Demonstration of an Ultra-Miniature Turboalternator for Space-Borne Turbo-Brayton Cryocooler

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ABSTRACT

A key component of turbo-Brayton cryocoolers is the cryogenic expansion turbine. This component removes work from the cycle gas at cryogenic temperatures, producing refrigeration. The design objectives for this component are high aerodynamic efficiency and low overhead losses. The latter objective is achieved in dual-temperature turbines by minimizing heat leak from the warm to cold end of the unit, and in isothermal turbines by minimizing electromagnetic and viscous losses. Turbines for space systems have adopted the use of electrical alternators (i.e., turboalternators) for turbine speed control and reduced cryocooler input power by recovering the turboalternator generated power. Creare recently built and demonstrated its smallest isothermal turboalternator. The turbine rotor diameter is 0.23 in. (5.8 mm), the shaft diameter is 0.08 in. (2.0 mm), and the rotor operates at nominally 3000 to 9000 rev/s. This turboalternator was optimized for operation in a cryocooler providing 1 to 2 W of refrigeration at 35 K. It is a 2/3-scale version of the flight-qualified NICMOS turboalternator and is our first turboalternator utilizing new fabrication techniques that enable higher aerodynamic efficiency. This turboalternator was tested at temperatures from 19 to 60 K and demonstrated high efficiency at all temperatures. In this paper, we review our turboalternator technology and heritage, and present results from the cryogenic testing of our ultra-miniature turboalternator.

INTRODUCTION

Creare Inc. began developing cryogenic turbines in 1979 when Herby Sixsmith joined the company. He brought with him gas bearing technology and precision fabrication techniques which are required for miniature high-speed turbomachines. Our initial turbine work was focused on turboexpanders. These devices operate with a cryogenic turbine rotor on a common shaft with an ambient temperature brake wheel. The brake wheel uses the turbine work to pump the cycle gas through an ambient flow loop. Turbine speed is controlled using a valve located in the ambient flow loop, which in addition consists of a heat exchanger and plumbing. The shaft is supported on gas bearings to eliminate contamination associated with oil-lubricated bearings and to allow high operating speeds required for peak aerodynamic efficiency.

During the period from 1983 through 1995, Creare produced eleven turboexpanders that produced refrigeration from 3 W at 4.2 K to 2.5 kW at 80 K. These units were developed for

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cryogenic systems at DOE National Laboratories as well as an emerging market for space cryocoolers to cool electro-optical sensor systems. In reverse turbo-Brayton cryocoolers, the turbine is used as the expansion device to produce refrigeration in the cycle. The high operating speeds of the turbomachines (compressors and turbines) precludes vibration at low frequencies which impacts pointing accuracy and image quality from sensor systems.

At low temperatures and at refrigeration capacities below say 10 W, the heat leak associated with an ambient brake wheel produces a significant refrigeration penalty. To mitigate this penalty, the brake wheel and gas bearings may operate at cryogenic temperatures reducing the temperature difference between the warm and cold ends of the turbine. This approach was demonstrated using the 3 W at 4.2 K turboexpander, but is difficult to implement in many systems because of the need for cryogenic heat rejection from the warm end.

In 1995, Creare investigated the use of a cryogenic alternator in place of a brake wheel to convert the shaft work to electrical power. Here, the speed control is provided by varying the electrical impedance or the back voltage in the control electronics. For space cryocoolers, a turboalternator is the preferred approach for the expansion turbine due to (1) the increased reliability and ease of operation associated with the elimination of the adjustable valve, (2) the reduction in system mass associated with the elimination of a secondary flow loop and heat exchanger, and (3) the reduction in system input power offered by recovering the electrical power generated by the turbine.

