150K Pulse Tube Cooler for Micro-Satellites

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ABSTRACT

A recent need for "high temperature" cryogenic applications [150 K-200 K] has emerged in Europe with observation missions on-board micro-satellite platforms. Passive cooling can be used for some of these applications but sometimes, the operational constraints together with the mass and the size are too critical and require the use of mechanical cryocoolers.

To address this need for mechanical cryocoolers, the CNES has awarded Air Liquide a contract for the development of the overall system (cooler and electronics) with Absolut System SAS responsible for development of the cold finger and cryocooler optimization. The aim is to develop an evolution of the existing 80K MPTC (Miniature Pulse Tube Cooler) towards higher operating temperatures. This cooler has been re-designed and optimized to meet micro-satellite budgets (input power, mass, size) and lifetime.

The paper presents the optimization of the cooler design with discussions on the cooler configuration trade-offs. The expected performances of the cooler are finally described.

INTRODUCTION

Recent needs in "high temperature" cryogenics (150 K-200 K) have emerged in Europe with missions such as MICROCARB and HYPXIM. Such temperatures are achievable by passive solutions but require a fixed satellite attitude (with high baffling and/or mirror sighting change, and generally large size). For missions with high agility, active cooling solution must be considered and in some cases seems mandatory. Another major constraint of these missions is to embark on micro-satellites with limited size/weight/power resources. Typically a satellite of the Myriade-type has a nominal allowance of 60 W/60 kg for its payload. A version called "Myriade new generation", currently in Phase A, proposes to extend this potential to about 80 W/80 kg. On the other hand, the evolution of the standard chain is accompanied by an increase in the mission lifetime. Lifetimes of 5 to 7 years are proposed today, instead of 2 to 3 years before.

It is therefore essential to have a cooler operating in the range mentioned above while optimizing these resources, especially electrical consumption, a strongly penalizing parameter for the carriage of active cryogenics on platforms with limited resources.

Today, no cryocooler is completely adapted to the Myriad type micro-satellite exists:

- Tactical miniature rotary Stirling cryocoolers embarked on-board inter planetary probes present a limited lifetime,

- "Small Scale Cooler" Technological Research Program from the European Space Agency is oriented to 80 K applications,
- Miniature high frequency coolers developed in Europe are not mature enough for short term space qualification and flight.

Finally, the optimization of the existing 80 K MPTC (Miniature Pulse Tube Cooler) towards higher operating temperatures ranges [150-200K] has been found to be the best compromise.

SPECIFICATIONS

The required cooling capacity has been specified at 2.5 W in the 150-180 K temperature range. The objective of the present development is to achieve this cooling capacity with a 30% decrease in the compressor input power compared to the existing 80 K MPTC performance, as shown in the summary Table 1. Besides power constraints, a decrease in the overall cooler mass has also been targeted.

The main constraint driver in the thermodynamic optimization of the 150 K pulse tube cooler is to reuse the existing MPTC compressor. This active component is considered the main critical one in the overall MPTC Cooler Mechanical Assembly and benefits today of 5 years experience in lifetime test in continuous operation at 35 W input power without failure.

The cold finger mechanical assembly shall be redesigned according to standard mechanical launch environment and be compatible with the thermal environment for minimum 5 years lifetime.

PULSE TUBE CRYOCOOLER MODELING

High fidelity pulse tube cryocooler numerical model has been established by Absolut System SAS experts, originators of the 80 K Miniature Pulse Tube Cooler MPTC [1,2]. Absolut System SAS recently developed, manufactured and tested Small Scale Coolers producing 1.5 W@77 K for the RAPID project (Revolutionary Avalanche Photodiode Infrared Detector presented in a companion paper) using a modified LSF91xx type commercial compressor from Thales Cryogenics BV.

To start with the redesign, the model has been run with the MPTC design and the predictions have been confirmed by the test results provided by Air Liquide.

The conmparison of the model predictions to the test results is illustrated in the Figure 1. In this graph, the predicted electrical input power is obtained by considering a MPTC compressor efficiency of 74% above the calculated PV compression power. As shown, the model provides

Table 1. 150 K cooler technical specifications summary.

