Cryocooler with Cold Compressor for Deep Space Applications

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ABSTRACT

Future NASA deep space and planetary missions will require cryocoolers providing cooling capacities of 0.3 W at 35 K with heat rejection temperatures as low as 150 K and input powers up to 10 W. Presently, there are no qualified cooler systems operating at this low rejection temperature.

Madison Cryogroup was awarded a Phase I SBIR contract from NASA GSFC to demonstrate this capability. Lockheed Martin's STAR lab was subcontracted to provide the compressor for this development using its "Mini" pulse tube cooler with its integral cold head as a concept demonstration.

Initial concept feasibility tests were conducted with the compressor operating at 150 K; cooling of 0.17 W at a cold tip temperature of 35 K was achieved, although the system was not optimized for the 150 K compressor operation.

Predictions of the required modifications to optimize the performance were conducted and indicate that the goal of 0.3 W at 35 K with 10 W of power input can be achieved at the rejection temperature of 150 K.

This paper presents test results for the tested "unoptimized" system and predictions for a compressor optimized for 150 K. The team was recently awarded a Phase II SBIR contract to develop an Engineering Model (EM) of the system; plans for this phase are also described.

INTRODUCTION

The compressor for operation at 150 K is to be based on LMATC's existing Mini Compressor technology. This line of pulse tube compressors employs a linear-motor, flexure-bearing, clearance-seal architecture with opposed pistons for low vibration. The Mini pulse tube cooler was initially developed by LMATC under a contract awarded by NASA-GSFC. The cooling requirements were 0.3 W at 65 K with 15 W of compressor power at a rejection temperature of 310 K (cooling performance was assessed from 250 to 310 K). In all, the flexure-bearing Mini compressor has been demonstrated to be a reliable, robust technology. It uses a moving magnet compressor that has simplified assembly, reduced cost, and enhanced reliability compared with the standard Oxford-heritage compressor with moving coil.

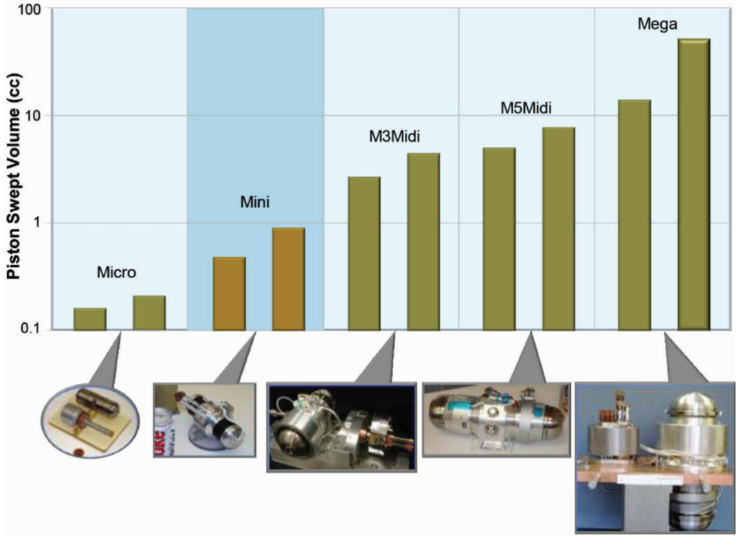

Figure 1. LM compressor family spans a 400 to 1 range in swept volume.

Researchers at LMSTAR Labs have developed linear compressors and cryocoolers for more than 30 years; the range of compressors developed is shown in Figure 1.

PHASE I DEVELOPMENT

Although not optimized for these operating conditions, Lockheed's existing "Mini" cryocooler was utilized in this Phase I SBIR to demonstrate the viability of operating the compressor with a 150 K heat rejection temperatures and achieving a significant fraction of the required cooling at 35 K.

Configuration of Existing "Mini Cryocooler"

Figures 2 and 3 are photographs of the Mini compressor and its associated U-tube pulse tube coldhead. The weight of the cryocooler shown is 1.2 kg, with a compressor envelope of 55 mm D × 184 mm L. Eight of these units were built and tested including three units delivered to NASA GSFC. The NASA unit has been qualified to TRL6.

The compressor incorporates several self-aligning features for the piston/motor/flexure assembly and has a low part count, which simplifies and shortens assembly time. The critical piston-cylinder tight clearance seal utilizes a simple alignment adjustment mechanism that rapidly and repeatedly performs this task. No electrical feed-throughs penetrate into the working gas space. The moving magnet linear motor with external potted coil eliminates most of the organics from the working gas space.

