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(54) COMPACT LOW-POWER CRYO-COOLING SYSTEMS FOR SUPERCONDUCTING ELEMENTS

(71) Applicant: **The Regents of the University of Colorado, a body corporate**, Denver, CO (US)

(72) Inventors: **Vincent Kotsubo**, Superior, CO (US); **Joel Ullom**, Boulder, CO (US); **Sae Woo Nam**, Boulder, CO (US)

(73) Assignee: **National Institute of Standards and Technology (NIST)**, Gaithersburg, MD (US)

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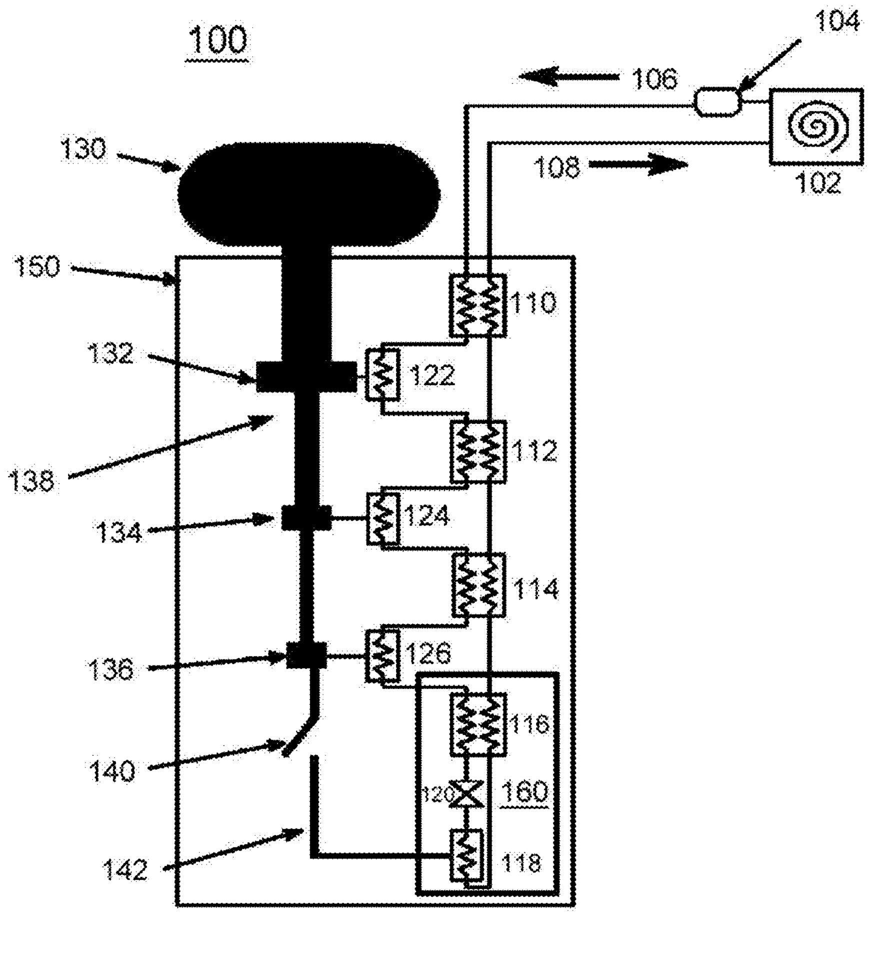
(57) **ABSTRACT**

A compact, low power cryo-cooler for cryogenic systems capable of cooling gas to at least as low as 2.5 K. The cryo-cooler has a room temperature compressor followed by filtration. Within the cryostat, four counterflow heat exchangers precool the incoming high-pressure gas using the outflowing low-pressure gas. The three warmest heat exchangers are successively heat sunk to three stages of a pulse tube to absorb residual heat from the slight ineffectiveness of the heat exchangers. The pulse tube cold head also absorbs loads from instrumentation leads and radiation loads. The pulse tube stages operate at around 80 K, 25 K, and 10 K. The entire system—cryo-cooler, drive and control electronics, and detector instrumentation, fits in a standard electronics rack mount enclosure, and requires around 300 W or less of power.



