IOP Conference Series:
Materials Science and
Engineering

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To cite this article: K. Cragin *et al* 2020 *IOP Conf. Ser.: Mater. Sci. Eng.* **755** 012032

Development of a Cryogenic Compressor for Airborne Cryocoolers

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Abstract. Superconducting electronics and spectral-spatial holographic systems are being developed for advanced digital communications. These devices must operate at cryogenic temperatures of near 4 K. Liquid helium is undesirable for mobile missions due to logistics and scarcity, and commercial low temperature cryocoolers are unable to meet size, weight, power, and environmental requirements for many missions. Creare is developing a turbo-Brayton cryocooler that provides refrigeration at 4.2 K and rejects heat at 77 K to an upper-stage cryocooler or through boil-off of liquid nitrogen. The cooling system is predicted to reduce size, weight, and input power by at least an order of magnitude as compared to the current state-of-the-art 4.2 K cryocooler. For systems utilizing nitrogen boil-off, the boil-off rate is reasonable. This paper reviews the development of the cryo-compressor, a key cryocooler component. The cryo-compressor has heritage in the cryogenic circulator used in the space-borne NICMOS cryocooler. To produce the pressure ratios and mass flow rates required by the cryocooler, the cryo-compressor must operate at much higher operating speeds than the cryogenic circulator while still at cryogenic temperatures. This operating condition presents a challenge for stable operation of gas bearings at low viscosities. The approach to overcome this challenge and the testing of the compressor at cryogenic temperatures are the focus of this paper.

1. Introduction

Superconducting electronics and spectral-spatial holography have the potential to revolutionize digital communications. Digital receivers have been demonstrated utilizing these technologies with extremely high bandwidth, but must operate at cryogenic temperatures. The most challenging applications in terms of size, weight, and power (SWaP) are those targeting small airborne platforms. Passive refrigeration using liquid helium can produce the required thermal environment, but is not practical in the field. Commercial mechanical refrigerators are too large and inefficient and hence are unable to meet SWaP requirements. Space cryocoolers are smaller and lighter, but inefficient at low temperatures and expensive. A new type of mobile refrigeration system is required that has high thermodynamic efficiency, is able to approximate the SWaP requirements for small platforms, and is optimized for operation with cryogenic electronics.

The technical approach we are developing is a two-stage turbo-Brayton cryocooler that provides refrigeration at 4.2 K and 25 K, and rejects heat at 77 K through boil-off of liquid nitrogen. The cryocooler design and projected performance were reported previously [1]. The cryocooler is predicted to reduce each SWaP parameter by an order of magnitude as compared to the current state-of-the-art 4.2 K cryocooler. In addition, liquid nitrogen is relatively safe, non-toxic, easy to handle, and unlike liquid helium, readily available and transportable. The nitrogen boil-off rate does not limit the operational duration between refilling. This paper reviews the design and test results for one of the key cryocooler components, the cryogenic compressor.

2. Compressors for Turbo-Brayton Cryocoolers

The compressor is one of the major components of a turbo-Brayton cryocooler. The refrigeration cycle for this system uses a single-phase gas as the working fluid. One or more compressors provide the pressure ratio and circulate the cycle gas, and inter/aftercoolers remove the heat of compression from the cycle gas and compressor dissipation losses. The compressor(s) typically operate and reject heat around ambient temperature. The key development in the current work is the ability of the compressor to operate at cryogenic temperatures.

Operating the compressor at cryogenic temperature instead of room temperature provides substantial benefits. First, because the density of the helium cycle gas is four times higher at 77 K than at room temperature, it enables the compressor to achieve the target pressure ratio in a single compression stage. To produce the equivalent pressure ratio at room temperature would require three compressor stages in series with total input power of 250 to 300 W. In addition to decreasing the input power by a factor of five or six, the overall size and mass of the compression stage(s) is substantially reduced. Second, the high pressure ratio achievable at cryogenic temperature significantly increases the specific cooling power of helium flow, which reduces the required gas flow rate in the system and thereby the size and mass of the recuperators. Most importantly, the input power to compress the helium is reduced due to the lower enthalpy rise that is required to compress a gas at lower temperatures. These features substantially reduce the size, mass, and input power of the cryocooler, making it ideal for SWaP-sensitive applications.

