High Frequency Coaxial Pulse Tube Microcooler

M. Petach, M. Waterman, G. Pruitt, and E. Tward

Northrop Grumman Space Technology
Redondo Beach, California, 90278

ABSTRACT

This paper describes the continued development of a cryocooler which uses a coaxial pulse tube cold head designed for high operating frequencies. This high frequency design results in a small, low mass, fast cool-down pulse tube cryocooler. The cooler's coaxial pulse tube cold head was optimized to utilize the small swept volume and high resonant frequency of a scaled down version of Northrop Grumman Space Technology's existing line of flight qualified compressors. It was also thermodynamically and mechanically optimized for rapid cool-down, and it has a very low inherent thermal mass at the cold end. At a reject temperature of 298 K, this cooler can lift 1.3 W at 77 K. This paper describes this cooler's performance over a range of frequencies up to 144 Hz and discusses the estimated losses.

INTRODUCTION

A very small, high reliability and high capacity cryocooler is beneficial for payloads such as large infrared focal planes, filters or cold optics that currently use heavier cold radiators. The development of the very small, low mass cryocooler with a coaxial pulse tube and a flexure bearing compressor has continued past the previously described laboratory version[1] to an Engineering Model maturity. The compressor is directly scaled from Northrup Grumman's TRL-9 flight heritage compressor product line.[1,2,3,4] The cryocooler has been implemented with an all welded compressor, small lightweight tactical drive electronics and a flight-like cold head that can interface with an integrated dewar assembly. This more mature implementation of the cooler has been subjected to random and sinusoidal vibration while operating and has shown no permanent performance change. It has operated under severe vibration, demonstrating only minor performance variation while the vibration is applied. It has been tested for thermal performance and shown to repeat the performance of the earlier development model.

CRYOCOOLER

The back to back linear flexure bearings compressor is connected to the coaxial pulse tube cold head in a split configuration. A photograph of the compressor, cold head and tactical drive electronics is shown in Figure 1 and the key design parameters are found in Table 1. The inertance and reservoir tank are not shown in this photograph, but are shown in the outline drawing of Figure 2. The compressor's heat is rejected at its cylindrical mounting surface. Similarly, the cylindrical

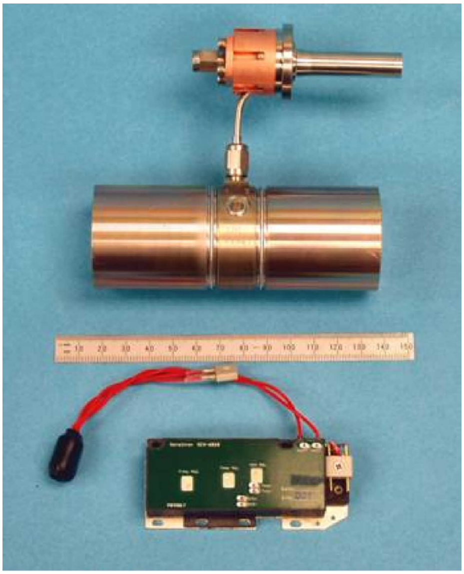

Figure 1. Photograph of the microcooler with engineering model all welded compressor and sine wave drive electronics.

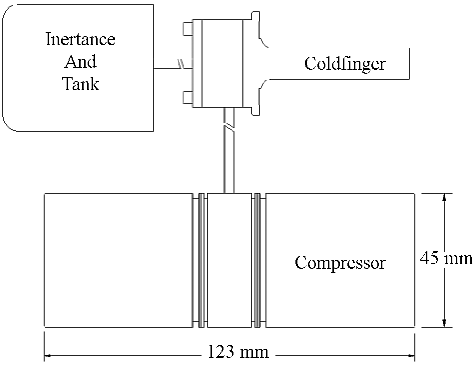

Figure 2. Outline dimensions of the micro cooler.

Table 1. Key parameters of the micro cooler.

Cooler mass	857	gm
Compressor envelope		
diameter	45	*mm*
length	123	*mm*
Coldfinger envelope		
coldfinger diameter	11.2	*mm*
coldfinger length	48	*mm*
Electronics Envelope		
LxWxH	50x38x20	*mm*
Max Cooling at 77K (298K reject)	1.3	*W*
Max Cooling at 150K (298K reject)	4.0	*W*

surface at the base of the cold head is used for rejecting its heat. The drive electronics are a modified COTS sine wave drive from Sensitron Semiconductor.[5] These drive electronics are capable of providing in excess of 50 Wac to the compressor from a nominal 28 Vdc bus. The estimated mass of the mechanical cooler in a flight configuration is 900 grams, including the COTS drive electronics.