This initial turboalternator work resulted in the development and testing of brassboard [McCormick et al. 1997] and engineering model [Zagarola et al. 2004] turboalternators at refrigeration temperatures of nominally 30 to 80 K. This turboalternator has become known as the low-capacity (LC) turboalternator. The LC turbine rotor diameter is 0.30 in. (7.6 mm), the shaft diameter is 0.14 in. (3.6 mm), and the rotor operates at nominally 2000 to 6000 rev/s, the optimal speed depending on cycle gas, refrigeration temperature, and cycle pressure ratio. The alternator in this unit was sized to produce up to 30 W of output power though typical operation is at 5 to 20 W. The brassboard unit was later used for gas bearing demonstrations at temperatures as low as 12 K [Zagarola et al. 2002].

In 1997, NASA's NICMOS cryocooler program provided the opportunity to quickly mature the LC turboalternator to space flight quality. The NICMOS turboalternator provides 10 W of turbine refrigeration at about 70 K which results in 7 W of net refrigeration to the circulation system that cools the NICMOS instrument on the Hubble Space Telescope. The cryocooler and turboalternator were launched as part of Servicing Mission 3b and operated in space for 6.5 years without change in performance [Swift et al. 2008].

In the remainder of this paper, we describe the extension of the NICMOS turboalternator to other loads and temperatures. In particular, we review the advancements in technology that led to the development and demonstration of an ultra-miniature turboalternator for space-borne turbo-Brayton cryocoolers.

ADVANCEMENTS IN TURBINE FABRICATION AND ANALYSIS METHODS

Prior to 2009, Creare's miniature turbomachine rotors were fabricated using a precision plunge electro-discharge machine (EDM). This process uses an extremely fine electrode to machine the precise features of the rotor flow passages. The EDM was initially developed by Herby Sixsmith in the 1970's and evolved with ever-changing rotor designs and fabrication requirements. The EDM can only produce line-of-sight blade features limiting the aerodynamic performance of the turbine. More recently, Creare developed and qualified a new rotor fabrication process that utilizes an ultra-precision, 5-axis CNC mill and produces extremely precise and varied blade geometries in a fraction of the time of the EDM [Hill et al. 2010]. In the new process, we machine the blades and then diffusion bond a shroud to it to complete the rotor assembly. The rotor assembly is then machined to final dimensions and coated with a wear-resistant hard coating. Figure 1 shows the rotor parts fabricated using the new process, and Figure 2 shows the final rotor. The turbine rotor diameter is 0.60 in. (15 mm).
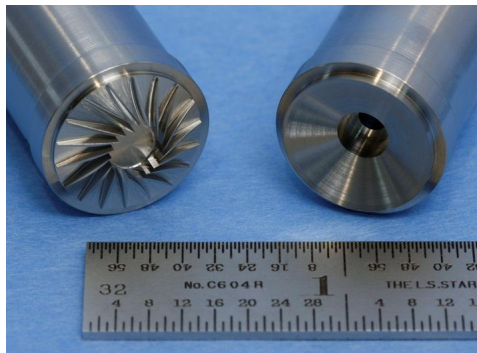
Figure 1. Turbine rotor parts prior to diffusion bonding
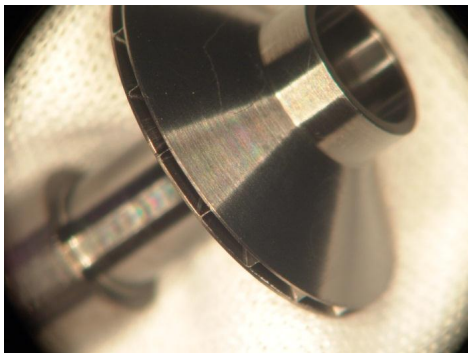

Figure 2. Turbine rotor assembly after diffusion bonding and coating

To take full advantage of the new rotor fabrication techniques, we implemented commercial aerodynamic design and performance predictions tools using ANSYS® CFX computational fluid dynamics software. The rotor designs can be simply ported to tool post programs used by the 5-axis CNC mill, structural analysis software, and mechanical design software. This software integration provides a timely and cost-effective method to optimize turbine rotors for new operating conditions.   To corroborate performance predictions, we performed companion experiments.   The experiments were performed at ambient temperature using a compressed nitrogen supply to provide the turbine pressure ratio. The testing was performed in a water-cooled housing to absorb the alternator and drag losses so that these losses did not impact the enthalpy decrease across the turbine.

