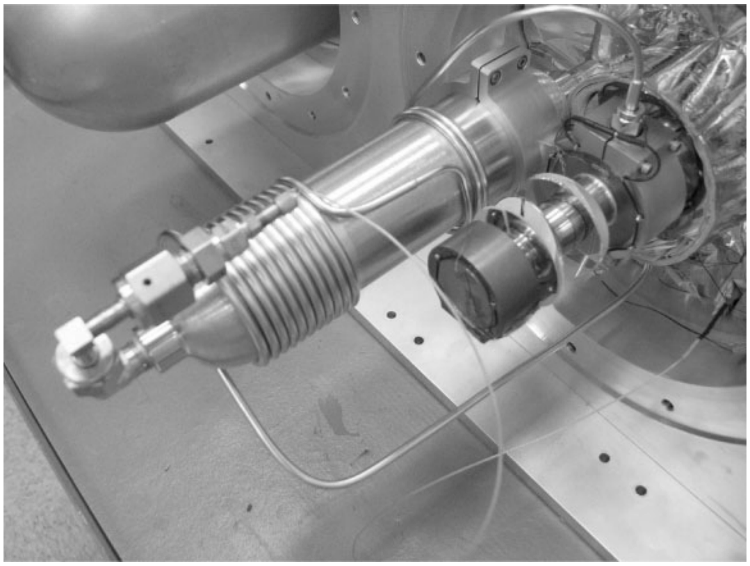

FIGURE 3. 12K / 60K cold head used for 2008 testing. Only change relative to Figure 2 is the inertance tube and surge volume; same Stirling / pulse tube cold finger.

During testing the machine was allowed to stabilize at a first stage temperature of 60 K and a second stage temperature of 12 K, after which a frequency sweep was performed. Load shifting was employed to maintain the first stage heat load at 60 K throughout the frequency sweep, and the second stage load was allowed to change in order to maintain a cold tip temperature of 12 K. The observed system optimal frequency matched the modeled value within a Hertz. This frequency was significantly lower than that of the original HC-RSP2 configuration, providing substantially increased cold head efficiency at the cost of ~10% of compressor efficiency. Further testing was performed at slightly higher power in order to characterize the second stage heat lift capacity as a function of temperature. Capacity at 12 K was found to be 380 mW. This represented a large increase compared to the value of 220 mW obtained during testing of the original HC-RSP2 configuration, and agreed extremely well with the predicted value of 410 mW. The observed no-load performance of ~10.4 K was limited by the heat rejection ability of the expander module; additional compressor capacity was available to drive the temperature lower, however it was not possible to maintain a constant expander housing temperature at higher input powers.

The results of the HC-RSP2 low temperature testing are provided in TABLE 1. This includes the 2008 data with the "custom" low temperature inertance tube and surge volume (Fig. 3) and the 2007 test results obtained with the "as built" test configuration, that is, with the original 35 K / 85 K inertance tube and surge volume (Fig. 2).

Two noteworthy pieces of information were gained in the course of the modified HC-RSP2 testing. First, a significant amount of confidence was gained in the ability of the system thermodynamic model to accurately predict performance at temperatures below 15 K. Second, the ability of a two-stage RSP2 type cold head to operate efficiently at low temperatures, using existing regenerator technology, was proven. The HC-RSP2 had proven to be a valuable test asset, despite its known deficiencies at low temperatures. These deficiencies are a result of the optimization of the machine at higher temperatures, and are addressable in a straightforward way.

TABLE 1. HC-RSP2 Low Temperature Test Data.

