Performance Investigation on SITP's 60K High Frequency Single-Stage Coaxial Pulse Tube Cryocoolers

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ABSTRACT

A series of 60 K high frequency single-stage coaxial pulse tube cryocoolers has been developed to provide reliable low-noise cooling at 60 K for space-borne long wave infrared focal plane arrays. The development goal and a simulation model are briefly introduced, then optimizations of the novel heat exchanger configurations, the temperature mismatch of the tubes, and the characteristics mismatch of the compressor and cold finger are described. Experiments show that our best cooler prototype has achieved 8.0% of Carnot at 60 K, and can typically provide 2 W at 60 K with 104 W of electric input power and a 300 K reject temperature. It is also shown that 2.5W of cooling capacity at 60K can be achieved when the input power is increased to 127 W. Further optimization is underway, and it is feasible that the thermodynamic performance goal can be realized in the near future.

INTRODUCTION

The past two decades have seen a rapid advancement and maturation of high frequency pulse tube cryocooler (PTC) technology.[1,2] The absence of moving mechanical components at the cold end provides the PTC with several meaningful intrinsic merits such as low noise, high reliability, and the potential for long life, all of which have a strong appeal to the space industry.

The year 1998 witnessed the PTC's first successful application in space[3], and since then, progress has been bounding forward. To date, Northrop Grumman Aerospace Systems (NGAS) has twelve vibrationally balanced PTCs currently in orbit, and two have been in continuous operation for more than 11.5 years.[4] This has encouraged more and more researchers to consider high frequency PTCs as a space qualified cryogenic technology.

Several specific space-borne long-wave infrared focal plane arrays (LWIRFPAs) need about 2 W of cooling power at around 60 K with a low noise level and high reliability to enhance the FPA's signal to noise ratio and sensitivity. Since spacecraft are limited in terms of allowable payload power, mass, and volume, there are additional rigorous requirements on the cryocoolers for high thermodynamic efficiency, lightweight structure, and compact volume. These bring a strong demand for a light, compact, and high efficiency PTC systems.

Table 1. Summary of key development goals of the SITP's 60 K PTC

	Design goal
Cooling capacity	≥ 2W@60K (at 300K reject temperature)
Power consumption	100 W (electric input power)
Refrigeration efficiency	≥ 8.0 % of Carnot at 60 K
No-load temperature	≤ 40 K
Mass	≤ 10 kg (including cooler control electronics)
Ambient temperature adaptability	-30 ºC to +50 ºC
Temperature stability	± 0.1 K
Vibration output of the cold head	≤ 0.1 Nrms
Expected lifetime	≥ 50,000 hours

DEVELOPMENT GOAL AND SIMULATION

SITP/CAS is developing a series of single-stage coaxial 60 K PTC prototypes to meet the above mentioned requirements. The main reason for the selection of the coaxial rather than the in-line arrangement, although a number of NGAS in-line space PTCs have already achieved success in space, is to ease the complexity of system integration and to minimize cost. Our development goal is to provide over 2 W of net reliable cooling power at 60 K with an electric input power of less than 100 W at a 300 K reject temperature. This requires a high refrigeration efficiency of at least 8.0% of Carnot at 60 K. The overall weight including cooler control electronics should be controlled to below 10 kg. Table 1 gives a summary of key development goals. Parameters like ambient temperature adaptability, temperature stability, vibration output of the cold head, and expected lifetime are especially important to future practical applications and are being emphasized in the engineering model development program.

