INVESTIGATION OF FLOW NONUNIFORMITIES IN A LARGE 50 K PULSE TUBE CRYOCOOLER

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ABSTRACT

A single-stage pulse tube cryocooler was optimized to provide 50 W of net refrigeration power at 50 K when driven by a pressure oscillator that can produce up to 2.8 kW of acoustic power at 60 Hz. The cryocooler was designed with the ability to provide rapid cooldown. The rapid cooling technique makes use of a resonant phenomenon in the inertance tube and reservoir system to decrease the flow impedance and thereby increase the acoustic power and refrigeration power in the system when the cold end is near room temperature. Initial experimental data produced no-load temperatures of about 100 K and showed large azimuthal non-uniformities in temperature profiles around the center plane of both the regenerator and the pulse tube. Inadequate diffusion bonding in the initial aftercooler resulted in non-uniform temperatures in the aftercooler and regenerator warm end where temperatures were as high as 350 K. Jetting into the pulse tube through both the warm and cold heat exchangers also contributed to the poor performance. This paper discusses the performance after an improved aftercooler and pulse tube modifications are added. The steps taken to eliminate the non-uniformities and their effect on the cooler performance are discussed.

KEYWORDS: Active Denial System, Fast Cooldown, Superconducting Magnet

INTRODUCTION

The purpose of the pulse tube cryocooler is to provide cooling for a high-temperature superconducting (HTS) magnet made with second-generation wire (YBCO) that is part of a gyrotron required for the generation of high-power (5 MW) mm-wave (95 GHz) beams. Such beams are used in the nonlethal weapons system known as the Active Denial System (ADS). The optimized cryocooler is designed to provide a minimum of 50 W of net refrigeration power at 50 K and is to be driven with a commercial pressure-wave generator.

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The pressure-wave generator can produce up to 2.8 kW acoustic power at 60 Hz. The fast cooldown technique makes use of a resonance phenomenon in the inertance tube and reservoir system to decrease the flow impedance and thereby increase the acoustic power flow through the system by about a factor of four above the optimized steady-state value used when the cold tip is at 50 K. This resonance phenomenon is produced by reducing the volume of the reservoir, accomplished by valving off a portion of the reservoir. The reduced flow impedance obtained with the resonance phenomenon also provides a better impedance match to the pressure oscillator when the cold end is warm, which results in high compressor efficiency even during initial cooldown.

For a fixed acoustic power at the cold end, the acoustic power required from the pressure oscillator decreases at higher cold-end temperatures. The maximum acoustic power at the cold end of Stirling cryocoolers is fixed by the stroke of the displacer, but pulse tube cryocoolers can accept higher acoustic powers whenever the impedance of the inertance tube is reduced.

APPLICATION

Cryocoolers for use in tactical military environments are required to have a rapid cooldown to enable rapid deployment. Presently one such system is the Active Denial System (ADS) being developed by the Air Force [1], which is illustrated in FIGURE 1. The principle of operation is that high-power millimeter waves are used as a non-lethal method for repelling personnel in crowd control or battlefield scenarios.

The ADS consists of a gyrotron that generates high-power millimeter waves at a frequency of 95 GHz by utilizing a 3 T high-temperature superconducting magnet. For the system to operate correctly, the magnet (YBCO) must be cooled to the design temperature of approximately 50 K and maintained steadily. The mass of the YBCO magnet is about 20 kg. By use of conventional Stirling and Gifford-McMahon based coolers, the typical cooldown period is on the order of roughly 16 hours when a cryocooler designed for high efficiency and low mass is used. For the active denial system considered here, cooldown periods less than four hours are often desired. Additionally, one of the operational parameters is the system is to be mobile, facilitating rapid deployment. For mobile applications the system mass and power requirements should be minimized. The net refrigeration power required at 50 K to maintain the magnet at 50 K is estimated to be 50 W. The cooldown period is inversely proportional to the refrigeration power provided by the cryocooler over the entire temperature range from ambient down to the operating temperature. Typically, the net refrigeration power available at room temperature is much larger than that available at the cold operating temperature, partly because the thermal losses, such as conduction, radiation, and regenerator ineffectiveness, are negligible at room temperature.

