ABSTRACT

A Large heat lift 40 to 80 K Pulse Tube Cooler (LPTC) is currently under development in partnership between AL/DTA, CEA/SBT and THALES Cryogenics. The Engineering Model (EM) foreseen is aiming to provide 2 W of cooling capacity at 50 K for 10°C rejection temperature and for 125 watts input power to the compressor’s motors.

Cold finger development models (DM) have been manufactured with an in-line architecture and connected to an off-the-shelf compressor from Thales Cryogenics. The DM results are discussed in the present paper while operating in inerterance mode for various PV work and are used to complete the design of the future EM.

In parallel, the compressor motor has been optimized and prototyped. The results have been confronted to the simulated performances and are also discussed herein.

The various trade-offs, performed both on the cold finger and compressor side, during the development phase are presented as well.

This work is funded by the European Space Agency (ESA/ESTEC Contract N°18433/04/NL/AR) in the frame of future Earth Observation instruments development. The LPTC Engineering Model will be delivered to ESA/ESTEC in September 2006.

INTRODUCTION

The previous Miniature 50-80 K Pulse Tube Cooler (MPTC) successfully demonstrated in 2003 the capability to provide high performance in the 80 K range: 1W@80K for 288 K rejection, 35 W input power and 2.8 kg mass [1]. Beside this development, high cooling capacities in the 50 K temperature range are needed to cool down thermal infrared detectors for future earth observation missions. The overall objective of the work is to design, manufacture and test at pre-qualification levels a Large 40-80 K Pulse Tube Cooler (LPTC). The resultant LPTC shall require no or only minor delta qualification for direct use in future earth observation missions. It shall be commercially competitive in performance, mass and cost within the current space cryocooler market. The technical specifications from ESA/ESTEC are summarized in the following table 1.
**COLD FINGER DEVELOPMENT PHASE**

A 10 months duration development phase has been performed. During this phase, pulse tube cold fingers Development Models (DM) have been designed and manufactured, based on a unique regenerator dimension (coming from previous heritage) and simulated optimal pulsation tube dimensions. In order to modify easily the geometry for optimization of the DMs, an in-line configuration has been traditionally used for the cold finger. Common materials such as stainless steel and OFHC copper have been implemented for the tubes and for the heat exchangers as shown in the FIGURE 1. Water-cooling is provided at both the hot ends of the regenerator and the tube. The regenerator was filled with stainless steel gauzes for which both homogeneous stacking and graded stacking have been experienced.

All the DMs have been operated in an inertance mode and in vertical orientation with cold finger pointing up to suppress natural convection. The standard tests conditions were as follows: 107 W PV work chosen for max 140 W input power with 75% targeted compressor efficiency (PV work meaning here Winput – Rif²), 13°C rejection temperature (water cooling), 30 bars filling pressure, 230 cc buffer volume, 5 layers MLI wrapping and 300 mm specified transfer line.

---

**Table 1: LPTC technical specifications summary table.**

<table>
<thead>
<tr>
<th>Specification</th>
<th>Specification Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cooling power / rejection temperature</td>
<td>2W @ 50 K / 283 K</td>
</tr>
<tr>
<td></td>
<td>3W @ 60 K / 283 K</td>
</tr>
<tr>
<td>Cooler efficiency @ 50 K</td>
<td>60 to 70 W/W</td>
</tr>
<tr>
<td>OFF state parasitic heat losses (50K / 300 K)</td>
<td>&lt; 1 W</td>
</tr>
<tr>
<td>Mechanical cooler mass</td>
<td>&lt; 5 kg</td>
</tr>
<tr>
<td>Transfer line</td>
<td>30 cm</td>
</tr>
<tr>
<td>Design lifetime</td>
<td>3 years ground + 7 years in-orbit</td>
</tr>
<tr>
<td>Static loads onto the cold finger</td>
<td>500 N / 5 N / 1000 N.mm</td>
</tr>
<tr>
<td>Sinus loads, 3 axis, 2 oct/min</td>
<td>[5-20.5 Hz] +/- 12 mm peak</td>
</tr>
<tr>
<td></td>
<td>[20.5-60 Hz] +/- 20 g peak</td>
</tr>
<tr>
<td></td>
<td>[60-100 Hz] +/- 8 g peak</td>
</tr>
<tr>
<td>Random loads, 3 axis, 13.1 grms composite</td>
<td>[20-100 Hz] / +/- 3 dB/oct</td>
</tr>
<tr>
<td></td>
<td>[100-300 Hz] / 0.3 g²/Hz</td>
</tr>
<tr>
<td></td>
<td>[300-2000 Hz] / -5 dB/oct</td>
</tr>
</tbody>
</table>

