Development of a Miniature 150 Hz Pulse Tube Cryocooler

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ABSTRACT

A miniature, high energy density, pulse tube cryocooler has been developed to provide appropriate cooling for size-limited cryogenic applications demanding fast cool down. This cryocooler design was optimized using REGEN 3.2 for an operating frequency of 150 Hz and an average pressure of 5.0 MPa, in order to maintain high cooling power and high efficiency in a very compact package. The regenerator has dimensions of 4.4 mm inside diameter by 27 mm long and is filled with #635 mesh stainless steel screen. A net refrigeration power of about 0.5 W at 80 K is calculated for this miniature pulse tube cryocooler. Various design features, such as the use of compact heat exchangers and a miniature linear compressor relying on a resonant "moving magnet" actuator and having a swept volume of 0.5 cm^3, resulted in a remarkably compact pulse tube cryocooler. In this study, we present the important design parameters of this particular cryocooler. The effect of fill pressure and operating frequency on cryocooler performance are presented along with the tested cryocooler cool down characteristics.

INTRODUCTION

A clear gap exists between the rapid development and miniaturization of low-temperature applications and the availability of applicable miniaturized cryogenic refrigeration systems. Many of these applications are optical and electronic based technologies, which by nature only require relatively small cooling powers (typically 1 watt or less) but do need quick, efficient, and reliable cooldown. As a result of inherent difficulties which have hampered the development of miniature closed loop Joule-Thomsom (J-T) cryocoolers, which were originally thought to be best suited for these applications, there has been a growing interest in reducing the size of a regenerative devices, such as the Pulse Tube cryocooler. A regenerative cycle operates with a relatively small cyclic pressure wave oscillating around a large mean fill pressure, rather than the constant large pressure gradient needed in a recuperative (JT) cycle. The guiding principle in the transition to regenerative cycles is that the compression necessary in regenerative cycles may be realized on considerably smaller scales if the cycle frequency and fill pressure is increased to compensate for the decrease in working fluid volume.

Much of the initial foundational theoretical work to this effect was to set forth the basic operating parameters and characteristics of such miniature high frequency Pulse Tubes. Peterson et al.[1,2] and others[3] began this effort in the mid 1990's by trying to establish a theoretical lower size limit, in general, to regenerative cryocooler miniaturization. Other work, more specific to actual cryocooler design, such as Radebaugh[4,5] focused more specifically on heat transfer phenomena in small dimensions and at higher frequency operation. Here issues of cycle frequency with respect to mass flow rates and thermal penetration into the heat transfer mediums were studied to establish the necessary operating parameters and regenerator geometries necessary to design viable high frequency miniature cryocooler.

The attempts to realize these concepts experimentally however in working high frequency miniature Pulse Tube cryocoolers[6,7,8] have been met with many difficulties and it was not only until recently that a high frequency miniature cryocooler, built at NIST, was able to successfully reach sub 100 Kelvin temperatures. In this work[9], a miniature Pulse Tube cryocooler designed to operate at a frequency of 120 Hz and an average pressure of 3.5 MPa achieved a no-load temperature of 50 K and provided 3.35 watt of cooling power at 80 K. While this cryocooler having a regenerator length of only 30 mm, did fulfill its stated purpose and showed how such a miniaturized cold head would perform at higher operating frequencies and pressures, it was driven by a relatively large compressor and did not reach the higher frequencies which are in theory possible in such systems.

To demonstrate this incompatibility, Figure 1 shows the disparity between the cold head of this cryocooler and the attached compressor with another miniature commercially available compressor shown on the side for comparison. To be entirely fair however, as the focus of this initial work was to establish an experimental foundation to the operation of miniature Pulse Tubes at these higher frequencies, there was not much of a reason to complicate matters by including the limitations and design considerations of a miniature compressor and higher frequencies at the outset of this project. Now however, in the wake of this first experimental success, it is necessary to continue the miniaturization process by incorporating a design approach which not only includes a miniature compressor but also allows for an increase it its operating frequencies up to those that are feasibly allowed while still retaining an efficient thermodynamic cooling cycle.