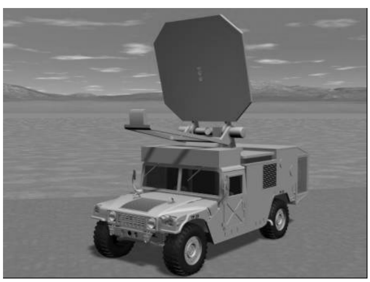


FIGURE 1. Artist's concept of mobile Active Denial System (ADS) to beam high power mm-waves (95 GHz).

To decrease the cooldown period the net refrigeration power should be increased beyond that normally available over the whole temperature range. However, the size and mass of the system will depend on the maximum net refrigeration power the system can provide at the cold-end operating temperature. The PV power provided by the compressor is highest when the cold end is at its lowest temperature, even though the net refrigeration power is higher at the higher temperatures. Ideally we need to find some method for increasing the net refrigeration power at the higher temperatures without increasing the size and mass of the system, and still keep the input power less than the maximum capability of the compressor. For Gifford-McMahon and rotary Stirling cryocoolers, an increased speed at the higher operating temperatures can be used to provide a faster cooldown period. For Stirling and pulse tube cryocoolers using linear-resonant pressure oscillators (compressors), the speed cannot be increased significantly, because the off-resonant condition of the compressor would lead to low efficiency. We describe here a method that should allow the cold head of a pulse tube cryocooler driven by a linear-resonant compressor to accept nearly the full PV power output of a compressor over the entire temperature range of the cold end. As a result, the cooldown period may be decreased by a factor of two or three compared with that of a Stirling cryocooler or a pulse tube cryocooler not utilizing the fast cooldown technique [2].

NOMINAL SYSTEM DESIGN

In the design of this system, multiple software packages and numerical codes were utilized. The regenerator component was optimized using REGEN3.2. The design of the regenerator incorporated individual copper screen within the stainless steel screen matrix to enhance the regenerator performance by increasing the transfer heat conduction of the regenerator matrix. [3]. The required compressor swept volume was computed using the program PHASOR, which uses the conservation of mass and energy to tie all components together and visually indicate the relative phase of different oscillating parameters such as flow, pressure, and compressor volume variation when all these parameters are assumed to vary sinusoidally with time. The aftercooler, cold heat exchanger, and warm heat exchanger were designed by use of ISOHX, which computes the heat transfer and pressure drop in the three isothermal heat exchangers. The inertance tube was designed using the program INERTANCE, which is a transmission line model for optimizing the inertance tube geometry to yield the proper flow impedance. The optimal system design and configuration is illustrated in FIGURE 2.

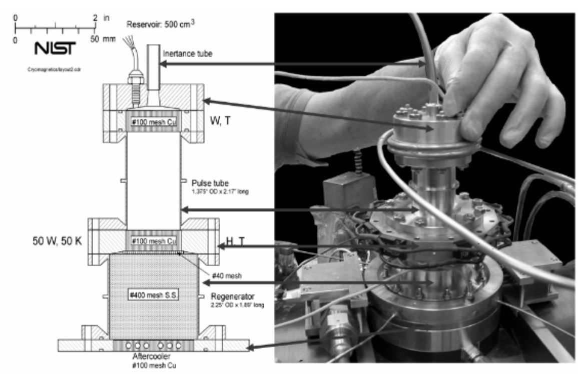


FIGURE 2. Schematic of the rapid cooldown pulse tube cooler with photograph illustrating the manufactured components.