3. Compressor Design

The compressor design is based on a prior compressor design that operates at 300 K [2]. Key changes were made to optimize the aerodynamic features of the impeller and diffuser for optimal performance at 80 K, and material selection and design changes were made to enable operation at cryogenic temperatures. A cross-sectional view of the compressor design is shown in Figure 1. The journal bearing diameter is nominally 4 mm (0.2 in.) and the impeller diameter is nominally 13 mm (0.5 in.). Operational speed is 8,000 rev/s. The overall compressor assembly is 9.4 cm (3.7 in.) in diameter and 10 cm (3.9 in.) long and weighs 2.3 kg.

Advanced fabrication techniques were developed to construct the impeller and shaft from a single piece of material, eliminating joints that can cause difficulties at high rotational speeds or over large temperature changes. The impeller is made from a titanium alloy to provide the strength required to accommodate the large centrifugal stresses at high operating speeds. A permanent magnet is installed within the shaft to form the electro-magnetic rotor. Self-acting gas bearings provide radial and axial support for the rotor. These bearings are aligned in a bearing cartridge that also locates the motor stator. The clearances between internal components are minimized to enable heat dissipated in the motor windings and by the bearings to efficiently conduct to the outer housing. Heat is conducted through the bearing housing and transferred to the cycle gas via an integrated heat exchanger. In the cryocooler, this heat is removed from the gas in an external aftercooler attached to the 77 K heat rejection interface. The impeller and diffuser blade designs were developed and refined using CFD to optimize the aerodynamic efficiency of the compressor at the cryogenic operating point. Close-clearance seals are employed forward and aft of the impeller to minimize bypass leakage from the impeller exit to the inlet pressure. Figure 2 shows a photograph of the completed compressor rotor assembly. The shaft and impeller surfaces are coated with a hard coating for low friction and to reduce wear during start-up and coast-down when contact occurs with the bearings. The assembled compressor is shown in Figure 3.

