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Cite as: AIP Conference Proceedings **710**, 1285 (2004); https://doi.org/10.1063/1.1774816

Published Online: 16 July 2004

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RAYTHEON STIRLING/PULSE TUBE TWO-STAGE (RSP2) CRYOCOOLER ADVANCEMENTS

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ABSTRACT

Raytheon is developing a two-stage hybrid Stirling/pulse tube cryocooler for long life space infrared (IR) sensor applications. The first expander stage is a conventional Oxford-class Stirling expander. The second expander stage is a U-turn pulse tube mechanically and thermodynamically extended from the first stage Stirling cold end.

Recent development work has been devoted to improving the math model correlation in preparation for construction of an optimized expander design. We have obtained a large performance database using a variety of phase shifters and surge volume arrangements with the cryocooler operating over a wide range of operating conditions and temperatures. This database not only forms the foundation of the model correlation effort, but it also is being used to identify the optimum phase shifter and surge volume designs for various operating regimes of interest.

Development testing of the first unit is now complete and a second, improved expander is being developed based on lessons learned. The new expander is being designed for cold stage service in the 30 K to 80 K range.

INTRODUCTION

FIGURE 1 shows the first RSP2 brassboard test unit [1-3]. The dual opposed compressor is a conventional Oxford class mechanism with a hermetically sealed housing, tangential suspension flexures, linear voice coil drive motors, linear variable differential transformer (LVDT) position sensors, and non-contacting clearance seals. These design features meet long life and high reliability requirements and have been widely demonstrated in the space cryocooler industry. A transfer line separates the compressor and expander modules, which assists in isolating the first stage from vibration. The expander first stage is a conventional Oxford class Stirling with spiral suspension flexures, an integral balance mass, linear voice coil control motors, and LVDT position sensors. The compressor can be configured for 6.0 cc to 7.5 cc swept volume; the first stage expander has a maximum swept volume of about 1.0 cc. The entire unit weighs about 8 kg.

CP710, *Advances in Cryogenic Engineering: Transactions of the Cryogenic Engineering Conference - CEC, Vol. 49,* edited by J. Waynert et al.
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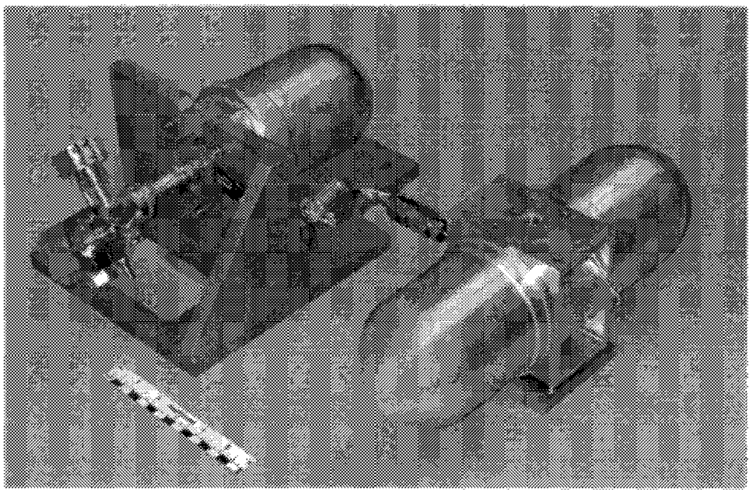

FIGURE 1. RSP2 brassboard test unit has accumulated over 2000 hours operation in the laboratory without failure or anomaly.

The second stage is a U-turn pulse tube pneumatically in series with the first stage. The U-turn configuration provides compactness, structural rigidity, and allows the second stage to orient at any angle versus the first stage. The expander has efficient flow through heat exchangers between the first stage outlet and second stage regenerator inlet, between the second stage regenerator outlet and the pulse tube, and at the pulse tube outlet leading to the phase shifting device. The first stage thermally anchors warm ends of the regenerator tube, pulse tube, phase shifter and surge volume. Measured performance includes 1.75 W at 58 K (second stage) with simultaneous 5.0 W at 110 K (first stage).