	Current MPTC	Objectives
Input power at design point 2.5 W@150 K / 273 K reject	*16 Wac*	11 Wac
Input power at design point 2.5 W@180 K / 273 K reject	*11 Wac*	8 Wac
Compressor and split pipe mass	*2.5 kg*	identical
Cold finger mass	*0.6 kg*	0.4 kg
Lifetime	5 years min. 10 years goal	
Sine loads (3 axis, 2 oct/min, 1 sweep-up)	[5-20.5 Hz] +/- 12 g peak	
	[20.5-60 Hz] / 20 g peak	
	[60-100 Hz] / 8 g peak	
Random loads (3 axis, 13.1 grms, 2.5 min/axis)	[20-100 Hz] / +3 dB/oct.	
	[100-300 Hz] / 0.3 g^2/Hz	
	[300-2000 Hz] / -5 dB/oct.	
Cold finger Eigen frequency	> 140 Hz	
Cold finger Non-Operating temperature limits	Min. -20 °C / Max. +50 °C	
Survival cold finger design temperature limits	Min. -50 °C / Max. +80 °C	

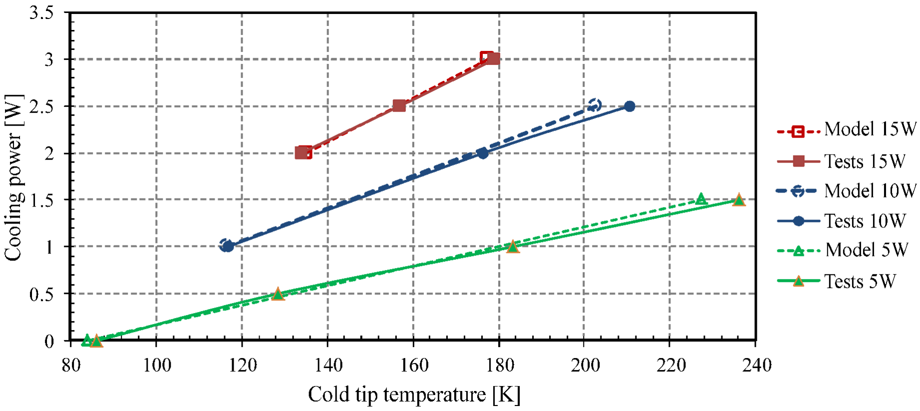

Figure 1. Cooling power versus cold tip temperature for various compressor electrical input power – Comparison of the model prediction to the MPTC test results – 273K rejection temperature.

high fidelity to the tests results. A discrepancy of less than 5% has been found on the overall temperature range (150-200K cold tip, 273-303K rejection) and power range (5-15W). Our model is then considered valid for the redesign.

The current MPTC cooler performance on the specified design points are:

- 2.5 W@150 K / 273 K: 16Wac, 11.9 W PV, 12.6 N piston force
- 2.5 W@180 K / 273 K: 11.6Wac, 8.6W PV, 12.6 N piston force

Three steps have been considered in the optimization of the 150 K pulse tube cold finger. In all cases, the numerical optimisation is made by minimisation of the compression PV work of the resonant system at minimum force.

- Optimization ①: the MPTC cold finger dimensions are not changed. The regenerator matrix is optimized, together with the phase shifter (inertance re-optimized for a constant buffer volume) and the split pipe. The MPTC compressor is strictly conserved with max 40 bars inlet pressure. This presents the shortest (and cheapest) way towards prototyping.
- Optimization ②: this is a step further than previous case. Here the cold finger dimensions (regenerator and pulsation tubes diameters) together with the aftercooler and cold heat exchanger dimensions are relaxed. A geometrical constraint is set to keep the pulsation tube and the regenerator at the same length for construction simplification in coaxial shape. The MPTC compressor is also strictly conserved with max 40 bars inlet pressure.
- Optimization ③: in this case, the cold finger dimensions are also relaxed as per the previous case. Due to a small gain reported by the numerical simulations, the buffer volume is replaced by the compressor's back volume. Furthermore, the compressor's piston diameter is decreased to allow for an overall system damping decrease.

The gains obtained over the current 80K MPTC performance for the three cases are reported in the Table 2 hereafter for the two design points considered. In order to reach the objective of 30% decrease, it is found mandatory to redesign the cold finger at 150 K and to modify the MPTC compressor in order to get resonance at lower force (thus lower damping and lower input power). This is even more important at 180 K where the MPTC compressor is not matching the cold finger.

