SERVICE DES BASSES TEMPERATURES

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FIRST - SORPTION COOLERS

Discussion on 4 liters versus 6 liters STP unit & Ultimate temperature improvement

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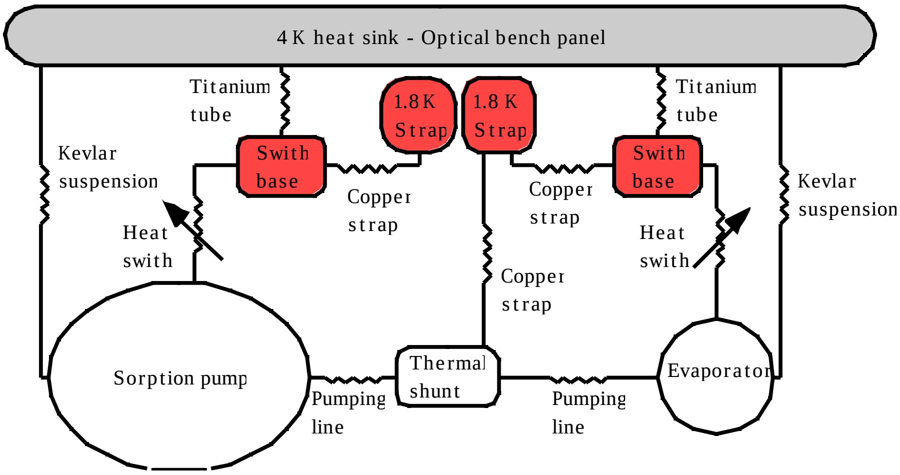
1 Introduction

The purpose of this short technical note is to discuss the possibility of increasing the size of the cooler from a 4 liters STP to a 6 liters STP unit for both the SPIRE and PACS instruments, and additionally to discuss on a possible improvement of the ultimate temperature.

The original sizing of the cooler for the SPIRE instrument (BOL at that time) was performed back in 1997 on the basis of a 10 µW net heat lift for about 46 hours, a recycling period of less than 2 hours, and an overall average power dissipation on the satellite cryostat of a few mW. At that time there was no specific thermal architecture for the cooler.
Later on the architecture was modified such that the cooler is now mounted off a 4 K structure with a couple of thermal straps to the superfluid helium cryostat (required for its operation). This architecture is represented on the figure below.



A detailed design has been performed and in the framework of an ESA TRP contract a prototype has been manufactured, assembled and tested. This prototype has successfully undergone a qualification program (mechanical, thermal, bake out and vibration tests). The cooler is operating pretty much to specifications.

One important aspect is that this system must be seen as an energy device, i.e. it will provide a given amount of Joules at 300 mK. The hold time then depends on the applied load (+ the parasitic), and obviously the average power dissipated is also a function of this load. Consequently if the applied load is increased or the parasitic is larger than expected, the cooler will still work but will need to be recycled more often.

2 Present status – 4 l STP

Following a recent meeting (Thermal summit - QMW 25 & 26 Sept. 2000) it turns out that to meet the SPIRE specifications with a 4 liters unit would be critical. A recent modeling shows that the load from the detector system may exceed 10 µW, and the mechanical structure temperature may raise from 4 K to above 5 K.
In addition during the mission the idea is to download the data and send whatever operating commands on a 48 hours cycle basis (possibly 24 h). Thus a cooler providing a 40 hours hold time because of an excess of applied load would not be adapted. On the contrary a hold time

of a finite number of days and possibly more than 2, would fit. This information is new for us and questions again the 4 liters STP design, which was originally decided on the basis of the 48 hours cycle.

It then clearly appears that a 6 liters unit would provide some margins. In fact as shown in the following discussion if it finally turns out the margins are not used a larger cooler will still be more efficient.

2.1 Heat lift, hold time, energy and margins

Under the nominal operating conditions, i.e. a heat sink temperature of 1.74 K and a mechanical structure at 4 K, the present design (4 l STP) provides the following (theoretical predictions) :

Condensation efficiency	94% (sorption pump heated to 45 K)
Net heat lift	10 µW
Parasitic load	line + switch : 10.5 µW, support : 1.4 µW
Ultimate temperature	287 mK
Hold time	1 day 21 hours 51 minutes (!)
Total overall timing	1 day 23 hours 22 minutes
Total energy dumped on superfluid bath	526 Joules
Average power over entire cycle	3 mW

Obviously there are no margins, any increase of the heat sink or structure temperature, or of the applied load will substantially affect the performance. This is illustrated in the following table where we have reported the influence on the performance of a variation of 10 and 20% of the heat sink temperature and of a variation from 4 to 4.5, 5 and 5.5 K of the structure temperature. In each cell are given the hold time assuming the load from the detector system is 10 µW, the average load and finally in italic the net heat lift if the hold time was to be kept at 46 hours.