In application, the RSP2 cryocooler can simultaneously cool a space-based IR detector system's focal plane array, optical components, and thermal shielding. Such components are normally at two different temperatures and sometimes with varying heat loads. The RSP2 is flexible enough to support these heat loads due to its hybrid configuration. By adjusting the mechanical phase angle between the compressor and expander piston, refrigeration capacity can be shifted between stages in real time. Additional advantages of a combined Stirling and pulse tube expander over current state-of-the-art approaches include higher efficiency, compactness, weight reduction, and lower production costs.

The development of a second RSP2 cryocooler began by correlating experiments with the brassboard unit to predicted performance data generated by an updated thermodynamic model developed using a combination of commercial codes. The results are first discussed. An explanation of changes made to the experimental RSP2 that evolve it into a new and improved design follows. Numerical studies focused on the new cryocooler are then presented. Finally, images are provided to illustrate an important design difference between the brassboard and improved cold head designs.

MODEL CORRELATION

A major objective of the testing was to correlate cooler performance with that predicted by the new thermodynamic code. A broad range of performance data was obtained with various operating conditions. During the course of the testing, the first stage regenerator and second stage heat exchangers were physically optimized within the constraints available of the existing hardware. A sample of correlated data is shown in TABLE 1, which compares heat loads at the two stages and compressor PV power for the RSP2 brassboard cooler versus the math model. Model quality is determined by a normalized error, defined as:

$$Error = \frac{|\ measured - predicted\ |}{measured} \tag{1}$$

First stage refrigeration capacity (Q1), second stage capacity (Q2), and compressor input thermodynamic (i.e., pressure-volume) power (PV comp) all generally agree within ten to fifteen percent. TABLE 2 illustrates that higher order model correlation involving internal operating characteristics of the device provides similarly satisfying results. (Note that the tabular values are dimensionless, i.e., the entries have been normalized to 1.0 based upon the measured performance of the cooler.)

TABLE 1. RSP2 cryocooler performance data correlation results with math model over a wide range of operating conditions.

Operating Parameters			*Measured Performance*			*Predicted Performance*			*Normalized Error*		
T1 [K]	**T2 [K]**	**Phase [deg]**	**Q1 [W]**	**Q2 [W]**	**PV comp [W]**	**Q1 [W]**	**Q2 [W]**	**PV comp [W]**	**Q1 [W]**	**Q2 [W]**	**PV comp [W]**
120	120	74.3	3.70	3.26	66.5	3.61	2.86	63.8	0.03	0.12	0.04
120	120	80.1	4.59	3.18	70.2	3.96	2.81	68.1	0.14	0.12	0.03
100	100	79.9	2.76	2.70	74.9	2.67	2.39	74.7	0.03	0.12	0.00
100	100	80.5	2.75	2.71	73.0	2.41	2.33	72.4	0.13	0.14	0.01
100	100	80.0	2.88	2.73	75.0	2.56	2.35	74.5	0.11	0.14	0.01
110	58	73.3	3.55	1.26	80.9	3.42	1.13	82.3	0.04	0.10	0.02
110	58	89.6	4.85	0.91	77.0	4.13	0.94	78.6	0.15	0.03	0.02
90	45	84.9	2.79	0.44	77.7	2.25	0.48	77.9	0.20	0.08	0.00

TABLE 2. Higher order model correlation comparing various phase angles and pressure ratios for line 1, Table 1 operating point. PV = thermodynamic power, PR = pressure ratio (Pmax/Pmin), P-V phase = local phase angle between pressure and volume, T = temperature. Measured values all normalized to 1.0. Worst case error = 20% for P-V phase, exp1 (first stage expansion).