The cold head is a single stage coaxial pulse tube of the single inlet inertance tube type. The cold tip's internal heat exchanger is configured to remove heat from the circumference of the end of the cold tip. This allows direct integration of an FPA onto the cold head. The cold head base at the warm end of the cold tip is configured to allow welding of an integral dewar. If an integral dewar is not needed, a modest performance increase can be achieved by the use of a lower conductivity cold head material. The cold head's regenerator is packed similar to Northrop Grumman's other flight proven cold heads.

The measured cryocooler lift vs. cold tip temperature is shown in Figure 3 for constant power line at a reject surface temperature of 298 K. At full drive power, a "no load" temperature of 47 K and 1.3 W of lift at 77K are achieved.

These data are also plotted on a performance map in Figure 4. This plot format shows the ac input power required to achieve a given cold tip temperature for a given user load.

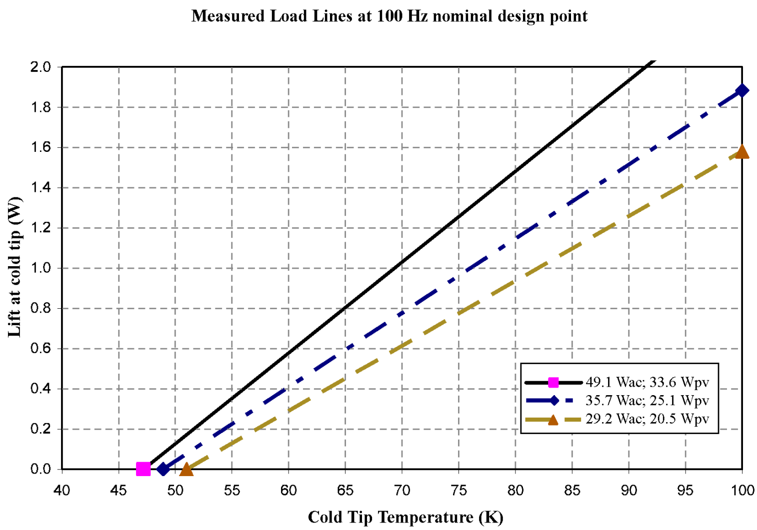

Figure 3. Measured load lines at 298 K reject temperature.

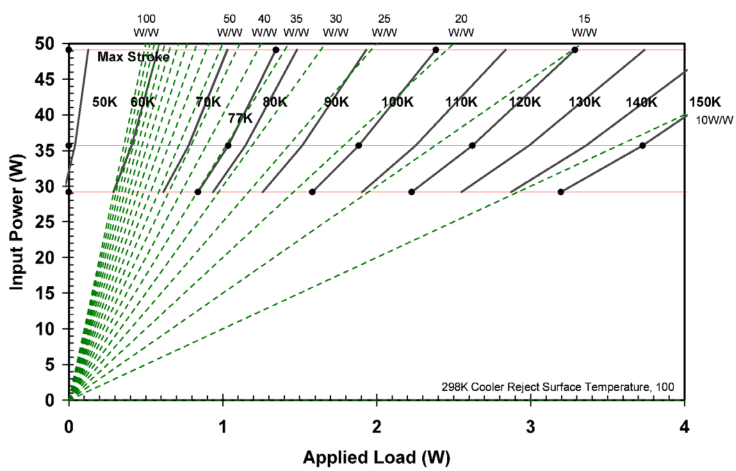

Figure 4. Cooler performance Plot at 298 K cooler reject temperature.

The rapid cool-down capabilities of this cooler are illustrated in Figure 5, which shows measured cool-down curves with 325J and 885J copper thermal masses added to the cold tip to simulate the heat capacity of typical large FPA sensors for this class of cooler.

If the cryocooler needs to be operated during launch and for some non-space applications, it will be subjected to vibration while operational. The inertial forces imparted on the gas in the pulse tube from these vibrations have the potential to disrupt the flow and convect parasitic heat. To quantify this effect, the cooler was subjected to vibration while operating at low power, where it is more susceptible to pulse tube flow disturbances. The cold tip temperature under load during various levels of both random and sinusoidal vibration is shown in Figure 6. For the nominal levels of 7.5 G_{rms} and 0.85 G_{pk}, there is less than 5% performance variation when the vibration is applied.