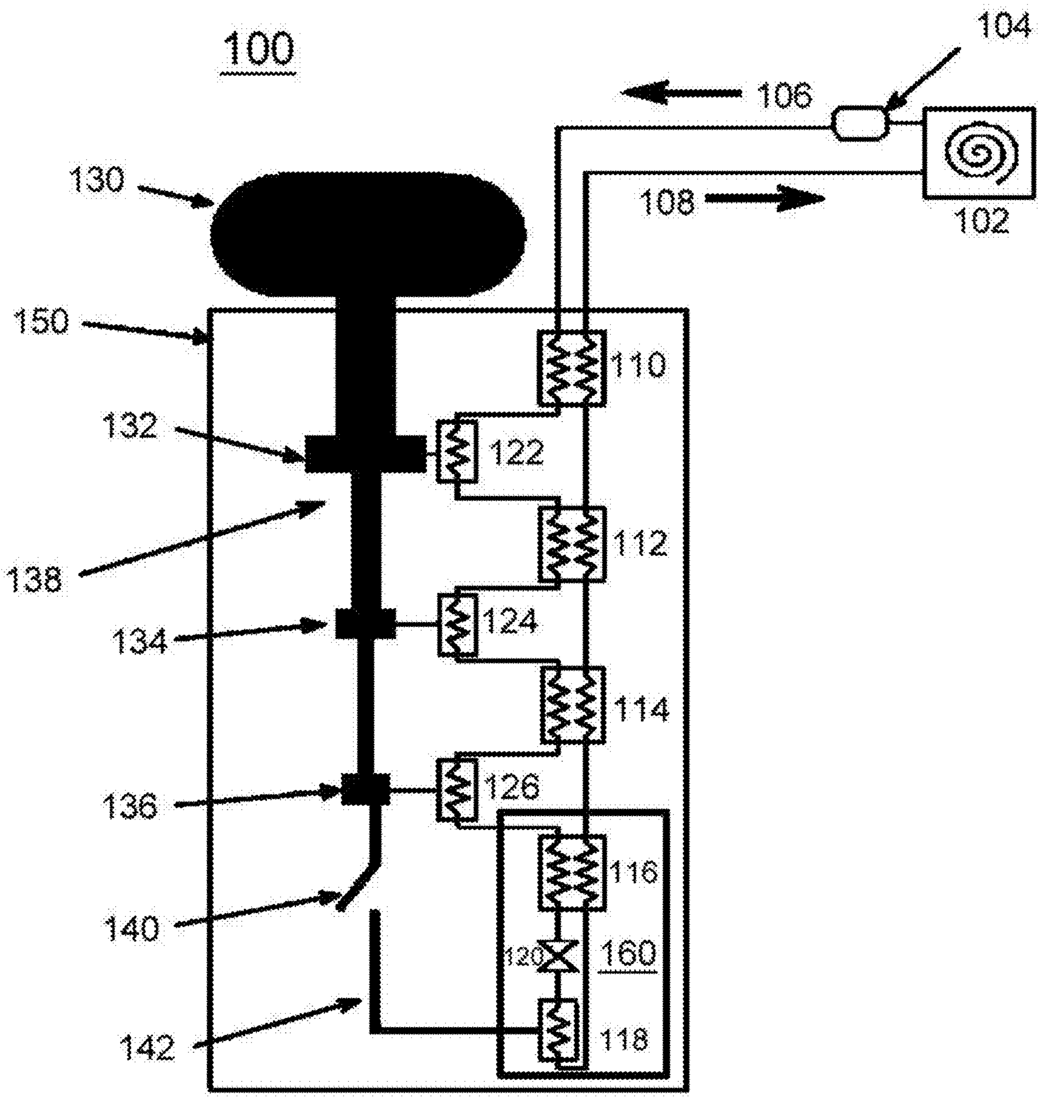

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Figure 1

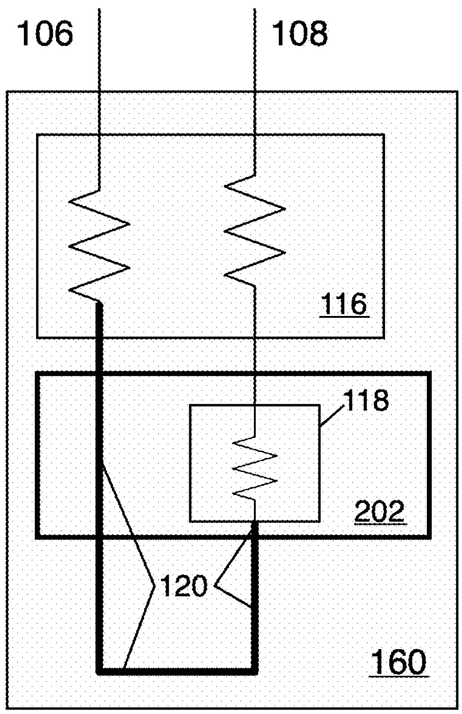

Figure 2A

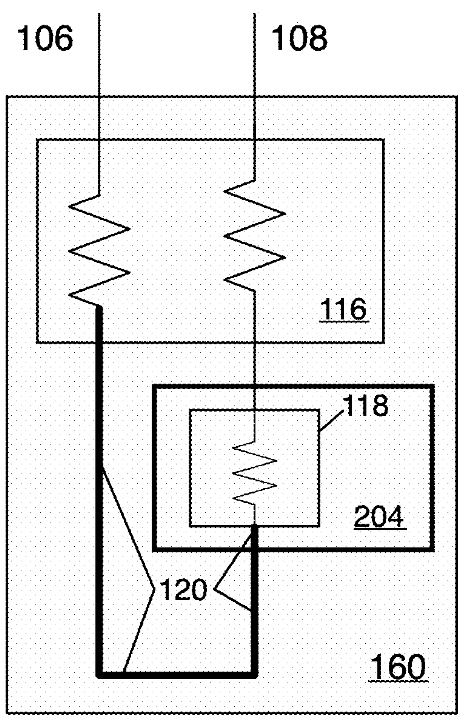

Figure 2B

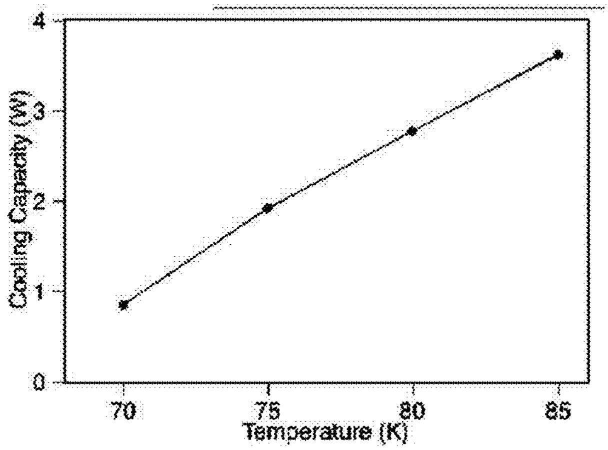

Figure 3

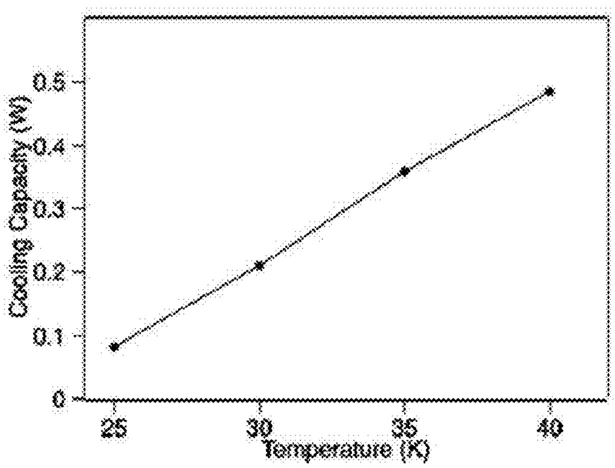

Figure 4

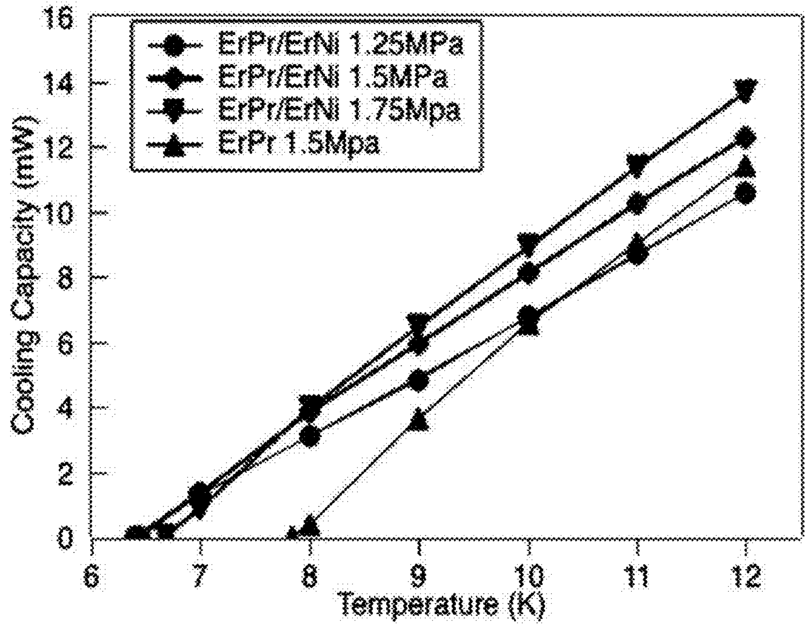

Figure 5

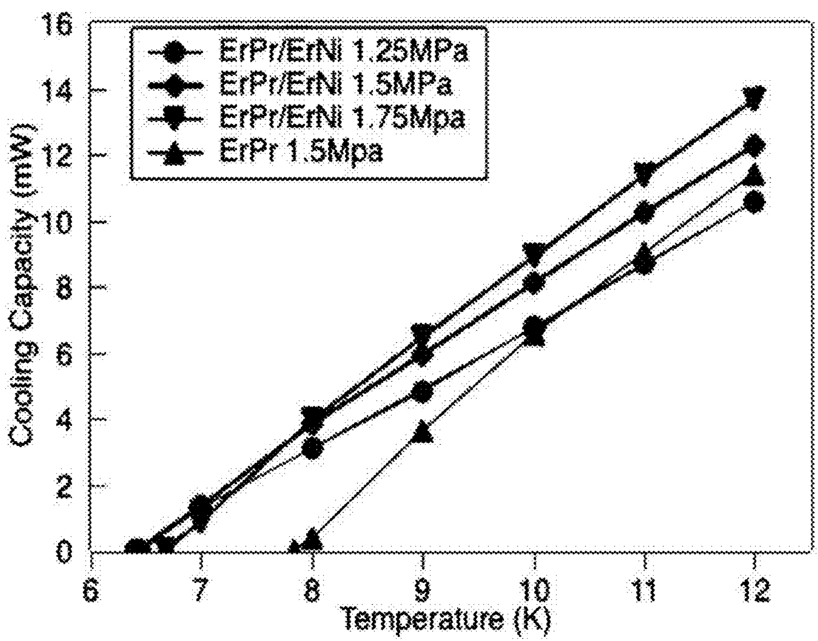

Figure 6

COMPACT LOW-POWER CRYO-COOLING SYSTEMS FOR SUPERCONDUCTING ELEMENTS

BACKGROUND OF THE INVENTION

[0001] This application incorporates by reference parent applications 62/621,483, filed Jan. 24, 2018, and 16/256,736, filed Jul. 24, 2019.

[0002] This invention was made with government support under grant number 70NANB14H095 awarded by NIST. The government has certain rights in the invention.

FIELD OF THE INVENTION

[0003] The present invention relates to compact, low-power cryo-cooling systems for superconducting elements. In particular, the present invention relates to such systems having a pulse tube compressor and a Joule-Thomson (JT) cooler.

[0004] DISCUSSION OF RELATED ART

[0005] Applications for superconductor-based sensors and electronics have been steadily increasing over the past several years in diverse areas such as astrophysics/cosmology, X-ray spectroscopy, gamma-ray spectroscopy, quantum information, and photon science. These systems operate at temperatures ranging from above 4 K to lower than 50 mK. Although these systems are developed by scientists with low-temperature expertise, end users typically have minimal