Design of a 4K Hybrid Stirling/Pulse Tube Cooler for Tactical Applications

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ABSTRACT

Iris Technology is leading a three-partner consortium in the development of a low temperature cryocooler for superconducting electronics (SCE) applications. The intended applications are tactical, so cost is a primary concern, as are package size and mass.

Due to the lack of a correlated thermodynamic model for this new type of three-stage Stirling / pulse tube / pulse tube hybrid cryocooler, the modeling efforts are premised upon using *multiple* proven cryocooler software modeling tools to enhance modeling confidence. Each tool is used for the aspect of the modeling for which it is particularly well suited. Computational domains are deliberately overlapped. Using this technique, a thermodynamic design was developed that predicts net refrigeration of 4.0 W at 60K, 0.30 W at 11.5K, and 0.2W at 4K for 800 W of input pressure-volume (PV) power. The next step in the development is the build and test of a sufficiently representative cryocooler to correlate the model.

Detailed design of a prototype cryocooler for this purpose is underway. To reduce development cost, the prototype will make heavy reuse of existing assets. These constraints, how the team is working around these constraints, the resulting projected performance of this demonstration prototype cooler, and the analysis methods used are presented. The initially targeted performance is 50 mW @ 5K, which the authors assert is sufficient to both correlate the model and demonstrate the three-stage Stirling/pulse tube hybrid cryocooler concept.

INTRODUCTION

The 4K Tactical Cryocooler (4KTC) Concept Demonstration Program, a Phase II STTR, was awarded to Iris Technology Corporation by the Office of Naval Research (ONR) in September 2011. The primary objective of the program is to demonstrate a 3-stage hybrid Stirling/pulse tube/pulse tube "Demonstration Cryocooler" that is a pathfinder for a future productized 4KTC intended for use in low temperature superconducting electronics applications.

The objective operating point for the productized 4KTC is 200 mW of cooling at 4K. The cooler will additionally provide nominal capacity at the two higher temperature stages to intercept heat load and possibly cool intermediate electronics, such as low noise amplifiers. The

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objective cryocooler will be compact, lightweight, and affordable, with a price point substantially closer to that of a present day tactical cryocooler ($5000-$15,000) than a space cryocooler (>$2M). In short, ONR sought a solution that is substantially more compact, efficient and reliable than traditional commercial Gifford-McMahon type cryocoolers, which are the present standard for closed-loop 4K cryogenic refrigerators.

At the core of the effort is the development of a numerical model premised upon using three proven cryocooler software modeling tools. Each tool is used for the aspect of the modeling for which it is particularly well suited. Computational domains are deliberately overlapped so that in key areas (like the 3rd stage regenerator) the results from different software tools can be compared to enhance modeling confidence.

The primary objective of Phase II is to obtain experimental test data to substantiate the analytic predictions. In a cost savings move, this is to be accomplished by leveraging a new purpose-built pulse tube cold head (i.e., the 2nd and 3rd stages) with existing cryocooler assets, namely a compressor and Stirling expander 1st stage and warm end. Successful build and test of this cryocooler will represent the first build of this type of three-stage cryocooler, thus establishing the viability of the concept. Furthermore, it will provide an invaluable tool for the model correlation to inform the productized versions to follow.

OBJECTIVE CRYOCOOLER DESIGN

A thermodynamic design was developed during Phase I that provides net refrigeration of 4.0 W at 60.0K, 0.30 W at 11.5K, and 0.20W at 4.0K for 800 W of input pressure-volume (PV) power. This performance was achieved using helium-4 (He-4) as the working fluid throughout the cryocooler, which is better supportive of meeting the affordability objective than, say, a design requiring the use of helium-3 (He-3).

The preliminary mechanical layout is depicted in Fig. 1. The design is based strongly on the Raytheon High Capacity Stirling/Pulse Tube Two-Stage (HC-RSP2) Cryocooler[1] with the fundamental difference being the additional of a third pulse tube stage[2]. The resulting 3-stage cryocooler has been coined the “RSP3.” A dual-opposed piston, linear compressor generates a pressure wave to drive the expander. The Stirling expander motion, i.e., the stroke amplitude and phase angle relative to the pressure wave, is controlled by a linear motor in the expander warm end. The pulse tube expansion phase angles are set for the second and third stages by the inertance tube – surge volume assemblies, as shown.