The cryocooler optimization process combines our relevant experience in pulse tube modeling, Stirling development, and linear compressor technology.[5] The regenerator simulation software REGEN 3.2 and the machine model SAGE are also employed for verification and further optimization. The heat exchanger designs are based on a simplified theoretical CFD model and design method developed by the analyses of thermodynamic behaviors of gas parcels in oscillating flows.[6]

In the models, the piston diameter of the dual-opposed compressor is set as 20 mm, and the metallic tube connecting the compressor to the cold finger has a length of 20 cm and a diameter of 4 mm. The fill pressure was chosen to be 3.2 MPa, and the operating frequency is 55 Hz. The inertance tubes together with the gas reservoir serve as the only phase-shifting mechanism. The model optimizes the dimensional, geometrical, and operating parameters together. Figure 1 shows the simulation results for net cooling capacity versus cold-end temperature for input electric powers of 70 W, 100 W and 120 W. It should be mentioned that during the optimizations, the primary focus is on maximizing the COP at 60 K, while the no-load temperatures are not the main targets. The goal is to achieve a PTC that has an optimum performance at 60 K, not a cryocooler which is able to provide cooling at a much lower temperature, or a cooler that can work over a wide temperature range. However, while satisfying the main optimization goal, we also characterize the cooler's performance at typical temperatures, say 55 K, 70 K, 85 K and 100 K, as shown in Figure 1. The typical simulation performance run provides a cooling power of 2.33W at 60K and reaches a no-load temperature of 40K with 100W input electric power.

DESIGN AND FABRICATION OPTIMIZATION

Novel Cold and Warm Exchangers

Since losses become more difficult to control at lower cryogenic temperatures, one of the important aspects in designing a PTC for operation at or below 60 K, in contrast to designing one for operation at 80 K, is the need to reduce the reversible losses as much as possible. In particular, high efficiency cold and warm-end heat exchangers are necessary. The concentric arrangement of the regenerator and the pulse tube makes the radial slit heat exchanger a very good choice for the

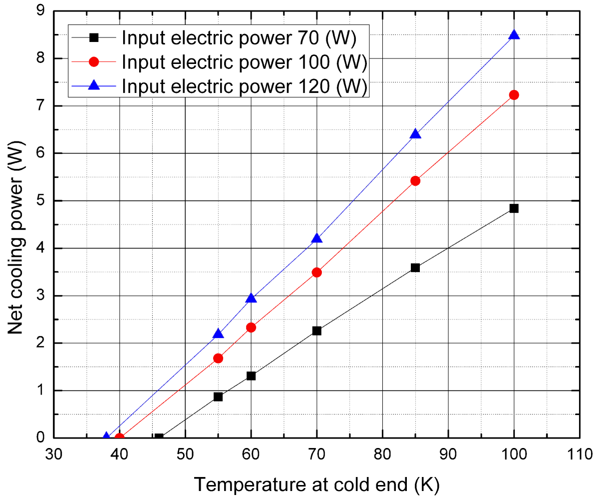

Figure 1. Simulation results of cooling capacity versus cold end temperature.

coaxial PTC. In our design, both cold and warm-end exchangers are made of OFHC copper and adopt a type of novel integral annular slit configuration. Figures 2 and 3 show the internal and external configurations of the cold and warm-end heat exchangers. Uniform slits are cut in both exchangers using a wire electro-discharge machine (EDM). The cold end heat exchanger is composed of 32 radial gas flow passages, each with a gap width of 0.2 mm. The internal structure of the warm-end exchanger is composed of 48 radial gas flow passages, each with a gap width of 0.2 mm. The integral slit configuration of the heat exchangers not only increases the heat exchanger area,

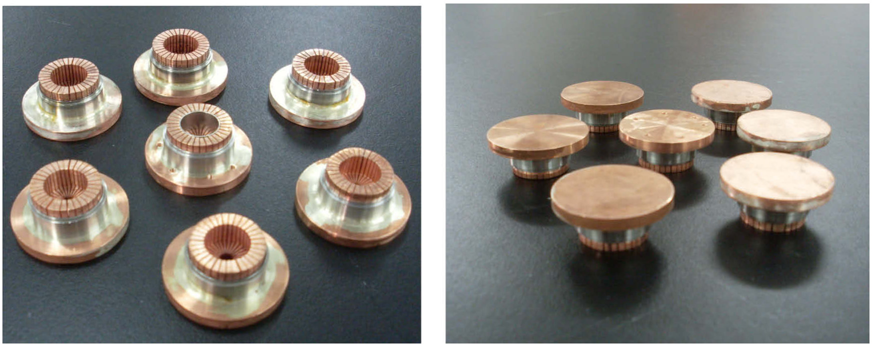

Figure 2. Internal and external configurations of the novel cold-end exchangers.