cryogenic experience, and therefore acceptance has been greatly facilitated by push-button, closed-cycle cryo-coolers. The majority of these systems depend on precooling with commercial Gifford-McMahon or valved pulse tube coolers, which were originally developed for applications requiring relatively large cooling capacities such as shield cooling in magnetic resonance imagers or cryo-pumping. As a result, for many, if not most of these systems, the size and power consumption of the system are orders of magnitude larger than necessary. For example, many transition edge sensor (TES) systems are cooled using adiabatic demagnetization refrigerators (ADRs) with cooling capacities of hundreds of nanowatts at temperatures below 100 mK. At the 4 K reject temperature for the ADR, the heat rejected is on the order of a milliwatt, yet precoolers with capacities on the order of a watt with a power draw of several kilowatts are used.

[0006] While there are advantages for using excessively large coolers such as rapid cool-down times and flexibility in cryostat design, some applications of these types of systems would benefit from cooling systems having reduced size and power. Such applications include secure quantum encrypted communication links that utilize superconducting nanowire single-photon detectors (SNSPD), and superconducting transition-edge sensor microcalorimeters for electron microscope microanalysis.

[0007] The smallest capacity, commercially available closed cycle cooler is a Gifford-McMahon cycle with 100mW of cooling at 4 K, and drawing about 1.5 kW wall power. This unit is considerably larger and consumes substantially more power than necessary for such applications. Its minimum temperature is about 2.5 K, so that lower temperatures require a second stage of cooling such as helium-4 sorption refrigerator.

SUMMARY OF THE INVENTION

[0008] It is an object of the present invention to provide a compact, low power cooler for cryogenic systems. One embodiment consists of a pulse tube compressor and a He Joule-Thomson (JT) cooler operating at temperatures down to 2.2 K, precooled to 10 K using a 3-stage linear compressor pulse tube operating at 35 Hz. Although a He JT would reach temperatures approaching 1 K, He was selected for the initial development because of the relative high cost and rarity of He. This cooler embodiment has a room temperature compressor followed by particulate and condensable gas filtration. Within the cryostat, four counterflow heat exchangers precool the incoming high-pressure gas using the outflowing low-pressure gas. The three warmest heat exchangers are successively heat sunk to the three stages of the pulse tube to absorb residual heat from the slight ineffectiveness of the heat exchangers. The pulse tube cold head also absorbs loads from instrumentation leads and radiation loads. The pulse tube stages operate nominally at 80 K, 25 K, and 10 K. The entire system—cryo-cooler, drive and control electronics, and detector instrumentation, fits in a 7U (0.31 m) tall standard electronics rack mount enclosure approximately 0.61 m long, and does not require water cooling.

