Experimental Investigation of a Unique Pulse Tube Expander Design

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ABSTRACT

The performance of a pulse tube expander in which the regenerator volume is distributed among three parallel tubes arranged symmetrically around the pulse tube has been experimentally investigated. This "4-tube" expander configuration, which was recently patented by Raytheon Systems Company (formerly Hughes Aircraft), is of interest because it has structural advantages over the more common U-tube design. The improved strength of the 4-tube design permits reduced wall thickness, higher allowable side loads, and related design and system integration advantages. Furthermore, the 4-tube expander does not have the problem of conductive coupling between the regenerator and pulse tube, which is a source of lost refrigeration capacity in the traditional concentric configuration, a competing rigid expander design. Experiments were performed with and without the regenerator tubes linked by conductive straps, and the expander was shown to work more efficiently with the straps. The data reveal that the thermodynamic efficiency of the 4-tube expander is comparable to that of the more common pulse tube expander configurations.

INTRODUCTION

As pulse tube cryocoolers have become competitive with the more well-established Stirling cryocoolers in the realm of thermodynamic efficiency and have thus become a practical alternative in many applications, there has been an increased interest in pulse tube cryocooler implementation and system integration issues [1,2,3]. Allowable side load, residual vibration amplitude, and cold mount location are examples of common mechanical integration concerns. With respect to the latter, locating the cold region at one end of the expander, as in a Stirling or folded pulse tube, eases the design integration of the cooler into the rest of the system. Residual vibration control is typically accomplished through an active balancer in the compressor module, a balanced dual-piston compressor design, a split expander-compressor configuration connected via a transfer line, or a combination of these techniques. The maximum allowable side load concern has been traditionally addressed through the use of light-weight, flexible cold straps to minimize the mass which must be supported and the structural loads which are transmitted to the cold tip.

The 4-tube pulse tube expander configuration specifically addresses two of these system integration issues, cold mount location and allowable side load. Like the concentric and U-tube designs which have been discussed previously [4,5,6], the 4-tube is a variation on the folded

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pulse tube theme with a distinct cold tip at one end of the expander. The rigidity of the 4-tube expander is theoretically higher than that of a similarly-sized U-tube because of the symmetry of the design, and analysis is presently underway to substantiate and quantify that assertion. Though the concentric expander yields these same system integration advantages, the thermodynamic performance of the 4-tube is not hampered by radial conduction between the pulse tube and regenerator, as in the standard concentric. (A variation on the concentric design in which the pulse tube and regenerator are separated by an annular vacuum has been discussed [7], and such a design is obviously not affected by pulse tube-to-regenerator radial conduction).

Though the system-integration advantages of the 4-tube are potentially significant, certain aspects of the 4-tube expander design suggest its maximum cooling efficiency may be less than that of the U-tube. The concern stems from the 73% larger perimeter-to-cross sectional area ratio caused by arranging identical regenerator volumes as three parallel tubes versus a single tube of the same length. First, the total regenerator wall cross sectional area is increased proportionately, assuming identical wall thicknesses, so the relative size of the parasitic loss due to conduction through the regenerator walls is higher in the 4-tube. Also due to the larger perimeter-to-area ratio, the percentage of regenerator volume affected by possible boundary layer effects in the vicinity of the walls is greater in the 4-tube. Though the former can be accurately quantified analytically, the latter is not as yielding to simple analytic tools. For these reasons, experiments were performed to measure the performance of a 4-tube pulse tube expander. Experimental data were obtained for three slightly varying expander designs: a baseline design, an intermediate configuration with the three regenerator tubes thermally linked with conductive straps, and a final configuration which includes the thermal straps and an increase in the cross sectional area of some internal flow passages.

EXPERIMENTAL SETUP

4-Tube Pulse Tube Expander Design

A side view of the patented Raytheon 4-tube expander is provided in Figure 1. The total regenerator volume is distributed among the three identical outside tubes, and the smaller center tube is the pulse tube component. The four identical-length stainless steel tubes are brazed into copper heat exchanger manifolds at either end. The large rectangular block is the warm end heat exchanger manifold, and it houses the inlet and rejection heat exchangers. The three inlet heat exchangers, which are located between the compressor and the regenerator in the flow circuit, are located at the warm ends of the regenerator ports. The single rejection heat exchanger is located at the warm end of the pulse tube. This particular expander is of the double-inlet configuration [8]; two external metering valves on the warm-end manifold (not shown in the figure) serve as the bypass and surge valves. The small manifold with the three fingers at the opposite end of the expander houses the cold, or load, heat exchanger. Three "fingers" link the regenerator tubes to the cold heat exchanger at the manifold centerline to allow the four tubes to shrink asymmetrically, a phenomenon which occurs due to uneven cooling, without inducing inordinately large stresses in the braze joints. Such structural compliance is important for any folded tube expander because pulse tubes and regenerators have different temperature profiles [9, 10]. For this particular expander, this design feature is particularly important because the three regenerator tubes themselves also tend to develop differential temperature gradients, as is discussed herein. Other aspects of the design are provided in the patent [11].