INITIAL RESULTS

The initial experimental data for the prototype cooler were achieved by use of a mean pressure of 2.5 MPa and 3.0 kW of electrical input power. The initial experiments yielded a low-end steady state temperature of about 102 K. Clearly this temperature was much higher than the anticipated temperature arrived at by use of the design models. These results indicated that there were significant parasitic loads on the cryocooler. To determine the source of the parasitic loads on the test system, multiple parameters were varied to determine system sensitivity. The primary diagnostic tools utilized were four equidistant azimuthally spaced thermocouples located at the aftercooler exit, center plane of the regenerator component, and center plane of the pulse tube component.

Analysis of the experimental data by use of the diagnostic temperature sensors indicated three areas where performance was being significantly affected. The first area was the gas entering the warm end of the regenerator or exiting the aftercooler. At this location, the regenerator inlet temperature was approximately 350 K, with a maximum azimuthal temperature gradient of 12 K. The system design target for the regenerator inlet temperature was a uniform inlet temperature of 300 K. The results indicated that the aftercooler was not efficiently rejecting the heat of compression and was subsequently undersized for this application.

The second area of performance degradation was in the temperature homogeneity in the regenerator. Analysis of the temperature sensor data at the mid plane of the regenerator component indicated the presence of non-uniform flow leading to large azimuthal temperature gradients. These initial tests showed gradients as high as 30 K. The exact cause of this temperature non-uniformity was not exactly clear, as there were competing effects from the aftercooler temperature profile and the pulse tube temperature profile. Non-uniform flow at either end could be responsible for the large temperature gradient experienced.

The final area of performance degradation was in the temperature homogeneity of the pulse tube component. Analysis of the temperature sensor data at the mid plane of the pulse tube indicated the presence of non-uniform flow leading to large azimuthal temperature gradients. The experimental measurements showed gradients as high as 55 K. Much like the regenerator, the exact cause of this temperature non-uniformity wasn't clear, as the regenerator temperature profile could be influencing the pulse tube component. In order to ascertain the exact location and reason of the flow non-uniformity, the system was thermo-graphically imaged by use of a commercial infrared camera.

To perform the thermographic imaging, the system was energized at room temperature and cooled to a temperature of ~240 K; for this is the lower imaging limit for the thermographic system utilized. At this temperature, the wall of the pulse tube component was imaged circumferentially, and the results are presented in FIGURE 3; the color scale is white indicates hot, black indicates cold. The results in FIGURE 3 clear illustrate the severity of the flow non-uniformity in the pulse tube component and the presence of an impinging jet of hot gas on the pulse tube wall. From this analysis, it was clear that some aspect of the warm end heat exchanger or transition configuration was responsible for this flow non-uniformity.

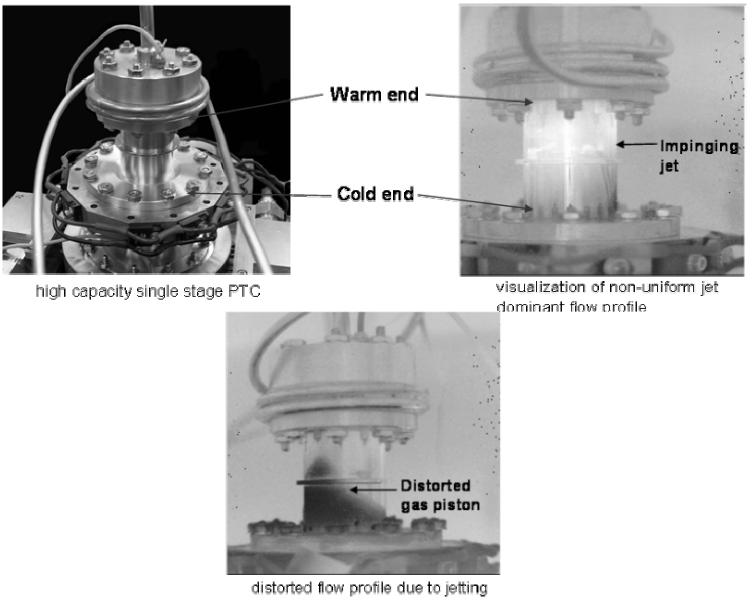
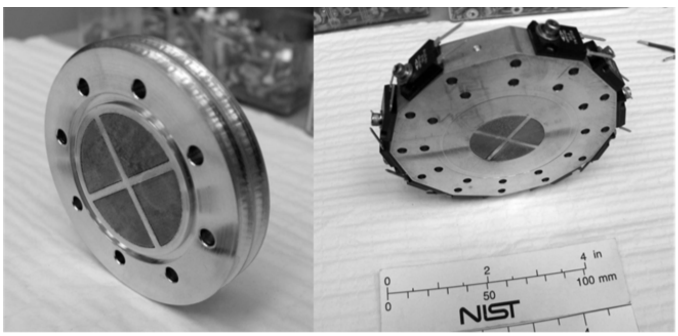