[0009] Another embodiment also has a form factor and low power draw of a standard rack-mountable electronics instrument. This embodiment increased the cooling capacity over the 2.2 K embodiment at 10 K by 35% and reduced the minimum temperature from 2.2 K to 1.7 K. For this embodiment, the cooler is a pulse tube/Joule-Thomson (PT/JT) hybrid, with the JT stage achieving 1.4 mW of cooling at 1.7 K, and the three-stage pulse tube providing cooling at 80 K, 25 K, and 10 K.

[0010] The cooler design was a compromise between thermodynamic efficiency, production costs, and risk. Preferred embodiments avoid complex, costly components and fabrication processes typically used in spaceflight coolers, resulting in thermodynamic efficiencies lower than those coolers. For example, standard size stainless steel tubing was used for regenerator and pulse tube walls instead of machined titanium, which is commonly used in aerospace coolers. Replacing the stainless-steel tubes with titanium tubes would result in 1.8 W, 0.07 W, and 0.001 W of additional cooling on the 80 K, 25 K, and 10 K stages, respectively.

[0011] The JT stage operates between a high pressure of 0.2 MPa and a low pressure of 1.3 kPa, with a flow rate of 0.6 mg/sec. The thermodynamic power required to recompress the gas is only a few watts, which is such a small fraction of the total power budget that a highly thermodynamically efficient design is not required.

[0012] Preferred embodiments utilize geometries that minimize the effects of secondary flows within the pulse tube and regenerator.

BRIEF DESCRIPTION OF THE DRAWINGS

[0013] FIG. **1** is a schematic block diagram of a preferred embodiment of the present cooler system, including a pulse tube compressor and a JT cooler.

[0014] FIGS. **2**A and **2**B are schematic block diagrams illustrating a more detailed view of the JT stage of the cooler of FIG. **1** for two variations of the embodiment of FIG. **1**.

[0015] FIG. **3** is a plot showing load curves for the final stage of the pulse tube compressor.

[0016] FIG. **4** is a plot showing cooling capacities of the next to final stage of the pulse tube compressor.

[0017] FIG. **5** is a plot showing load curves for the final stage of the pulse tube compressor for various configurations.

[0018] FIG. **6** is a plot showing load curves for the JT cooler at various high-side pressures.

DETAILED DESCRIPTION OF THE INVENTION

[0019] FIG. **1** is a schematic block diagram of the present cooler system **100**. System **100** is a compact cooler having a form factor and low power draw of a standard rack-mountable electronics instrument. The cooler is a pulse tube/Joule-Thomson (PT/JT) hybrid, with the JT stage **160** achieving 1.4 mW of cooling at 1.7 K or 2.2K (depending on the configuration, see FIGS. **2**A and **2**B), and the three-stage **132**, **134**, **136** pulse tube **138** providing cooling at 80 K (**132**), 25 K (**134**), and 10 K (**136**).

[0020] System **100** comprises a cryo-cooler compressor **102** providing high-pressure gas out **106** and low-pressure gas in **108** and includes a filter **104**. Pulse tube cooler **130** includes an 80K stage **132**, a 25K stage **134**, and a 10K stage **136** housed within vacuum shell **150**. Also within vacuum shell **150** are thermal busbar **142**, heat exchangers **122**, **124**, **126**, and **118**, as well as counterflow heat exchangers **110**, **112**, **114**, and **116**. Counterflow heat exchanger **116** and heat exchanger **118** are part of JT stage **160**, as is JT expansion capillary **120** (see FIGS. **2**A and **2**B). The counterflow heat exchangers could be tube-in-tube heat exchangers. Since one important application of the present invention is the cooling of quantum encrypted communication links that utilize superconducting nanowire single-photon detectors (SNSPD), and superconducting transition-edge sensor microcalorimeters for electron microscope microanalysis, the stages of this embodiment were developed with estimated cooling loads as shown in Table 1. Those skilled in the art will appreciate that specific applications will result in different cooling load requirements and thus variations in the cooler design within the spirit of the present invention.

TABLE 1

Stage	Radiation (mW)	JT Precooling (mW)	Conduction (mW)	Total (mW)	Design Target (mW)
80K	1000-2350	400	150	1550-2900	3000
25K	5-12	28	7	40-47	100
10K	0.2	2	1	3.2	5
2.0K	0	N/A	0.28	0.28	0.5

[0021] Estimated cooling loads on each of the cooler stages are presented in Table 1 along with the design goals. The cooling load on the JT stage 160 arises primarily from low thermal conductance bias/readout coaxial cables, since SNSPD's dissipate negligible power and photons are coupled in through optical fiber. Targeted cooling capacities included design margin at each stage. The 80 K stage **132** radiation load estimate has considerable uncertainty due to unknown ambient temperature since the cryo-cooler will generally be enclosed in an equipment rack-mounted box, and because of the uncertain surface emissivity of the radiation shield.