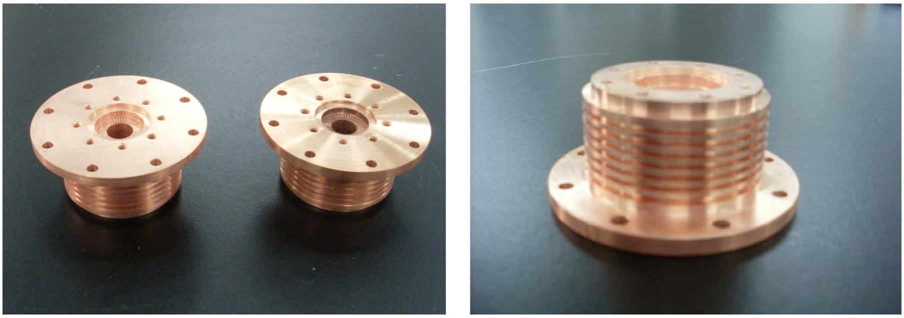

Figure 3. Internal and external configurations of the novel warm-end exchangers.

Figure 4. A visual comparison of the lengths of the regenerator (middle) and pulse tube (right) in a practical 60 K coaxial PTC developed in SITP.

but also straightens the turbulence introduced by the flow reversal. The external part of the warm heat exchanger uses the finned configuration to enhance the heat transfer. In our experiments, the COP of a prototype that uses these novel exchangers increased a remarkable 5.3% at 60 K compared to ones with conventional heat exchangers.

Optimum Temperature Match of the Regenerator and Pulse Tube

Numerical simulation and experimental investigation suggest that the axial temperature mismatch of the regenerator and tube has a considerable effect on the cooler performance. The simulation model shows that a steady radial thermal conduction exists between the pulse tube and the regenerator and the heat transfer has a negative effect on the fluid dynamics and thermodynamics in the system. Experimental results show that the outer wall temperature profile is a sign of the cooler performance. The respective lengths of the two tubes have been designed carefully and rearranged deliberately to acquire an optimum temperature match. This contributes notably to the enhancement of the thermodynamic efficiency. Figure 4 shows a visual comparison of the lengths of the regenerator and pulse tube in a practical 60 K coaxial PTC developed at SITP.

Phase Characteristics Investigation

To achieve a PTC with the high reliability needed for space missions, the phase shift mechanism was limited to an inertance tube and its accompanying gas reservoir. The key phase-shift contributors include the mass flows, the pressure amplitudes, and the phase shifts between them. These are decisive factors for cooler performance and are strongly dependent on variations of the inertance tube. Theoretical and experimental investigations of different inertance tubes were conducted, and the results show that for the determined cold finger dimensions, a single inertance tube with a constant diameter has great difficulty in obtaining the desired phase relationships. The tube is often too long to be used in a practical application. A double-segmented inertance tube with different diameters and lengths, respectively, achieved a much more satisfactory phase shift and acceptable volume. A three-segmental inertance tube was also investigated experimentally, but there was no noticeable enhancement of the efficiency.

Compressor and Cold Finger Integration

A dual-opposed piston compressor with a maximum swept volume of 8.2 cc was chosen to generate the necessary pressure wave. As the model suggests, each piston has a diameter of 20 mm. Successfully matching the compressor with the cold finger with high efficiency is always tricky. In our experimental prototype, LVDTs (linear variable differential transformers) are employed to trace and monitor the displacement of the piston in order to investigate its dynamic characteristics.