MODELING AND DESIGN

Sub-Components

To practically implement further cryocooler miniaturization beyond what had already been accomplished in the 120 Hz Pulse Tube, a miniature compressor capable of providing the necessary PV power at 150 Hz must be incorporated. A solution was found in a commercially available miniature linear compressor from RICOR which relies on a resonant “moving magnet” actuator and

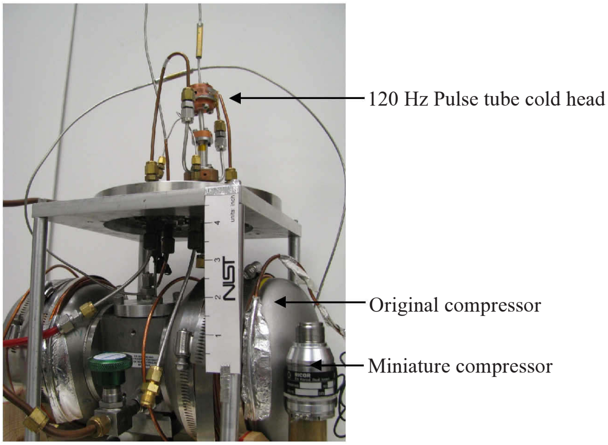

Figure 1. The NIST 120 Hz cryocooler showing the disparity between the size of the cold head and the attached compressor, another more miniature commercially available compressor is also shown for size comparison.

has a swept volume of 0.5 cm³. This compressor is ideally suited for a miniature high frequency cryocoolers not only due to its small size but also due to its dynamic structure which allows for the resonant frequency of the compressor to be tuned to match the optimum cycle frequency of the Pulse Tube. This leads to an improvement of the overall, electrical power to cooling load, efficiency. Other benefits of this compressor include the wire windings that are outside of the Helium space thus preventing any outgassing of hydrocarbons into the Helium during operation. This compressor, according to the manufacturer, is rated to safely operate at pressures up to 5.0 MPa.

In terms of regenerator, the smallest commercially available structured regenerator matrix material is currently the #635 stainless steel woven mesh with a hydraulic diameter of 19 mm, wire diameter of 20 μm, and porosity of 0.60. Given the hydraulic diameter of this mesh and the aforementioned pressure limit of 5.0 MPa, one can determine the frequency range that would still allow for effective thermal penetration into Helium. In Equation 1, we write the thermal penetration depth, λ, as a function of thermal diffusivity, α_m:

$$\lambda = \sqrt{\frac{2\alpha_m}{\omega}} \quad \text{Where} \quad \alpha_m = \frac{k}{\rho \cdot Cp} \tag{1}$$

where ω is the frequency, k is the thermal conductivity, ρ is the density, and Cp is the specific heat. We solve this function for Helium at 50 atm, as Helium is the "limiting" material. With respect to heat penetration at increasing frequencies stainless steel retains considerably better thermal penetration depth than Helium at temperatures below 300 K. Results are shown in Figure 2. From this figure we clearly see that in order to maintain a good measure of thermal penetration depth ($\lambda >> Dh$) as compared to the hydraulic diameter of 19 μm for the #635 matrix it is necessary to limit the operating frequencies. At frequencies much greater than 200 Hz, the Helium at the cold end of the regenerator (80 K) will have thermal penetration depths on the order of the hydraulic diameter. The heat transfer will not be completed quickly enough and in effect; choke the cycle. As a result, in order to still maintain a good ratio of thermal penetration into the hydraulic diameter of the #635 screen mesh we limited the cycle frequency to 150 Hz.

REGENERATOR OPTIMIZATION USING REGEN 3.2 MODELING

The regenerator modeling for this design, a regenerator able to perform 1 W of cooling at 80 K, was performed using REGEN 3.2 in an iterative optimization process. To begin, initial guesses were made with respect to the regenerator length L_{reg}=*22.0 mm*, inner diameter D_{id}=*5.84 mm*, and an optimal phase shift (between pressure and velocity) at the cold heat exchanger of θ=*-30°deg* (these values will be adjusted later on as the iterative optimization sequence progresses). Figure 3 shows the optimization curve for COP with respect to fill pressure along with the corresponding mass-flow and the area to mass-flow ratio. We show in this graph that, given the initial "guessed" regenerator geometry, mesh filling, and operating frequency, the COP at a fill pressure of 5.0 MPa