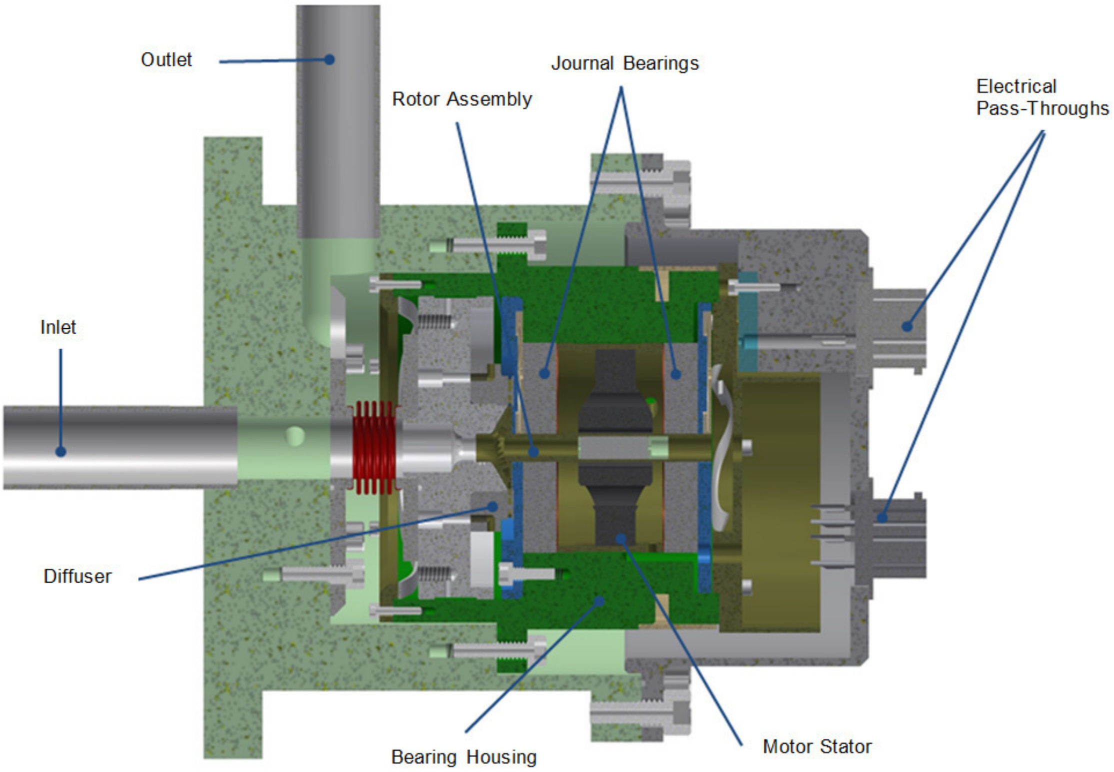

Figure 1. Solid model of cryogenic compressor design.

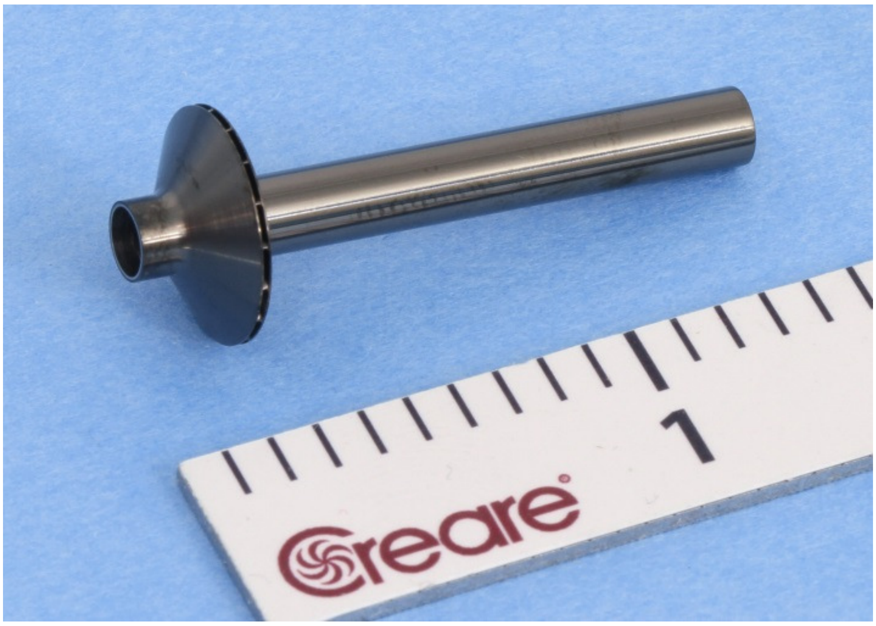

Figure 2. Completed compressor rotor assembly.

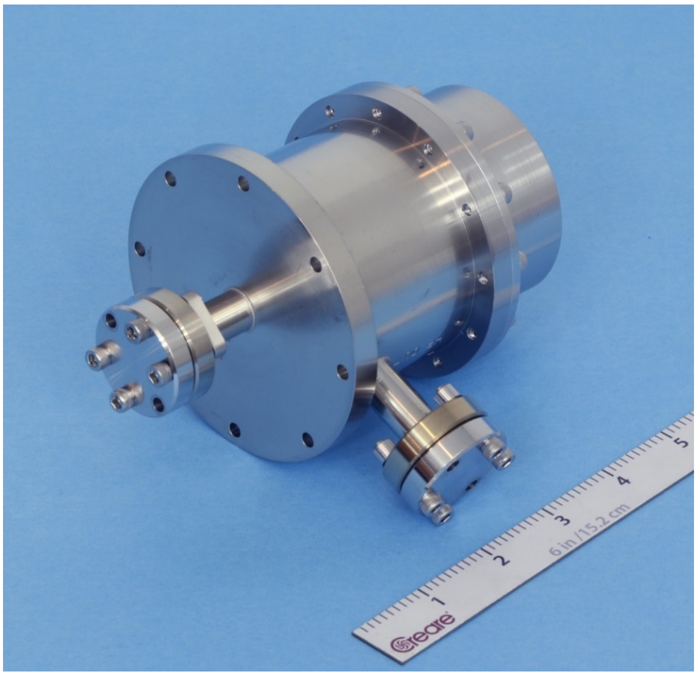

Figure 3. Compressor assembly prior to installation in test facility.