Test Apparatus and Instrumentation

The test apparatus consists of the 4-tube expander and a convenient wall-powered laboratory compressor. The compressor, which has been used in other similar studies, is a fixed swept volume (20 cc), fixed frequency (21 Hz) rotary device [12]. The compressor and expander were connected with a seven centimeter transfer line. The expander was specifically sized for use

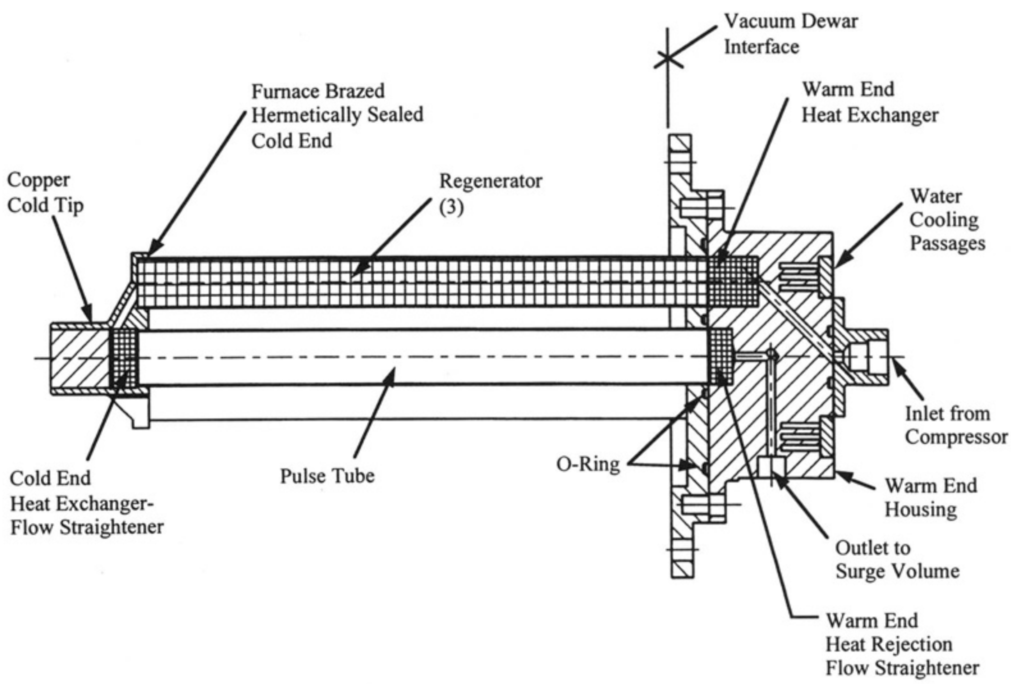

Figure 1. Side view of a 4-tube pulse tube expander.

with this compressor, so the performance data provided herein are a fair assessment of the expander's capabilities.

Figure 2 illustrates the cryocooler assembly and associated instrumentation. The rejection temperature at the warm end heat exchanger manifold was maintained at 291 K using a recirculating glycol-water bath for all of these experiments. The cold end heat exchanger is temperature-controlled with a LakeShore controller, DT470 temperature diodes, and an **80 Ω** resistive heater. The pressure waves in the transfer line, the surge tank, and on the pulse tube side of the surge valve are measured using high-rate Paine strain gauges and amplifiers. A piezoelectric differential pressure transducer in the surge tank provides the amplitude of the small surge volume pressure fluctuations, a direct measurement which facilitates an indirect measurement of the gross refrigeration power produced within the expander. The phase of the compressor piston stroke is provided by a proximity sensor. A cam is mounted to the drive shaft such that it is identically in phase with the drive piston, and the proximity sensor measures the position of the cam, hence providing the piston phase angle. The vacuum environment is maintained by a Leybold vacuum system. Temperatures are measured at the locations shown using welded type-T thermocouples. Note that the temperature of the three regenerator tubes is measured at an identical axial location (1.0" from cold end) to provide a means of evaluating the thermal balance of the regenerator during operation.