Operating Parameters		*Measured Performance*	*Predicted Performance*
Q1	[-]	1.00	0.97
Q2	[-]	1.00	0.88
PV compressor	[-]	1.00	0.96
PV expander	[-]	1.00	1.12
PV pulse tube	[-]	1.00	0.96
PR transfer line	[-]	1.00	0.99
PR surge volume	[-]	1.00	1.00
PR stirling cold tip	[-]	1.00	0.99
P-V phase, comp	[-]	1.00	0.99
P-V phase, exp1	[-]	1.00	1.20
T transfer line	[-]	1.00	1.00

RSP2 IMPROVEMENTS

Applying multi-stage cooler optimization techniques previously discussed [4], the correlated model has been used to design a second, improved RSP2 expander. The model has been used to optimize heat exchangers and regenerators and to identify and remedy flow restrictions. Most notably, the brassboard was retrofitted with an inertance tube and the resulting data correlated to the math model. TABLE 3 illustrates the performance improvement in the brassboard test unit with the introduction of an inertance tube. Even greater improvement is possible in the improved cold head because the change to an inertance tube necessitates a change to the pulse tube for optimum performance, an option not available for the brassboard retrofit. This is discussed further in the pages to follow.

The new design is shown in FIGURE 2. The first round of testing will utilize the same compressor and many of the same expander parts, but will have a new cold head as shown. Predicted performance for the upgrade is 1.1 W at 40 K plus 6.6 W at 110 K for 150 W input power to the motor.

TABLE 3: Numerical analysis of brassboard improvement with introduction of inertance tube. Results correlate well to experimental data.

	110 K/58 K; 38 Hz		*110 K/40 K; 45 Hz*		*80 K/40 K; 45 Hz*	
	orifice	**inertance tube**	**orifice**	**inertance tube**	**orifice**	**inertance tube**
Q1 [W]	3.421	3.421	4.779	4.270	2.107	2.107
Q2 [W]	1.130	1.502	0.0632	0.5424	0.4306	0.8381
PVcomp [W]	82.34	85.38	110.5	120.1	116.8	123.2

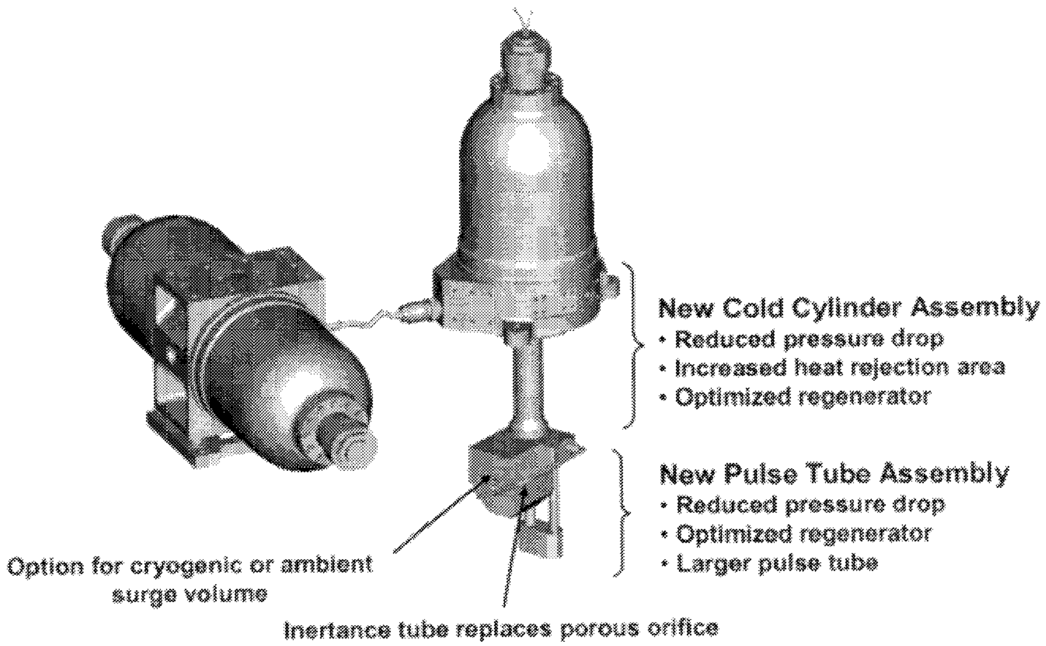

FIGURE 2: New and improved RSP2 cryocooler cold head design. Note option for cryogenic or ambient surge volume (cryogenic shown). Brassboard expander shown (left); flight version currently in development.