Cooler Configuration:	Units	Stock HC-RSP2	HC-RSP2 with 12K-Optimized Surge Volume / Inertance Tube				
Test Data / Model:		Test	Model:	Test	Test	Test	Test
Expander Housing Temperature:	K	300	296	296	296	296	295
1st Stage Temperature:	K	65.5	65	65	65	65	65
1st Stage Heat Lift:	W	1.11	1.00	1.00	0.93	0.98	0.96
2nd Stage Temperature:	K	12.0	12.0	12.0	11.4	10.5	12.0
2nd Stage Heat Lift:	W	0.22	0.41	0.38	0.23	0.00	0.29
Compressor Input Power:	W	566	551	551	552	555	505

LOW TEMPERATURE HC-RSP2 DESIGN

Following the encouraging results of HC-RSP2 low temperature testing in 2008, internal research and development funding was awarded in 2009 for the purpose of designing, fabricating and testing a fully-optimized Low Temperature RSP2 cryocooler (LT-RSP2). While this machine draws heavily from the HC-RSP2, known low-temperature deficiencies have been addressed and improvements have been implemented in both the compressor and expander modules in order to improve performance and producibility.

FIGURE 4 contains solid model images of the HC-RSP2 and LT-RSP2 compressor modules with the module masses indicated. The LT-RSP2 carries over several features from the legacy compressor design, including the effective heat rejection system and the inclusion of piston position sensors. As with the previous design, the new compressor includes secondary motor coils that allow for extremely high-fidelity vibration cancellation forces to be injected into the motor despite the high-power nature of the system. Typical of most long-life linear cryocoolers, the LT-RSP2 compressor contains dual-opposed, flexure-borne moving mechanisms in order to minimize vibration and ensure robustness. Though significant similarity exists between the two compressors, the LT-RSP2 version does incorporate an advanced internal architecture with two major benefits.

First, the parts count of the new compressor design is approximately 40% lower than that of the predecessor, with an associated reduction in build and alignment complexity. This improvement was achieved largely by incorporating several design features from our low cost Dual Use Cryocooler architecture [2]. The internal components have been arranged into two major subassemblies, and all of the fine alignment operations take place at the subassembly level instead of the module level; the LT-RSP2 compressor is simpler, more robust and more cost effective than the previous design. Second, the LT-RSP2 compressor mechanical design permits the use of much more powerful motors while achieving a reduction of overall package size. Specifically, the new compressor increases motor strength by a factor of two while also allowing the module length to be reduced by approximately 30% relative to HC-RSP2. The increase in motor strength is critical for the LT-RSP2 cryocooler due to the necessity of maintaining high efficiency while operating at low frequency and high power. This design is applicable to other, higher-frequency designs as well and is anticipated to be able to deliver excellent efficiency while operating with input powers well over 600W.

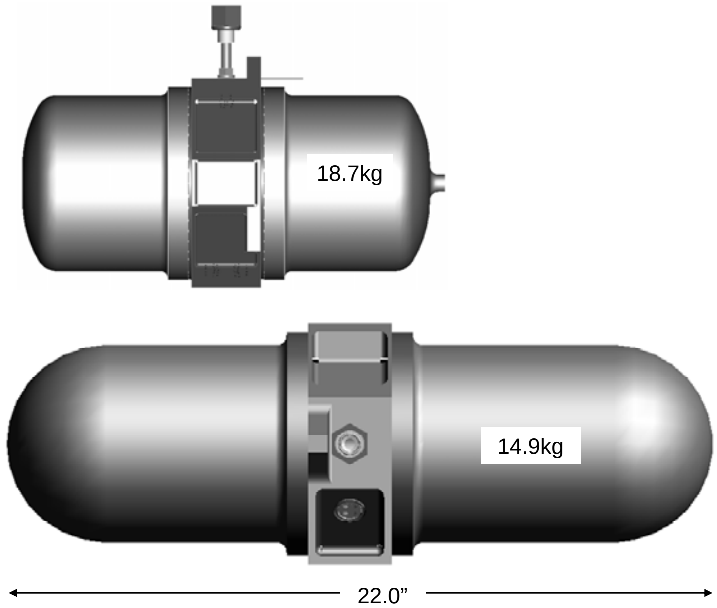

FIGURE 4. Comparison of HC-RSP2 (bottom) and LT-RSP2 (top) compressor modules. Scale same for both images.