In the model, the use of constant 400-mesh stainless steel screens for the regenerator matrix

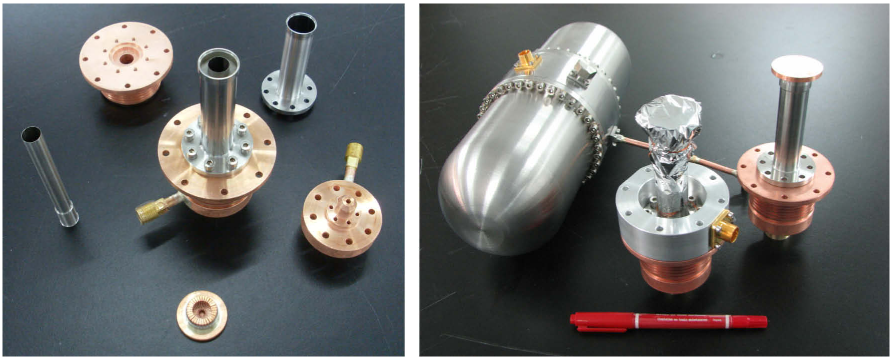

Figure 5. (Left) The key components and an unfinished cold finger. (Right) An integrated cold finger for experimental measurements and a compressor connected to a finished cold finger (without the phase shift mechanism).

was sufficient to achieve the desirable performance at 60 K. However, a hybrid multilayer regenerator with higher mesh screens at the coldest end could achieve a better performance at lower temperatures. Considering the main development goal, we chose the use of the 400-mesh screens throughout the matrix.

Figure 5 (left) shows the key components and an unfinished cold finger, while the right side shows an integrated cold finger for experimental measurements and a compressor connected to a finished cold finger (without the phase shift mechanism).

EXPERIMENTAL THERMODYNAMIC PERFORMANCE

Experimental Setup

The first and most important experiments were to evaluate the prototype's thermodynamic performance. Figure 6 shows a schematic of the experimental setup. Three pressure sensors, P1, P2 and P3 were installed at the compressor outlet, the warm end of the pulse tube before the inertance tubes, and at the bottom of the gas reservoir. They are used to monitor the dynamic pressure and phase change between them. An LVDT sensor is used to monitor the compressor piston movement, which can help to evaluate the PV power transferred from the compressor to the cold finger. Cold-head temperatures below the liquid nitrogen range are accurately monitored by Lakeshore Cernox temperature sensors. The reject temperatures at the warm end are controlled by water circulating in cooling tubes. An active temperature controller is employed at the cold end to acquire the cooling power at the exact desired temperature points. Date acquisition and processing are assisted by a Keithley 3706 system.

Experimental Results and Discussions

Figure 7 shows the experimental results for one of the best prototypes. For this unit, 2.0 W of net cooling power was achieved for 104 W of electric power into the compressor and a 300 K warm-end reject temperature kept constant using water cooling. Or in other words, if the input power is keep constant at 100 W, the cooling power is 1.93W at 60 K.

Comparing Figure 7 with Figure 1, the experimental performance is clearly poorer than the simulation results. The irreversible losses in the practical system are somehow underestimated. However, the general trends between the model and experiments confirm each other, especially in the case where we optimized for maximum performance at 60 K.

The experiments verify the design and optimization method to a large extent. However, it should be pointed out that there still exists an important disadvantage for the prototypes. That is, the performance is remarkably sensitive to the reject temperature. In the experiments, the reject temperature at the warm end is controlled by circulating water at 5ºC to 40ºC. Figure 8 shows the