NUMERICAL STUDIES

The hybrid Stirling/pulse tube has the ability to dramatically reallocate refrigeration capacity between stages at constant input power by varying the mechanical phase angle of the expander piston [2]. In contrast, multistage pulse tubes have no mechanism for active phase control; single-piston, two-stage Stirlings have phase-locked expansion zones, so any "load shifting" that occurs with phase angle is a second-order byproduct of regenerator saturation [5]. Thus the unique combination of decoupled expansion mechanisms and active first stage phase control in the Stirling/pulse tube provides for a substantial capability to shift capacity without varying the input power. This is of interest because it allows the cooler to perform over a wide range of temperature and load requirements, as well as simultaneously manage cyclical loads at two different temperatures.

Numerical studies were conducted to quantitatively investigate this phenomenon for the new cold head design. As discussed previously [4], the first stage net refrigeration can be increased while second stage refrigeration decreases by simply increasing the motor-controlled phase angle (and vice versa). FIGURES 3 through 5 show the predicted performance of the improved RSP2 design for a range of Stirling piston phase angles.

The new brassboard RSP2 cryocooler is designed to operate with a nominal 75° phase angle. At this baseline, the refrigeration is approximately 6.6 W at 110 K and 1.1 W at 40 K. When the phase is lowered by fifteen degrees, Q1 drops to 4.0 W. However, Q2 increases to about 1.3 W. The reverse trend occurs when the cooler is operated at 90 degrees, where Q1 increases to 8.0 W while Q2 drops to 0.9 W. Similar performance trends occur when the temperatures of the two stages are increased, although the refrigeration capacity is greater overall. The amount of input PV power required by the compressor only increases a small percentage as the mechanical phase angle increases.

Load shifting provides advantages to the user over the entire life cycle of the program. The operating point of the cryocooler can be adjusted as thermal requirements evolve during the formulation phase. Real time adjustments can be made in system-level thermal vacuum testing without having to disassemble the test setup nor rework the cryocooler. Finally, the phase shift command can be uploaded on orbit, so the cooler can react to thermal environments different from that predicted prior to launch.

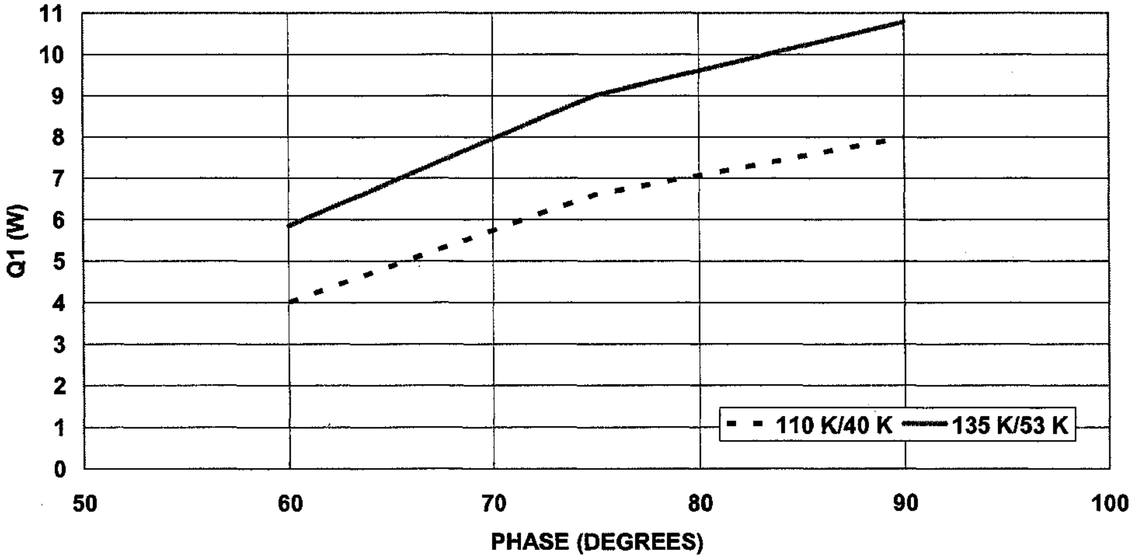

FIGURE 3. RSP2 parameter study; first stage refrigeration versus mechanical phase angle.