FIGURE 3. (*upper left)* Photo of a high-capacity single-stage PTC for 50 K operation; *(upper right)* thermographic image of wall temperature profile showing the hot spot created by jetting effects in the flow field; *(center)* corresponding thermographic image to upper left photo that shows the effect of the jetting on the shape of the gas piston in the pulse tube; from Garaway et al. [4]. Note that the temperatures depicted are qualitative, and the temperature scale runs from white (hot) to black (cold).

The most logical source of this flow nonuniformity was attributed to the design of the heat exchangers at the warm and cold ends of the pulse tube. In the design of the heat exchangers, due to the large diameter of the regenerator and pulse tube, large radial temperature gradients can exist in the heat-exchange matrix, decreasing its effectiveness. To mitigate this radial temperature gradient, the heat exchangers were designed in a quadrant configuration by which solid sections at the core of the heat exchanger would enhance the radial transport of heat; the heat exchangers are illustrated in FIGURE 4.



(a) (b)

FIGURE 4. Photos illustrating (a) the warm heat exchanger and, (b) the cold heat exchanger. Note the quadrant configuration of the heat exchange matrix in each case.

While in principle this should be a superior design from a strictly heat transfer standpoint, when compared to a solid screen heat exchange matrix, the hydraulic performance can be severely degraded. This degradation is attributed to the hydraulic resistance in each quadrant. If the hydraulic resistance is not equal amongst the quadrants, the gas exiting the regenerator or inertance tube will take the path of least resistance and enter the pulse tube component asymmetrically, setting up non-uniform flow.

DESIGN MODIFICATIONS

In an effort to mitigate the problems realized from the initial experimental results, two modifications were performed. The first modification to the system design was the re-design of the aftercooler. The initial results indicated the aftercooler was thermally overloaded beyond a specific mass flow/power input, thus introducing high temperatures into the regenerator warm end. Additionally, analysis of the experimental data from the four thermocouples in the flow stream at the inlet to the regenerator indicated there was temperature non-uniformity. To mitigate the problems, the aftercooler was re-designed to aid in maintaining thermal uniformity of the gas, in addition to a larger heat exchange area. The new aftercooler utilized a diffusion bonded #100 mesh copper screen stack that was twice the thickness of the original design (guided by ISOHX) that was brazed into a copper flange. Passing through the horizontal plane of the screen stack were two sets of in-line water tubes which were brazed to the screen stack to enhance thermal communication between the cooling water and the helium gas flow. FIGURE 5 illustrates the newly designed aftercooler.

The second modification to the system design was the addition of a small screen stack at the warm end of the pulse tube component. The initial results indicated the pulse tube component was experiencing large flow non-uniformity due to large azimuthal temperature discrepancies at the pulse tube center plane. The goal of using the screen stack was to provide a moderate axial flow resistance with minimal radial resistance to induce any jets of gas exiting the warm heat exchanger to redistribute before entering the pulse tube. Recent work by Taylor et al [5] has indicated the flow and how it is transitioned at the warm end of the pulse tube is critical to maintaining high system efficiency. The screen stack consisted of two layers of #40 mesh copper backed with a diffusion-bonded screen stack consisting of two #400 mesh stainless steel screens and a single #24 mesh stainless steel screen.