[0022] The WSi SNSPD detectors currently require cooling to 1.25 K for maximum quantum efficiency, but this temperature is difficult to achieve with a 4He JT refrigerator that utilizes a low power, compact compressor, and generally temperatures up to 2.5 K are acceptable. This 2.5 K minimum temperature is easily reachable by various embodiments of the present invention. Thus, this embodiment operates at a slightly higher minimum temperature of around 1.7 K/2.2 K. As an alternative, 3He does reach 1.25 K at the same pressure that 4He reaches 1.7 K, and can be used when the lower temperature is especially important.

[0023] Similarly, the 10 K for the precooling temperature for the JT stage **160** is a lower temperature than the thermodynamic optimum for this hybrid, but was chosen to reduce the high-pressure requirement for the JT loop, which in turn reduces stress on the JT compressor. This was done because the JT compressor is the largest risk to reliability. Final test results shown below show that the actual operating temperature of this stage was below 7.7 K. Again, for other applications, the operating conditions will be adjusted slightly.

[0024] The cooler is shown schematically in FIG. **1**. The pulse tube **138** is a three-stage (**132**, **134**, **136**), cooled inertance tube configuration **138**, resulting in an easy-to-integrate coldhead. Because the present invention is intended to provide a low cost, easily produced cooler, the mechanical design sacrificed performance to reduce costs. The most impactful tradeoff was the use of standard sized stainless-steel tubing for the regenerator and pulse tube walls, rather than machined titanium, which is commonly used in aerospace coolers. Replacing the stainless-steel tubes with titanium tubes would result in 1.8 W, 0.07 W, and 0.001 W of additional cooling on the 80 K, 25 K, and 10 K stages, respectively.

[0025] The upper two-stage inertance tubes were dual diameters, whereas the third stage used only a single diameter because modeling indicated minimal improvement in performance with dual diameters. Regenerators used standard materials; the first stage used die-punched stainless steel screens with #150 mesh, 66 μm wire diameter in the warm side and #400 mesh, 25.4 μm wire diameter in the cold side. The second stage used #400 mesh, 25.4 μm wire diameter stainless steel screens in the warm side and #400 mesh, 25.4 μm wire diameter phosphor bronzes screens, flattened to produce 0.55 porosity, in the cold side.

[0026] The third stage used high heat capacity microspheres. In the previous embodiments, 100 um diameter 50-50 erbium-praseodymium (ErPr) alloy spheres were used. The present embodiment replaced the lower half of the regenerator with 100 μm diameter erbium-nickel (ErNi) spheres to improve cooling capacity below 10 K, and comparative tests results are presented below.

[0027] A critical mechanical design objective was geometries that minimize the effects of secondary flows within the pulse tube and regenerator, because these flows can seriously degrade performance. Most well known are turbulence/convection in the pulse tube and streaming in regenerators. Somewhat less appreciated are secondary flows in the displacer gap in Stirling coolers. A net circulation can flow through the displacer gap and return through the regenerator partially unregenerated and cause cooling loss unless there is sufficient lateral thermal conduction. Alternatively, because the displacer gap is dynamic, the local gap dimensions can vary during the operating cycle, causing local oscillating flow to not exactly reverse motion, again leading to unregenerated flow and cooling loss.

[0028] While it is well documented that, in large diameter regenerators, net circulation can lead to significant degradation in coolers, we were concerned that even in small diameter regenerators, some level of circulation can exist, causing measurable cooling loss. In multi-stage coolers, circulation can be generated in the regenerators near the junctions between stages, where the phasing of the flows between the upper stage regenerator, lower stage regenerator, and buffer tube will necessarily lead to circulation in the regenerators if the upper and lower stage regenerators are in-line with each other because flows entering the regenerator can come from either the adjacent regenerator or flow channel from the buffer tube. To mitigate this, some embodiments locate the second stage regenerator such that the warm entrance is in the connecting channel between the first stage regenerator and first stage pulse tube rather than directly in-line with the first stage regenerator.

[0029] Thus, the flow entering the cold end of the first stage regenerator always comes from the channel. This arrangement was not workable in this embodiment for the junction between the second stage and third stage regenerators because the second and third stage regenerator packing access was through an attachment flange for the third stage regenerator which was directly in line with the second stage regenerator. However, the small diameters of the second and third stages likely mitigated any effects of circulation.

