High Effectiveness Micro-Tube Recuperators for Low-Capacity Turbo Brayton Cryocoolers for Space

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ABSTRACT

In 2014, Creare, with our partners at Edare and Mezzo Technologies, started to develop a micro shell in tube recuperator for high capacity turbo Brayton cryocoolers. This technology was initially pursued for a 20 W at 20 K cryocooler because the cost, size, and mass of existing recuperator technologies did not scale well to high mass flow rates. The microtube technology has inherently high heat transfer rates per unit volume, which is ideal for high capacity cryocoolers because several kilowatts of heat must be transferred in a turbo Brayton recuperator. The challenge of using this technology for high performance cryogenic recuperators is minimization of two critical performance penalties: axial conduction within the core structure and flow maldistribution. Axial conduction is typically a non factor for high capacity cryocoolers. Flow distribution was addressed in the case of the 20 W at 20 K cryocooler by a combination of design features of the headers and core and using five modules in series to allow mixing. We recently applied the microtube technology to a low-capacity cryocooler producing less than 500 mW at 10 K. Here mass flow rates are an order of magnitude less than at high capacities, and thermal effectiveness requirements are commensurate (0.99 per module). This paper reviews the design features, analyses, and thermal performance test results of a low capacity microtube recuperator and compares the technology to prior space proven technology.

INTRODUCTION

Future NASA astronomical observatories will require efficient, long life, mechanical cryocoolers for cooling electro optical payloads and advanced space instruments including detectors, sensors, shields, telescopes, and sub Kelvin cryocoolers to temperatures of 10 K and below. Cooling loads for these applications will range from 50 mW to 500 mW at the primary load site. Concurrently, additional refrigeration loads at higher temperatures will be required for cooling shields, electronics, and/or optics. Mission durations may exceed 10 years, making stored cryogens an unacceptable option due to their mass. Current low temperature mechanical cryocoolers are too inefficient for space borne applications, requiring more than 500 W of electrical input power per watt of cooling at 10 K (i.e., 500 W/W). Brayton cycle coolers are well suited to this purpose, as they provide continuous flow in a closed loop, obviating the need for regenerative materials and allowing for remote cooling. This feature gives the Brayton cycle superior performance at low temperatures. In addition, Brayton cryocoolers produce negligible vibration that could create jitter of imaging systems or precision instruments.

A conceptual design of an optimized 10 K cryocooler to address this need and its predicted performance are given in Fig. 1. Performance predictions indicate that this two-stage turbo Brayton cryocooler

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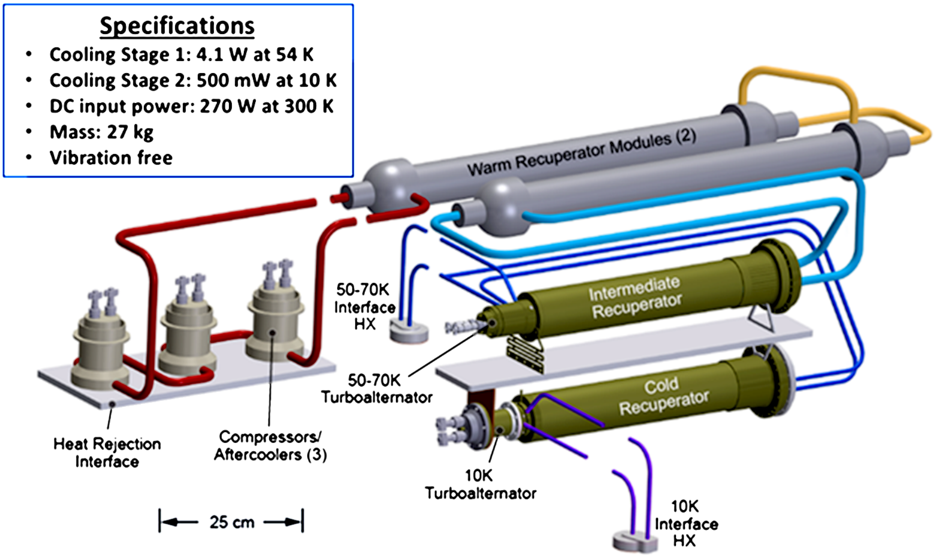

Figure 1. Optimized two-stage 10 K turbo Brayton cryocooler.

can provide over 500 mW of refrigeration at 10 K and 4.1 W at 54 K, with a heat rejection temperature of 300 K and a DC input power of 270 W. The estimated specific power at 10 K is less than 300 W/W when accounting for the Stage 1 load using Carnot scaling. This specific power is far less than any existing low temperature cryocooler. The mass of the cryocooler is nominally 27 kg. The low input power and mass will reduce overall payload mass (i.e., photovoltaics, heat rejection radiator, and cryocooler), enabling future space missions that require low temperature cooling.