Considering 8.8 W PV compression work and a force of 8.7 N for the fully optimized design (optimization ③ at 150 K), and keeping constant the MPTC motor design, the calculated compressor efficiency increases to 78%. At 180 K, the motor efficiency is maintained at 74%.

Table 2. Predicted gains on PV power and force and comparison to the MPTC current status.

2.5 W@150 K / 273 K reject	PV [W]	Gain/MPTC [%]	Force [N]	Gain/MPTC [%]
Current MPTC performance	*11.9*	-	*12.6*	-
Optimization ①	9.4	21	11.5	9
Optimization ②	8.7	26	11.2	11
Optimization ③	8.8	26	8.7	31
2.5 W@180 K / 273 K reject	PV [W]	Gain/MPTC [%]	Force [N]	Gain/MPTC [%]
Current MPTC performance	*8.9*	-	*12.6*	-
Optimization ①	7.0	21	11.5	2
Optimization ②	6.5	27	11.2	4
Optimization ③	6.6	25	8.6	25

The predicted optimized performance will then be:

- 2.5 W@150 K / 273 K: 11.3Wac, 8.8 W PV, 8.7 N piston force
- 2.5 W@180 K / 273 K: 8.9Wac, 6.6 W PV, 8.6 N piston force

PULSE TUBE COOLER DESIGN

Cold Finger Design

For the design, the split coaxial architecture using an inertance mode has been selected due to high relevant experience gained in previous 50-80 K developments and qualifications. Some CAD views of the cold finger design are attached in Figure 2.

The hot flange sub-assembly is composed of two parts: the warm-end flange made with aluminium 6061 temper T6 (solution heat treated) and the Ti-6Al-4V titanium alloy lower inertance flange. The aluminium used for the warm end is class 1 of ECSS-Q-ST-70-36 with high resistance to stress-corrosion cracking. Moreover, this materials benefit from previous experience on Electron Discharge Machining (EDM) technique for the manufacturing of the warm end heat exchanger gained on several programs. The warm end also integrates the heat rejection zone for heat sinking. The heat transfer will be performed via conductive mode to the evaporator of a miniature loop heat pipe.

The Ti-6Al-4V lower inertance flange integrates the pulsation tube connection, the inertance tube connection (interface with SS316L upper inertance flange) and the warm flow straightener.

The cold tip is made of Oxide Free High Conductivity copper (OFHC Cu) material optimized with respect to the thermal conductivity at 150 K and brazing process with the titanium alloy regenerator tube. This material exhibits the highest thermal conductivity at 150 K (about 418 W/m.K, RRR=100, density 8960 kg/m3). The cold heat exchanger is designed also with EDM slots in the copper material. In order to reduce the radiative heat exchange between the cold tip and the surroundings at ambient temperature, a gold coating will be used. This coating enhances the conductance of the thermal contact during the thermal heat transfer via conduction with the attached load (or thermal strap).

The regenerator and pulsation tubes are made of titanium alloy Ti-6Al-4V grade ELI (Extra Low Intersticials) material optimized with respect to the conductivity integral (lowest possible) and the mechanical resistance (as high as possible). The ELI grade provides the characteristics of low crack propagation and is recommended for low Stress Corrosion Cracking. Titanium alloy Ti-6Al-4V grade ELI also benefits from previous experience on all pulse tube cold fingers manufactured. The regenerator tube is 17 mm ØOD and 0.15 mm thick, which is a compromise between mechanical characteristics, manufacturing and conduction heat losses. The regenerator flange is machined together with the regenerator tube in the same part and is bolted and sealed to the warm end flange using aluminum plated Inconel metallic spring energized rings. At the cold side, the Ti-6Al-4V regenerator tube is brazed to the copper cold heat exchanger. A high vacuum silver brazing process is used. This process has been matured and qualified for all other pulse tube cold fingers manufactured.

The pulsation tube wall is reduced to 0.10mm thick because it is not submitted to a pressure difference (lower wall thickness being difficult to manufacture). The pulsation tube and the cold heat exchanger are associated with a flow straightener to stabilize the mass flow at the entrance of the tube.

The regenerator matrix is made of annular 316L stainless steel gauzes with apparent porosity of 75% which results from previous numerical optimization.