The key design and operational challenges for the design of a compressor operating at cryogenic temperatures are (1) maintaining precise alignment between parts across a very large temperature range as the machine cools to the operating temperature, and (2) maintaining adequate support for the rotor to operate at very high rotational speeds while at cryogenic temperatures. Alignment between components is maintained by the use of a precision-machined bearing housing to align the bearings, motor, thrust bearings, and diffuser; and these parts are all made from the same raw material to eliminate CTE mismatches. Low temperature operation of self-acting gas bearings is difficult because the low gas viscosity substantially reduces the restoring forces produced by the bearing. In order to maintain bearing stability and enable the very high rotational speeds required for the compressor design while at cryogenic temperatures, the bearing design employs sophisticated adjustment features through which the clearance to the rotor shaft can be altered. Creare has developed an advanced bearing adjustment protocol capable of accurately resolving clearances at the shaft/bore interface. This technique enables very precise bearing adjustments to be made in response to observed rotor behavior. Initially, the bearings are adjusted with a generous clearance to the shaft. A spin test is conducted, and rotor stability and runout in the bearings is recorded. If necessary, the bearing clearances are incrementally reduced until full-speed operation is attained. These adjustments enable very close control over the bearing clearances and thus the properties of the supporting fluid film. By tailoring the clearances to the precise requirements of each application and each individual machine, high rotational speeds can be achieved at low temperatures with low drag.

4. Compressor Performance

Compressor performance testing was performed in an existing open-loop helium test facility. We performed initial checkout tests and spin tests at ambient temperature to assess journal bearing stability up to the design operational speed. Several tests resulted in a maximum speed that was less than the design speed due to bearing instability. After each test, we reduced the clearance in the journal bearings to enable higher operating speeds. After a few iterations to the journal bearing settings, the rotor attained the design operating speed of 8,000 rev/s.

We assessed the compressor performance at ambient temperature to verify the test instrumentation and to screen for a gross performance shortfall created by an assembly error. Finding acceptable performance, we proceeded to cryogenic performance testing.

The test is conducted using liquid nitrogen to attain the test temperatures. High-purity helium gas passes through an adsorber to remove water vapor and other impurities that could freeze out inside the compressor and impair performance. The flow then passes through a coil at the bottom of a dewar that is filled with liquid nitrogen. This coil cools the gas flowing to the compressor to nominally 80 K. The compressor is suspended in the dewar above the coil and is cooled by the nitrogen boil-off. As the system cools down, the liquid level is raised to submerge the compressor and attain an isothermal operating environment.

The compressor performance is characterized by a nondimensionalized head-flow curve and its net efficiency. These parameters are defined as:

Head Coefficient: $$\psi = \frac{\Delta h_s}{U^2} \tag{1}$$

Flow Coefficient: $$\phi = \frac{Q}{AU} = \frac{\dot{m}}{\rho\left(\frac{\pi}{4}D_{eye}^2\right)U} \tag{2}$$

Net Efficiency: $$\eta_{net} = \frac{\dot{m}\Delta h_s}{W_{AC}} \tag{3}$$

where Δh_s is the isentropic enthalpy rise, U is the impeller tip speed, Q is the volumetric flow rate, A is the flow area defined by the inlet diameter, $\dot{m}$ is the mass flow rate, ρ is the inlet density, D_{eye} is the impeller inlet eye diameter, and W_{AC} is the input power to the compressor.

Figure 4 shows the compressor head-flow curve measured at 80 K along with the predicted performance from CFD analysis. In this initial test, the compressor was operated at 4,000 rev/s instead of the 8,000 rev/s design speed. Similar to the adjustments performed to attain the design speed at ambient temperature, the compressor bearings require further adjustment to attain the design operating speed at 80 K. However, the nondimensionalized head-flow performance is independent of operating speed, and data taken at lower operating speeds are representative of the compressor performance. The compressor performance looks excellent and exceeds our performance predictions. The shapes of the measured and predicted performance curves match quite closely, but the head rise achieved in the test exceeds predictions.