EXPERIMENTAL RESULTS

Baseline Expander Design (No Copper Straps)

The initial experiments were performed with the expander as shown in Figure 1, i.e., without copper straps thermally linking the tubes together. A variety of mean operating pressures and valve settings were investigated in an effort to identify the optimum operating condition, defined

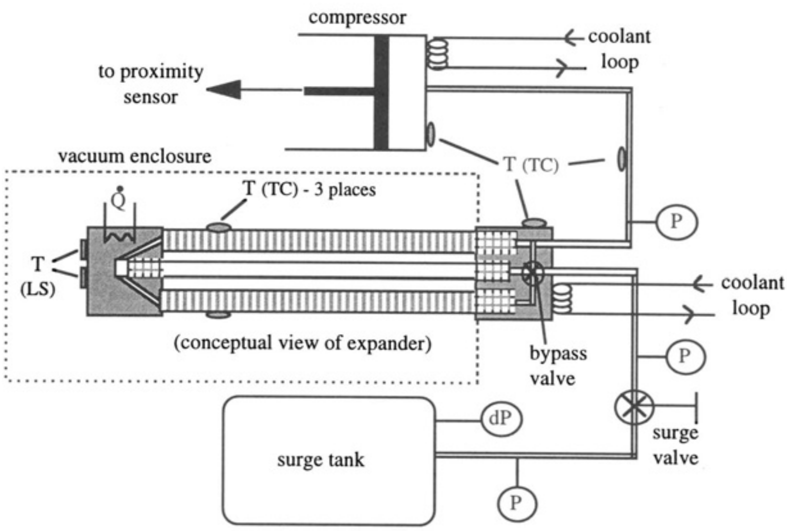

Figure 2. Experimental test setup and instrumentation. P = absolute pressure, dP = differential pressure, T = temperature, LS = LakeShore diode, TC = type-T thermocouple, Q = electrical heat load. See fig. 1 for actual expander geometry.

as that combination of settings which yielded the maximum net refrigeration at 70 K. The piston swept volume and operating frequency are fixed by the design of the compressor, so these parameters were obviously not part of the optimization process. A mean operating pressure of 25.4 atm and a 5.5:1 ratio between the bypass valve and surge valve flow coefficients were identified as the optimum operating parameters. This same operating point was used for the other two configurations to maintain a relevant basis for comparison between the data sets.

The net refrigeration and input PV power versus temperature for the optimum operating condition are shown in Figure 3. (The input PV power is estimated from the measured values of the mean pressure, pressure ratio, and piston phase angle. See reference [12] for details.) The cryocooler yielded 1.5 W net refrigeration at 70 K for 125.4 W input PV power, yielding a specific power (SP; input PV power/net refrigeration) of 83.6. This performance fell well short of the 3.2 W and SP of 41.2 which had been projected for this cold tip temperature from the numerical model (model described elsewhere [13]).

A likely explanation for the shortfall was revealed by the regenerator thermocouples, located at an identical axial location on each tube (1.0" from the edge of the cold-end copper manifold). Figure 4 shows the temperatures on the three regenerator tubes for each of the load points from Figure 3. Note the large temperature variations between the three tubes for each operating point, the difference between the maximum and minimum temperatures ranging from 73.1 K at no load to 62.5 K for a 90 K cold tip. This operating condition was neither predicted by nor accounted for in the numerical model. Given the identical size, environmental loading, and heat exchanger interfaces for the three tubes, the most plausible explanation is an uneven flow distribution between the three tubes due to subtle differences in tube diameter and/or packing density. The next section describes an effort to mitigate this impact of this effect.