This technology requires a multi stage recuperator set to recuperate heat between passing flow streams. The low power cryocooler requires a low capacity, high effectiveness recuperator with gas thermal capacitance on the order of 3 W/K. For space borne technologies, Creare has typically designed such recuperators using a slotted-plate technology, comprised of a core of conductive slotted plates separated by a thermal resistor to prevent axial conduction. A copper stainless slotted-plate design has traditionally been utilized in the higher temperature first stage, while the lower stages utilize either copper stainless or silicon plate technology, with the use of the latter owing to the very high conductivity of pure silicon at very low temperatures.

In 2014, Creare, with our partners at Edare and Mezzo Technologies, started to develop a micro shell in tube recuperator for high capacity turbo Brayton cryocoolers [1]. This technology was initially pursued for a 20 W at 20 K cryocooler because the cost, size, and mass of existing recuperator technologies did not scale well to high mass flow rates. The 20 W, 20 K recuperator transferred 5.2 kW down to 20 K, with an overall effectiveness of greater than 0.995 in a five module set. The capacitance of the flow stream was around 18 W/K, whereas the slotted-plate technologies were typically suitable for 1 3 W/K. In the present study, it was determined that the microtube recuperator performance and fabrication methods scale well to the low capacity application as well, providing a low pressure loss and low cost alternative to the legacy copper stainless technology. Keys to the successful adaption include minimization of axial conduction, and flow maldistribution performance penalties.

SIGNIFICANCE OF DEVELOPMENT

The 10 K microtube recuperator concept leverages legacy microtube recuperator development work while also implementing key design advancements, thus adding to the range of applications for which this technology is suitable. This class of recuperators provides the advantages of scalability in capacity; low pressure drop; and a lightweight, compact, hermetically sealed core. In addition, the use of all-metal components minimizes outgassing and enables high temperature bakeout for removing system contami-

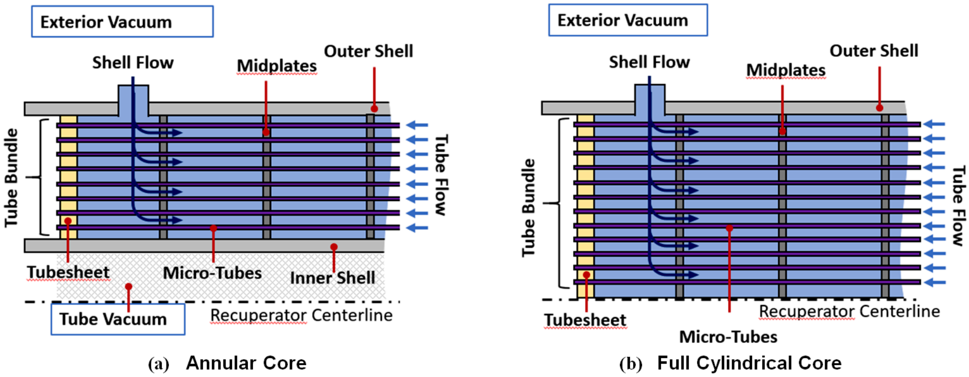

Figure 2. Illustration of microtube recuperator core layout. Relative to the legacy annular core arrangement (a), the fully cylindrical core (b) populates the core interior with microtubes, leading to a more compact, lightweight design. The cylindrical core also eliminates the inner shell, reducing parasitic axial conduction and further reducing weight.

nants. The lack of polymers also reduces the chance for structural performance degradation or failure due to cold flow, and the lack of exotic materials in the design reduces fabrication cost.

RECUPERATOR DESIGN AND ANALYSIS

Like previous designs, the 10 K shell and microtube recuperator has a core consisting of thousands of stainless steel microtubes that are laser welded to a stainless steel tubesheet. A thin, welded stainless steel shell provides the hermetic boundary. Welded stainless steel headers on the ends of the module provide the flow manifold boundary and transition flow to the core.

