Performance Testing of a High Capacity Compressor for a 20K 20W Cryocooler

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ABSTRACT

Creare is currently developing for NASA a high capacity 20 K turbo-Brayton cryocooler for long-term cryogen storage in space. The cryocooler utilizes three compressors with intercooling to compress and circulate the helium cycle gas. The compressors operate on self-acting journal bearings, are driven by brushless permanent magnet motors at speeds exceeding 6,000 rev/s, and have a nominal input power capacity of 500 W. As a result of the high speed and flow rates, the predicted aerodynamic efficiency is extremely high for all stages, resulting in net efficiencies of greater than 60% for the three compressor stages. This is significantly higher than prior permanent magnet motor compressors developed by Creare which have shown peak efficiency of around 50%. The high compressor efficiency results in a cryocooler predicted input power below 1.6 kW, corresponding to a specific power of 80 W/W. This performance is significantly better than any 20 K cryocooler existing or (to our knowledge) under development. In preparation for cryocooler integration, the compressors were tested at 300 K using helium at design flow rates. This paper describes the design, fabrication, and initial test results for these high-capacity compressors.

INTRODUCTION

To support long-duration space exploration missions, NASA must store cryogenic propellants in space for long periods of time. However, heat radiated from the sun and the Earth causes the liquid cryogen tanks to pressurize. Currently, to control the pressure inside a cryogenic storage tank, the boil-off is vented, releasing otherwise useful cryogen into space. While cryogen boil-off is acceptable for short-duration missions, long-term storage of cryogen in space requires a system to avoid loss of the propellant.[1] One solution to this problem is to employ a cryocooler to intercept the heat load on the tank and maintain the cryogen at a constant temperature. Due to the large size of the cryogen tanks, the cooling capacity is large for space cryocoolers, exceeding the current state-of-the-art by an order of magnitude.

Creare is currently developing for NASA a high capacity 20 K turbo-Brayton cryocooler (Fig. 1) that will be used in a demonstration of a zero-boil-off hydrogen system. At high cooling loads, the turbo-Brayton cycle has a clear advantage over other cycles in terms of performance, size, and mass.[2] In addition, the continuous-flow nature of the Brayton cycle is ideal for cryogen storage missions where the cycle gas can be directly interfaced with a Broad Area Cooling (BAC) system attached to the storage tank. On a recent program, Creare delivered a single-stage

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turbo-Brayton cryocooler to NASA/GRC to support a ground demonstration of a zero-boil-off liquid oxygen storage system and a reduced-boil-off liquid hydrogen storage tank.[3] This cryocooler provides up to 20 W of refrigeration at 90 K. The cryocooler currently under development (and subject of this paper) provides cooling at 20 K to enable zero-boil-off liquid hydrogen storage. The cryocooler utilizes three compressors with intercooling to compress and circulate the helium cycle gas. This paper reviews our progress on developing a next-generation compressor for turbo-Brayton cryocoolers. Key developments include a high-capacity 500 W motor and advanced aerodynamics for high efficiency.

COMPRESSORS FOR TURBO-BRAYTON CRYOCOOLERS

The compressor is one of the major components of a turbo-Brayton cryocooler in addition to the turboalternator, recuperative heat exchanger(s), and electronics. The refrigeration cycle uses single-phase helium gas as the working fluid, which was selected for optimal thermodynamic performance at the 20 K load temperature. The compressors provide the pressure ratio and circulate the cycle gas in turbo-Brayton cryocoolers and inter/aftercoolers remove the heat of compression from the cycle gas and compressor dissipation losses from each compression stage. The 20 K, 20 W cryocooler utilizes three compression stages with intercooling. The serial arrangement allows centrifugal compressors to obtain the desired cycle pressure ratio without exceeding material stress limits, and intercooling reduces input power.