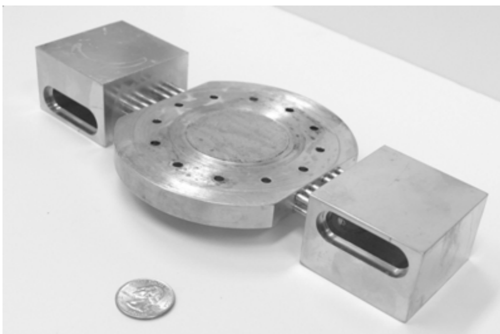


FIGURE 5. Photo illustrating the newly designed aftercooler for the experimental pulse tube cryocooler.

EXPERIMENTAL RESULTS

To quantify any performance gains from the system modifications, the system was experimentally characterized by first replacing the original aftercooler with the redesigned component. The system was then cooled again under maximum power to determine the steady state no-load temperature. The results of this test in comparison to the original configuration are presented in TABLE 1. The results from the test indicated a significant improvement in the system performance as witnessed by the ~18 K lower cold end temperature of 85 K. However, observation of the temperature profile around the regenerator and pulse tube components still indicated the presence of severe nonuniform flow. While the new aftercooler design did improve system performance, this performance increase is attributed simply to the reduced regenerator loss from the lower warm-end regenerator temperature. This indicated that the source of the flow non-uniformity was not attributed to the aftercooler component.

The second experimental characterization involved the use of the redesigned aftercooler in conjunction with the screen stack placed at the warm end of the pulse tube component. Again the system was cooled to determine the steady-state no-load temperature and also observe the temperature uniformity in the pulse tube and regenerator components. The results of this analysis are presented in TABLE 1 in comparison to the original design as well as the modified design using the new aftercooler. The results of this test showed a cold end temperature of 65 K, which is significantly better than that achieved with the original design. Observation of the temperature profiles of the regenerator and pulse tube showed a marked improvement in the flow uniformity. In this case, the azimuthal temperature gradient in regenerator was still ~30 K, while the pulse tube gradient was reduced from 55 K to 8 K. These results show that that the warm end heat exchanger design is flawed and must be replaced to aid in better flow uniformity. The cause for the minimal change in the regenerator temperature profile can also be attributed to the cold heat exchanger. In this case, no efforts were used to redistribute this flow, and the results was only minimal improvement in the temperature profile.

CONCLUSIONS

This paper has presented the design of a large-capacity fast cool-down pulse tube cryocooler. Initial experimental results indicated the presence of large parasitic heat loads that severely degraded the expected performance. Upon analysis of the experimental data

TABLE 1. Azimuthal Temperature Measurements for the Regenerator and Pulse Tube

Temperature Location	Original Design T(K)	New Aftercooler Design T(K)	Aftercooler and Screen Stack T(K)
Regenerator (South)	214	197.4	191.8
Regenerator (East)	196	178.5	160.5
Regenerator (North)	205	203.3	192.8
Regenerator (West)	203	207.2	198.4
Pulse Tube (South)	221	193.1	122.9
Pulse Tube (East)	260	249.3	130.3
Pulse Tube (North)	261	221.6	129
Pulse Tube (West)	237	209.7	129.5
Cold End	102	85	65

from the system by use of diagnostic temperature sensors and thermographic imaging, two critical design flaws were identified. The first flaw was the use of an undersized aftercooler, while the second was the use of a quadrant heat exchanger design with non-uniform hydraulic resistance. The aftercooler was re-designed for this application and the experimental results showed significant improvements in system performance. The results of this analysis confirmed the initial hypothesis of severe flow mal-distribution due to improper flow transitioning in the pulse tube component. Future work will be to re-design the pulse tube, cold heat exchanger, and warm heat exchanger components to eliminate the losses due to flow mal-distribution.

Future work on this project will be to redesign and manufacture the cold and warm heat exchangers, in addition to a pulse tube with a larger aspect ratio. Based upon the results presented in this paper, these modifications should allow the system to achieve the design performance.

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