Intermediate Test Results (With Copper Straps)

In an effort to balance out the regenerator temperature distribution, copper straps were added at four locations to thermally link the three tubes. An improvement in the balance of the flow distribution and a reduction in the regenerator parasitic losses were anticipated with an improved

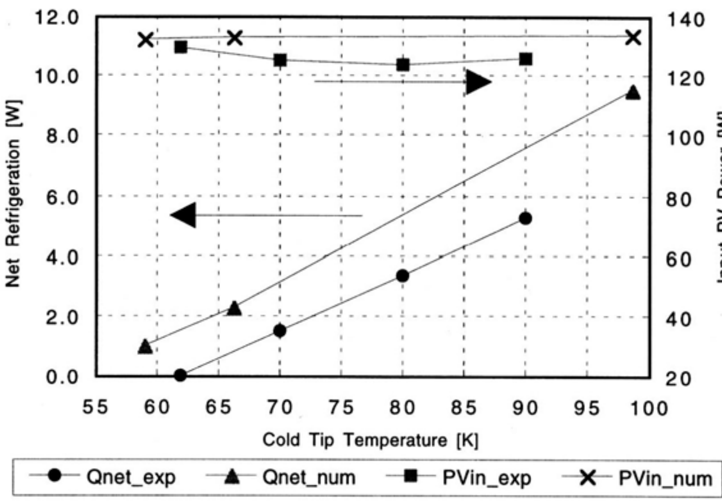

Figure 3. Comparison of experimental and numerically-predicted load and power curves for baseline expander design (no thermal straps). 'exp' = experimental; 'num' = numerical.

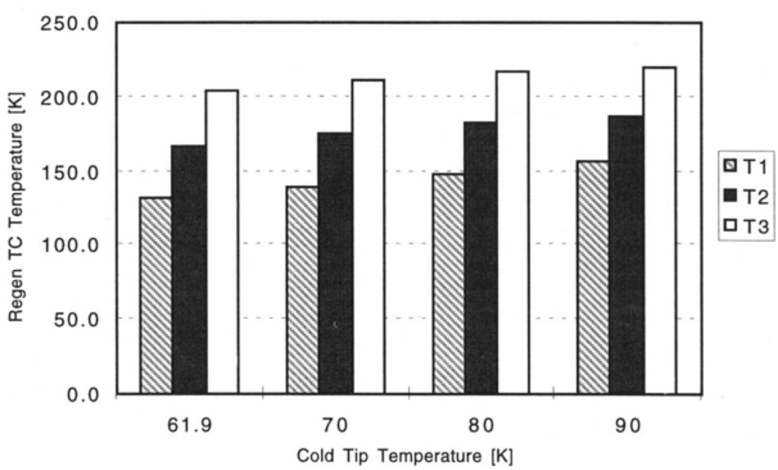

Figure 4. Regenerator temperatures versus cold tip temperature for baseline expander configuration. Cold tip temperatures correspond to experimental load points from Fig. 3. T3–T1 = 73.1 K at no load (61.9 K).

thermal balance. The exact locations of the straps are shown in Figure 5. Each strap was constructed of 3 layers of 9/32 inch wide, 10 mil thick OFHC copper strips with the layers bonded to each other and to the regenerator tubes with "Scotch" 966 "Hi-Temperature" acrylic laminating adhesive. A higher conductivity adhesive could possibly have been used to obtain better performing thermal straps, but the results obtained were impressive, nevertheless.

The reduction in temperature differentials between the regenerator tubes is demonstrated in Figure 6. The maximum temperature delta between tubes was reduced from 73.1 K to 17.2 K, and this translated into improved cryocooler performance. The no-load cold tip temperature was reduced by 3.3 K to 58.6 K, and the load capacity at 70 K was increased from 1.5 W to 2.3 W. Furthermore, the improved flow balance reduced the overall system pressure drop, so the input

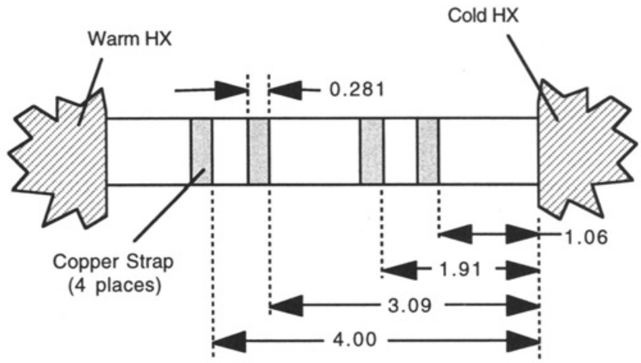

Figure 5. Location of regenerator thermal straps. Each strap constructed of (3) layers of 10 mil copper, bonded to regenerator tubes and bonded layer-to-layer. Straps located at identical axial location on all three regenerator tubes. Dimensions given in inches.

PV power dropped to 117 W, yielding a specific power of 50.9 at 70 K. Though still short of the design goal, this performance marked a significant improvement beyond that which had been obtained without the straps.