[0030] For the JT system **160**, the very low cooling capacity requirement allowed a design driven by ease of fabrication rather than by thermodynamics. The stage nominally operates between a high pressure of 200 kPa and a low pressure of 3.2 kPa, with a flow rate of 0.6 mg/s. The thermodynamic power required to recompress the gas is only a few watts, which is such a small fraction of the total power budget that a highly thermodynamically efficient design was not required.

[0031] All four counterflow heat exchangers **110**, **112**, **114**, **116** are simple tube-in-tube designs, with high-pressure gas flowing in the narrow annular space between the two tubes and the low-pressure gas flowing through the inner tube This configuration simplifies fabrication, plus has a large internal surface area on the high-pressure return side to allow condensable contaminants to freeze out without plugging the flow passage.

[0032] The expansion impedance **120** was a 2 m long, 50 μm inner-diameter stainless steel capillary.

[0033] At the cold ends of the warmer three counterflow heat exchangers **110**, **112**, **114**, a small heat exchanger consisting of fine mesh copper screen diffusion bonded to a copper body was used to heat sink the incoming gas to the pulse tube. The 1.7 K coldstage **116** also had a small copper screen mesh heat exchanger for thermal contact to the load (see FIG. **2**B).

[0034] To mitigate particulate contamination, sintered stainless steel particulate filters **104** were inserted in the high-pressure line prior to entering the cryostat, at the end of the heat exchanger **132** at 80 K (not shown) and just upstream of the JT expansion impedance (not shown). In addition, a small capsule of activated charcoal (not shown) was placed in the high-pressure stream on the 80 K stage to adsorb any condensable contaminants.

[0035] To expedite cooldown of the JT coldstage from room temperature, a heat switch 140 consisting of a small bar that clamped down on a small copper tab attached to the JT coldhead was used. The clamp was heat-sunk to the pulse tube 10 K stage **136** and was actuated by pulling on a fine stainless steel wire attached to the bar (not shown). The far end of the wire was fed through a bellows feedthrough through the room temperature vacuum flange to allow actuation of the heat switch.

[0036] In the 2.2 K embodiment of FIG. **2**A, both the inlet and outlet of the JT expansion capillary **120** were housed in the same copper block **202** and therefore at the same temperature. In the 1.7 K embodiment of FIG. **2**B, the warm and cold ends of the capillary were separated, and only the cold end was housed in copper block **204**. The result was a lower minimum base temperature of 1.7 K in comparison to the 2.2 K base temperature of the FIG. **2**A embodiment.

[0037] The JT compressor is a scroll compressor, selected because of the ability to achieve low suction pressures, compact size, and low per-unit cost. Because of the very low flow rates and low thermodynamic compression power, the JT loop did not require a high degree of optimization, so the counterflow heat exchangers were designed for ease of fabrication. They were modeled using a simple NTU analysis using temperature-averaged fluid properties, along with a separate calculation of thermal conduction down the heat exchanger tubes. There was considerable leeway in the design, allowing heat exchanger lengths to vary by a factor of two without significantly affecting performance. The main consideration was minimizing the demand on the JT compressor, so the system was designed for the lowest reasonable high-side pressure to minimize stresses on bearings and back-leakage through seals. Final flow and pressure requirements for the compressor were determined from test data.

[0038] Test results of the pulse tube:

[0039] FIGS. **3** and **4** show the cooling capacities of the 80 K stage and 25 K stage as a function of temperature. Each load curve was taken while holding the other two stages at constant temperature. These plots illustrate that the 80 K stage handles loads on the order of watts, while the 25 K stage handles loads on the order of tenths of watts.

[0040] FIG. **5** is a plot showing load curves for the 10 K stage and compares the performance of the all ErPr regenerator and the regenerator with ErPr in the warm half and ErNi in the cold half, with data for ErPr/ErNi regenerator combination shown at three different charge pressures. In all cases the ErPr/ ErNi regenerator outperforms the all-ErPr regenerator, except at temperature above 10 K with 1.25 MPa charge pressure for the ErPr/ErNi regenerator. The ErPr/ErNi combination also consistently reached a minimum temperature in the 6.3 K to 6.5 K range while the minimum temperature of the all ErPr regenerator reached a minimum temperature slightly below 8 K. These lower temperatures were expected because of the lower heat capacity of ErPr compared to ErNi at lower temperatures.

[0041] These data were taken with no applied load to the upper two stages such that the upper stages ran colder than the data in Table 6, which resulted in higher 10 K cooling capacity.

[0042] Testing on the integrated JT-pulse tube coldhead was conducted open loop using a regulated compressed gas storage bottle as a supply and a vacuum pump vented to atmosphere on the return. The low side pressure was controlled by adjusting a valve in front of the vacuum pump.