The LT-RSP2 expander module shown in FIGURE 5 is essentially a heavily optimized version of the HC-RSP2 expander (also pictured). The warm-end internal mechanical and motor designs are largely unchanged, as no architectural changes were required in order to support the newly-optimized cold head. The heat rejection portions of the LT-RSP2 expander housing have been enlarged in order to accommodate significantly larger bolted interfaces to which heat pipes or thermal straps can be directly attached. The gas flow paths within the module’s warm end have been redesigned in order to promote the effective transfer of heat from the working gas to the housing; these internal modifications are based on the results of computational fluid dynamics modeling efforts performed during the analysis of previous-generation machines.

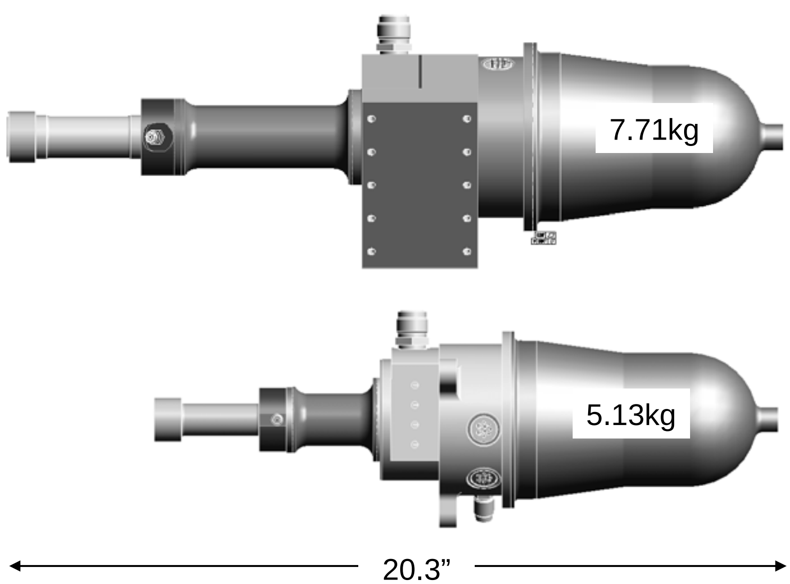

FIGURE 5. Comparison of HC-RSP2 (bottom) and LT-RSP2 (top) expander modules. Scale same for both images.

Though the LT-RSP2 cold head geometries have been completely reoptimized in order to maximize efficiency at first and second stage temperatures of 60-70 K and 10-15 K, the cold end of the LT-RSP2 expander is architecturally identical to that of the HC-RSP2. The first stage is an actively-driven Stirling system while the second stage consists of a concentric pulse tube arrangement employing a cryogenic inertance tube / surge volume phase shifter. For reasons of risk reduction the regenerator types, materials and build processes have been left unchanged from those of the HC-RSP2. The material used to construct the pressure vessel thin-walled tubes has been changed to a lower thermal conductivity alloy in order to reduce the parasitic load of the second stage, and wall thicknesses have been appropriately modified to ensure a high first mode frequency (>300 Hz) and significant safety margin (1.25) with respect to proof-pressure testing.

FIGURE 6 contains two load curves illustrating the expected LT-RSP2 second stage capacity as a function of temperature at medium and high input powers. The low-temperature optimized machine will be capable of providing capacity and efficiency that is vastly superior to that than the HC-RSP2 when operated below 15 K. For instance, it is expected that the LT-RSP2 will be able to match the HC-RSP2's 220 mW at 12 K heat lift while requiring ~50% less input power. Alternatively, for a similar amount of input power the LT-RSP2 will be able to produce much high levels of heat lift while completely avoiding any heat rejection issues related to operating at power levels exceeding 550 W. The high-power second stage no load temperature of the LT-RSP2 is expected to occur well below 10 K, with a heat load of 5 W at 65 K applied to the first stage simultaneously. FIGURE 7 contains a high power load shifting plot for the LT-RSP2, illustrating the ability of the machine to actively shift refrigeration capacity between the two stages through simple software commands issues to the drive electronics. The first stage has a load shifting range of ± 42 % while the second stage has a range of ± 26 %. The load shifting curve was generated under the condition of nominally constant compressor power (+/- 5 W), and load shifting occurs with nominally constant overall efficiency (+/- 10%).