The overall cryocooler efficiency and weight is strongly affected by the compressor performance. The compressor performance is characterized by the net efficiency, which is the ideal compressor work divided by the AC input power to the motor (Equation 1). This performance parameter accounts for losses in the motor and bearings, and aerodynamic losses in the impeller and diffuser. Therefore, the efficiency can also be represented as the product of the motor efficiency, shaft efficiency, and the aerodynamic efficiency and is given by Equation (1):

$$\eta_{net} = \frac{\dot{m}\Delta h_s}{W_{AC}} = \eta_{Motor}\eta_{Shaft}\eta_{Aero} \tag{1}$$

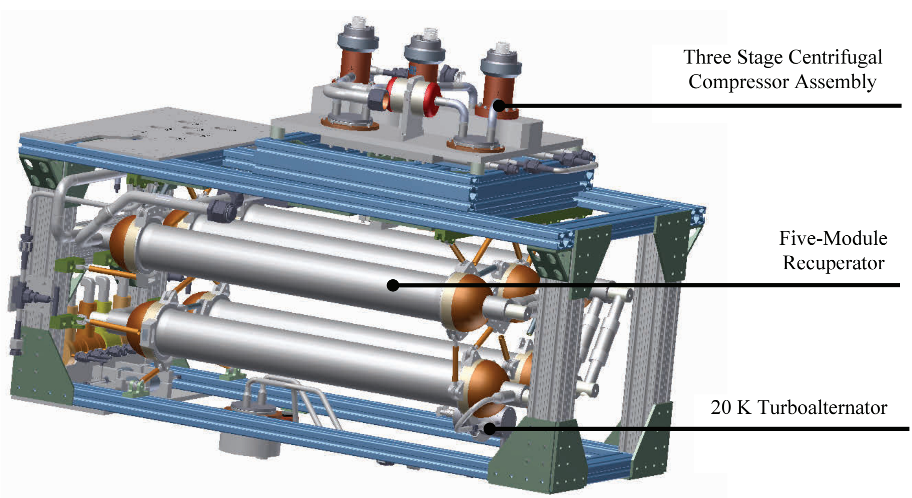

Figure 1. Creare's 20 K, 20 W cryocooler for liquid cryogen storage.

COMPRESSOR DESIGN

The compressor design is a modified version of Creare's 400-500 W class permanent magnet compressor shown in Fig. 2.[4] Each compressor is identical except for the aerodynamic design of the impeller and diffuser. The design is based on a prior compressor design with key upgrades to optimize the aerodynamic features of the impellers and diffusers for optimal performance at each compression stage and a new permanent magnet motor design that extends motor capacity from 400 W to 500 W. The journal bearing diameter is 6.4 mm (0.25 in.) and the impeller diameter is 19 mm (0.75 in.). Operational speed is 6,300 rev/s. The overall compressor assembly is 9.53 cm (3.75 in.) in diameter and 16 cm (6.32 in.) long and weighs 4 kg.

The compressor uses a multi-part rotor. A permanent magnet is installed within the shaft to form the electro-magnetic rotor. A precision machined impeller is mechanically fastened to the shaft. Self-acting gas bearings provide radial and axial positioning of the rotor. These bearings are supported 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 heat generated by the bearings to efficiently conduct to the outer housing. Heat is conducted through the outer housing to the base plate where it is removed by the spacecraft thermal management system.

We developed impeller and diffuser blade designs to optimize the aerodynamic efficiency of each compression stage. The performance predictions at the system operating point for each stage are shown in Table 1.