The parameter of interest is the aerodynamic efficiency, which is the ratio of actual work transfer to the rotor to the isentropic work transfer. The test set-up was designed to permit the determination of the actual turbine work using two methods. In the first method, the output power is measured and adjusted for losses associated with electrical losses, leakage losses and viscous drag losses. The electrical losses are relatively small quantities that are calculated using standard electro-magnetic relationships.   The leakage losses are calculated from the measured seal clearances. The viscous drag losses are determined by measuring the motor input power rotating a blank rotor over a range of speeds and gas densities to cover a Reynolds number range that encompasses the test conditions.   The sum of electrical output power measured during the aerodynamic tests and the electrical, drag, and leakage losses equals the work transfer from the process gas to the turbine rotor. In the second method, the mass flow rate and the temperature (enthalpy) decrease across the turbine are measured directly. The product of these two quantities is equal to the turbine work transfer in the absence of alternator and drag losses, which were adsorbed by water cooling. The two methods for determining the work transfer provide an independent verification of test data.

The aerodynamic efficiency results are plotted in Figure 3 as a function of velocity ratio, the ratio of the turbine tip speed to isentropic spouting velocity. Also shown in Figure 3 are error bars based on known measurement uncertainties and a predicted curve from our ANSYS CFX simulations for this rotor design. The agreement between the experimental data (derived by both methods) and the predictions is extremely good. The peak efficiency is about 82% and occurs at a velocity ratio of about 0.70 to 0.75, which is consistent with theory.
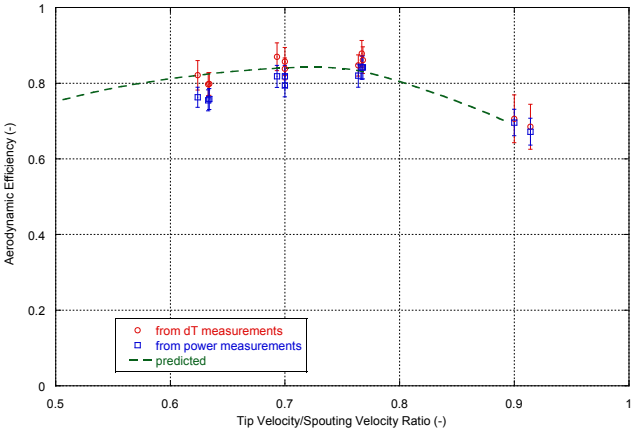
Figure 3. Comparison of aerodynamic performance data and predictions from CFD validation experiment

DEMONSTRATION OF AN ULTRA-MINIATURE TURBOALTERNATOR

Our first opportunity to utilize the new aerodynamic design, analysis, and fabrication techniques on a turboalternator was during the development of an ultra-miniature turboalternator. This turboalternator is a nominal 2/3-scale version of the NICMOS turboalternator. The smaller size is intended for operation at turbine refrigeration levels of 1 to 6 W. The turbine rotor diameter is 0.23 in. (5.8 mm), the shaft diameter is 0.08 in. (2.0 mm), and the rotor operates at nominally 3000 to 9000 rev/s, the optimal speed depending on cycle gas, refrigeration temperature, and cycle pressure ratio. Figure 4 shows a photograph of the rotor.

An existing open-loop cryogenic test facility was used to test the turboalternator. A schematic of the test facility is shown in Figure 5. Ultra high-purity helium or neon gas is supplied from high pressure bottles to a cryogenic adsorber that removes moisture. The test gas then enters the test facility, is precooled using two recuperators, cooled using a Gifford-McMahon (GM) cryocooler, and then enters the turboalternator. The two recuperators are used to reduce the heat load on the GM cryocooler. Trim heaters on the GM coldhead are used to control the temperature of the gas entering the turboalternator. The minimum turbine inlet temperature achievable in this test facility is nominally 19 K. The key measurements are the turbine inlet temperature, turbine pressure difference, turbine exit pressure, mass flow rate, rotational speed, and output power.