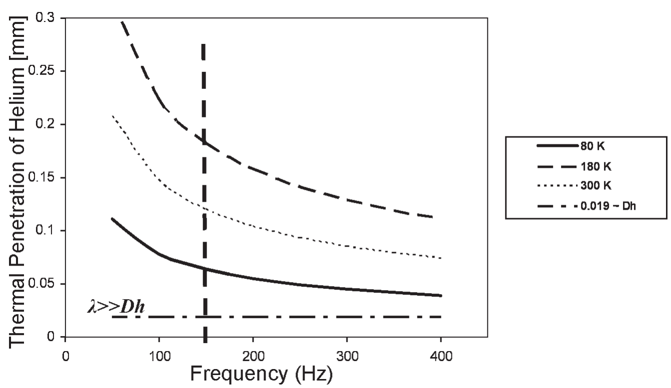

Figure 2. Thermal penetration depth of heat into Helium at 5.0 MPa as a function of oscillating frequency in comparison to the hydraulic diameter of #635 SS regenerator screen.

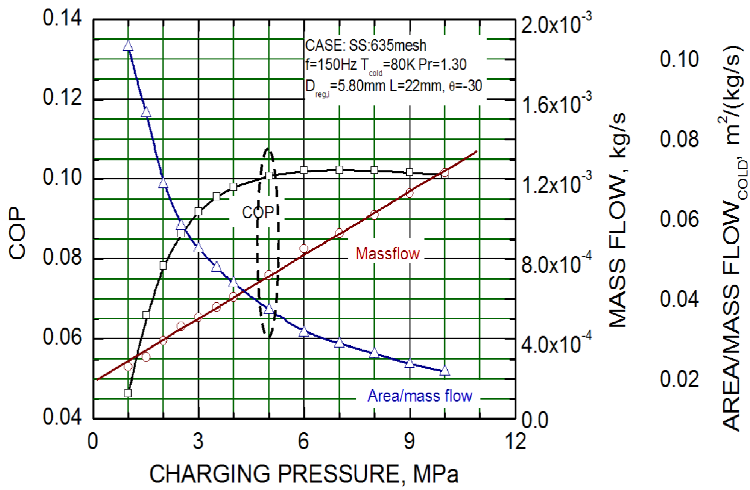

Figure 3. Regen 3.2 calculations by Z. Gan showing the effect of pressure on COP and the corresponding mass-flow and area/mass ratio, optimum values for 5 MPa are within dotted line.

should be about 0.10. It is interesting to note that further increasing the fill pressure beyond the 5.0 MPa limit (established by the compressor) will only slightly improve the COP of the regenerator (with the maximum being ~ 0.105 and appearing at around a fill pressure of 6.5 MPa).

We now begin the optimization process for the regenerator geometry by setting the fill pressure to 5.0 MPa and initially solving for the optimal area to mass-flow ratio for varying phase shifts at the cold heat exchanger. We then are able to show, in Figure 4, that in fact the optimum area to mass-flow ratio is pretty much independent of the phase shift at the cold heat exchanger and occurs at about 0.023 $m^2/(kg/s)$. This optimum, corresponds to a mass flow of approximately 0.72 g/s (given the initial guess for the inner diameter and length of the regenerator). We can then determine what the optimal phase shift at the cold heat exchanger should be given this optimum mass flow. In Figure 5, we show that this phase shift should optimally be ~ 45°.

Now with the optimum phase shift and mass flow at the cold heat exchanger known, we change the initial guess for regenerator length of $L_{reg}=22.0\ mm$ to determine the actual optimal length.

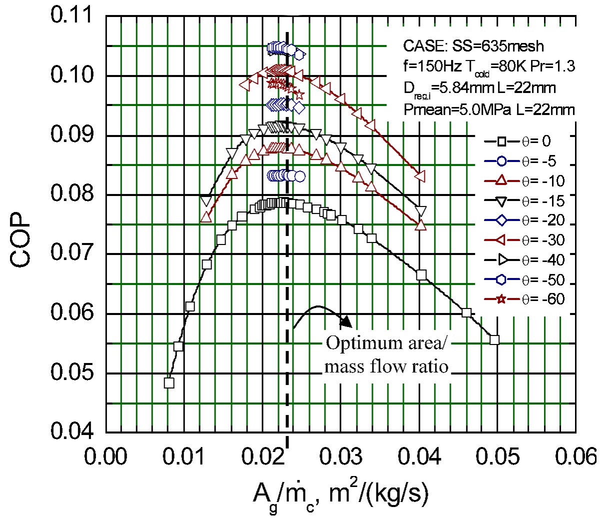

Figure 4. COP as a function of area to mass-flow ratio and phase shift at the cold heat exchanger by Z. Gan.

