A 40 K Turbo-Brayton Cryocooler for Earth Observation Applications

J. Tanchon, J. Lacapere, J.C. Rey, A. Molyneaux[1], T. Tirolien[2]

Absolut System SAS, Seyssinet-Pariset, France
[1]Ofttech, Gloucester, United Kingdom
[2]European Space Agency, Noordwijk, The Netherlands

ABSTRACT

Several types of active cryocoolers have been developed for space and military applications in the last ten years. Performance and reliability have continuously increased to follow the requirements evolution of new generations of satellites: less power consumption, more cooling capacity, and increase life duration.

The microvibrations exported by the cryocooler are becoming the main source of microvibrations on board the satellite. For some applications on board earth observation satellites, the existing Stirling or Pulse Tube cryocoolers exceed the targeted microvibration levels, even when active control is used on those cryocoolers.

To offer an alternative solution, Absolut System has developed a 40 K turbo-Brayton cryocooler using very high speed turbomachines in order to avoid any generated perturbations below 1000 Hz. This development has been performed as part of an ESA Technical Research Program - 4000113495/15/NL/KML.

As the design and expected performance for this cryocooler have been published previously, this paper presents the manufacturing outcome and preliminary performance results.

INTRODUCTION

In recent years in Europe, several cryocooler developments for Earth Observation (IR Detection) applications have been conducted to provide significant cooling power at 50 K. Those single-stage Stirling or Pulse Tube coolers are mechanisms that involve moving parts at low frequency (in the compressor and in the Cold Finger for the Stirling) that induce microvibrations. Relatively low exported microvibrations (<100 mN in all directions at all harmonics) have been achieved with active microvibration cancellation and careful screening and manufacturing of the cooler's parts. However, meeting ever more stringent microvibration requirements has led to very complex solutions to overcome those vibrations (e.g. suspended coolers and radiators, and flexible thermal link assemblies that degrade the overall thermal performance).

To further decrease the exported vibration level generated by cryocoolers, vibration-free cooler technologies need to be developed for the next generation of instruments. This includes both Sorption and turbo-Brayton based cryocoolers. The Sorption cooler offers the advantage of being free of vibrating mechanisms (source of vibration) while the turbo-Brayton only generates exported vibration far above the frequency bandwidth (0-1000 Hz) sensitive to most instruments.

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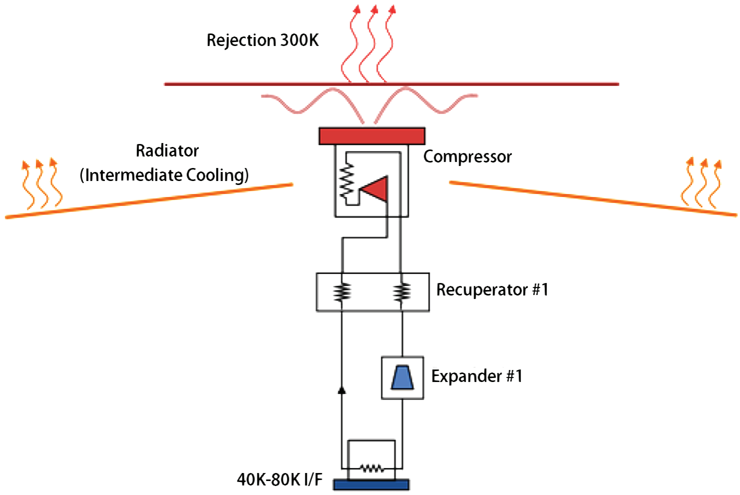

Figure 1. Baseline architecture of the 40-80K vibration-free Brayton cooler

The turbo-Brayton cycle cryocooler uses miniature, high-speed turbomachines and high-effectiveness recuperators to provide efficient cooling with low vibrations and high reliability. Gas bearings are used in the miniature machines to support the rotors, which operate at speeds of 100,000 to 600,000 rpm [1]. The low-mass rotors are the only moving parts in the systems, and because they are precision balanced, the systems are inherently vibration-free. No additional electronics or hardware is required to suppress vibrations. The gas bearings also provide non-contact operation, so performance degradation resulting from wear or the accumulation of debris is absent. These systems are generally capable of maintenance-free operating lives of 5 to 20 years [2][3].