Figure 2. Lockheed Martin's Mini compressor

Figure 3. U-Tube Cold Head

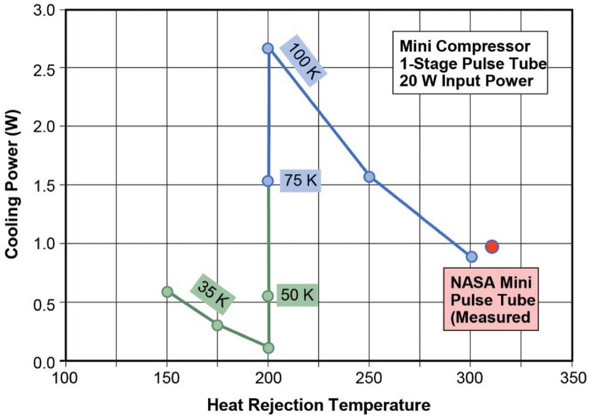

Figure 4. Predicted cooling of optimized system over a range of heat rejection temperatures

Predicted Performance of Optimized System at 150K Compressor Temperature

Modeling studies were conducted at both MCG and LM to predict the performance of the optimized mini pulse tube cooler at 150 K heat rejection temperature. The programs were run for Lockheed Martin's Mini-class compressor which was utilized for the testing. The models were used to explore a complete range of parameters. The variables that were varied were the following: 1) Charge pressure, 2) Operating pressure, 3) Piston diameter, 4) Operating stroke, 5) Rejection temperature, 6) Inertance tube dimensions 7) Regenerator material, 8) Dimensions of cold head tubes, 9) Operating frequency, and 10) Input power. These coupled variables were simultaneously varied in the models to establish their effect on cooling load at various temperatures and power input. Emphasis was placed on cooling at 35 K with minimum power input at the 150 K heat rejection temperature.

The predicted cooling power for the optimized system is shown in Fig. 4. Data are presented for a range of cold head temperatures, and also shown is the measured performance of the original pulse tube system developed under NASA GSFC funding. The performance predictions are for the existing regenerator construction which utilizes stainless steel gauzes. The remaining parameters, such as operating pressure, piston diameter, and regenerator and pulse tube sizes were optimized to achieve maximum cooling. These predictions indicate that the optimized cooler should deliver 0.6 W cooling at 35 K with 150 K heat rejection temperature and 20 W of compressor power input.

One critical consideration is the optimization of the working gas pressure at the 150 K heat rejection temperatures and how it relates to the fill pressure at room temperature. Figure 5 shows the optimum operating pressure at a range of heat rejection temperatures and cold tip temperatures

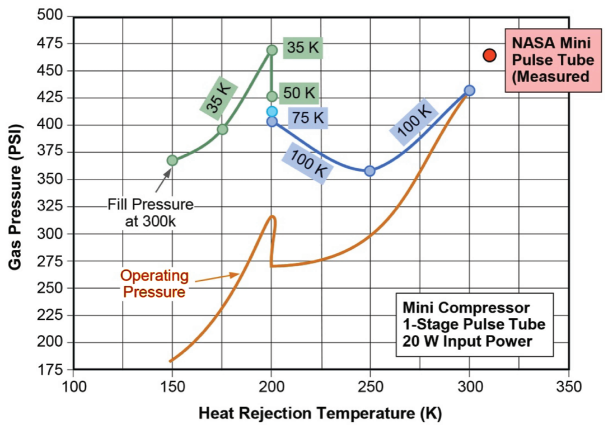

Figure 5. Optimum charge and operating pressure for a range of cold tip and heat rejection values

Figure 6. Overall cryostat system

Figure 7. Cold plate for mounting cryocooler

from 35 K up to 100 K. The upper curve shows the charge pressure at the fill temperature of 300 K, which is required to give optimum operating pressure when the system is cooled to the heat rejection temperature and cold tip temperature shown. The highest optimum charge pressure for any combination of cold tip and heat rejection temperatures is 475 psia, which is close to the original design pressure of the mini pulse tube system. At the baseline cooling goal of 35 K cold tip and 150 K rejection temperature, the fill pressure at 300 K is 370 psia, which leads to the optimum operating pressure of 185 psia. This pressure is well below the proven capability of the system.

Phase I Testing of the "off the shelf" Cryocooler System

Prior to testing of the cryocooler, a study was made of the various components that might affect the performance of the system when operating at cryogenic temperatures. A preliminary survey of the system components was made and the following components were identified for screening tests before subjecting the complete system to cryogenic temperature.

1) Motor coil assembly: This assembly was one of the only assemblies with materials having various coefficients of thermal expansion. The complete housing assembly with coil and magnetic iron works was cooled to LN2 temperature while coil resistance measurements were made. There was no indication of any issue before during or after the LN2 cooling.