Figure 2 provides an enhanced view of the cold head to illustrate the key thermal interfaces. The driving requirement for the cryocooler is the 200 mW @ 4K refrigeration load. The upper stage temperatures have been set at 11.5K and 60K somewhat arbitrarily. The Sage™ “Low Temperature” cryocooler design software (version 8) was used to help establish intermediate temperatures which minimize the input power to achieve the objective 4K refrigeration[3], although this optimization was admittedly not exhaustive. When the objective system thermal requirements are defined, the temperature setpoints will be revisited.

Similarly, the upper stage refrigeration loads were set according to thermodynamic-based design rules in the absence of a complete system context. The approach was, in the manner described by Kirkconnell and Price[4], to balance the “normalized” refrigeration capacity of the third and first stages:

$$\dot{Q}_i^{norm} = \frac{\beta_{c,L}}{\beta_{c,i}}\left(\dot{Q}_i\right) \tag{1}$$

where $\dot{Q}_i^{norm}$ is the normalized refrigeration capacity, $\beta_{c,L}$ is the Carnot efficiency of the lowest temperature stage in the system, and $\beta_{c,i}$ is the Carnot efficiency of the i^{th} stage. Carnot efficiency is in all cases based upon the cryocooler heat rejection temperature (nominally ambient), T_H, and the cryogenic temperature of the i^{th} stage, T_i:

$$\beta_{c,i} = \frac{T_i}{T_H - T_i} \tag{2}$$

For T_H = 300 K, the normalized refrigeration capacities are 200 mW at the 3rd stage and 216 mW @ the first stage, so the stages are indeed balanced. Finally, the second stage capacity was set at a value substantially above 0, in this case 300 mW, to help assure that indeed that stated 11.5K temperature will be reached with some margin. This is critical because if either of the upper stages "no load" at a higher temperature than predicted, then the entire model will be off because the temperature distribution within the cryocooler, and hence the working fluid mass distribution, will be incorrect. This would result in inaccuracies in the parasitic loss calculations, phase angles, mass flow rates, etc.

DEMONSTRATION CRYOCOOLER DESIGN

Hardware Design Constraints

The primary objective of the present research is to correlate the numerical model for the RSP3 Cryocooler. This requires the build and test of a complete cryocooler, the cost of which unfortunately exceeds the available budget. Therefore, a creative programmatic approach was devised to leverage the reuse of existing cryocooler assets, specifically a compressor and Stirling expander first stage, and the warm end. Thus, the only new hardware that is being designed

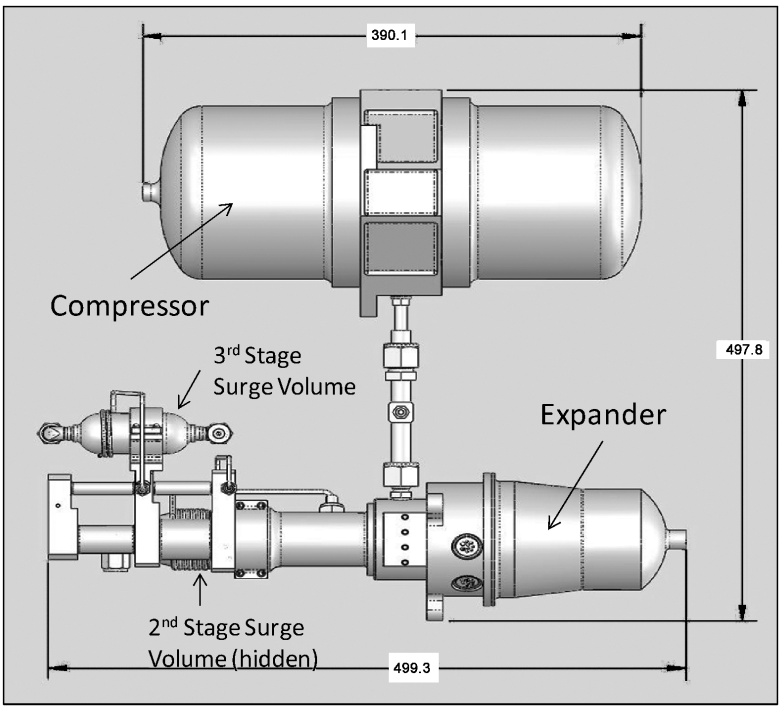

Figure 1. Preliminary 4KTC Objective Cryocooler Design with RSP3 Configuration. Dimensions shown are in mm. Sized to achieve 200 mW at 4K. Second and third stages shown as U-tubes, but eventual product is expected to utilize a concentric design once the final pulse tube and regenerator dimensions are established experimentally.