Finally, as shown in Figure 2, the design makes use of a Ti-6Al-4V launch support to suppress any lateral motion of the cold tip for maximal robustness during launch phase. It suppresses the shocks that can occur in design which makes use of a snubber (which allows small lateral motions due to non-contacting parts). During the cooling down phase, due to thermal contraction of the cold tip, there is no more (or much reduced) contact and the thermal performance is optimal. The launch support is composed of a conical tube which is bolted to the regenerator flange on the warm end side. 4 Peek^{TM} 450G rods, screwed into the launch support tube, are used to lock the cold tip of the cold finger under warm conditions during launch mechanical loading. Each peek rod is clamped at each ends into dedicated groves: one in the screw located on the launch support tube, the other in a Ti-6Al-4V ring adjusted to the cold finger with a reduced gap. This launch support design relies on reproducible dimensioning adjustment of the peek rods length and is locked with controlled torque on the screw (positive locking can be added if required). This launch support design has been qualified to high random spectra density (over 0.5 g^2/Hz) providing efficient damping.

Compressor and Cooler Mechanical Assembly Design

As mentioned previously, the MPTC compressor shall be conserved as far as possible in order to take benefit of the experience gained from the on-going lifetime test. The motorization and the flexure assemblies are then conserved. The only modifications gathered in the design are linked to a swept volume decrease (reduction of the piston's diameter) and connection of the inertance to the compressor back volume.

The decrease of the piston diameter allows for decreasing the overall cooler system damping by increasing the piston stroke (compared to the stroke developed by the current MPTC compressor design driving the 150 K pulse tube cold finger).

The connection to the back volume of the compressor allows for suppressing the static buffer volume nominally attached to the cold finger warm end. The connection is made by the modification of the end plate of one half compressor. An access port is created and will be sealed by a spring energized ring and a bolted flange machined in the inertance assembly (see Fig. 3). The inertance assembly comprises internally a big part of the total inertance length. A dedicated flange is also machined into the inertance assembly to connect the inertance pipe from the inertance

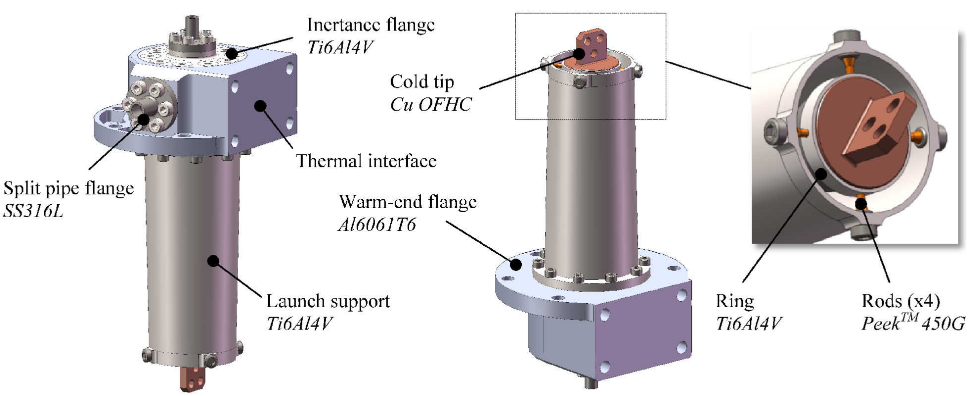

Figure 2. CAD views of the 150 K Pulse Tube cold finger.

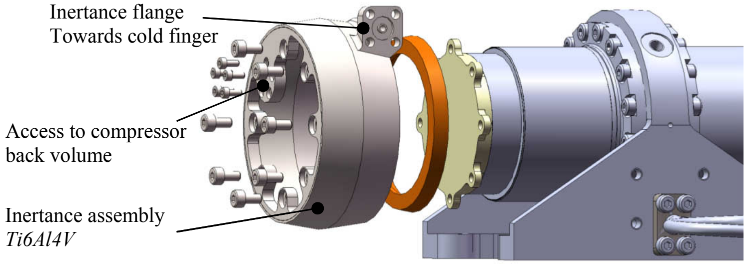

Figure 3. Mounting of the inertance assembly to one MPTC compressor half side.

assembly to the cold finger inertance flange as shown in the assembly attached in the Figure 4. The total mass transferred to the compressor half for the inertance assembly is 340 g.

