

Demonstration of a Two-Stage Turbo-Brayton Cryocooler for Space Applications

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ABSTRACT

Turbo-Brayton cryocoolers have inherent technical advantages for many space applications because of their extremely low emitted vibration and simplicity of integration with payloads and spacecraft heat rejection systems. The technology was demonstrated in space in March 2002 when the NICMOS Cryocooler was installed on the Hubble Space Telescope. The NICMOS Cryocooler is a single-stage unit that supplies 7 W of refrigeration at 70 K and has exceeded six years of space operations without degradation or malfunction. More recently, the need for efficient cooling at multiple temperatures has driven the development of a two-stage turbo-Brayton cryocooler. An initial brassboard two-stage cryocooler was built and tested. This unit relies heavily on NICMOS technologies for the mechanical hardware, but uses new control electronics that have been designed for precise and independent control of two load temperatures. The cryocooler was optimized to simultaneously provide at least 1.8 W of refrigeration at 65 K and 12.6 W of refrigeration at 100 K, and the mechanical hardware weighs approximately 21 kg. The unit underwent extensive thermal vacuum testing to demonstrate thermodynamic performance and flight-like operations. This paper describes the design features and performance of the cryocooler.

INTRODUCTION

This decade has seen the maturation and increased acceptance of mechanical cryocoolers for space applications, in particular, the cooling of infrared detectors. Refrigeration temperatures typically range from about 10 K for silicon-doped focal plane arrays to around 120 K for longwave optics. The refrigeration to cool these objects may be provided by either active mechanical cryocoolers or passive techniques such as cryogenic radiators or stored cryogenes. Mechanical cryocoolers provide numerous advantages over competing passive approaches. Cryocoolers have significantly lower mass than stored cryogenes of comparable capacity. Cryocoolers also provide much longer life than stored cryogen systems, which must rely upon a finite source of either stored cryogen or high pressure gas that must be expanded through an orifice to create refrigeration. Cryocoolers are also typically lower in mass than cryogenic radiators, and do not require complex spacecraft maneuvers to assure one side of the spacecraft maintains deep-space pointing. Despite these advantages, the acceptance of mechanical cryocoolers for space applications has been slow, owing to concerns with both emitted vibrations that would impact pointing accuracy and the inherent limitations in reliability of mechanical devices.

Creare Inc. has developed and demonstrated a unique class of mechanical cryocoolers, turbomachine-based reverse-Brayton cryocoolers, which have inherent advantages over competing active cryocoolers when it comes to addressing these concerns. Turbo-Brayton cryocoolers utilize miniature turbomachines to compress and expand the working gas as part of a reverse-Brayton thermodynamic cycle. The only moving parts in these cryocoolers are miniature turbomachine rotors which have masses of 1 to 10 grams. The rotors have diameters of a few millimeters and operate at speeds of several thousand revolutions per second. These small sizes and high speeds are required for high thermodynamic efficiency at the low power levels associated with space cryocooling applications. The rotors are supported on non-contacting gas bearings, and the turbomachines use clearance seals so there is no mechanical contact or wear mechanism during normal operations. Contact between the rotor and bearings occurs briefly during start-up and shutdown operations, but the extremely low contact forces prevent any appreciable wear during thousands of start-up and shutdown cycles. Numerous demonstrations support this claim.¹ In addition, the rotors are precisely balanced for high-speed operations, and the cycle gas flow is continuous. The result is that emitted vibrations are extremely small and occur at high frequencies where the predicted mechanical displacement caused by vibrations is on the order of nanometers.

The initial space demonstration of a turbo-Brayton cryocooler occurred during NASA's NICMOS Cooling System (NCS) Program.² The NCS consists of a single-stage turbo-Brayton cryocooler coupled with a cryogenic circulator and capillary pump loop. The system provides 7 W of cooling at 70 K to the NICMOS instrument on the Hubble Space Telescope and replaced the solid nitrogen cryogen that had depleted prematurely after less than two years in space. The NCS was deployed and integrated with NICMOS by astronauts during STS-109 (Space Shuttle Columbia) in March 2002 and has operated nearly continuously for over six years without performance degradation. On-orbit operations have shown that emitted vibrations and temperature stability are within the stringent requirements of the telescope. In addition, the on-orbit integration epitomizes another advantage of the technology—the continuous flow nature of the cryocooler greatly simplified integration with the payload; in this case, permitting on-orbit integration with the payload.