---

**Figure 1: LPTC cold finger Development Model (CEA/SBT).**
A total of 8 DM configurations have been tested. The optimized performances for each configuration are reported in the FIGURE 2. The first configurations (from config 1 to 4) were implementing standard homogeneous stainless steel regenerator matrix and were used to perform the optimization of the pulsation tube dimensions. Once the pulsation tube frozen, various gradings of the regenerator were experienced. For the configurations 5 and 6, the cold part of the regenerator has been filled with higher surface density end lower porosity meshes than the standard previous one. It is noticed that such regenerator allows for the lowest no-load temperature due to lower regenerator losses (improved surface heat transfer). As expected, the cooling slope is slightly decreased (compared to config 4) due to larger pressure drops. Finally, the configurations 5 and 6 have been modified with lower surface density and higher porosity meshes in the warm part of the regenerator, resulting in the configurations 7 and 8. With such regenerators, the no-load temperature and the cooling slope were slightly increased, which could be very interesting for applications requiring to be cooled at higher cryogenic temperatures.

The best performances achieved during the prototyping campaign are summarized in the TABLE 2 for the configurations 6, 7 and 8. For the complete cooler trade-off, the configuration 6 is the one finally retained for the design of the LPTC Engineering Model.

Figure 2: LPTC cold finger Development Model performance results.
The configuration 6 provides the lowest OFF-state parasitic heat losses. The operating frequency of 55 Hz is also frozen for the final compressor due to the smallest required swept volume (about 10% less for 55 Hz than for 50 Hz driving frequency), which will result in a higher motor efficiency and lower compressor mass. The small decrease in the cold finger performance (1930 mW @ 55 Hz compared to 1970 mW @ 50 Hz) will be recovered by the increased compressor efficiency (2% gain from 50 Hz to 55 Hz with the foreseen motor). Finally, the configuration 6 will provide more cooling capacity in the 40K range (>660 mW) due to the lowest no-load temperature and presents the simplest regenerator (graded regenerator in the cold part only).

During the cold finger development phase, the influence of some operating and/or design parameters has also been experienced. A gain of about 100 mW cooling capacity at 50 K has been measured with an increase of the filling pressure from 30 bars to 35 bars. However, 35 bars will impact too much the mechanical design of the compressor. The cooling capacity has been measured to be linear with the PV work in the experimental range of 87-127 W. At 50 K cold temperature, the sensitivity to the rejection temperature has been measured to be 30 mW/K (which corresponds to about 1K lost at the cold end for an increase of 4 K of the rejection temperature). Finally, the decrease of the transfer line length from 300 mm to 200 mm provided a small gain of about 0.5 K at 50 K (which is 60 mW).

### COMPRESSOR DEVELOPMENT PHASE

The compressor design to be used for the LPTC EM will consist of two pistons that reciprocate head-to-head into the same compression chamber (dual opposed pistons) to optimise the compactness of the system. Each piston will be supported by a bearing that consists of two flexure spring packs (each one made of two flexures), located at the front and at the rear of the piston.

In parallel to the cold finger development, the compressor’s motor has been simulated and optimized. The main drivers for the selection of the motor concept were tailored requirements such as a 75% minimum efficiency for a maximum compressor mass of 3.9 kg. The required generated force is 75 N and the maximum system damping is 52 kg/s for 55 Hz operating frequency (output from the cold finger test campaign previously reported). 6 motor concepts have been studied: axially magnetized moving magnet motor, axially magnetized moving magnet motor with 2 flux paths, 1 radially magnetized moving magnet motor, 4 radially magnetized moving magnets motor, moving iron motor and moving coil motor. From the simulations, the 4 radially magnetized moving magnets motor was the most promising motor concept. Beside the high efficiency and low current density, this concept offers the advantages of high reliability (no flying current leads, no hermetic glass feed-throughs, coils outside the helium working gas) and high design flexibility mainly due to the contribution of the magnets to the axial stiffness.
Due to the lack of heritage on this new motor concept, a test campaign was decided in order to validate the simulated axial force generated and the simulated motor losses, especially the hysteretic and eddy current losses. The single motor used during the breadboarding phase is shown in the FIGURE 3, equipped with the LVDT for the stroke measurement. The motor breadboard is completely dismountable (except the magnets) in order to experiment various stator materials: AISI 430F, Armco Ingot Iron RFe100, iron powders and slotted stators. The motor has been tested under atmospheric circumstances. There is no gas spring effect, because there is no piston and all volumes are connected by means of additional holes. The maximum possible piston amplitude in the test set-up is 5mm. The simulations are performed with PC Opera®.