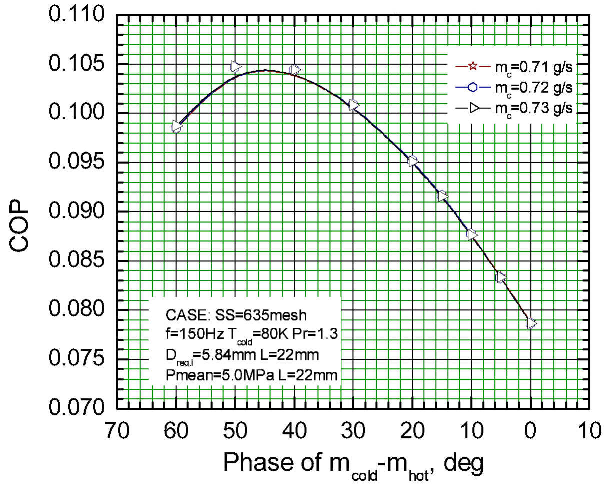

Figure 5. COP as a function of phase shift (between the pressure and velocity at the cold heat exchanger) for three different mass-flows by Z. Gan.

Figure 6 shows the REGEN3.2 solution for the length optimization. This length is about 27.0 mm, given the optimal mass flow of 0.72 g/s.

MINIATURE PULSE TUBE DESIGN PARAMETERS

With the regenerator optimization sequence described above in conjunction with a transmission line model for the inertance tube, the cryocooler parameters for a 150 Hz Pulse Tube operating at a fill pressure of 5.0 MPa are obtained. The central design parameters are tabulated in Table 1.

In order to evaluate and further establish the validity of the design parameters reached by REGEN 3.2, SAGE by Gedeon Associates was used as a comparative tool. The results from running the "SAGE 4.0 Pulse Tube solver" confirmed that a cryocooler built with these design parameters should be able to achieve cooling at 80 K but predicted only about 0.5 W vs. the 1 W predicted in the REGEN 3.2 model. This discrepancy, might be a result of SAGE 4.0 taking into account the "real behavior" loss factors in the buffer tube that were not adequately addressed in the model using REGEN 3.2.

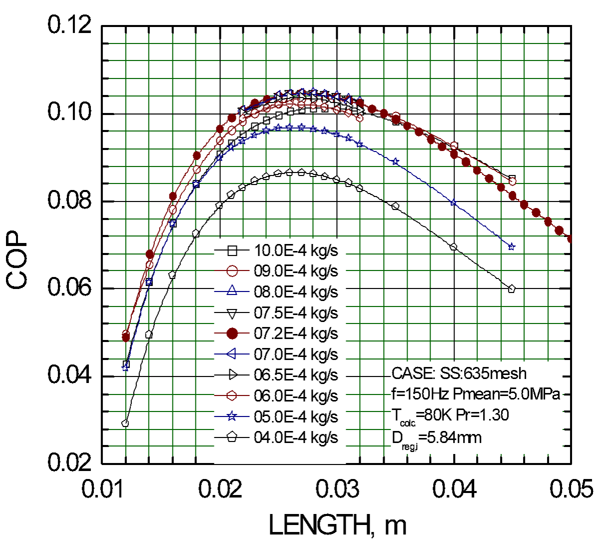

Figure 6. COP as a function of regenerator length for various mass flows by Z. Gan.

Table 1. Central design parameters of the 150 Hz Pulse Tube Cryocooler

	After Cooler	Reg.	Cold HX	Buffer Tube	Warm HX	Inertance Tube #1	Inertance Tube #2	Reservoir
Length [mm]	3.3	27.0	2.7	40.4	2.1	23.5	61.0	30.0
Inner Diameter [mm]	4.63	4.5	2.1	2.1	2.0	0.45	0.80	25.0
Filling	#200 Copper	#635 S.S	#200 Copper	-	#100 copper	-	-	-