**Figure 4.** Ultra-miniature turbine rotor
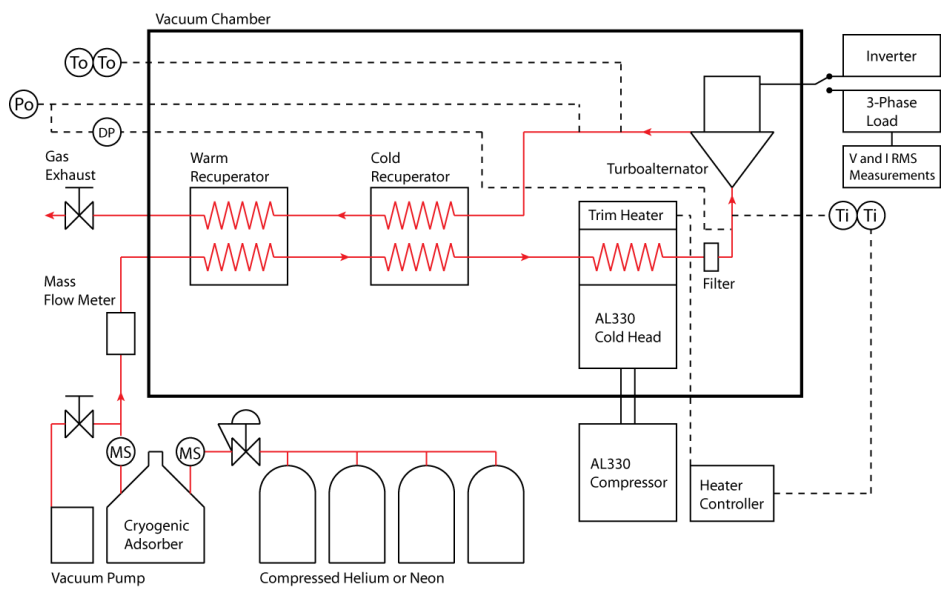
**Figure 5.** Schematic of open-loop cryogenic test facility

The test data were reduced and analyzed. The output power data were corrected for the electrical losses in the power harness between the alternator and the data acquisition system using the measured resistance. The key performance parameter is the net efficiency. This parameter is primarily a function of the operating speed and power level. The net efficiency of the turboalternator is plotted as a function of isentropic power in Figure 6. Here only data at the optimum speed are shown. The peak efficiency was 69% and the efficiency at the design operating conditions was 66%. The measured performance of the turboalternator far exceeds the net efficiency performance target of 52% at the design operating conditions.

The current turboalternator is labeled "ULC" in Figure 6, which stands for ultra-low capacity. Also shown in Figure 6 are data from testing our low capacity (labeled "LC") turboalternator at similar power levels. The net efficiency of the LC turboalternator is 18 to 28 efficiency points below the ULC unit at comparable power levels. The improved performance of the ULC unit is the result of utilizing a properly sized machine for a given capacity range and advancements in rotor design and fabrication techniques described above.