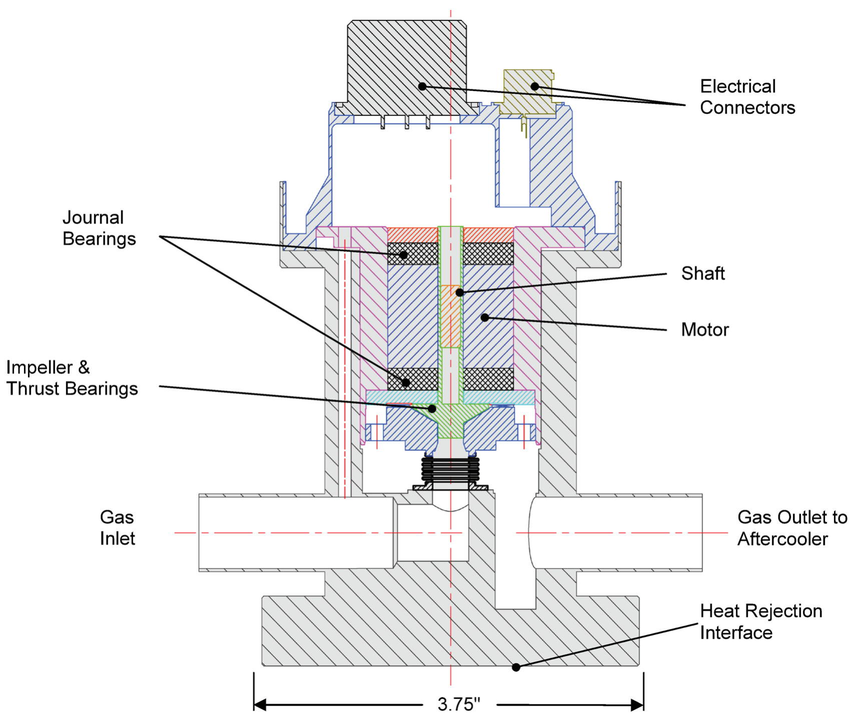

Figure 2. 400–500 W class permanent magnet motor compressor design [Hill et al. 2007].

Table 1. Compressor operating conditions and performance predictions

	Stage 1	Stage 2	Stage 3
Fluid	Helium	Helium	Helium
Inlet Pressure (atm)	4.9	5.6	6.3
Inlet Temperature (K)	300	300	300
Mass Flow Rate (g/s)	3.62	3.62	3.62
Pressure Ratio (-)	1.138	1.132	1.130
Rotational Speed (rev/s)	6300	6300	6300
Motor AC Input Power (W)	450	450	450
Head Coefficient (-)	0.58	0.56	0.55
Tip Flow Coefficient (-)	0.043	0.038	0.033
Compressor Efficiency – Predicted (%)	67	65	63

COMPRESSOR FABRICATION

Most of the compressor components are machined from bar stock. The compressor impellers are made from titanium to provide the strength needed to accommodate the large centrifugal stresses at high rotational operating speeds. Figure 3 shows a photograph of the compressor rotor after the impeller has been attached to the shaft using a threaded fastener. The shaft and impeller are coated with a hard coating for low friction and to reduce wear during start-up and coast-down when contact occurs with the bearing surfaces.

The outer housing assembly and cover with electrical passthrus for power lead wires and shaft runout probe signals is shown in Fig. 4.

The motor consists of a concentrated 3-phase winding on a 6-tooth core. The core is made from stacked laminations of Carpenter Alloy 49. The primary constituents of the alloy are nickel and iron in nearly equal proportions (about 50%). Its key feature is low core loss, compared with more commonly used silicon steel. The laminations are cut from sheet stock into the desired geometry via wire EDM. They are then coated with an oxide layer to provide electrical resistivity, stacked in a close-clearance motor housing which provides precise alignment between the laminations, and then the motor is wound with magnet wire. A photo of the completed stator assembly is shown in Fig. 5.

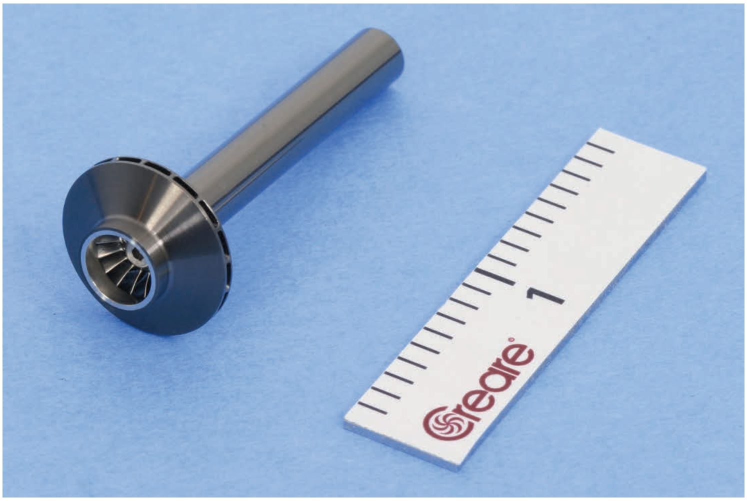

Figure 3. Completed compressor rotor assembly.