CONSTRUCTION

Cryocooler Hardware

The small size of this Pulse Tube coupled with the need to include an adequate amount of sensors to perform testing brought about some design considerations which will be discussed here. The cold head subcomponents were all braze welded together as their small size made the inclusion of "screwable" flanges on each subcomponent quite impossible. As it was, even brazing these small subcomponents together correctly, required designing brazing jigs to keep the parts axially aligned and supported throughout the brazing process. In addition two supporting bars were mounted within the vacuum can to align the weight of the warm heat exchanger and phase shifting components so that the these components would not buckle the thin wall (0.08 mm) of the buffer tube, see Figure 7. This support mechanism is designed to limit any planar movement and yet allow for axial contraction as occurs with cooling. These same supporting rods also double as the conduction cooling path for the warm heat exchanger and reservoir. Other design considerations include connecting the reservoir to the supporting rod alongside the warm heat exchanger so that all the heat would need to travel down these same supporting rods to be dissipated on the bottom vacuum plate, see Figure 8. This single conduction path provides a means by which the total heat dissipated from the warm end of the Pulse Tube may be measured. The compressor was mounted just below the vacuum plate, relatively close to the after-cooler so as to minimize unnecessary dead volume in transfer lines. The after-cooler is mounted directly onto the outside of the vacuum plate so as to thermally sink it to the much larger thermal mass of the plate and still allow convection cooling from external fans if necessary.

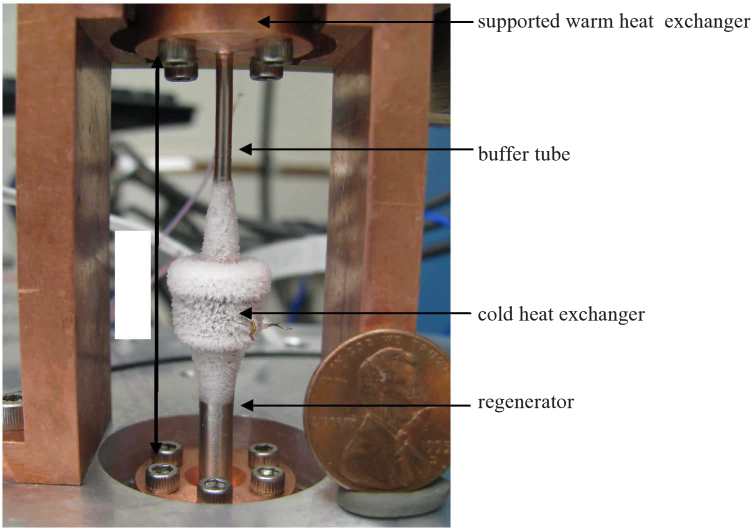

Figure 7. Cold head assembly, operating in exposed conditions and showing signs of frost.

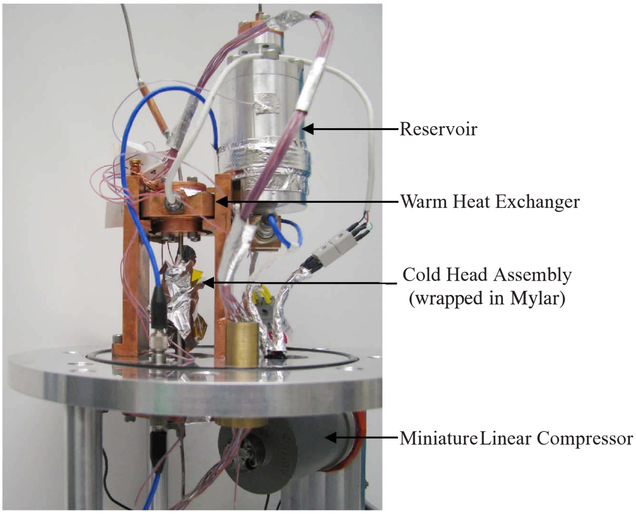

Figure 8. Picture of 150 Hz Pulse Tube assembly with attached sensors

Sensors and Interface

Figure 9 summarizes the sensors included in the test bench to provide feedback as to the cryocooler performance:

Testing

The Pulse Tube was slowly ramped up with increasing input PV powers. Figure 10 shows the resulting temperature profiles of the cold heat exchanger as a function of time from start-up for increasing pressure ratios at the cold end.