MECHANICAL ANALYSIS

Regarding the geometry of the different components of the 150 K cold finger, a 3D finite element model has been built. HEXA8 finite elements have been used as they give more accurate results. 2D plate elements have been used for thinner components and 3D Tet10 (quadratic) finite elements have been used elsewhere. It is assumed that all the materials are isotropic and exhibit a linear behavior. Total mass of the finite element model is comprised of 418 g for the cold finger and 50g additional mass attached to the cold tip to take into account the contribution of a thermal braid.

Femap v10.3 (Siemens) has been used to mesh the finite element model composed of 117,346 nodes and 85,462 finite elements. The cold finger meshed is shown in the Figure 5 hereafter. NX Nastran v8.0 (Siemens) has been used to solve the different problems and Femap v10.3 has been used to pre-process the finite element models.

The FEM model has been loaded with 40 bars internal pressure. For the dynamic mechanical load cases, the signal in each direction is input to the cold finger at the screw locations through the use of a rigid element (RBE2 NX Nastran) as shown in Figure 5. The damping assumption is Q = 50.

A normal modes analysis has been conducted on the whole structure showing a first natural bending mode of the cold finger at 865 Hz which is far above the specification of 140 Hz. An illustration of one of the cold finger shape mode at 900 Hz is attached in the Figure 5 as well.

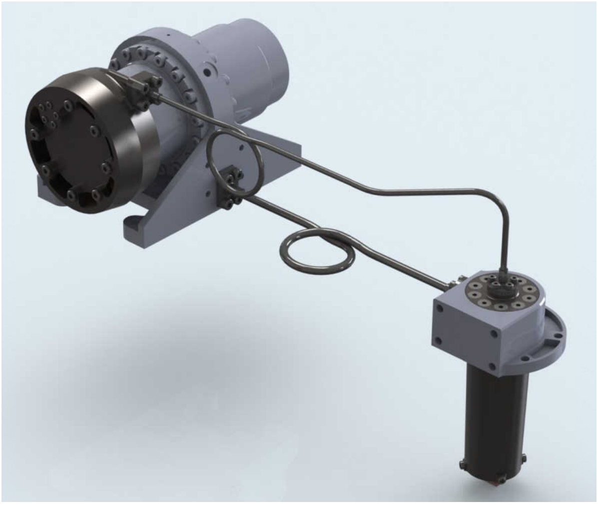

Figure 4. Complete 150K pulse tube cooler mechanical assembly.

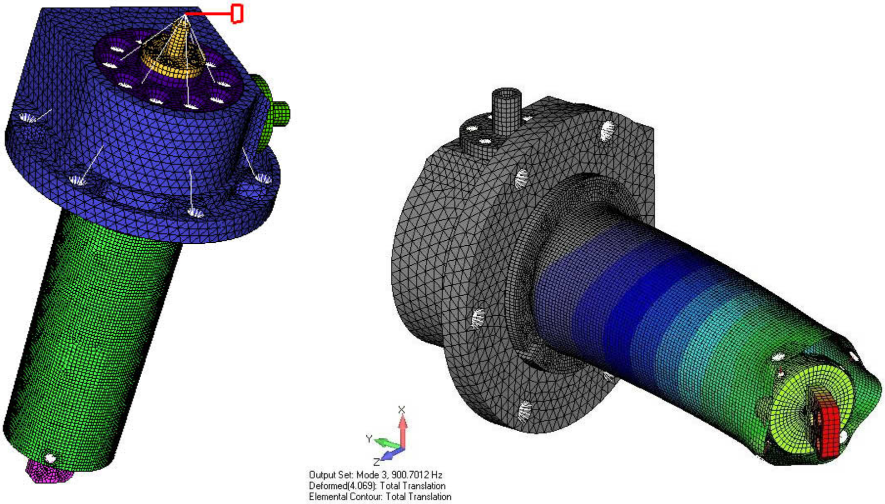

Figure 5. Overall FEM model of the 150 K cold finger and cold finger mode illustration

For both the sine and random dynamical load cases (refer to profiles attached in the Figure 6), the maximal Von Mises stress has been found at 261 MPa in the Titanium alloy regenerator pipe element which corresponds to the pressure load stress. The 5-100Hz sine loading case does not excite the structure due to its very high natural modes. Considering a Safety Factor of 1.5 on ultimate stress (896 MPa for Ti6Al4V), a comfortable Margin of Safety (MOS) of 1.3 is thus exhibited by the system under sine load. In the case of the [20-2000Hz] random load case, the overall structure is excited, and thus the launch support is stressed. For this load case, the minimum MOS of 0.07 is found for the Peek™ rods. It shall be noticed that this MOS is obtained for pessimistic Qfactor of 50 (experimental figures of 30 are generally measured) and huge thermal braid mass of 100 g (shared between the cold tip and the focal plane).