An innovation implemented in the 10 K microtube recuperator design is a fully populated cylindrical core arrangement of microtubes to reduce size and increase specific conductance of the recuperator. Legacy annular cores integrated an inner vacuum shell that was used to blank the central core and reduce radial flow travel distance from the flow manifolds into the shell space of the core. The cylindrical core eliminates the inner shell and populates the volume with additional microtubes. Elimination of the inner core also saves on weight and cost and reduces axial conduction penalty. Figure 2 illustrates the layout of both the annular core and cylindrical core. To enable the cylindrical core arrangement without causing excessive maldistribution in the shell side flow, it is critical to control the flow distribution and pressure losses within the shell side flow. Excessive flow restriction in the radial direction can cause a flow imbalance in the shell side of the recuperator. Maldistribution of the flow between the shell flow and the tube flow would diminish heat transfer in the imbalanced regions, resulting in a lower overall recuperator effectiveness. The shell side flow balance is achieved with the use of frequent and regularly spaced shell side mid plates that introduce axial flow restriction in the shell space and tightly control the shell side flow paths around the microtubes.

The recuperator was designed based on the requirements of the 10 K cryocooler. The nominal operating conditions of the recuperator are 0.6 g/s of helium, shell side inlet pressure of 3.6 atm and inlet temperature of 300 K, and tube side inlet pressure of 2.6 atm and inlet temperature of 140 K. An existing recuperator performance model was modified for this application and was exercised to minimize the mass of the recuperator while achieving the effectiveness and pressure drop required by the cryocooler thermodynamic cycle. The baseline model has been validated using test data from previously fabricated microtube recuperators. Performance loss due to maldistribution was estimated based on historical test data, with an appropriate margin on ineffectiveness added to account for unavoidable maldistribution losses.

To enable similarity to past builds, constraints were placed on the design configuration, including restrictions on length for ease of fabrication and a microtube size of approximately 0.6 mm. To meet length constraints for manufacturability and compact packaging, the recuperator was divided into two modules in series. Each module has a shell outer diameter of 6.5 cm, a core of length 58 cm, and an overall length of 72 cm. The core contains over 3,800 tubes.

The mass per module is 3.4 kg, giving a total mass of 6.8 kg (not including structural supports). The predicted nominal performance parameters are an overall effectiveness 0.996 and a total non dimensional pressure drop (dP/P) of 2.7%. The predicted recuperator loss is 1.6 W. The recuperator loss is defined as the cold end stream to stream enthalpy difference multiplied by the recuperator mass flow rate. This loss must be absorbed by the turboalternator of the cryocooler, in addition to the cryocooler heat load. Parasitics should also be considered in this loss estimate, but these are highly dependent on the packaging and configuration of the cryocooler and insulation, which were not sufficiently defined to include in this analysis model.

RECUPERATOR FABRICATION

To demonstrate recuperator fabrication, a single module of the two module design was built. Fabrication challenges include fabricating a thin shell to reduce axial conduction, attaining a rapid and high reliability welding process for joining microtubes to tubes to the tubesheet, attaining uniform shell side flow, and preventing bypass flow between the shell and core.

The shell must be hermetic and able to withstand both elevated internal pressure and launch vibration loads without failure, yet thin to keep the heat exchanger lightweight and reduce parasitic axial thermal conduction from hot end to cold end. The recuperator shell was fabricated from a honed cylinder of stainless steel. The honed cylinder for the shell was then turned on a lathe to attain a wall thickness of less than 1 mm. Mid plates were fabricated to provide a tight gap to the shell inner diameter to prevent bypass flow between the core and shell, a form of flow maldistribution. With the shell complete, the microtubes, mid plates, and spacer components were assembled to create the core.

To join microtubes to the tubesheet, a CNC laser welding process was used that was developed on a previous recuperator build [1]. High reliability of the laser welding must be attained to create leak tight joints and prevent cross stream leakage. After the tubesheets are welded, a bubble point leak check is performed to identify non hermetical joints. In the event of a failed weld joint, several techniques to repair the joint have been developed: (1) additional laser weld passes to seal joints; (2) cutting failed welds, extracting the tubes, inserting and rewelding new tubes; and (3) weld sealing failed tubes. The weld sealing method is avoided, if possible, because the blocked tube creates a local flow imbalance in the recuperator that results in a loss of heat transfer. Only a few failed tubes can significantly degrade recuperator performance.