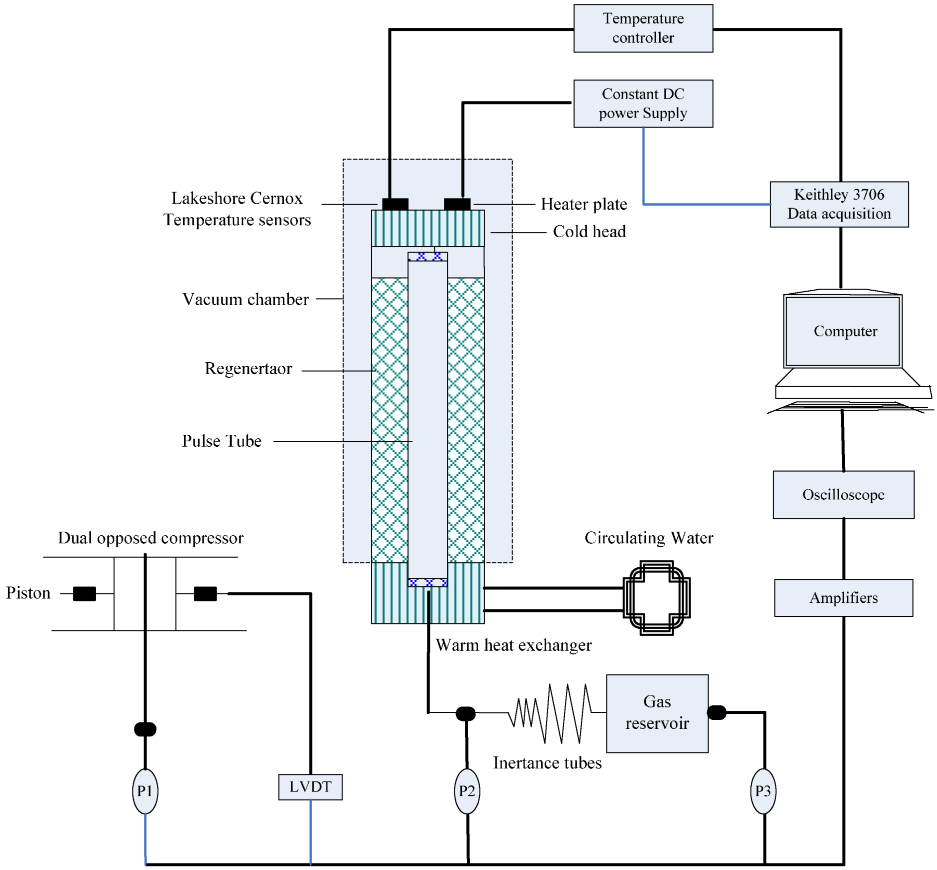

Figure 6. Schematic of the experimental setup for thermodynamic performance.

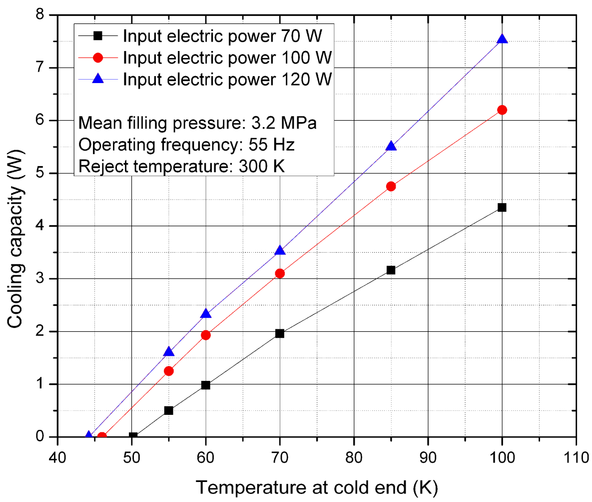

Figure 7. Experimental results of cooling capacity versus cold-end temperature.

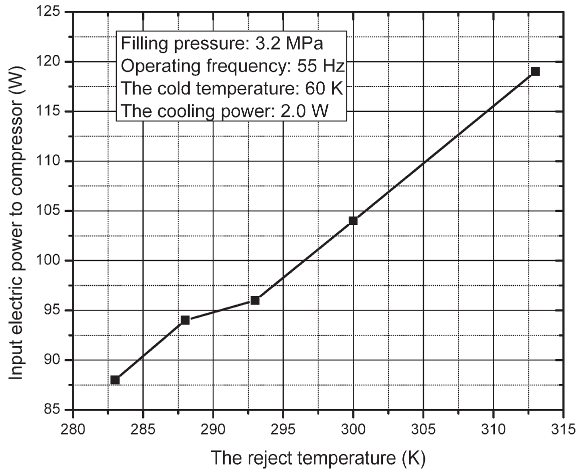

Figure 8. Dependence of the input power at 60 K on the reject temperature.