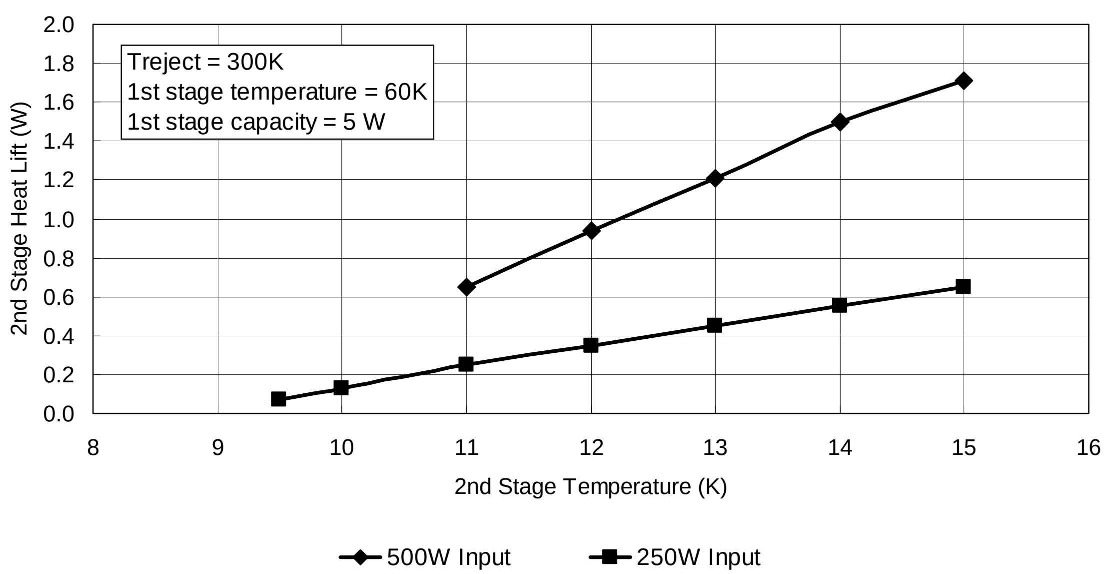

FIGURE 6. LT-RSP2 load second stage load curves.