Following the successful laser welding of microtubes (no tubes required weld sealing), the tubesheet and shell were sealed closed by welding, then headers and flow tubes were welded onto the core ends to complete the recuperator build. The completed 10 K recuperator module is shown in Fig. 3.

RECUPERATOR TESTING

The completed recuperator was tested for cross stream and external leaks using a helium vacuum leak check, and proof pressure tested based on the requirements of the cryocooler. These tests verified workmanship and established viability of the heat exchanger for installation into a cryocooler or test facility. The fabricated unit passed all tests.

Figure 3. Completed microtube recuperator assembly.

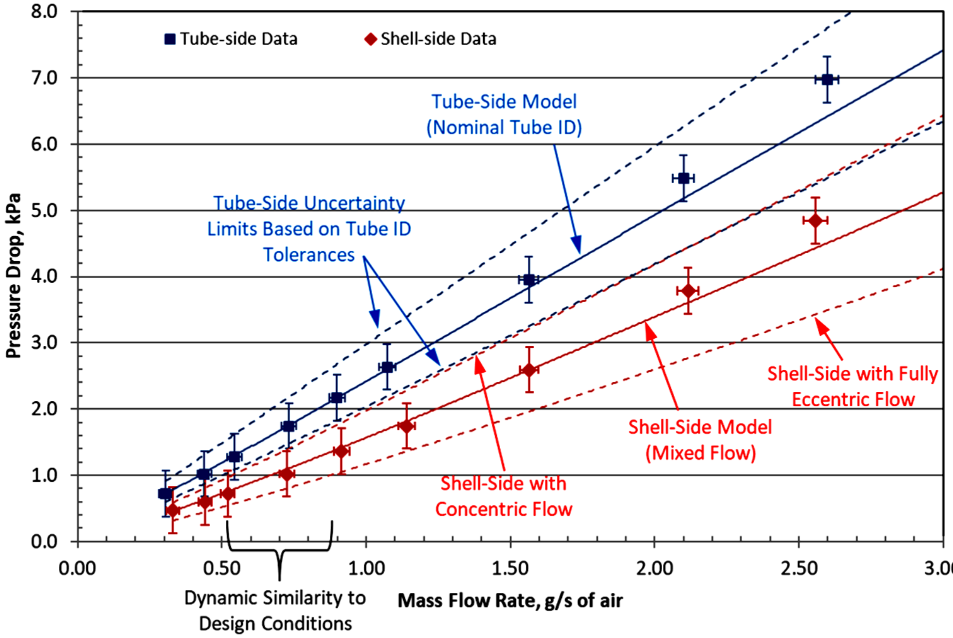

Figure 4. Pressure flow test data for microtube recuperator, with data compared to model prediction for a room temperature test with air. Error bars provide the uncertainty from test instrumentation. Model predictions are shown by solid lines for tube side (blue) and shell side (maroon) flow streams. Uncertainty of pressure loss prediction from fabrication tolerances is highlighted by dashed lines.

Pressure flow testing of the recuperator was completed using a shop air supply at room temperature. A pressure transducer provided pressure drop across the unit, and a laminar flow meter measured the air flow rate and temperature going into the recuperator. The high pressure (shell side) and low pressure (tube side) flow streams were tested separately. Following testing, the readings were corrected to design conditions by matching the Reynolds number using the average pressure and temperature of the prototypical operating conditions.

A wide range of flow rates around conditions dynamically similar to the design point were tested, and the test results were compared to model predictions, as shown in Fig. 4. Uncertainty bounds of the data points are also included, calculated from the uncertainty of the instrumentation datasheets.

Figure 4 also shows the ranges of pressure loss that are possible based on fabrication tolerances. Pressure loss on the shell side is predominately through annular passages between through holes in the mid plates and the outer diameter of the microtubes. Based on the long aspect ratio of the annulus, the pressure drop relation was modeled as flow passing through an annular tube (eccentric or concentric) per Kacaç, et al. [2], combined with entrance and exit losses per Idelchik [3]. The pressure loss varies depending on whether the flow is assumed to be passing through the mid plate holes eccentrically, concentrically, or a mix of concentric and eccentric condition. For the model correlation, the mixed condition was chosen (plotted as solid line), but both fully eccentric and concentric loss predictions were also plotted as dashed lines. This resulted in good agreement between model and data for the shell side flow; the difference was less than 7% in the region of dynamic similarity between test data and design conditions. Likewise, the tube side data was in excellent agreement between test and design conditions. The percent difference was less than 2% in the region of dynamic similarity. The range limits for pressure loss for the tube side data were based on the published tolerances for the microtube diameter and wall thickness; the dashed lines represent pressure loss through tubes at maximum and minimum tolerance conditions, while the solid line provides the pressure loss through nominal tube inner diameter.