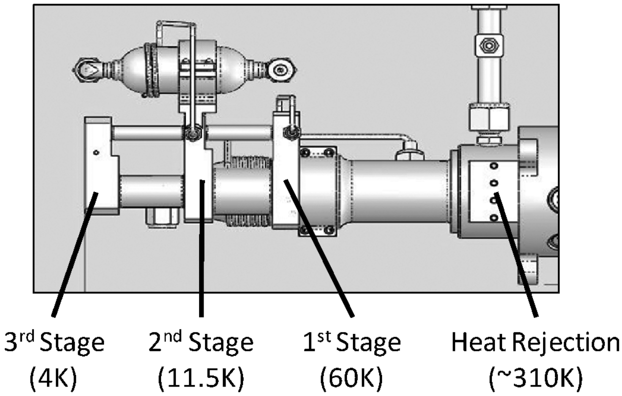

Figure 2. RSP3 Expander Thermal Interfaces. Note that the 2nd stage surge volume resides at the 1st stage temperature. By that same token, the 3rd stage surge volume resides at the 2nd stage temperature.

and fabricated is the passive two-stage pulse tube cold head, which is inexpensive as compared to the compressor and expander moving mechanism.

The compressor that is being used was designed and built by Raytheon for the United States Air Force on a previous program. Called the High Capacity Dual Use Cryocooler (HC-DUC) compressor, it is essentially a scaled up version of the compressor disclosed in a previous paper[5]. The HC-DUC compressor has been extensively characterized at Raytheon and demonstrated to operate safely at up to 650 W of input electrical power, which is sufficient for the Demonstration Cooler's purposes. The primary regret with respect to its applicability for this application is that its designed mechanical resonance is about 15 Hz higher than the targeted resonance speed for the 4KTC.

The warm end of the expander, which primarily consists of the displacer and balancer motors and associated flexure suspension assemblies, is also being provided by the United States Air Force[1]. The first stage Stirling displacer piston is also being provided, which means that the key first stage thermodynamic dimensions, in particular the regenerator length and diameter and piston stroke amplitude, are fixed. These obviously become constraints on the Demonstration Cooler design. Fortunately, the normalized refrigeration (i.e., the exergetic heat lift) required of the Stirling stage in the 4KTC Demonstration Cooler is less than but of the same order as required for the original HC-RSP2 application, so using the HC-RSP2 Stirling first stage is actually a good, conservative fit for this application.

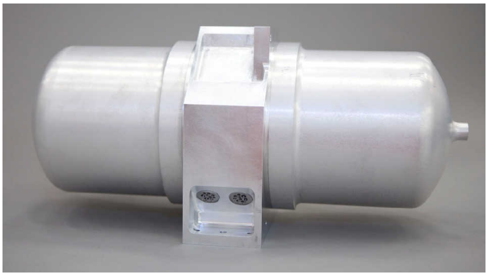

Figure 3. Raytheon HC-DUC Compressor. Nominal dimensions are 381 mm length and 185 mm diameter.

Numerical Modeling

The basic numerical modeling approach is described in Figure 4. As the first step in our analysis, REGEN version 3.3 was used to develop a preliminary design for the 3rd stage regenerator. More specifically, REGEN was used to help select the optimum rare earth material, objective hydraulic diameter (i.e., sphere particle size), length, and diameter.

The next step was to use Sage (version 8 with the Low Temperature module) to develop thermodynamic dimensions for the overall cryocooler to achieve the targeted 50 mW at 5 K. Per the previously described hardware constraints, the compressor and Stirling first stage dimensions were fixed in accord with the reuse hardware dimensions. For a starting point, the existing HC-RSP2 2nd stage dimensions (pulse tube, regenerator, inertance tube, etc.) were used for the 4KTC 2nd stage. These dimensions were subsequently allowed to vary. The 3rd stage regenerator initial guess was based on the REGEN 3.3 result as indicated, and rough estimates for the 3rd stage pulse tube and inertance tube were made based upon the mass flow rates indicated as optimum from the REGEN result.