Discussion

The results show us that with increasing pressure ratio at the cold heat exchanger the Pulse Tube performance improves considerably. At the highest pressure ratio reached during this initial experimentation (1.17) the Pulse Tube achieved a low temperature of 97.5K. The speed at which the cooldown is achieved is also to be noted with the cryocooler dropping below 100K in under 100 seconds. Due

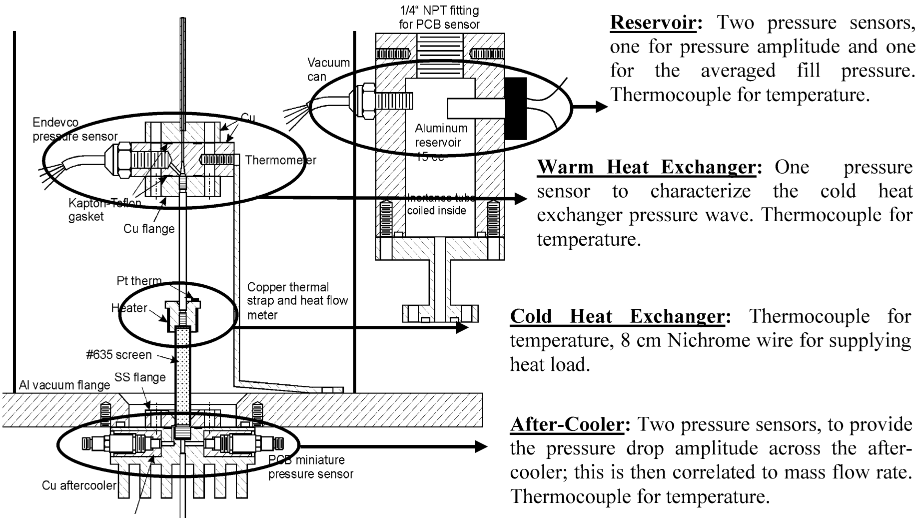

Figure 9. 150 Hz Pulse Tube system sketch including short description of sensor components.

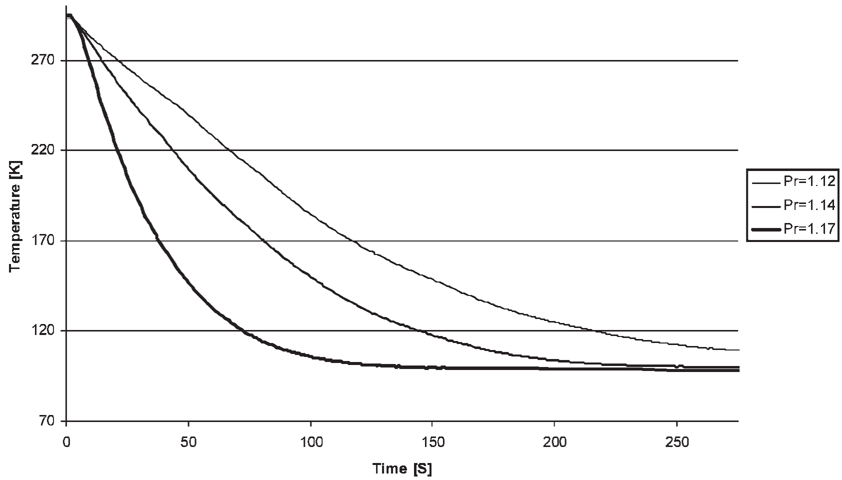

Figure 10. Cooldown times for the 150 Hz Pulse Tube while operating with increasing pressure ratios

to some problems that surfaced with the specific compressor used in this test we could not achieve the needed pressure ratio of 1.3 set forth in the design parameters. This prevented any testing at the design point of 1.3 pressure ratio. It is the intention of the research team to replace this compressor out with another miniature compressor better fitted for these pressure ratios at these frequencies. It is clear however that this cryocooler can effectively and quickly perform cooldown.

CONCLUSIONS

The results of these preliminary tests confirm that the cryocooler with this specific compressor attached is quite underpowered at this higher frequency and as a result is not performing as well as expected. The pressure ratio at the cold heat exchanger is approximately half of that which was designed for in this configuration (1.17 vs. 1.30). However, even with this lack of supplied PV power from the compressor the cryocooler is still able to drop below 100K in about 100 seconds and as expected from the model it seems to work decently well as a result of the elevated pressures and increased frequency of the cycle. Further testing should be conducted however after incorporating a miniature compressor manufactured by RICOR which is better matched for operation at 150 Hz so as to provide the necessary pressure ratio at this frequency. It is expected that by increasing the pressure ratio to the design point the cryocooler performance will improve to that which was expected in the model.

ACKNOWLEDGMENT

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