More recent space applications have required the development of cryocoolers that can provide cooling at multiple temperatures. For longer wavelength applications, say >10 microns, the optics must typically be cooled in addition to the focal plane assembly (FPA) to reduce background radiative flux on the FPA and thus achieve the required sensitivity in the bandwidth of interest. A technologically straightforward approach is to use separate cryocoolers operating at distinct temperatures: one to cool the FPA; and a second, operating at a warmer but still cryogenic temperature, to cool the optics. Another approach is to use a single multistage cryocooler. This approach has mass and performance benefits, though the cryocooler controls must be adapted to simultaneously accommodate two temperature set points. The goal of providing simultaneous FPA and optics cooling with a single cryocooler has driven the development of a variety of multistage cryocoolers intended for space applications.³

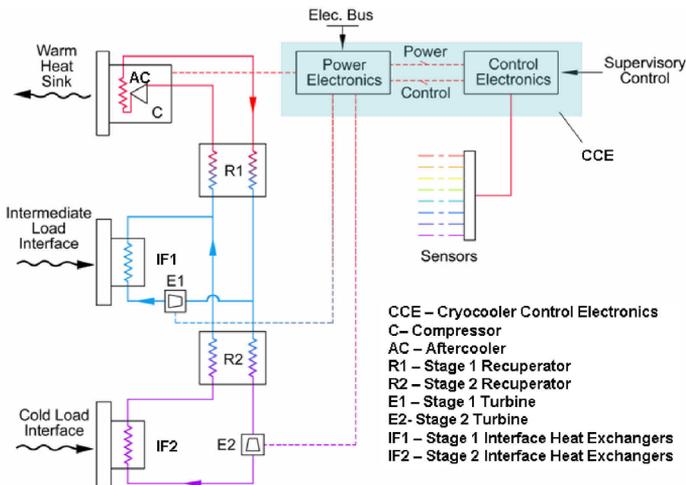
Several multistage Stirling-class cryocoolers, which includes Stirling, pulse tube, and combination Stirling/pulse tube systems, have been developed and demonstrated for space. These cryocoolers are generally compact and efficient due to the high cycle pressure ratio and the inherent efficiency of the thermodynamic cycle at low to moderate temperature ratios. The moving components within the linear compressors that drive these systems, and also the displacer in the case of the Stirling cryocoolers, create vibrations that are exported to the supporting payload structure. Vibration cancellation techniques have been demonstrated to reduce exported vibration to below 100 mN⁴, but this value is still unacceptable for some missions.

Recently, Creare Inc., Raytheon Space and Airborne Systems, and Jackson & Tull Chartered Engineers have collaborated on the development of a two-stage turbo-Brayton cryocooler to support a technology demonstration program. All mechanical components have heritage in the NICMOS cryocooler. The electronics are based on a combination of NCS heritage brought by Jackson & Tull, Creare's patented rotating field inverter (RFI) drive circuits, and Raytheon's modular flight electronics packaging techniques and turboalternator power recovery circuits. The cryocooler was designed to provide 12.6 W of cooling at the first stage temperature of 100 K and 1.8 W at the

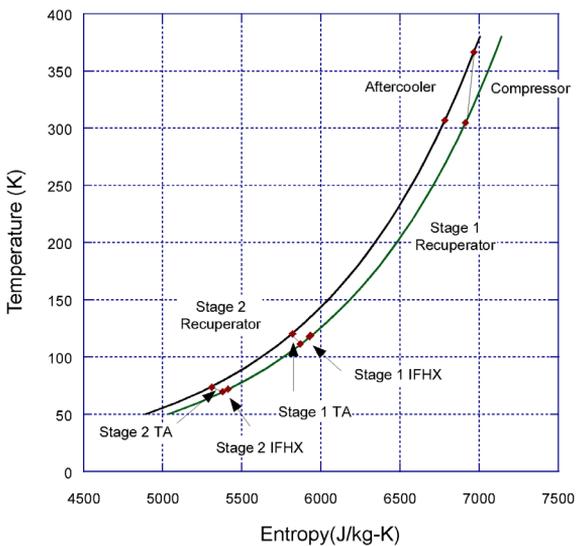
second stage temperature of 65 K with an AC compressor input power of 400 W. Other key requirements that have driven the design are negligible vibration, significant separation distances between thermal interfaces, and a lifetime of greater than eight years. The features of the cryocooler and electronics, and the results of system testing are described in this paper. The design and development of the electronics is described in more detail in a separate paper at this conference.⁵