The maximum fill pressure was constrained to proven-safe limits of the existing reuse hardware. The drive frequency was allowed to vary; the dynamics of the suspension system were modeled in Sage, along with the thermodynamic dimensions, to the power penalty of driving off-resonance was taken into account. The electrical input power to the HC-DUC compressor was limited to the proven-safe level of 650 W.

The key optimization target was the 3rd stage capacity. Additionally, we required that positive refrigeration capacity be achieved at both upper stages. As previously stated, it is critically important that the targeted upper stage temperatures be achieved, even if the targeted capacity is not, if the 3rd stage capacity is to be achieved. For this experiment, the 3rd stage capacity is indeed by far the most critical objective. We initially targeted a minimum of 1 W of cooling at the first stage and 200 mW at the second stage.

To our knowledge, a Stirling / pulse tube / pulse tube of this type has never been built. With certainty we can state that our Sage model for this cryocooler has yet to be validated. Therefore, following the establishment of the initial Sage model, it was decided to recheck the regenerator design and performance prediction against REGEN for independent confirmation. This was accomplished for the 3rd stage regenerator. The net enthalpy flux predictions at both ends of the regenerator agreed to within 1 mW. We plan to do a similar check on the 2nd stage regenerator in the near future.

The key remaining step as of this writing is the utilization of FLUENT to help provide additional confidence in the design. We presently plan to use FLUENT, at a minimum, to help guide the design of the more complicated flow transitions. In particular, at the interface between the second and third stages, the flow splits. Some of the flow exiting the second stage regenerator gets expanded in the second stage pulse tube, and the remainder passes into the third stage regenerator to drive the third stage refrigeration cycle. FLUENT will be used to help optimize the design for low pressure drop, uniformity of flow into and out of the regenerators, and straightening of the flow into the second stage pulse tube.

RESULTS

To date, the first three major steps depicted in Figure 4 have been completed. The hardware constraints have made it very challenging to achieve the targeted performance to date, but it does appear that the construction of a Demonstration Cooler that supports the accomplishment of the key objectives, namely model correlation and proof of the RSP3 concept, is achievable.

The current predictions are provided in Table 1, along with a comparison to the original targets for the Demonstration Cooler. Although the margins are not as large as desired, positive refrigeration is predicted for the upper stages. Similarly, the third stage capacity is not as high as desired, but 5K is nevertheless achieved with some margin.

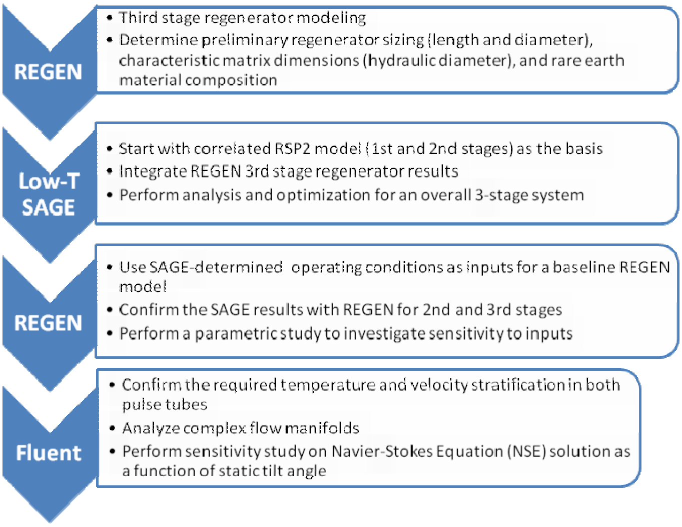

Figure 4. Modeling Methodology for the Demonstration Cryocooler.

Table 1. Performance Predictions for the Demonstration Cryocooler.

	Units	Target	Predicted
1st stage temperature	K	50-60	55
2nd stage temperature	K	10-15K	11.5
3rd stage temperature	K	5.0	5.0
1st stage heat lift	mW	1000	600
2nd stage heat lift	mW	200	180
3rd stage heat lift	mW	50	28

CONCLUSION

The initial design for a three-stage 5K Cryocooler has been completed. To date, the key software design tools, Sage and REGEN, have been employed. In the key thermodynamic region of the 3rd stage regenerator, the two software tools have been shown to provide essentially identical results, enhancing modeling confidence. The next step is to employ FLUENT to help guide the design of key flow transitions.

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