The measured compressor net efficiency during the test was 56 to 59%, slightly lower than the design target of 61%. Unlike the head-flow performance, the efficiency is strongly affected by operating speed and power level, and was reduced at least in part to operating lower than the full design speed. Further testing at the design speed is needed to quantify any true differences from predictions. Nevertheless, this net efficiency is quite good and more than adequate to meet overall cryocooler performance targets.

5. Conclusions and Future Work

The design and fabrication of a lightweight, high-efficiency cryogenic compressor for a 4.2 K turbo-Brayton cryocooler is complete, and initial testing has commenced. This paper discussed the design and initial test results for the compressor. The ability of the compressor to operate at 80 K instead of 300 K has enabled an immense reduction in size, weight, and input power for the cryocooler under development compared to the current state-of-the-art 4 K cryocooler. The measured compressor performance exceeds performance predictions and is a promising step toward meeting the performance targets for the cryocooler. Future work involves performance characterization of the compressor at the full operating speed to verify the net efficiency, integration with the other components to assemble a complete cryocooler, and testing the cryocooler.

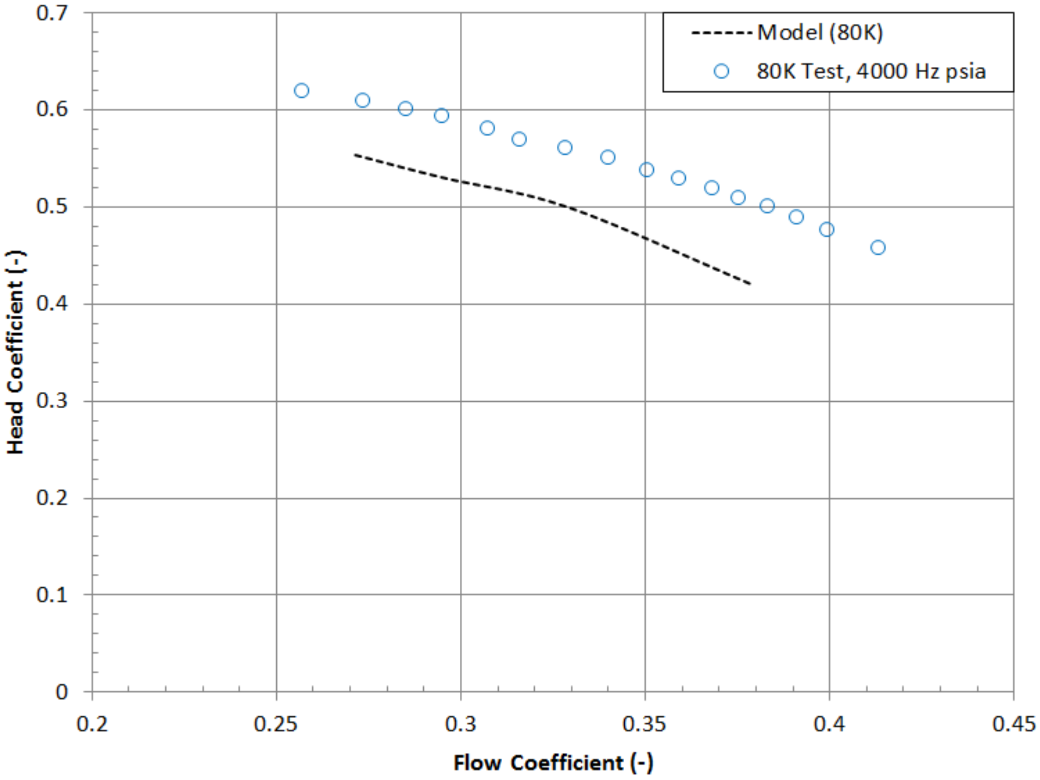

Figure 4. Preliminary head-flow curve from cryo-compressor testing.

6. References

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7. Acknowledgment

The support of the Office of Naval Research for this work (Contract N00014-14-C-0210) is gratefully acknowledged.