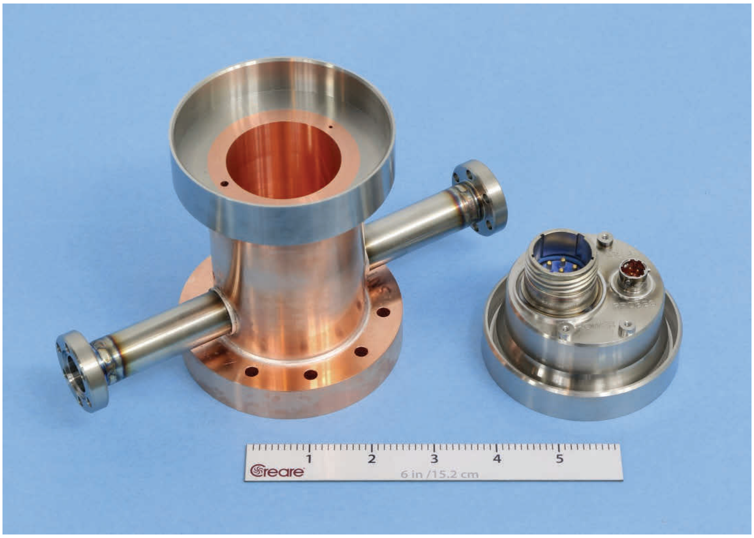

Figure 4. Compressor outer housing assembly with aft cover.

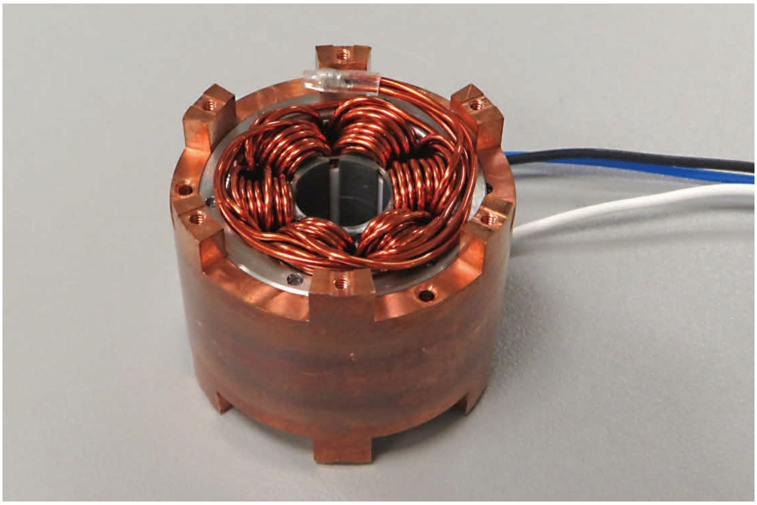

Figure 5. Compressor motor stator assembly.

The remaining compressor components are machined from copper or bronze. Figure 6 shows the three compressor assemblies ready for initial performance testing. During initial spin and performance tests, the aft covers are secured using compression clamps as shown in the photo. Once performance has been verified, a closeout weld is performed to hermetically seal the covers are welded to the housings.

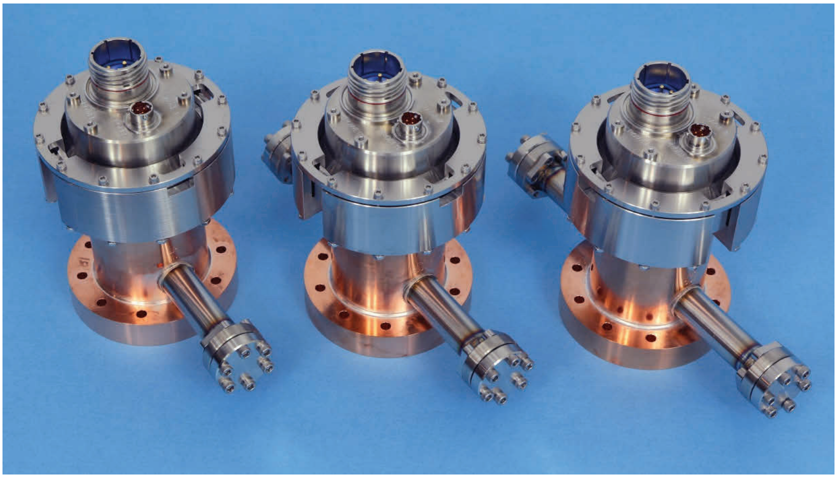

Figure 6. Stage 1, 2, and 3 compressor assemblies. The aft covers are temporarily attached using clamps during initial performance screening tests.