We also compared the mass flow rate through the turboalternator with CFD predictions (not shown). The comparison indicates that the measured mass flow rate is consistently 18% greater than the predicted value. This means that 18% of the turboalternator flow is bypassing the rotor and not being used to produce refrigeration. The discrepancy between the measured and predicted flow rate is partially attributable to internal leakage through the rotor clearance seals. This leakage is unavoidable in a gas bearing turbomachine and is estimated to be 3 to 5% of the rotor flow rate for this seal geometry. The additional leakage of 13% is believed to be associated with internal cross-stream leakage between the inlet (high-pressure) and outlet (low-pressure) manifolds. The impact on turboalternator performance is shown in Figure 7. A net efficiency of 75% appears possible which will lead to lighter and more efficient turbo-Brayton cryocoolers in the future and is readily achievable utilizing improved internal sealing methods.
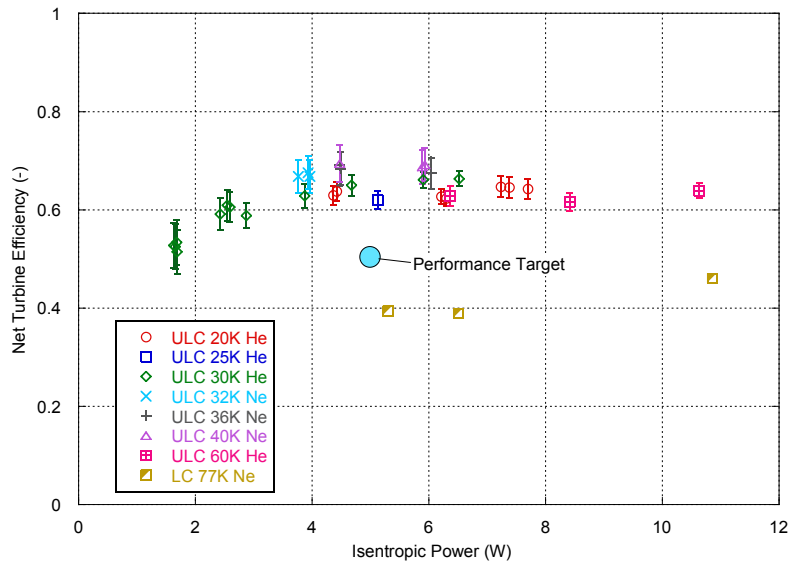
Figure 6. Net efficiency of ULC and LC turboalternators at low refrigeration capacity
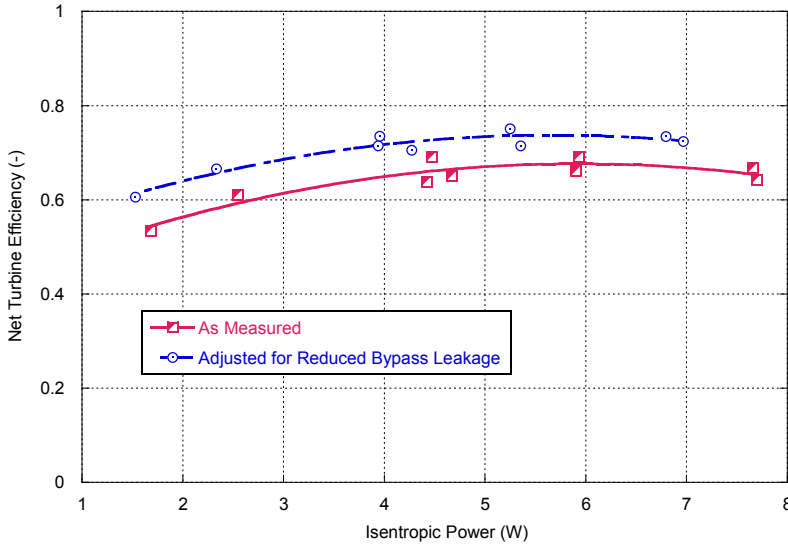
Figure 7. Impact of reduced cross-stream leakage on net efficiency

TURBOALTERNATORS FOR SPACE-BORNE CRYOCOOLERS

In addition to the ULC and LC turboalternators, Creare has built a mid-capacity (MC) turboalternator that utilizes a 0.40 in. (10 mm) turbine rotor [Zagarola et al. 2009] and has a basic design for a high-capacity (HC) turboalternator that utilizes a 0.50 in. (13 mm) turbine rotor. To date, we have built seven turboalternators for space applications. Current versions of the ULC, LC, MC, and HC turboalternators share common materials and processes which reduce qualification costs and development risks. Most parts are scaled in size to reflect the different power levels and rotor sizes, but share the same basic design.

The four turboalternator frame sizes can efficiently operate over a broad range of temperatures and refrigeration capacities. Figure 8 illustrates the nominal refrigeration range for each turboalternator frame size. The upper refrigeration limit is based on alternator power limits or cycle pressure limits. The lower refrigeration limit is based on efficiency. Lower refrigeration capacity is possible, but the efficiency of the turboalternator will decrease as the overhead losses become a larger fraction of the output power. The overlap region indicates that either frame size can be utilized, the final selection being based on cycle optimization and more detailed assessment of turboalternator performance. The ovals shown in Figure 8 indicate ranges where prior turboalternators have been tested at steady-state conditions. All turboalternators have operated from ambient or higher temperatures down to the steady-state test temperature. Helium is used as the cycle gas for operation at temperatures below 30 K, and neon or helium is used as the cycle gas at higher temperatures.