CONCLUSIONS

The design of a 150 K pulse tube cooler has been optimized with the constraint of using an existing 80K compressor. The pulse tube cold finger configuration retained is an inertance type coaxial shaped. The compressor back volume is used as the buffer volume. The pulse tube cold finger design has been validated against standard European launch load profiles.

The simulated specific power at 150 K is as low as 4.6 W/W (excluding cooler drive electronics efficiency) as shown in Figure 7 for 2.5 W cooling power and 273 K rejection

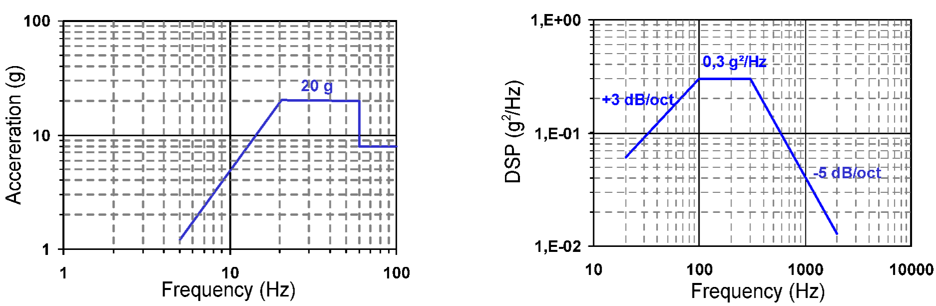

Figure 6. Sine and random mechanical load profiles.

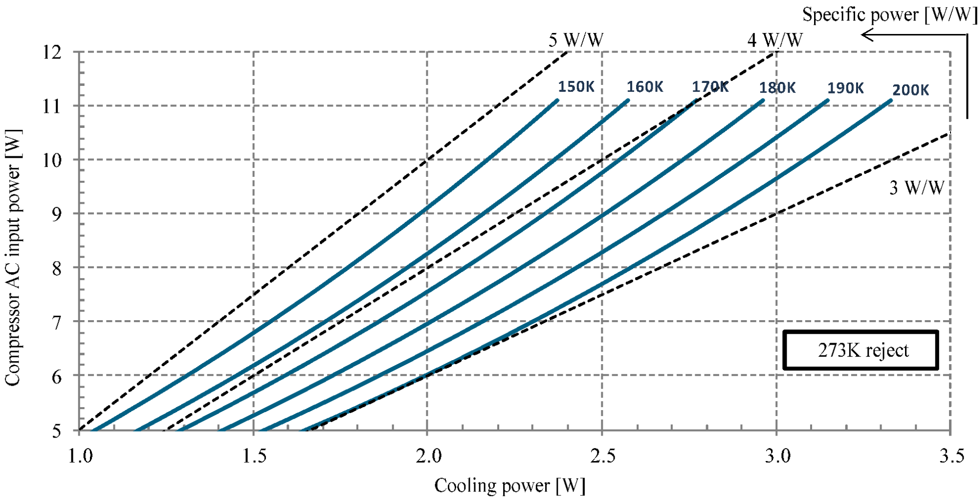

Figure 7. Compressor AC input power versus cooling power - Ron Ross type diagram of the optimized 150 K pulse tube cooler – 273 K rejection temperature**.**

temperature which represents a gain of about 29% compared to the current MPTC performance (cooler designed for 80 K operation).

The mass of the optimized cold finger is 420 g with overall dimensions of Ø75 mm x 134 mm total length. The complete cooler mechanical assembly weighs 3 kg.

ACKNOWLEDGMENT

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REFERENCES

1. Trollier, T., Tanchon, J., Buquet, J. and Ravex, A. “Status of Air Liquide Space Pulse Tube Cryocoolers,” *Cryocoolers 15*, ICC Press, Boulder, CO (2009), pp. 115-123.

2. Tanchon, J., Trollier, T., Triqueneaux, S. and Ravex, A. “20–50 K and 40–80 K pulse tube coolers: Two candidates for a low temperature cooling chain,” *Cryogenics,* Vol. 50, Issue 1 (January 2010), pp. 55-60.