TWO-STAGE TURBO-BRAYTON CRYOCOOLER

A cycle schematic and corresponding temperature-entropy diagram for a two-stage turbo-Brayton cryocooler is shown in Figure 1. The cycle gas chosen for this application is neon. A single compressor compresses the neon, causing it to flow continuously through the system. The heat of compression and any losses within the compressor are rejected at the warm heat sink, which also rejects any losses associated with the electronics. The high-pressure gas flows through the Stage 1 recuperator and then splits into two streams. The first stream expands through the Stage 1



(a)



(b)

Figure 1. Two-stage reverse-Brayton cycle: (a) cycle schematic, (b) Temperature-entropy diagram.

turbine and then flows through a Stage 1 load interface heat exchanger. The second stream flows through the Stage 2 recuperator and then is expanded through the second stage expansion turbine. This stream continues through the Stage 2 load interface heat exchanger, through the Stage 2 recuperator, and then rejoins the first stream. The recombined streams then flow through the Stage 1 recuperator and back to the compressor.

The brassboard configuration of the cryocooler is shown in Figure 2. The compressor uses an induction motor to convert high-frequency, three-phase electrical power supplied by the electronics into shaft power. The compressor has heritage in the NICMOS unit with modification to address input power levels of 400 W, as compared to 350 W for NICMOS. The compressor and aftercooler are mounted on a liquid-cooled heat sink remote from the cold cryocooler components to minimize the thermal parasitic heat loads on the cryocooler and improve the efficiency of waste heat rejection. The first stage uses a slotted-plate recuperator similar to the NICMOS design, approximately 0.56 m (22 inches) in length and divided into two modules to meet packaging constraints. The temperature of the working gas is reduced from approximately 300 K to 200 K in the first module, and from 200 K to 100 K in the second module. The turbines are similar to the NICMOS design and have permanent-magnet alternators (i.e., turboalternators), which convert shaft power to electric power that is fed to the electronics. The first-stage turbine and first-stage load interface heat exchanger are remotely mounted away from the first-stage recuperator. An additional turbine and recuperator are used to reduce the working gas temperature from the first-stage temperature of 100 K to the second-stage temperature of 65 K. These components are mated into a single module with a total length of 0.23 m (9 inches). The cooled working gas is routed a short distance from the turbine to the 65 K interface heat exchanger.

The component masses are summarized in Table 1. The overall mass of mechanical cryocooler hardware is 21 kg (47 lbm). For testing, the cryocooler used a commercial-off-the-shelf compressor inlet filter and aftercooler, which is not reflected in Table 1. The configuration shown in Figure 1 was selected to allow testing in a small thermal vacuum chamber. The tubing lengths and number of bends between components were selected to represent a worst-case for a flight configuration. The cryocooler was tested with rack-mounted electronics. The flight version of these electronics was designed for full redundancy and has an estimated mass of 9 kg (20 lbm) per cryocooler.