From a user's perspective, the net refrigeration is a more useful parameter than the turbine refrigeration. The difference between the two values being the recuperator loss which is typically 20 to 40% of the turbine refrigeration. Therefore, nominally 60 to 80% of the turbine refrigeration shown in Figure 8 is available to the user for heat lift. The four turboalternator frame sizes are expected to meet the needs of future space missions except (1) for low-capacity, high-temperature applications where reverse-Brayton cryocoolers are not particularly efficient, and (2) for zero boil-off (ZBO) cryogen storage where heat loads of up to 80 W at 20 K and 500 W at 90 K are possible depending on the size of the cryogen tank and thermal design of the payload. Turbo-Brayton cryocoolers are ideal for ZBO applications because of (1) favorable mass and performance scaling to high capacities and (2) the ability to directly cool cryogen tanks and shields without the need for an intermediate heat transport loop. For higher refrigeration capacities, we have developed a turboalternator that provides 2 kW of refrigeration at 50 K for a terrestrial application. The 2 kW turboalternator is a scaled version of our space turboalternators. Figure 9 shows a comparison between the ULC and the 2 kW turbine rotors.
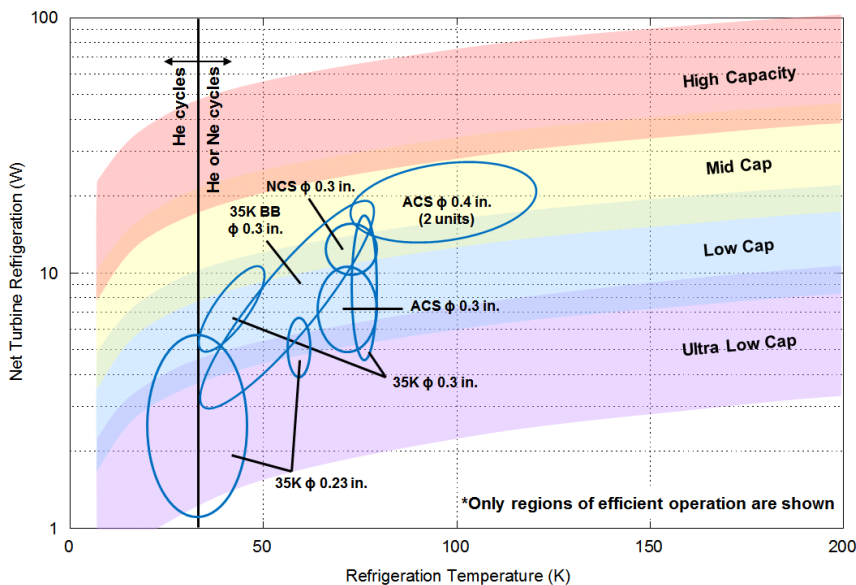

Figure 8. Refrigeration ranges for four turboalternator frame sizes

Figure 9. Comparison between ULC and 2 kW turboalternator rotors

SUMMARY

Creare has been developing cryogenic expansion turbines for over 30 years. Our work during the last 15 years has focused on turboalternators for space-borne cryocoolers. We have built seven turboalternators for space cryocoolers representing three frame sizes and have a basic design for a fourth frame size. The four frame sizes are suitable to handle the needs of most future space missions. The latest turboalternator is the ultra-low capacity, which is intended for operation at turbine refrigeration levels of 1 to 6 W. This turboalternator takes advantage of recent developments in rotor design, optimization and fabrication techniques to attain an unprecedented high efficiency for a miniature turboalternator. The impact of this development is lighter and more efficient turbo-Brayton cryocoolers in the future.

ACKNOWLEDGMENT

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