The Turbo-Brayton cryocooler developed by Absolut System is configured in a conventional way including a 2-stage compressor, a recuperative heat exchanger, an expansion turbine, and a thermal heat exchanger interfaced with the customer load (focal plane, detector, thermal shield…). These components may be integrated into a compact package or distributed over fairly large areas, interconnected by lengths of tubing. Refrigeration can be delivered to multiple loads either at a single temperature or at several different temperatures. The second type of delivery can be accomplished either by multi-staging an integral cooler or by combining several cryocoolers at the appropriate interfaces. Cooling loads and thermal interfaces may be separated by large distances without significant effects on overall system efficiency. Thus, the turbo-Brayton cryocooler can be implemented in a variety of ways in space applications. All these advantages appeared to solve many issues generated by cryocoolers in the space instrument arrangement.

The architecture of the reverse turbo-Brayton cycle proposed is presented in the Figure 1. This schematic constitutes the baseline architecture of the cooler. The centrifugal compressor is heat sunk on a 300 K radiator (the radiator surface required is below 1m^2). Then, the counter-flow heat exchanger is used to precool the gas before entering the turbine where it is expanded. The cold gas goes through a heat exchanger to be connected to the load to be cooled and goes back to the counter-flow heat exchanger up to the compressor. This concept is the simplest way to build a turbo-Brayton cooler.

SUMMARY DESCRIPTION OF THE CRYOCOOLER DESIGN

The 40-80 K vibration free cooler is composed of the following subsystems as detailed in Figure 2 and Table 1:

- A 2-stage centrifugal compressor system
- A microtube recuperator
- A cryogenic expander subsystem (including a cold electrical generator)
- An external supporting structure

The compressor consists in a pair of identical units in a series formation. Each unit contains a centrifugal compressor wheel optimized to be operated at 250,000 rpm, supported on gas bearings and driven by a

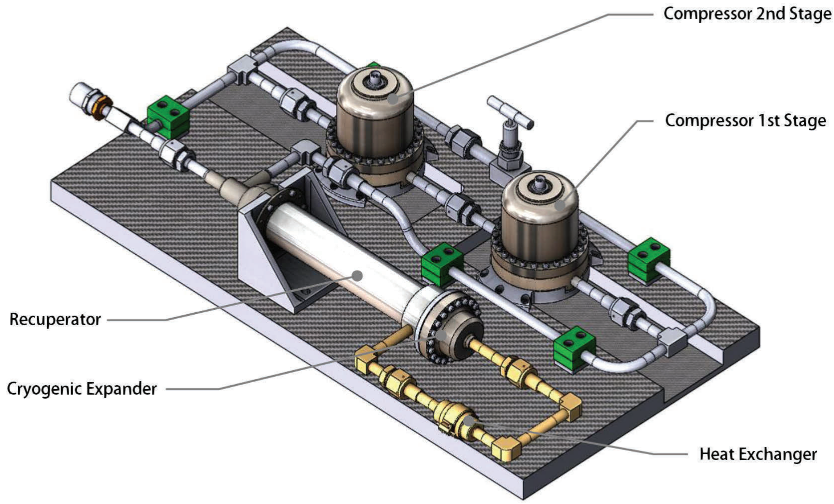

Figure 2. Overview of the 40-80 K vibration free cooler

Table 1. Summary of 40-80K turbo-Brayton characteristics

Components	
Working fluid	- Pure Neon

Compressor	- Use of 2 stage compression
	- Use of 2 identical compressors with the same wheels and the same motor in each compressor
	- Each compressor subsystem uses an aftercooler to reject compression work
	- Isentropic efficiency : 57.6 % for the following inputs
	    o 1.25 g/s
	    o (260 K, 1 bar) at the first compressor inlet
	    o 1.80 pressure ratio
	- Efficiency of the motor : 94.5 % considering a PAM electronic driver
	- Windage losses included in the isentropic efficiency and in the motor efficiency
	- Shaft diameter of 6 mm
	- Nominal rotational speed of 250 000 rpm
	- Hydrodynamic losses on the bearings (radial and thrust bearings) calculated for each case regarding rotation speed and shaft dimensions : 10 W per compressor at 370 K (nominal temperature of the bearings in steady conditions)

Expander	- Isentropic efficiency : 68 % for 40 K steady-state - Using the same wheel for other cold temperature, this efficiency is decreased down to 60 % for 80 K cold temperature - Shaft diameter of 4.2 mm - Rotation speed limited to 150 000 rpm - Bearing losses (including radial and thrust bearings) about 0.3 W at 40 K - Generator losses about 0.75 W at 150 kRPM and 40 K
Recuperator	- 1 recuperator with an efficiency of 98 % and total pressure losses of 1500 Pa for both sides

permanent-magnet motor. The Neon gas flows through each compressor stage; the first is a low pressure stage, and the second, a high pressure stage. The compression heat is rejected using an aftercooler integrated into the compressor body. The heat exchangers reside in the warm thermal interface of the cryocooler.