Final Test Results (With Copper Straps and Reduced Flow Restrictions)

As described above, the addition of the thermal straps reduced the thermal imbalance between the regenerator tubes and improved the cooling capacity and efficiency of the expander. With the link between a more balanced regenerator and improved performance having been established, an additional step was taken to further reduce this imbalance.

Consider the initial cool down of the refrigerator. At time zero, the entire expander is in thermal equilibrium, including each of the three regenerator tubes. Immediately upon starting the compressor, however, the temperatures between the tubes begin to diverge. Evidently,

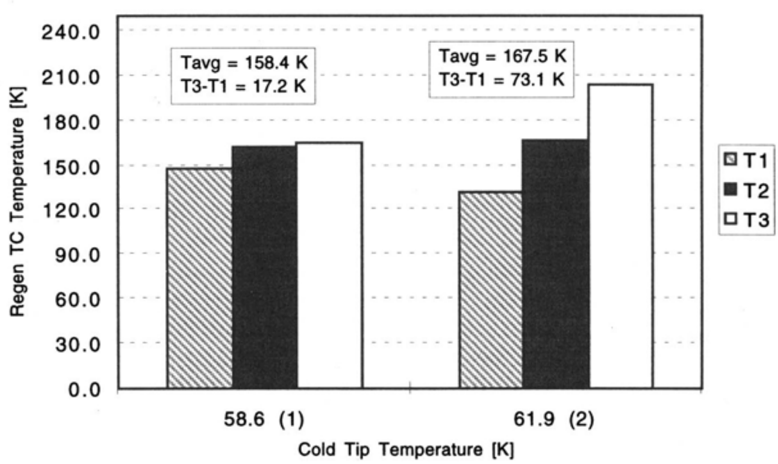

Figure 6. Comparison of regenerator temperatures with and without thermal straps. (1) No-load with thermal straps, (2) No-load without thermal straps. Note the significant decrease in temperature differences between the regenerator tubes due to the addition of the straps.

slightly more refrigeration is produced in the cold tip region of the more efficient tube(s). This results in a faster localized cooling of the gas, which in turn causes the density of the gas in that particular tube to decrease more rapidly, which results in a lower pressure drop and an even larger percentage of the flow being shuttled through the more efficient tube. This leads to an even more rapid cooling of this more efficient tube in relation to the other tubes, hence the temperature differentials develop and persist. Given this relationship between the fluid dynamic and thermal imbalances, it was hypothesized that a reduction in the regenerator pressure drops would reduce the magnitude and impact of any flow imbalances, hence retarding the development of the regenerator thermal mismatch. Towards this end, the regenerator was disassembled and re-packed with a slightly less restrictive screen pack design.

The effects of this change were modest. Figure 7 shows that the reduced pressure drops in the regenerator did indeed result in a further decrease of the regenerator thermal imbalance, with the maximum temperature differential at no load falling to 7.2 K. However, as revealed by the load curve provided in Figure 8, the cooling capacity was unchanged (2.3 W @ 70 K) from the previous configuration, and the input power was only slightly less (110 W vs. 117 W @ 70 K). The substantial improvements in expander performance versus the initial data set (compare Figures 3 and 8) must therefore be attributed to the addition of the thermal straps, not the modification of the internal regenerator design.

DISCUSSION AND CONCLUSION

A pulse tube expander in which the regenerator volume is divided between three parallel tubes has been constructed and tested. After the addition of thermal straps, which yielded a substantial improvement, and the implementation of a slightly less flow restrictive regenerator design, which provided only a modest increase in thermodynamic efficiency, the performance of this 4-Tube expander was shown to be comparable to that of the more conventional pulse tube designs. For example, a U-Tube expander of similar size described in a previous paper [12] provided 4.5 W @ 80 K for 117 W input PV power **(SP = 26.0),** while the 4-Tube yielded 4.3 W @ 80 K for 107 W **(SP = 24.9).** With respect to the modeling projection of **SP = 41.2** at 70 K, the SP of 47.6 achieved for the final configuration at 70 K is in reasonably sound agreement.