To evaluate thermal effectiveness, the recuperator was tested under cold temperature conditions. The warm end of the recuperator was held at 300 K, and the cold end was tested at 140 K and at 220 K. The larger temperature range (300 K to 140 K) increases the stream to stream temperature difference and

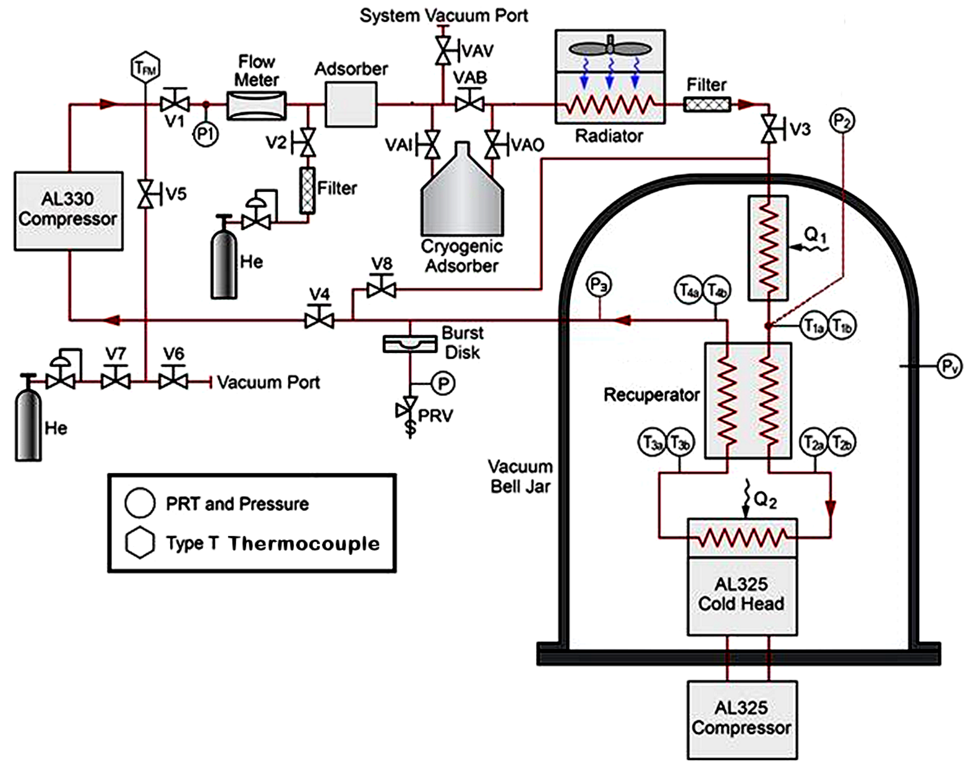

Figure 5. Cryogenic thermal performance test facility schematic.

improves fidelity in the thermal measurements. The test facility schematic is shown in Fig. 5. It consisted of a cryogenic closed circulation loop with a circulator. Helium was introduced into the loop and was cooled by an AL325 G M cryocooler at the cold end of the loop. A pair of adsorbers reduced water and other contaminants from the cycle fluid. At each cold end test temperature and flow rate, temperature was controlled to the desired value by manual and automatic heater adjustment. The recuperator and all cold end tubing components were housed inside of a vacuum bell jar. Temperatures were measured at the inlet and outlet ports of both flow streams of the recuperator using redundant, in line Platinum Resistance Thermometers (PRTs), and the flow rate was measured using a rotameter. The facility operates in a closed loop mode, allowing testing for a sufficient time period to reach steady state thermal conditions at each data point. Nine layers of multi-layer insulation around the cold components minimized heat leak to the environment (predicted heat leak was 210 mW).

During the thermal test, a total of five data points were collected at each temperature, with flow rates varied from the design flow rate upward. The key performance metric during testing was thermal ineffectiveness. Real gas enthalpy values calculated from temperature and pressure measurements, along with flow rate, were used to calculate ineffectiveness. The results of the test are shown in Fig. 6 for the cold end at 140 K, and Fig. 7 for the cold end at 220 K.