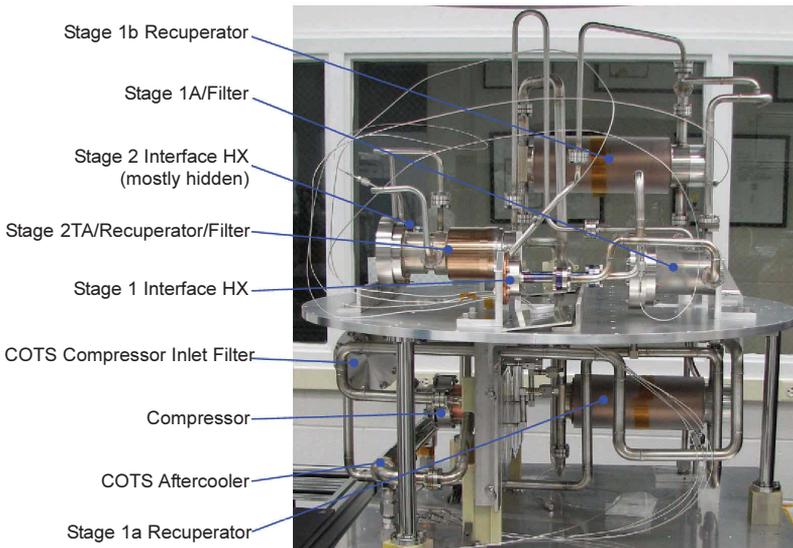


Figure 2. Brassboard two-stage reverse-Brayton cryocooler.

Table 1. Mechanical hardware mass.

Component	Mass (kg)	Mass (lbm)
Compressor/Aftercooler/Filter*	7.4	16.3
Stage 1 Recuperator (2 modules)	7.3	16.1
Stage 1 (100 K) Turboalternator	2.0	4.4
Stage 2 Recuperator/Turboalternator	2.7	6.0
Stage 1 (100 K) IFHX	0.3	0.7
Stage 2 (65 K) IFHX	0.1	0.2
Plumbing	1.4	3.1
Mechanical Cryocooler Total	21	47

*Commercial aftercooler and filter were used during system testing.

Performance

The scale-up from a single-stage cryocooler to multiple stages was straightforward from a thermodynamic viewpoint due to the continuous flow nature of the cycle. The performance of individual components was previously characterized and represented by empirical curves, which were included in a thermodynamic model of the two-stage reverse-Brayton cycle as shown in Figure 1. The key to obtaining the required load split between the two stages was specified by the geometric features of the turbine nozzles and the thermal performance of the recuperators.

The flow area of the turbine nozzles sets the flow impedance of each turboalternator, and therefore the mass flow rate delivered to each stage. The efficiency of each stage is governed by both the efficiency of the turboalternators and the thermal performance of the recuperators. The thermal performance of the recuperators is a design parameter and can be tailored by the number of heat exchanger plates in each recuperator. The thermal performance and size of the recuperators were selected through trade studies that also considered packaging issues and mass.

The maximum refrigeration delivered at each stage is predicted in Figure 3 for AC compressor input power levels of 200 W to 400 W. The load temperatures are 65 K and 100 K for all points. The cryocooler was optimized for 400 W of AC input power to the compressor and the design loads of 1.8 W at 65 K and 12.6 W at 100 K plus conservative margins. Input power levels less than 400 W are obtained by reducing the compressor rotational speed. The compressor input power

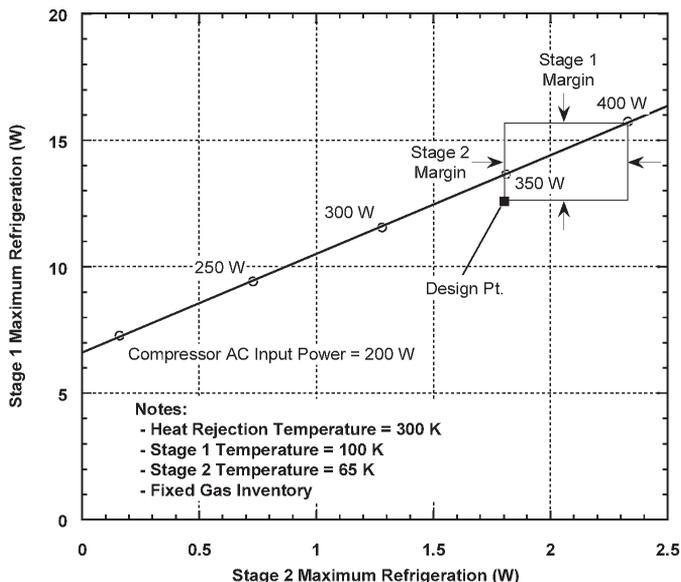


Figure 3. Maximum refrigeration at each stage for different input power levels.

varies as the cube of the rotational speed, so a reduction of input power from 400 W to 200 W nominally requires a 21% reduction in compressor speed.