dependence of the input power of the cooler at 60 K on the reject temperature. For reject temperatures below 300 K, the situation is very favorable. However, if the reject temperature increases, the input power has to increase remarkably to maintain the required cooling capacity. For example, at 313 K reject temperature, the input power has to increase up to 119 W. This phenomenon adversely affects our ability to meet the temperature adaptability requirement given in Table 1. To date, two solutions have been considered: 1) to try and remove the heat from the warm end more efficiently using a better design for the heat exchanger, and 2) to try and achieve a higher overall refrigeration efficiency. One of the measures that we are paying particular attention to is to increase the overall efficiency by achieving a better match between the compressor and the cold finger.

The Match between Compressor and Cold Finger

In the design of a high frequency PTC, the compressor and the cold finger are often considered separately. The compressor is usually considered for its dynamic characteristics, such as force, damping, and resonant frequency. While for the cold finger, the thermodynamic characteristics are always the primary focus. In fact, the two characteristics are not isolated. They may have a strong impact on each other.

In our experiments under the 2W at 60 K conditions, we have observed two phenomena: One phenomenon is that the compressor's efficiency remains quite low at reject temperatures from 10 to 43°C. Even at 10°C, it is still below 64.4%. However, the measurements show that the compressor is working very close to its resonant frequency. The obvious mismatch between the compressor and the cold finger means that there is still much room for increasing the overall efficiency.

The other meaningful phenomenon is that there exists a steady decrease in compressor efficiency with reject temperature as shown in Fig. 9. Although it is not very obvious, there is definite evidence that the thermodynamic parameters of the cold finger can influence the compressor. The mutual influence of the compressor and cold finger are being considered systematically, and a detailed investigation is underway.

CONCLUSIONS

SITP/CAS has developed a series of high frequency single-stage coaxial pulse tube cryocoolers to provide reliable low-noise cooling at 60 K for the space-borne long wave infrared focal plane array. Novel heat exchanger configurations at the cold and warm ends have been developed, while other investigations have made some progress on the temperature mismatch of the tubes, and the characteristics mismatch of the compressor and cold finger. At present, the cooler prototypes real-

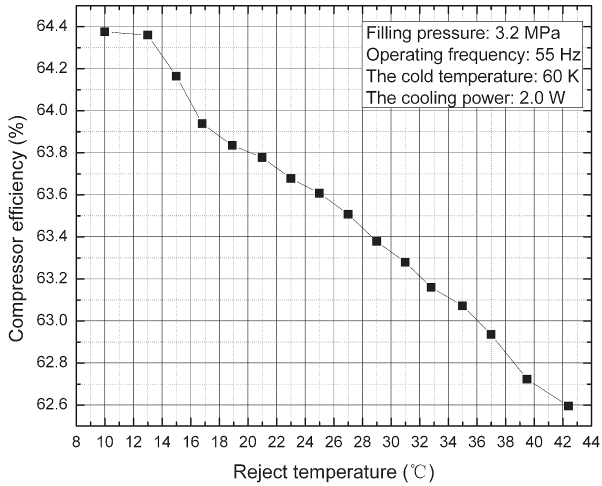

Figure 9. Variation of compressor efficiency at 60 K with reject temperature.

ize 8.0% of Carnot at 60 K, and can typically provide 2 W at 60 K with 104 W of electric input power at 300 K reject temperature. A cooling capacity of 2.5W at 60 K can be achieved when the input power is increased to 127 W. Further optimizations are underway, and it seems promising that the thermodynamic performance can meet the development goal in the near future. Temperature adaptability, vibration output, reliability, and life expectancy will be the next areas of research focus.

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