Uncertainty bars are based on published uncertainty in the rotameter flow meter and PRT measurements, as well as the various uncertainty contributions of the Lakeshore 218 temperature monitor used to convert temperature readings. The model predicted performance is also shown in the figures. The test results show good self similarity between the measurements of the warm end and cold end effectiveness calculations for all data points, with the exception of the 220 K, 0.6 g/s data point. The discrepancy at this data point is likely due to insufficient time taken to reach thermal stabilization. The data also show excellent agreement with the model for the nine stabilized data points; for these, the model predictions fall within the data uncertainty band. Good agreement exists between model predicted and measured ineffectiveness, with a maximum ineffectiveness difference of 0.0014 for the stabilized data points.

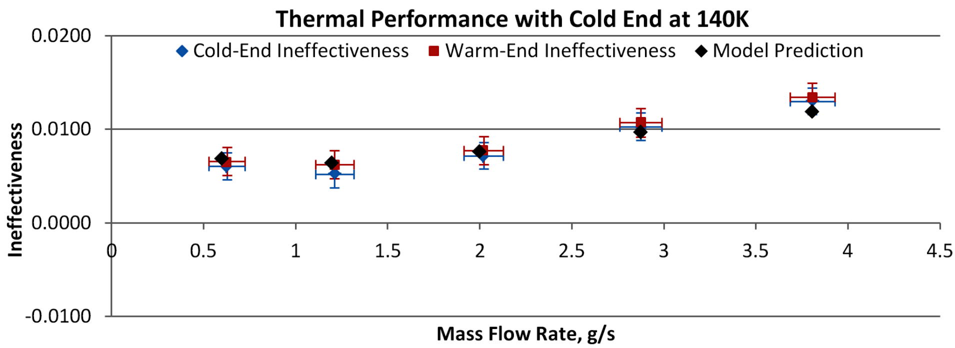

Figure 6. Recuperator ineffectiveness with cold end at 140 K, compared to model predictions.

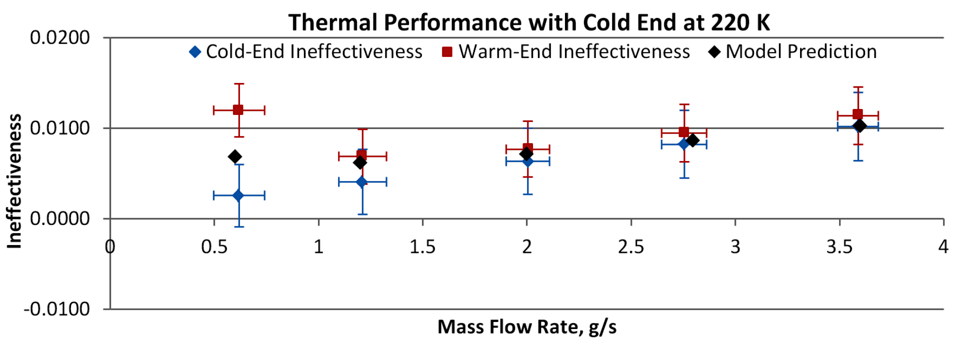

Figure 7. Recuperator ineffectiveness with cold end at 220 K, compared to model predictions.

CONCLUSIONS

Creare, in collaboration with Edare and Mezzo Technologies, has developed a high effectiveness shell and microtube recuperator for low capacity cryocooler applications, with capacity on the order of 3 W/K. This technology integrates well into the upper stage of the cryocooler upper stages and serves as a substitute for legacy copper stainless, stacked-plate units built for space borne applications. The low capacity microtube recuperator leveraged design, modeling, and fabrication techniques utilized for development of a high capacity microtube recuperator and further innovated the technology with a fully populated cylindrical core with design features to promote uniform flow. Thermal and performance testing of a built module confirmed model predictions for effectiveness and pressure drop. In the 10 K cryocooler application, the two module recuperator set is expected to provide an effectiveness of approximately 0.996, with a non dimensional pressure drop (dP/P) of 2.7%. The results confirm that the microtube recuperator is well suited for integration into future lightweight, low capacity turbo Brayton cryocoolers for space borne applications.

ACKNOWLEDGMENTS

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