The data shown in Figure 3 are for the maximum refrigeration values for each stage. This condition requires the turboalternators to operate at the rotational speeds corresponding to peak efficiency. The cryocooler can operate at all refrigeration values less than the maximum value by simply changing the turboalternator speed to a less efficient operating point.

The cryocooler control electronics adjust the turbomachine speeds to provide temperature control and to minimize input power. The controls are briefly discussed in the next section. For a more detailed description of the electronics, see McCormick, et al., 2008.⁵

Controls

The NICMOS cryocooler is a single-stage cooler. Temperature control at the load is provided by varying the compressor speed, which changes both the compressor input power and refrigeration in a manner analogous to that described above for a two-stage cryocooler. Independent temperature control for two loads requires the use of either trim heaters or independent control of the turboalternator speeds. For the development program described here, the team implemented turboalternator speed control which reduces overall input power relative to the use of trim heaters. This approach also provides a straightforward means to recover the turboalternator output power, further reducing power draw from the DC bus.

The load temperature set-points provide two control objectives, and the turbomachine speeds provide three independent control variables. The excess control variable allowed implementation of an algorithm to minimize compressor input power. The operation of the temperature control system is illustrated by the test data shown in Figure 4. The turboalternator and compressor speeds are continually adjusted to minimize input power and maintain the loads at specific temperatures. The performance over 120 minutes with nominally steady loads is shown in Figure 5. The span of the ordinate corresponds to the objectives of ± 1.0 K at Stage 1 and ± 0.25 K at Stage 2. The demonstrated temperature control was ± 0.1 K at Stage 1 and ± 0.05 K at Stage 2, well within the objectives and consistent with the resolution of the flight-like telemetry used. (The data shown in Figure 5 are from the test support telemetry, which had extremely fine resolution.)

System Testing

The cryocooler was extensively tested to (1) determine its performance at the design point and at off-design conditions, (2) correlate thermodynamic models, and (3) establish the correspondence between component performance data attained at the component-level and system-level. During

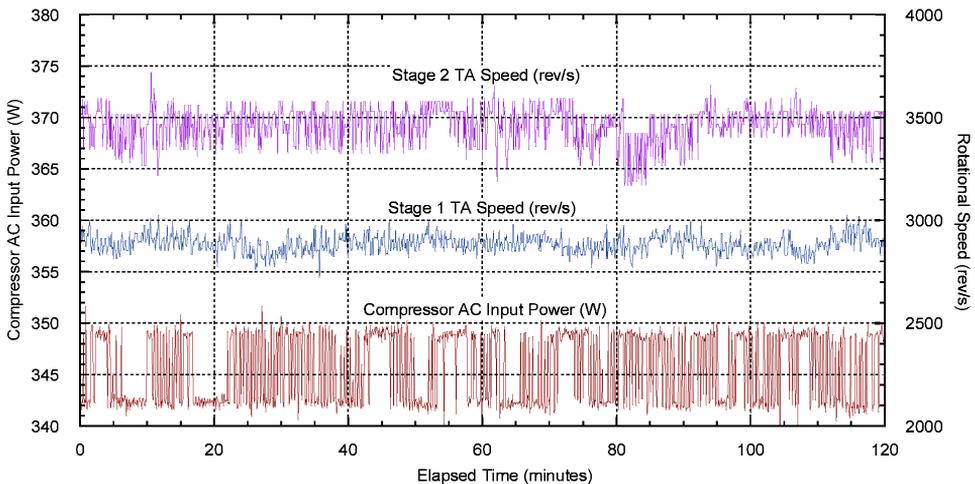


Figure 4. Operation of cryocooler temperature controls.