The cryogenic expander subsystem is dedicated to extract power to the fluid flow in order to decrease its temperature. This extraction is made by an expansion through a turbine wheel and permanent magnet generator. The internal design of the expander is very similar to the one used for the centrifugal compressor except for the dimensions of the components.

Both the compressor and expander operate using hydrodynamic gas bearing technology. The gas bearings used here are of the spiral groove type. These are identical in form to those used both in the MELFI project and Atlas Copco Helium expander [4]— all designed using the same methods by the same experts. These gas bearings guarantee stability and load capacity and require no adjustment or setting up once manufactured.

The recuperator is based on microtube heat exchanger technology. One of the advantages of this technology is that the recuperator is quite compact compared to other technologies. The recuperator has a cylindrical shape with the interfaces on the two extremities. This configuration is very interesting for structural aspects and also eases the implementation of radiative insulation. The recuperator is mounted rigidly on the mechanical support of the cryocooler. On the other side (cold side), the expander is directly integrated into the recuperator.

BREADBOARDING ACTIVITIES AND RECUPERATOR OPTIMIZATION

In order to mitigate the different technical risks highlighted during the preliminary design phase, different breadboard activities have been run. These activities concern different aspects of the cooler from material characterization down to elementary component testing and manufacturing technics validation.

The thermodynamic cycle used for the turbo-Brayton cooler is well known and benefits from a large background in industry. The main challenges regarding our development is to miniaturize the size of the components while operating at very high frequency. As a consequence, the manufacturing techniques used to produce the parts exceed the conventional manufacturing techniques and require specific processes and controls.

The main points addressed during the breadboard phase are:

- Manufacturing and tests of different recuperator configurations
- Characterization of core losses from 40 K to Room Temperature for different materials and high frequency 1000 to 5000 Hz
- Thermal contraction of permanent magnet materials and mounting process in the shaft
- Diffusion bonding process development and characterization
- Tests of specific coating on representative samples for gas bearings

In this paper, we focus on the activities performed on the recuperator (which is the more related to cooler performance). All the other activities are linked to the production aspects.

Recuperator Performance Evaluation

The recuperator is one of the critical points of a Brayton cycle. The efficiency of the cycle is directly impacted by the performance of the recuperator. The exchangers are characterized by:

- Their thermal efficiency (ratio between the enthalpy transferred and the total enthalpy transferable)
- Their pressure drop, which directly impacts the expansion work available in the expander (and therefore the cooling power)

A recuperator will be called “perfect” when all the enthalpy transferred by the fluid (High Pressure side) is transferred to the other fluid (Low pressure side) without temperature difference (between point 4 and point 6 of the TS diagram of Figure 3, T = 0 for the “perfect” case). The greater the difference, the less cooling power will be available for the application heat exchanger (user load).

This is illustrated in Figure 3, where the cooling power produced during the expansion is represented by the temperature drop between point 4 and point 5. It is clearly seen that the more the temperature difference across the recuperator increases, the less power is available for the cold heat exchanger.