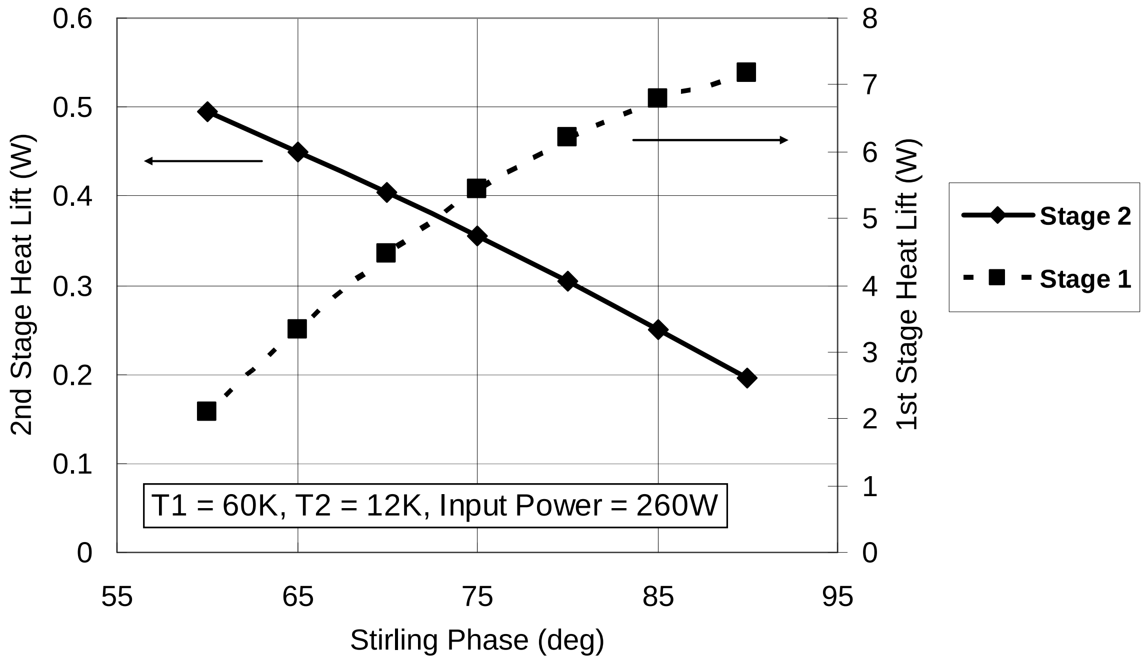

FIGURE 7. LT-RSP2 load shifting curves for moderate power (260 W input) operation. Capacity is software-commandable from 200 mW to 500 mW for 2nd stage and from 2.1 W to 7.2 W on the 1st stage over a constant input power control range.

DISCUSSION

Regenerative cryocoolers have traditionally been forced to operate at low frequency (<5 Hz) to achieve efficient operation in the 10 K regime and below to overcome the inherent design challenge posed by: 1) the rapid fall off in the specific heat of traditional regenerator materials, such as stainless steel, and 2) the limited heat exchanger matrix geometries available from rare earth elements and metallic compounds, which maintain high specific heat in this temperature range but are generally only available in powder form. There have been numerous papers on this subject over the last ten to fifteen years. Recently, this challenge was well discussed by Ladner, et. al [3] in which a theoretical parallel plate regenerator was posed as a means to achieve the desired low porosity while maintaining an acceptably low pressure drop. As another example, packed beds of rare earth powders with varying diameters have been used successfully to achieve lower porosities than the geometric 38% obtained with a packed bed of spheres of identical size [4]. Despite the interest and advancement in the art, however, a compact, high frequency, high efficiency 10 K-class linear cryocooler, such as would be required for a space-borne application, has yet to emerge.

For space applications, much of the recent focus on low temperature cryocoolers, particularly in the 6 K range, has involved the use of linear (Stirling or pulse tube) upper stage coolers and a Joule-Thomson cold stage [5-7]. This avoids the regenerator problem, but it introduces a new set of undesirable characteristics:

- Additional mechanical compressor;
- Additional electronics to drive the compressor;
- Inherent inefficiency of the purely dissipative J-T cycle.

The additional mechanical and electrical components increase cost, increase mass, and reduce reliability. In the 10 K to 12 K temperature range of interest for the LT-RSP2, the use of a J-T cold stage is particularly problematic because of the low Joule-Thomson coefficient of helium at these temperatures. See FIGURE 8. Very large pressure ratios are required to achieve appreciable temperature drops, which complicates the compressor design and decreases thermodynamic cycle efficiency.

A 3-stage, high frequency (30 Hz) pulse tube capable of carrying 200 mW at 10 K simultaneously with 1.3 W at 51 K for about 370 W input power was recently presented [8]. This cryocooler achieves respectable 10K performance in a compact size while avoiding the complexity of the auxiliary J-T cryocooler system. It is thus similar in approach and objective to the LT-RSP2. The LT-RSP2 is expected to compare favorably to the referenced 3-stage pulse tube cryocooler with significantly higher efficiency, more upper stage capacity, and more operational flexibility arising from the inherent Stirling/pulse tube load shifting capability.