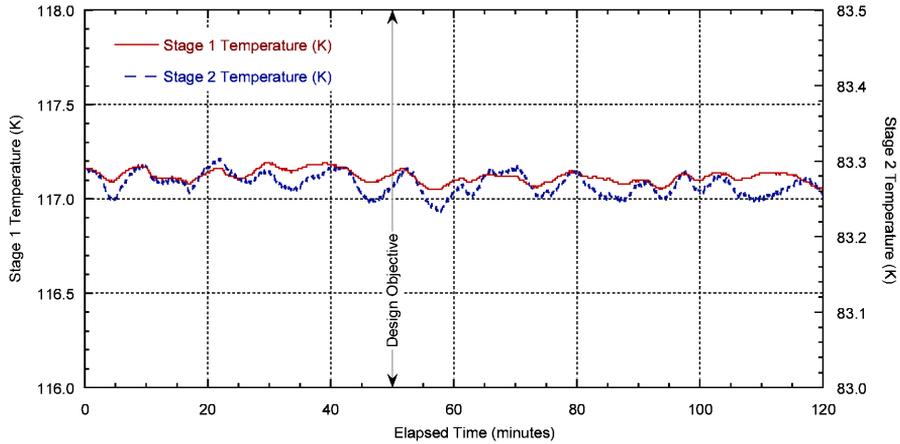


Figure 5. Performance of cryocooler temperature controls.

testing, the refrigeration loads at each stage were provided by electrical heater, and the heat rejection temperature was set by a liquid cooling loop. The cold-components of the cryocooler were suspended by low conductivity wires, and the radiative environment was thermally controlled using a Gifford-McMahon cryocooler to minimize external parasitics. In addition, multilayer insulation was applied to all external cryocooler surfaces. The approach to minimize parasitics allowed the thermodynamic performance of the Brayton cryocooler to be accurately quantified independent of influences from the test configuration. The external parasitics were determined during testing to be less than 100 mW at each stage.

The cryocooler was instrumented with flight-like sensors for safety and controls as well as extensive ground-test sensors for thorough performance characterization. The measurements included turbomachine power and speed, temperatures, pressures at key locations, and load heater power. Redundant pairs of platinum resistance temperature detectors (RTDs) were placed at 27 locations for performance characterization, and redundant pairs of thermistors and negative temperature coefficient RTDs were placed at six locations for safety and controls. A list of the key measurements and their corresponding uncertainty is given in Table 2.

The thermal vacuum testing involved several campaigns with incremental upgrades of the hardware, electronics, and controls between campaigns. The thermodynamic data from the final test is listed in Table 3. Data were obtained at seven thermodynamically stable test points. The input power was nominally 400 W for all test points. Data were obtained at two different heat rejection temperatures to determine the cryocooler sensitivity to that parameter, and six different combinations of Stage 1/Stage 2 load temperatures. The Stage 1 load temperature was varied between 100 K and 118 K, and the Stage 2 load temperature was varied between 65 K and 72 K. The

Table 2. Key measurements.

Performance Parameter	Location	Nominal Value	Measurement Uncertainty
Compressor AC Power	Standalone 3-phase power meter	400 W	± 6.4 W
Stage 1 Refrigeration Load	Calculated	12.6 W	± 0.024 W
Stage 1 Heater Voltage	4-wire measurement at heater	44.3 V	± 0.0026 V
Stage 1 Heater Current	Shunt resistor in test rack	0.284 A	± 0.00054 A
Stage 2 Refrigeration Load	Calculated	1.80 W	± 0.0085 W
Stage 2 Heater Voltage	4-wire measurement at heater	22.8 V	± 0.0016 V
Stage 2 Heater Current	Shunt resistor in test rack	0.0791 A	± 0.00037 A
Stage 1 Load Temperature	PRT on cryocooler side of thermal interface	100 K	± 0.014 K
Stage 2 Load Temperature	PRT on cryocooler side of thermal interface	65 K	± 0.014 K
Aftercooler Outlet Temperature	PRT on aftercooler outlet tube	300 K	± 0.025 K

Table 3. Steady-state test points.

	Test Pt. 1	Test Pt. 2	Test Pt. 3	Test Pt. 4	Test Pt. 5	Test Pt. 6	Test Pt. 7
Compressor Inlet Pressure (atm)	1.950	1.964	1.975	2.042	2.071	2.061	2.094
Aftercooler Outlet Temperature (K)	302.7	302.7	302.7	302.8	302.8	292.6	302.8
Stage 1 Load Temperature (K)	100.6	100.3	100.2	110.0	118.3	118.2	118.4
Stage 2 Load Temperature (K)	65.07	68.86	72.13	72.07	64.93	72.10	72.00
Compressor AC Input Power (W)	395	396	396	401	400	396	399
Net Stage 1 Refrigeration (W)	15.7	15.7	15.7	18.6	20.5	21.3	20.9
Net Stage 2 Refrigeration (W)	1.97	2.52	2.90	2.48	1.10	2.18	2.02
Carnot Performance w/o Electronics (-)	0.098	0.102	0.104	0.101	0.090	0.096	0.098

refrigeration loads are based on the electrical input power to the load heaters, and the values reported have not been adjusted to account for external parasitics, which were relatively small (order of 100 mW at each stage). The heat rejection temperature and load temperatures were measured on the cryocooler side of the interfaces. The test results are plotted in Figure 6.