[0043] FIG. **6** shows load curves at high side pressures of 0.2 MPa, 0.15 MPa, 0.125 MPa, and 0.1 MPa. A maximum of 1.4 mW was produced at the minimum temperature of about 1.7 K. The cooling capacity increase at lower temperatures is from increased mass flow through the capillary from higher gas density. Operating at 0.1 MPa does not provide sufficient cooling to cool down the JT stage from the initial temperature of around 7 K. If the coldstage was temperature-regulated slightly above the liquification temperature, no liquid accumulates, which stabilizes the cold stage temperature. the load on the pulse tube from circulating 4He would cause the third stage temperature to rise slightly. However, in all situations the temperature remained below 7.7 K, indicating that Nb-Ti wire can be used for electrical leads between the 10 K stage and the JT stage.

[0044] Test data described above show that the design point cooling powers for the pulse tube require a nominal 150 W of compressor power. We have tested a variety of commercial off-the-shelf power. Given conversion efficiencies of off-the-shelf power electronic modules for driving the compressor are generally better than 90%, the pulse tube will require a total power draw of about 165 W. The power draw by the compressor and drive electronics will be about 50 W. If the power for cooling fans and thermometry/diagnostic instrumentation is included, power requirements come in at under 300 W, usually about 250 W. In an embodiment using a smaller, commercially available, pulse tube compressor to reduce system weight, another 30 W-50 W of compressor power is required.

[0045] While the exemplary preferred embodiments of the present invention are described herein with particularity, those skilled in the art will appreciate various changes, additions, and applications other than those specifically mentioned, which are within the spirit of this invention. For example, for low heat load applications, the flow impedance can be increased, reducing the ass flow, which would reduce the power required for both the JT compressor and the pulse tube compressor. Conversely, there is substantial upside in the cooling power capability of the JT stage. The JT impedance can be reduced which would result in higher mass flow rates and higher cooling powers. The pulse tube has sufficient capacity to accommodate higher JT mass flow rates, so capacities approaching 5 mW are possible, although this will require development of a larger capacity JT compressor.

[0046] Lower temperatures using 4He are possible but require a combination of reworking the JT counterflow heat exchangers to reduce pressure drop, a JT compressor with lower suction pressure, and superfluid film creep mitigation. Direct substitution of 3He results in temperatures below 1 K.

What is claimed is:

1. A compact, low-power cryo-cooler system for cryogenic systems comprising:

a cryo-cooler compressor for providing a flow of hot high-pressure gas and receiving a return flow of cool low-pressure gas;

a series of counterflow heat exchangers configured to cool hotter incoming gas from the compressor with cooler gas returning to the compressor;

a pulse tube cooler including a series of closed-cycle pulse tube cooling stages configured to interact with the counterflow heat exchangers to pre-cool the high-pressure gas to at least as low as 10 K; and

a Joule-Thomson (JT) cooler configured to cool the pre-cooled gas to at least as low as 1.7 K, the JT cooler including an expansion capillary having a warm end and a cold end, wherein the warm and cold ends of the expansion capillary are physically separated enough to achieve sufficient thermal isolation between the warm and cold ends to achieve the 1.7K;

wherein the cryo-cooler and associated drive and control electronics fit within a 7U electronics rack and the cooler and associated drive and control electronics require less than about 300 W of power.

2. The cryo-cooler of claim **1** wherein the cryo-cooler requires less than 250 W of power.

3. The cryo-cooler of claim **1** wherein the input pressure to the JT cooler is between 0.1 MPa and 0.2 MPa and the output pressure from the JT cooler is between 0.2 MPa and 1.3 kPa and wherein the output pressure is lower than the input pressure.

4. The cryo-cooler of claim **3** wherein the flow from the JT cooler is on the order of half a mg/sec.

5. The cryo-cooler of claim **1** having three pulse tube cooling stages, the three stages pre-cooling the gas to about 50-100K, 20-30K, and 6-10K in turn.

6. The cryo-cooler of claim **5** wherein the three cooling stages are implemented with a 3-stage linear compressor pulse tube operating at 35 Hz.

7. The cryo-cooler of claim **1** wherein the gas is 4He.

8. The cryo-cooler of claim **1** wherein the gas is 3He and the JT cooler is configured to cool the gas to around 1.25 K.

9. The cryo-cooler of claim **1** wherein the pulse tube stages have regenerators and pulse tube walls comprising stainless-steel tubing.

10. The cryo-cooler of claim **1** wherein the coldest pulse tube cooling stage utilizes erbium-nickel spheres.

11. The cryo-cooler of claim **1** wherein the counterflow heat exchangers are tube-in-tube.

12. The cryo-cooler of claim **1** wherein a JT expansion element within the JT cooler is comprises a 1 m-3 m long, 40 μm-60 μm inner-diameter stainless steel capillary

13. The cryo-cooler of claim **1** wherein the cold end of the expansion capillary is housed in copper block and the warm end of the expansion capillary is spaced apart from the copper block.

* * * * *