As shown in the FIGURE 4, the simulations of the generated axial force are very accurate. At high current densities the experiments shown less degradation in generated force than the simulations. This is probably caused by the fact that the worst-case B-H curve of the stator material has been used in the FEM simulations. For the losses, the simulations are not fully representative of the experimental behavior and corrective terms of 1.35 and 2.2 shall be applied respectively for the eddy current losses and the hysteretic losses.
LPTC ENGINEERING MODEL

The coaxial configuration is retained for the EM cold finger. This configuration has been internally proven to be more efficient than U-shape configuration. In-line configuration is foreseen to be more efficient due to lower dead volume and pressure drop at the cold end. Nevertheless the efficiency is affected by the difficulty to remove the heat from the pulsation tube hot end, which is located into the cryostat. Even it has been demonstrated that inertance gas flow plays an active role in the heat transport, a huge amount of heat needs to be removed with the present LPTC cooler and this could consequently limit the efficiency. With the heat exchanger design currently used, the coaxial version allows for low pressure drops at the warm end and better gas distribution at the regenerator inlet. Furthermore, the coaxial configuration provides the advantages of low complexity tightness, easier integration and lower mass compared to other configurations.

A picture of the EM cold finger preliminary design is attached in the FIGURE 5. For the split configuration, radial injection line will be used in order to simplify the design. It has been shown that radial injection line does not impact the cryogenic performance compared to axial injection line. Integrated buffer volume (to the warm end) will be used as well for mass optimisation. A splitted buffer volume design would have added two critical extra tightness, mechanical support of the inertance and an increase of the compressor heat dissipation path (the buffer being attached to the compressor in this case in order not to add extra heat dissipation area). The warm end flange is made of aluminium alloy for thermal heat transfer and mechanical stiffness optimization. The heat exchangers are manufactured by Electron Discharge Machining directly in the flange material. The inertance tube is wound inside the buffer volume. Both the tube of the regenerator and the pulsation tube are made of thin walled titanium alloy Ti-6Al-4V in order to reduce the parasitic heat leaks. The design makes use of bolted flanges and metallic C-rings to seal the cold finger and the buffer to the warm end. At the cold side, a high vacuum brazing process is used for the assembly of the titanium tube onto the pure copper cold block of the cold finger.

Figure 5: LPTC cold finger Engineering Model design.
In order to limit the stress during the loading phase, a hot clamping of the cold finger will be implemented on the EM as a launch lock device. An example is shown on the FIGURE 6. Some peek rods attached to the cold tip are adjusted on a very stiff external tube. In the warm condition of the launch, the modal analysis performed shown a flexion mode of 620 Hz. In the cold condition, the thermal contraction of the materials (about 0.14 mm) will allow for very low added thermal load on the cold tip during nominal operation. Notice that this solution allows for optimization of the cold finger tube with very small wall thickness designed for the pressure loads.

For the moving-magnet EM compressor motor, iron powders have been retained for the stators due to very low electrical conductivity and thus very low eddy current losses. Several motor parameters have been traded-off with respect to the overall efficiency, mass and reliability. The magnet length has been optimized to reach constant axial magnet stiffness along the stroke. The stator shape has been modified till saturation is acceptable. The coil length has been limited due to huge mass impact on the final compressor. Finally, the trade-off concluded, for round copper coil wire, on a calculated efficiency of 80.7 % at 30°C coil temperature. An improvement of 0.5 % can be gained by using rectangular copper coil wire, and an extra 0.6 % by using silver material. Notice that higher efficiency can also be reached by increasing the magnet volume but this will increase drastically the moving mass, which has been limited to 300 g (critical for stress applied on the flexure bearings and generated vibrations), and the outer diameter of the compressor and thus the overall mass budget.
density. The two compressor halves are mounted on a dedicated nickel-plated aluminium alloy centre part that contains all the mechanical, thermal and electrical interfaces of the compressor and the two cylinders. Bolted flanges are directly machined in the titanium alloy block of the coil holder. The gas containment is achieved by means of metallic seals as well. The total mass of the EM compressor is estimated to be 3700 g. The outer dimensions are 120 mm total height and 196 mm total length. The EM compressor motor design is expected to meet MIL-STD-461E for the magnetic emissions.

CONCLUSIONS

A Large heat lift 40-80K Pulse Tube Cooler (LPTC) has been designed based on experimental results obtained on cold finger and compressor breadboards. The Preliminary Design Review hold successfully in mid July 2005. The expected mass of the LPTC is 5.5 kg including development margins. Thanks to a calculated efficiency well above the targeted 75% the specified cooling capacity of 2W@50K is expected to be provided with 125 Wrms input power to the compressor motors at 55 Hz driving frequency and 10°C rejection temperature. The implementation of high performance materials, together with the mechanical optimization of the cold finger highly contributes to this performance. It is also expected to decrease the off-state cold finger parasitic heat losses from 1030 mW down to 900 mW between 50 K and 300 K. The Critical Design Review is scheduled in mid November 2005. The Engineering Model of the LPTC will be delivered in September 2006 after extensive thermal and mechanical tests campaign.

ACKNOWLEDGEMENTS

This work is supported by the European Space Agency (Contract 18433/04/NL/AR).

1.1.1 REFERENCES