An energy budget for the microcooler system was derived from a combination of direct measurement and previously validated correlations. This is shown graphically in Figure 7, with values corresponding to resonant operation at 298K reject. Starting at the left in the figure, the DC input power was determined from measured DC voltage and current. The ac power, ac voltage and ac current into the compressor were measured with a digital sampling power meter (Yokogawa WT1600). The compressor coil resistance was measured at operating temperature by injecting a small dc current into the coil during operation. The measured AC current and the measured coil resistances

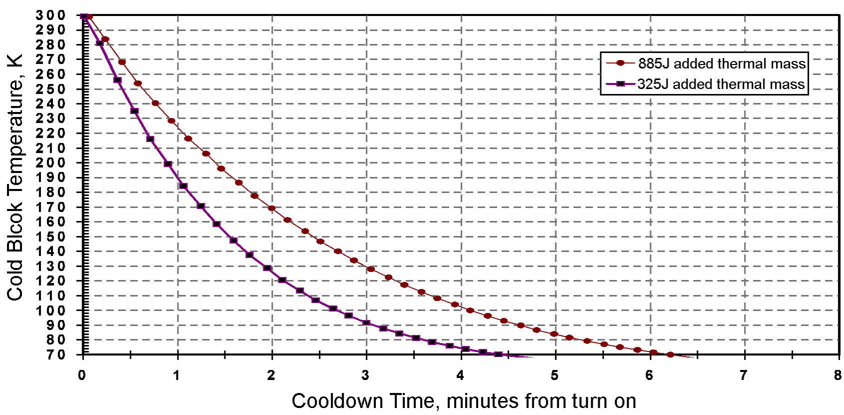

Figure 5. Measured cool down time curves with two different thermal masses added to the cold tip to characterize the effect of user load heat capacity.

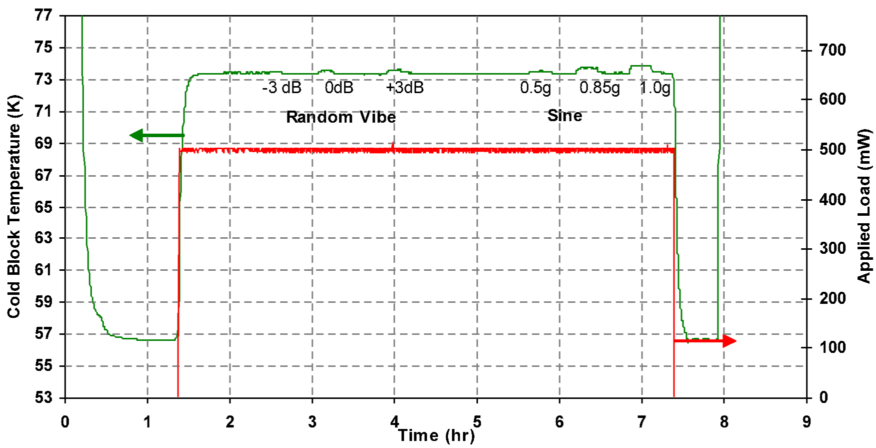

Figure 6. Measured effect of vibration on operating cooler temperature.

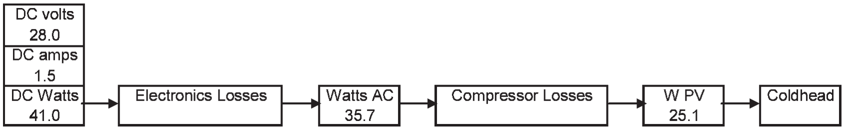

Figure 7. Cooler energy budget at a 298 K reject temperature, 100 Hz.

were then used to compute the I^2R loss. The "shaft" power is then computed as ac power - I^2R loss. The compressor losses due to hysteresis are often lumped into an equivalent mechanical resistance, R_m, term. Previously measured R_m term contributors in a larger compressor of this design[6] were scaled to this smaller compressor size. The seal loss was estimated from measured seal flow vs. pressure and measured pressure amplitudes in the compression space using standard seal loss formulations.[7] The remaining power after the measured Wac- I^2R has the hysteresis and seal loss subtracted is the estimated acoustic (or "PV" power at the compressor exit port.

The efficiencies of the various components are a function of frequency, and the best overall performance occurs when the mechanical resonance of the compressor coincides with the cold head tuning. Others have shown efficient performance of a cold head at frequency higher than the 100 Hz design point of this system[8], so cold head characterization measurements and modeling were done at higher frequencies. In order to extend the range of frequencies, an optimum length inertance line was used at each operating frequency. Test frequencies were limited by the increased I^2R dissipation in the compressor as the compressor was run off resonance. The measured load lines at 124 Hz and 144 Hz are shown in Figure 8, where it is evident that temperatures below 50 K are achieved