LOW TEMPERATURE RSP2 STATUS

The LT-RSP2 thermodynamic, electromagnetic and solid modeling efforts (all performed in-house at Raytheon Space and Airborne Systems) were completed in early June of 2009. Long-lead purchase orders have been placed, and the remainder of the piece part orders is in procurement. The majority of the test equipment and support equipment necessary for testing has been identified, and remainder is being procured in parallel with the cooler piece parts. The LT-RSP2 will be assembled in the 4th quarter of 2009, with the goal of demonstrating an initial cool down early in 2010.

CONCLUSION

Raytheon has demonstrated that a straightforward, two-stage, high frequency, linear cryocooler can be used for 10K to 12K refrigeration. Initial testing performed with a cryocooler optimized for 35K refrigeration yielded 380 mW at 12K for 550W input power. The fabrication of a Stirling/pulse tube hybrid cryocooler optimized for low temperature (10K to 12K) operation is now underway with projected performance of 350 mW at 12K and 5 W at 60K for 250 W input power. The LT-RSP2 provides an attractive option for space cryocoolers in this temperature range because it avoids the complexity of an auxiliary J-T cryocooler while providing the operational controllability and flexibility inherently available in the RSP2 architecture.

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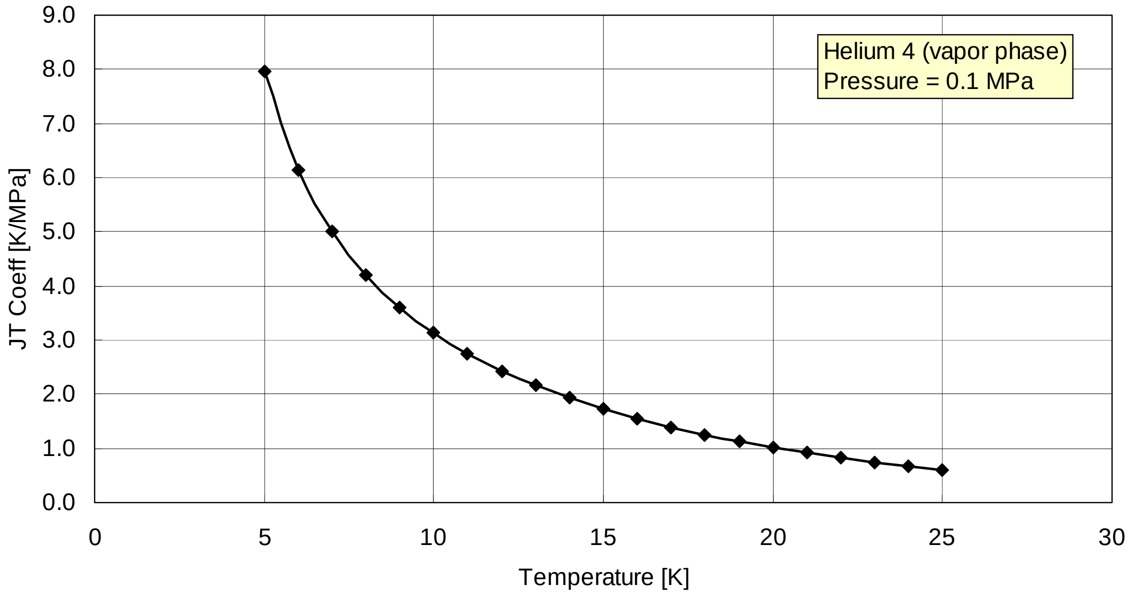

FIGURE 8. Joule-Thomson coefficient for helium 4 at P = 0.1 MPa. Rapidly declining J-T coefficient in the 10 K to 12 K range complicates efficient cooling by throttling.

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