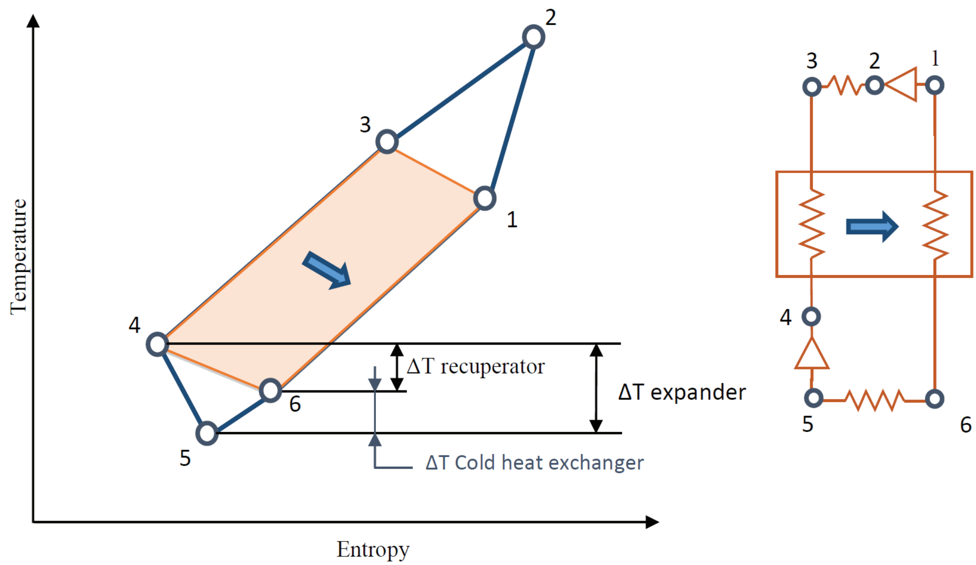

Figure 3. Impact of the recuperator efficiency

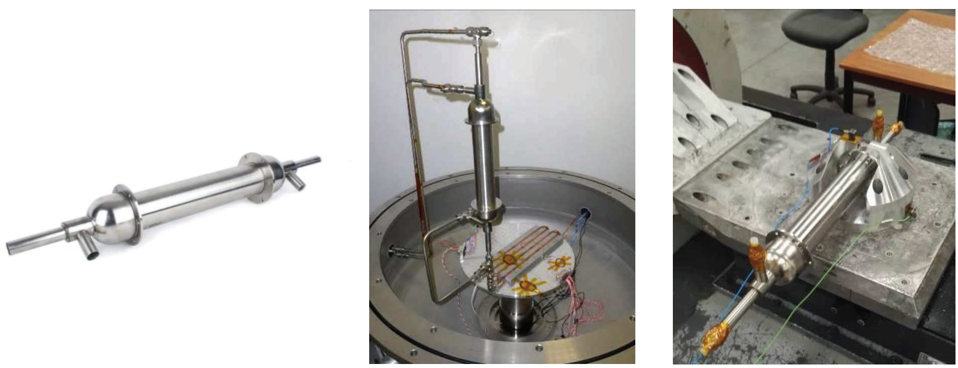

Figure 4. Picture of the recuperator prototype, test setup in vacuum test chamber and shaker

Several technologies could be used for the recuperator. A baseline has been selected using a micro-tube heat exchanger for its very good compromise between efficiency, pressure drop and mechanical robustness for launch loads. This technology has been prototyped in order to evaluate its performance and to compare it with the predicted performance from CFD models. Figure 4 shows the prototype of the recuperator manufactured by Mezzo Technologies and its implementation in Absolut System test facilities for characterization.

This recuperator has been designed to reach a 98.5% for the nominal flow rate and nominal temperature difference. Unfortunately, the efficiency measured during the test campaign was lower than expected. The efficiency measured was 95.6% instead of 98.5% for nominal conditions.

After analysis and discussion with Mezzo, the root cause has been identified and attributed to an abnormal flow distribution in the recuperator which leads to a limitation of the area used for heat transfer. This conclusion has been cross-checked with the pressure drop measurement which confirmed this conclusion.

Table 2. Impact of different recuperator options on the efficiency and cooler performance

Case	Shell OD [mm]	Exposed Tube Length [mm]	µtube OD [mm]	Eff. without axial conductance [%]	Eff. with axial conductance [%]	Cooling power@40K
OPTION 1	50.8	228.6 / 9"	0.56	99.0	98.1	1 W
OPTION 2	50.8	228.6 / 9"	0.41	99.4	98.1	1 W
OPTION 3	50.8	304.8 / 12"	0.56	99.2	98.5	1.6 W
OPTION 4	50.8	381 / 15"	0.56	99.6	98.9	3 W
OPTION 5	50.8	431.0 / 17"	0.72	99.7	99.3	4.1 W

As explained previously, the recuperator efficiency is a strong driver in the cooler performance. For this reason, a mitigation plan has been setup to limit the risk not to achieve the performance in the final recuperator to be produced for this cooler. Several design configurations have been compared and several recuperator lengths have been evaluated regarding structural constraints. The different options evaluated are:

- OPTION 1: breadboard design (9" long) + optimization of the internal flow distribution
- OPTION 2: breadboard design (9" long) + optimization of the internal flow distribution + smaller microtube length
- OPTION 3: Extended recuperator 12" long
- OPTION 4: Extended recuperator 15" long with launch support
- OPTION 5: Extended recuperator 17" long with larger microtubes and with launch support

For each design, the efficiency has been evaluated using analysis. The efficiency includes or not the axial conductivity in the tubes (external shell and microtubes). From Table 2, one can see that the nominal configuration corresponds to an efficiency of 98.1%. With the same design, but with smaller microtubes (OPTION 2), the net efficiency is constant due to higher thermal conductivity along the microtubes. This solution is then not interesting because we have more tubes to weld and no improvement in performance. This design could be interesting with smaller tubes of smaller thickness, but this option makes the production more difficult and the price prohibitive.

The OPTION 3, 4 and 5 are options where the length is increased while all the other dimensions are kept constant. The difference between OPTION 3 and 4 is that a mechanical support, called launch support, is needed for the OPTION 4 due to the length of the recuperator. This constraint is directly linked to the mechanical analysis performed in parallel for all the designs. The OPTION 4 has been selected for the next step of the project. This design allows a significant improvement of the cooling power with minor modifications of the design. The additional length requires the implementation of a launch support, but this support also adds an additional robustness to the shell.

MANUFACTURING AND TESTING OF THE SYSTEM

Following the detailed engineering supported with many different elementary breadboards, the manufacturing of the final 40-80 K vibration-free cooler started.

Compressors Manufacturing

Figure 5 presents different pictures of the centrifugal compressor at different steps in the manufacturing. The shaft is a monolithic part, and for this reason the impeller needs to be machined first before it is fully welded. After welding, the final shape of the shaft is machined. The final part can be seen on the middle view of Figure 5.

After magnet mounting, balancing, and many other intermediate manufacturing steps, the shaft is mounted into the compressor stator under ISO5 clean environment conditions.

Figure 5. Centrifugal compressor: impeller before diffusion welding, final shaft, compressor assembly with the rotor

Figure 6. Cryogenic expander: impeller before diffusion welding, final shaft, expander assembly without the rotor

Expander Manufacturing

The expander manufacturing (Figure 6) followed roughly the same manufacturing steps as the compressor, except that additional thermal treatment is implemented on the parts due to the criticality of the dimensional stability. The expander will operate at cryogenic temperature and the dimensions of the parts must be ensured from room temperature down to 40 K. For example, a bending of the shaft during cool down will generate a failure of the expander due to dynamic stability.

Recuperator Manufacturing

The recuperator is mostly produced by Mezzo Technologies located in Baton Rouge (LA, USA). The parts or processes linked to structural robustness of the recuperator have been handled by Absolut System to ensure a good match with the structural analysis. So the external shell (shown on the Figure 7) and its final welding on the flanges are under Absolut System responsibility. The final assembly is planned for summer 2018, and recuperator performance tests are scheduled for September 2018.

Figure 7. Recuperator: recuperator shell and internal microtubes support

Figure 8. ISO8 clean room with test TVAC test benches and ISO5 assembling hoods at Absolut System

Environmental Test Campaign

The 40-80 K vibration-free cooler will be submitted to a complete performance and environmental test campaign including:

- Physical properties measurement
- Performance tests:
  - Thermal performance optimization
  - Thermal performance mapping
  - Pressure loss characterization
- Mechanical test:
  - Proof pressure test
  - Natural frequency & sine sweep
  - Sine vibration
  - Random vibration
- Thermal Vacuum Test:
  - Reference test
  - Thermal cycling
  - Exported vibration measurement

CONCLUSIONS AND OUTLOOK

An alternative to regenerative cryocoolers for future European Earth observation missions is under development in the frame of an ESA contract. The development of a reverse Brayton cooler started two years ago and focused during the first half of the program on the design trade-offs and breadboarding activities. All the components constituting this cryocooler have been produced after a long manufacturing time amplified by the small size of the rotor components.

The cooler should be able to provide 2-3 W of cooling power at 40 K with 180 W of electrical input power (with an initial requirement of 1 W at 40 K).

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