2) Position sensor: The Hall effect position sensor was LN2 immersed to establish survivability. It returned to the original settings and provided a signal during cooling to LN2. Accuracy at low temperatures was not established, but will be accomplished during Phase II activities.

3) Piston/Cylinder assembly: This assembly contained some epoxy bonded elements so it was also LN2 cooled to determine the integrity of the bonds. There was no degradation of the bonds. After these component tests provided confidence in the viability of cryogenic operation, we started setting up for the complete system tests. In order to provide the low rejection temperatures in a controlled way, we utilized a vacuum container that had a cold plate cooled by a Gifford McMahon cooler. The complete assembly was welded, with the exception of the final closure on the fill port which had an epoxy bonded cap over it.

The configuration of the cold chamber is shown in Figs. 6 and 7. Figure 6 shows the overall system, while Fig. 7 shows the cold plate utilized for mounting the cryocooler.

The cryocooler, mounted to the cold plate, is shown in Fig. 8. The U-tube cold head is pointing down. A rack of electronics was utilized for testing and contained the drive electronics for the cryocooler and all instrumentation, including position sensor read out, operating frequency, temperature sensors, and piston motion. Figure 9 shows the rack electronics with the cryocooler during early check out.

Tests were conducted on the existing cooler at rejection temperatures from 300 K to 130 K to determine the ability of the compressor to operate at these reduced heat rejection temperatures and to determine the cooling achieved at 35 K cold tip temperatures. The results are shown in Fig. 10.

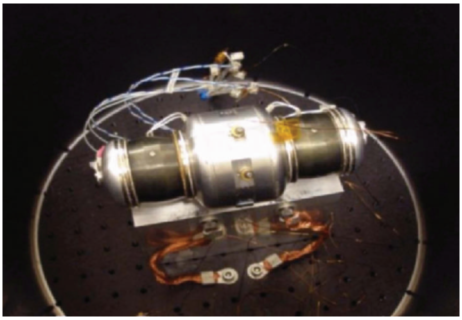

Figure 8. Cryocooler mounted to cold plate

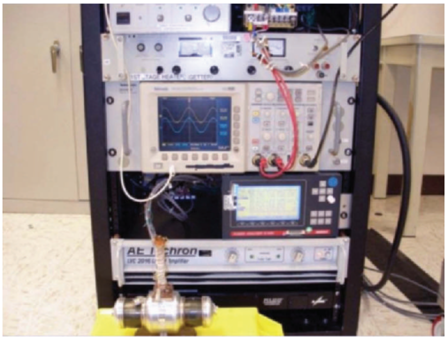

Figure 9. Rack electronics during initial check out of cryocooler

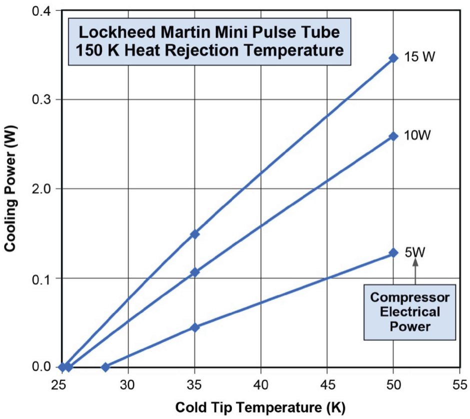

Figure 10. Summary of cooling at 150 K heat rejection temperatures and various power in puts for "off the shelf "cooler

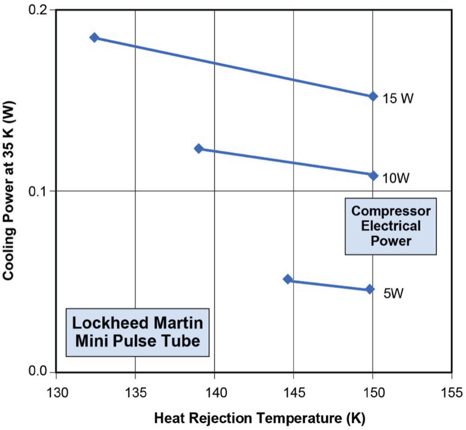

Figure 11. Effect of power and heat rejection temperature on cooling at 35 K

At 35 K with 15 W of input power 0.17 W of cooling was achieved. This was an excellent result for the "off the shelf" cooler without any optimization. Some hysteresis of one of the pistons at these conditions was noted. It is not known if this was a result of accuracy issues with the position sensor during cold operation.