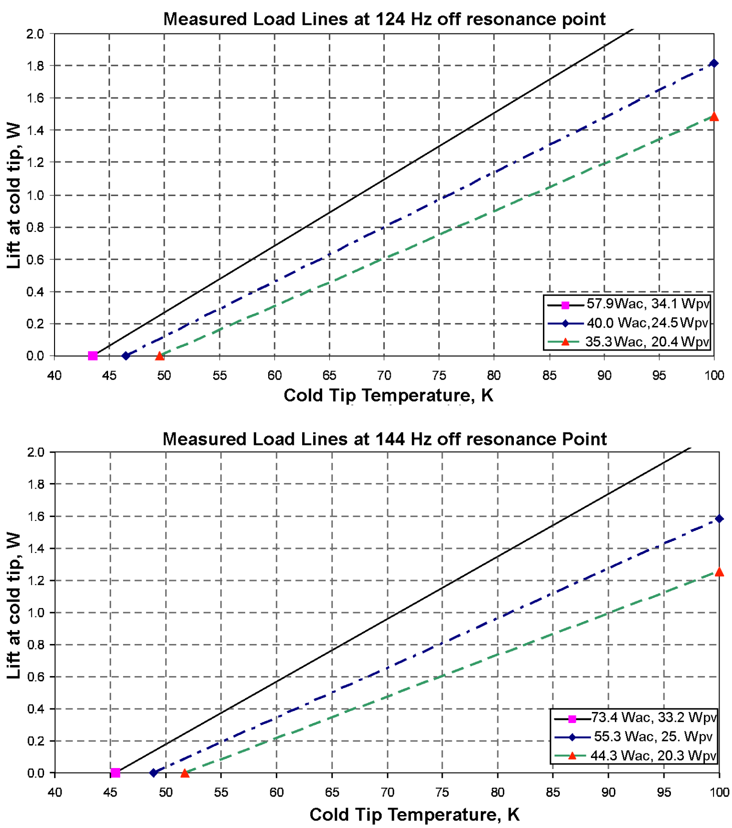

Figure 8. Measured load lines at 124 Hz and 144 Hz, 298 K , with the cooler operating off the compressor's resonance. The lines correspond to similar "PV" power conditions as the 100 Hz data shown in Figure 3.

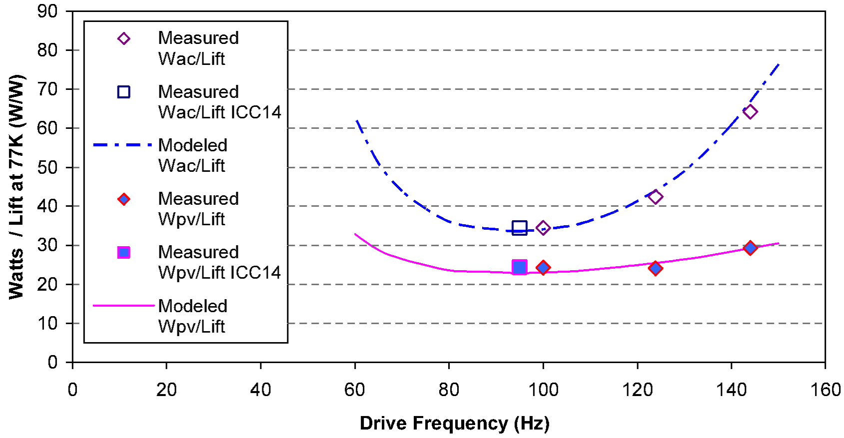

Figure 9. Measured and modeled cooler specific power as a function of frequency. Different inertance line lengths were used to tune the cold head at each frequency. The dashed line and open symbols show the watts of cooling at 77 K per watt of AC power into the compressor. The solid line and solid symbols show the watts of cooling at 77 K per watt of acoustic power leaving the compressor.

but at somewhat lower slopes than at the optimum 100 Hz. At 124 Hz the "no load" temperature of 44 K is actually slightly lower than the "no load temperature for the 100 Hz case, but the slightly lower slope results in approximately the same cooling per PV power at 77 K as the 100 Hz operation. At 144 Hz, while the "no load" temperature of 46 K is still lower than the 47 K at 100 Hz, the lower slope results in less lift at 77 K per watt of PV power. This frequency dependence of the specific power at 77 K for 25 W PV operation is shown in Figure 9 for both the PV referenced lift and the AC input power referenced lift. Also shown in this figure are the modeled performance curves with an optimum inertance line length at each operating frequency.

It is apparent that the cold head itself maintains its performance up to 124 Hz, but that it would need to be modified to achieve operation at 144 Hz with the same efficiency as at 100 Hz This modification would allow an increase in cooling due to the increase in drive frequency allowing a higher cooler input power.

CONCLUSION

A small, low mass pulse tube cryocooler capable of 1.3 W at 77 K which is compatible with tactical electronic drives as well as space qualified electronics drives has been developed. The cold head maintains its performance beyond the nominal system optimum of 100 Hz up to 124 Hz, raising the possibility for future increases in capacity by raising the compressor resonant frequency.

ACKNOWLEDGMENTS

The compressors and cold heads were built and tested using Northrop Grumman internal funds.

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