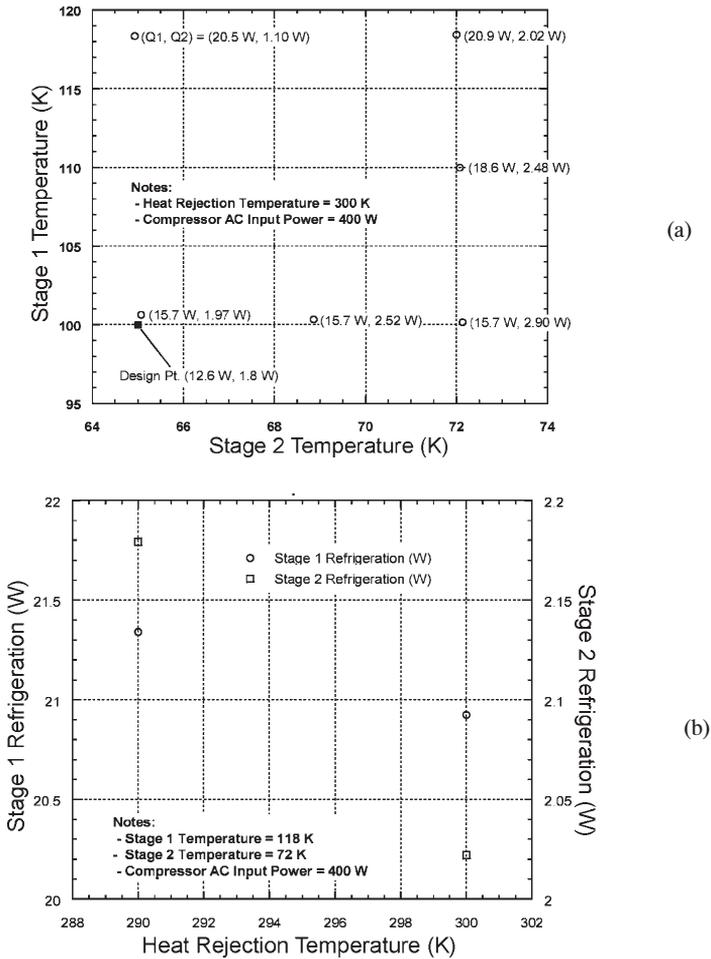


Figure 6. Performance of two-stage turbo-Brayton cryocooler, (a) Load map, (b) Variation of heat rejection temperature.

The cryocooler performance at the design point load temperatures of 65 K and 100 K was 15.7 W and 2.0 W at the first and second stage, respectively—for a compressor input power of less than 400 W, which met the requirements of 12.6 W and 1.8 W. The efficiency was nominally 10% of the Carnot cycle excluding the electronics for all test conditions. The maximum refrigeration delivered was 21.3 W at 118 K for Stage 1 and 2.9 W at 72 K for Stage 2, though not simultaneously. The test data were used to correlate a thermodynamic model of a two-stage turbo-Brayton cryocooler.

CONCLUSIONS

Recently completed work has advanced the flight maturity of a two-stage turbo-Brayton cryocooler to a technology readiness level of nominally five. The operation of the mechanical hardware was identical whether tested as components or within a system, which may lead to simplified testing campaigns during future programs. A thermodynamic model was developed and correlated using component and cryocooler test data. The model will allow more accurate cryocooler performance predictions for future applications. Power and control electronics were developed and demonstrated that allowed precise and independent control of two load temperatures while minimizing input power. The adaptation of turbo-Brayton technology from the single-stage NICMOS cryocooler to a two-stage cryocooler was successfully demonstrated.

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