Figure 11 shows the test data for a range of powers and rejection temperatures. The system was operated down to 132 K, which was the lower limit of our cold plate. The power input was limited to about 15 W because of piston stroke limitations at the lower operating pressure. The optimized system will utilize a larger piston to allow higher strokes, which also leads to more optimum performance.

Figure 12 presents the cooling for a range of drive frequencies showing the peak efficiency is achieved at 42-44 Hz.

Figure 13 presents the motor efficiency, which is 90% at the reduced heat rejection temperature and has a peak value near 54 Hz. The peak efficiency is improved due to the lower coil wire resistance and reduced Joule heating. These two sets of frequency-dependent data show that the inertance tube is not correctly sized for the 150 K heat rejection temperature. This was known prior to the testing; however the "off the shelf" cryocooler had an internal integrated inertance tube that could not be changed. Re-sizing the inertance tube to the correct value is expected to increase the cooling by 50 % with no other optimization changes.

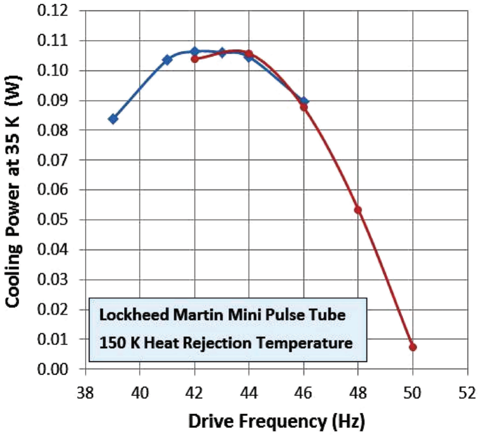

Figure 12. Cooling power vs. drive frequency

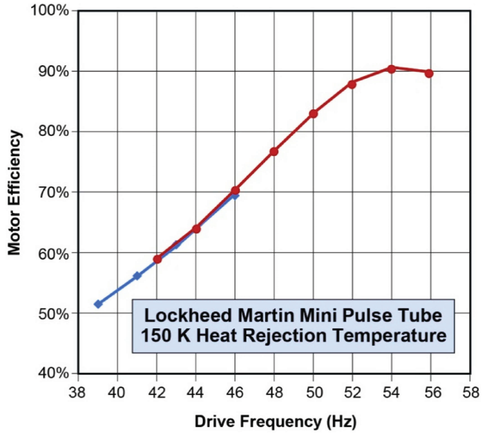

Figure 13. Motor efficiency vs. drive frequency

PHASE II PLANS

The overall objective of the proposed Phase II effort is to deliver a pulse tube cryocooler for the target application of providing 0.3 W of cooling at 35 K, with 150 K heat rejection temperature and at most 20W of input power. In this pursuit, MCG will design a pulse tube cryocooler coldhead that will be driven by the LM compressor.

In order to drive the optimal oscillating helium gas flow to the pulse tube coldhead, a LM Mini compressor will be reconfigured to optimize cooling with the 150 K heat rejection platform. To complement the reconfigured compressor, a matching pulse tube coldhead will be designed, built, and tested. The goal is to configure several versions of the coldhead, then test the respective integrated cryocooler systems, and reiterate to demonstrate the feasibility of achieving the requisite performance. Several pulse tube coldhead modeling methods of differing complexity have been applied during the first phase of the project. Each successively more involved model was built on the previous modeling that was done, ultimately leading to a prediction for a refined, optimized pulse tube coldhead configuration. For the current project this means a layout that is optimally suited for heat rejection to 150 K. The Phase I effort has afforded the initial iteration in the process of truly optimizing the PTC layout.

SUMMARY

This Phase I SBIR work resulted in the first demonstration of a cryocooler operating at a heat rejection temperature of 132 K, surpassing the program goal of 150 K. The cryocooler operated without any known problems and provided 0.17 W of cooling at 35 K. This testing utilized an "off the shelf" cryocooler that was not designed or optimized for 35 K cooling and 150 K reject temperature. Design and optimization of this type of pulse tube system is predicted to achieve 0.3 W of cooling with 10 W of power input to the compressor.

The basic cryocooler system, which was previously qualified to TRL6, can be optimized for the cooling required with relatively straightforward changes in the piston diameter and the inertance tube. In addition, predictions indicate the required cooling can be achieved with the existing regenerator materials (but with different regenerator and pulse tube dimensions). Development of a more efficient regenerator would further reduce the input power required if successfully achieved.

The next step in demonstration of the system is to build an engineering model unit in Phase II with optimized parameters to provide a full demonstration of the required cooling capability.

ACKNOWLEDGMENTS

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