COMPRESSOR PERFORMANCE

Compressor performance testing was performed in an existing compressor test loop. The test loop includes all the equipment—plumbing; valves; pressure, power, and temperature instrumentation; heat rejection loop; and drive inverter—that is required to execute a compressor performance test. Initial testing has been completed on the Stage 3 compressor and is presented below. Full performance characterization of the three compressors, alone and in series, is ongoing.

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

Head Coefficient:
$$\Psi = \frac{\Delta h_S}{U^2} \tag{2}$$

Flow Coefficient:
$$\Phi = \frac{Q}{A \cdot U} = \frac{\dot{m}}{\rho \cdot \left(\frac{\pi}{4} \cdot D_{tip}^2\right) \cdot U} \tag{3}$$

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 tip diameter, $\dot{m}$ is the mass flow rate, ρ is the inlet density, and D_{tip} is the impeller tip diameter. Compressor efficiency is defined by Equation 1.

Figure 7 shows the head-flow curve measured during initial testing on the Stage 3 compressor along with the predicted performance from CFD analysis. In this test the compressor was operated at 5,500 rev/s instead of the 6,300 rev/s design speed. The compressor bearings currently require further adjustment to attain the design operating speed and a few iterations where the bearing clearances are systematically reduced are typical in the development of these gas bearing machines. The head-flow performance is close to predictions, but the peak in head coefficient occurs at a lower than expected flow coefficient. The difference is modest, and can be ameliorated by increasing the system pressure slightly.

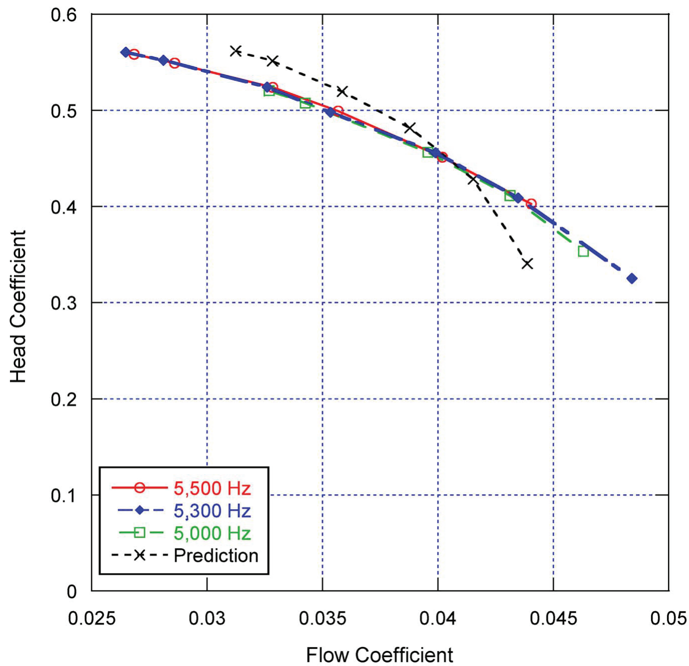

Figure 7. Preliminary head-flow curve from Stage 3 compressor testing.

The head-flow behavior is independent of operating speed, so data taken at lower operating speeds (Fig. 7) are representative of the compressor performance. However, the compressor net efficiency measured during the test was 57%, lower than the design target of 63%. The efficiency was reduced in part to the lower operating speed and further testing at the design speed is needed to quantify the difference from predictions, if any. Nevertheless, this net efficiency represents a considerable improvement over the efficiency of induction motor compressors of similar size (less than 40%), and corresponds to a reduction in input power of up to 30% as compared to prior Creare induction-motor compressors at this power level.

CONCLUSIONS AND FUTURE WORK

The design and fabrication of a 500 W class PMM compressor for a 20 K turbo-Brayton cryocooler is complete, and initial testing has commenced. This paper discussed the design, fabrication and initial test results for three such compressors for a 20 K cryocooler. The compressors have predicted efficiencies exceeding 60%, enabling the 20 K cryocooler to deliver 20 W of refrigeration with an input power below 1.6 kW, corresponding to a specific power of about 80 W/W. The compressor efficiency significantly exceeds that of prior induction motor compressors (less than 40%), resulting in a substantial reduction of input power to the cryocooler. Future work involves complete performance characterization for each compressor stage, integrating them into a cryocoolers, and testing the cryocooler.

ACKNOWLEDGMENT

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