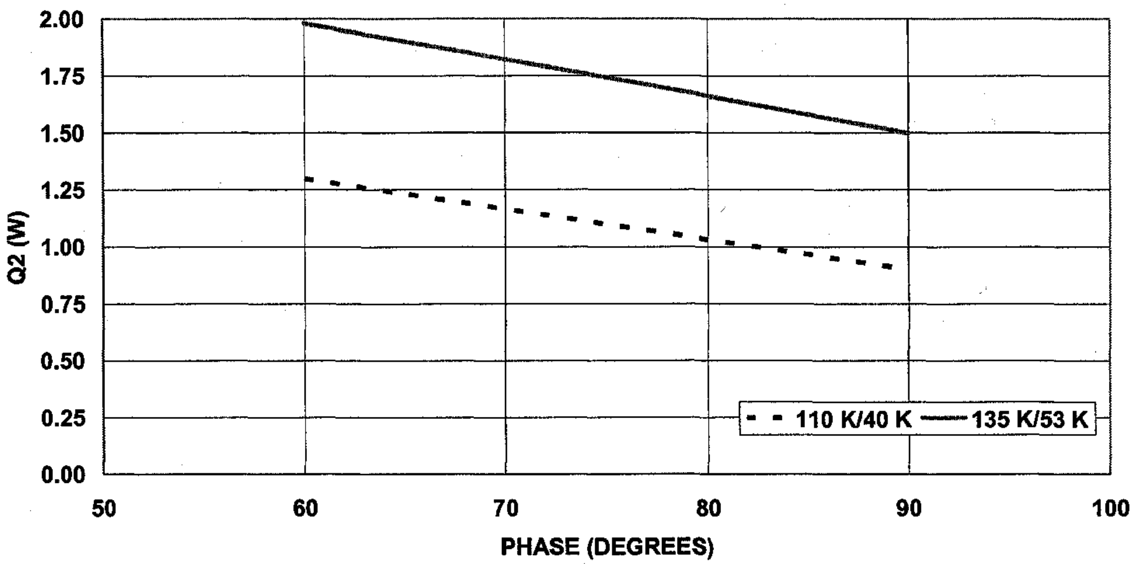

FIGURE 4. RSP2 parameter study; second stage refrigeration versus mechanical phase angle.

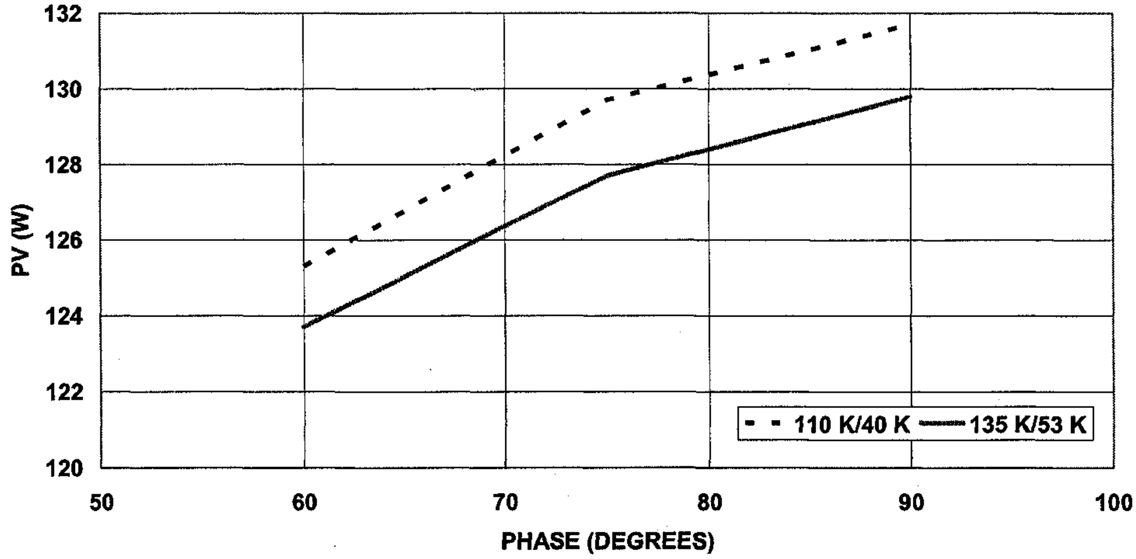

FIGURE 5: RSP2 parameter study; compressor input power versus mechanical phase angle. Small variation is power (~5%) for 30 degree phase shift.