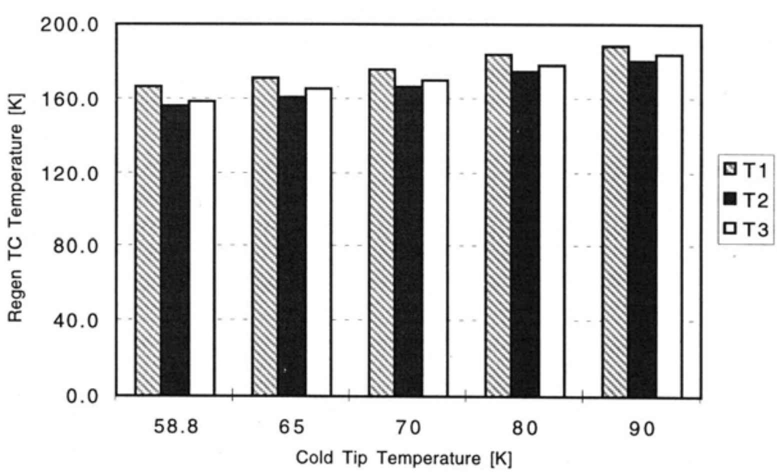

Figure 7. Regenerator temperatures versus cold tip temperature for final expander configuration. Note a further reduction in regenerator tube temperature deltas due to reduced pressure drops. See Fig. 5 for comparison.

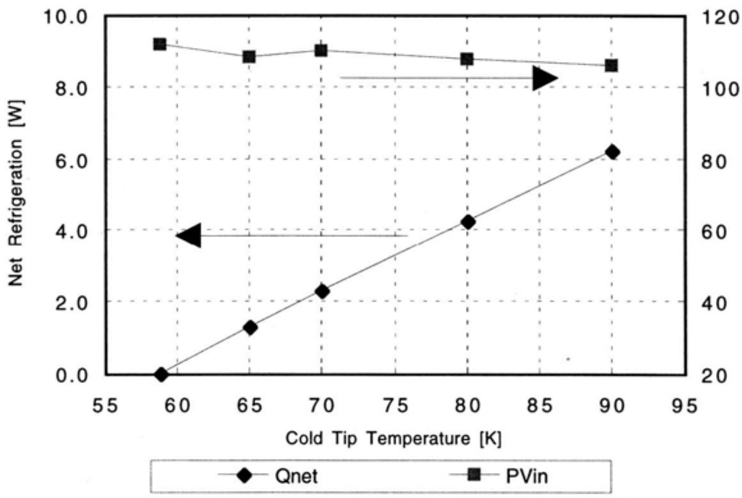

Figure 8. Load curves for improved expander design (thermal straps and reduced pressure drops). The significant improvement versus the baseline performance attributable primarily to thermal straps.

By virtue of simple thermodynamics, the observed increase in cooling capacity in going from the baseline to final configurations must be due to an increase in gross refrigeration produced, a reduction in parasitic losses (conduction, regenerator inefficiency, etc.), or a combination of the two. The relative performance of the initial and final expander configurations was evaluated to characterize the individual contribution of these two effects. The 70 K refrigeration data are provided in Table 1 as a typical example of the comparative measurements taken over the temperature range investigated (58 K to 90 K). Note that the gross refrigeration is essentially the same for both cases, with the strapped expander actually yielding slightly less gross capacity. Since the gross refrigeration value is a derived measurement, it is desirable to support the validity of the values reported with direct measurements. The pressure ratio measurement (Pmax/Pmin) before the surge valve provides that support. Since the pressure drops in the pulse tube and rejection heat exchanger are very small, the pressure ratio measured at the valve is representative of the pressure ratio in the cold expansion space. The data reveal that the amplitude of the pressure wave in the expansion space is the same for both configurations, which supports the assertion that the gross refrigeration is about the same. Therefore, the improved cooling capacity is evidently due almost entirely to a reduction in the parasitic losses. As a final note, the slight reduction in the amplitude of the compressor-side pressure ratio for the final configuration is consistent with the reported decrease in required input power discussed previously.

Table 1. Performance Improvement Evaluation

Configuration	Gross Capacity (W)	Net Capacity (W)	Total Parasitic Loss (W)	Pressure Ratio - Compressor	Pressure Ratio - Expander
Baseline - No Straps	9.92	1.47	8.45	1.28	1.15
Final - Straps; Reduced dP	9.82	2.30	7.52	1.27	1.15

The data obtained indicate that the 4-tube configuration is a viable part of the trade space available to Raytheon in considering pulse tube designs for particular systems, especially those applications requiring a robust expander design with a high allowable side load. On a more general note, the results reveal the difficulty in achieving an even, predictable flow distribution in parallel hydrodynamic systems, particularly those systems in which significant heating or cooling is taking place in localized regions, and how the flow distribution can be somewhat controlled by adding external thermal connections between the parallel circuits.

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