HARDWARE PROGRESS

The first test of the new cold head design will reuse the compressor module and expander motor assembly from the original RSP2 brassboard. Several components of the improved RSP2 cryocooler cold head have already been fabricated with delivery of the final piece parts and major subassemblies expected in November 2003.

Much of the recent development efforts have focused on optimizing the cold head assembly to meet projected refrigeration capacity requirements for various programs of interest. Braze coupons from the original brassboard unit and the new cold head are shown side by side in FIGURE 6 to illustrate one example of how the physical design has changed. The original brassboard is on the left, the new cold head on the right. Notice the difference in the geometric relationships between the regenerator and pulse tube for each assembly. For the new RSP2 cold head, the pulse tube is significantly larger. This change arises from the introduction of an inertance tube as a phase shifter. The inertance tube results in a larger displacement of the pulse tube gas column versus that observed with fixed orifice, thus the need for a larger pulse tube to maintain pulse tube flow losses (particularly enthalpy streaming) acceptably low.

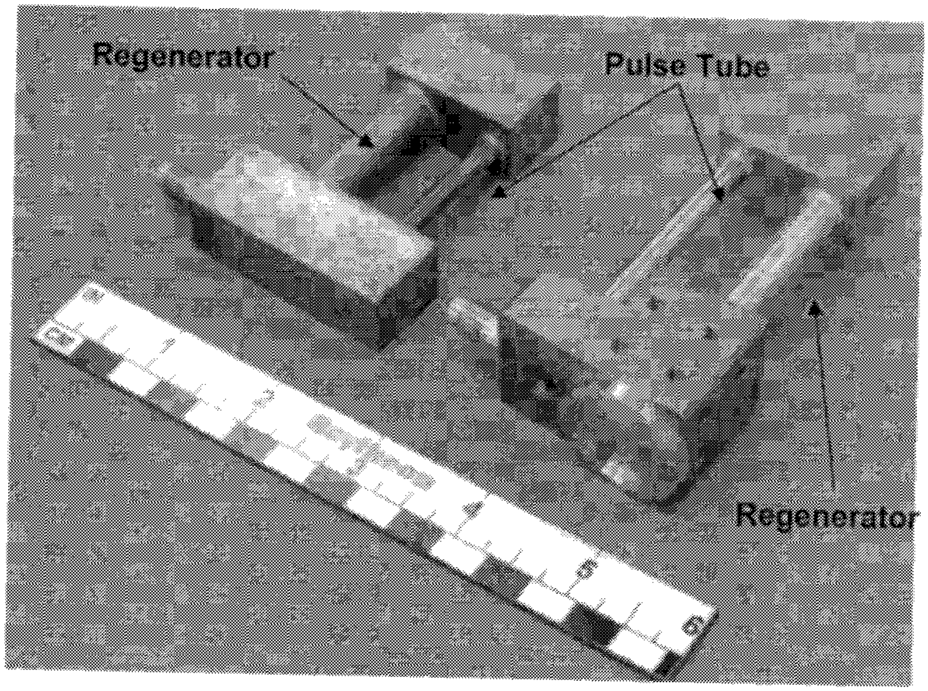

FIGURE 6. Comparison of braze coupons from original brassboard cold head assembly (left) and new RSP2 cold head (right). Tube sizes same as actual hardware. Cold blocks similar to actual unit but modified to meet objective test requirements.

SUMMARY

Engineering designs for an improved Raytheon Stirling/Pulse Tube Two-Stage cryocooler are now complete. The thermodynamic models have been successfully correlated against the brassboard RSP2 unit. Hardware components for major subassemblies have been received. The new RSP2 expander module will be assembled and integrated with the compressor by the end of 2003. The first cool-down test will be performed starting January 2004.

ACKNOWLEDGEMENTS

The authors would like to acknowledge the valuable contributions of the "ABC" design team, namely: Abram Alaniz, Mike Barr, and Ken Ciccarelli.

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