		Heat sink temperature (K)		
		Nominal / 1.74 K	+10% / 1.91 K	+20% (2.1 K)
Mounting structure temperature (K)	Nominal / 4 K	46 hours 3 mW *10 µW*	38.5 hours 3.6 mW *6 µW*	31.5 hours 4.4 mW *0.9 µW*
	4.5 K	45 hours 3.1 mW *9.5 µW*	38 hours 3.7 mW *5.5 µW*	31 hours 4.5 mW *0.4 µW*
	5 K	44 hours 3.2 mW *8.9 µW*	37 hours 3.7 mW *4.9 µW*	30.5 hours 4.5 mW XX
	5.5 K	42.5 hours 3.3 mW *8.3 µW*	36 hours 3.8 mW *4.3 µW*	30 hours 4.6 mW XX

Note : the additional contribution from the heat switches base (link between structure and 1.74 K straps is not taken into account here – this contribution is not part of the cooler budget)

3 Six liters STP unit

From the previous discussion it is clear a 4 liter STP unit does not provide any margin. We propose then a slight increase of the cooler to a 6 liters unit. It is important to note that for the present design (4l) the parasitic loads account for roughly 50%. In a larger unit the parasitic remains the same (the design of the switches, pumping line and Kevlar structure will not change) and consequently as a first approximation an increase in size of 50% yields to a 100% increase in net heat lift. In addition if the original specifications are met (10 µW from the detector system, 1.74 K heat sink and 4K structure) then a 6 liters STP unit will in fact also provide a gain in efficiency (see below).

3.1 Heat lift, hold time, energy and margins

We have reported below the performance for these two figures: additional heat lift but 46 hours hold time or 10 µW nominal heat lift and corresponding hold time.

46 hours hold time

Condensation efficiency	95% (sorption pump heated to 45 K)
Net heat lift	21.4 µW
Parasitic load	line + switch : 10.5 µW, support : 1.4 µW
Ultimate temperature	304 mK
Hold time	1 day 22 hours
Total overall timing	2 days
Total energy dumped on superfluid bath	711 Joules
Average power over entire cycle	4.1 mW

10 µW net heat lift

Condensation efficiency	95% (sorption pump heated to 45 K)
Net heat lift	10 µW
Parasitic load	line + switch : 10.5 µW, support : 1.4 µW
Ultimate temperature	287 mK
Hold time	2 day 21 hours 24 minutes
Total overall timing	2 days 23 hours 26 minutes
Total energy dumped on superfluid bath	726 Joules
Average power over entire cycle	2.8 mW

3.2 Design impact

We have already perform a preliminary analysis of the impact of a 6 liters unit on the present design. Basically the components that need to be slightly redesign are the evaporator and the sorption pump; however an increase in volume of 50% only requires to change the diameters by 14%.

Evaporator : for construction reason the internal volume of the evaporator is already oversized and is such it is compatible "as is" with a 6 liters unit.

Sorption pump : we do not expect any problem here. The attachment points for the pulleys can remain pretty much at the same geometrical location (angle to respect) while the pump volume is increased. In fact it will ease the integration of the activated charcoal.

Support structure : unchanged for the evaporator. Slight modification for the pump.

Mechanical aspects (proof pressure, resonant frequencies, …) : this modification does not lead to any significant change. Regarding internal pressure the pump exhibits a MOS of 6.8 (burst pressure at 68 MPa – MOP at 10 MPa), obviously we have some margin.
The resonant frequencies were found to be around 600 Hz, the slight increase in mass will not affect significantly these values (f_r % $m^{0.5}$).

Overall size : No problem expected.

3.3 Compliance with ESA requirements

As far as we know the specifications on the energy and average power dissipated have been defined in the SOW of the ESA TRP as 5 mW average (or 864 Joules) (Cryogenic Sorption Cooler - SOW - ref YCT/2487.BC Issue 1 rev. 2). In addition at that time the cooler was mounted off a 1.8 K cryostat. Later on however the thermal architecture was changed such that the cooler is now thermally operated from the 1.8 K cryostat (straps), but is mechanically mounted off a 4K structure. This in particular means that the heat switches are connected to both temperature (1.8 and 4 K). Of course the 1.8-4 K link is designed to be mechanically strong and thermally weak but yet it contributes to some additional load on the cryostat. This extra load is not part of the original specification.

It seems reasonable to separate out what is related to the cooler efficiency and what is related to the fact that we are mounted to a 4 K structure. If the temperature of the structure would raise to above 5 K the impact on the [1.8 – T] switch link would be significant (the integrated thermal conductivity of titanium varies like $T^{2.1}$]. In the present state (4 K structure) the specification on the additional contribution is that it shall not be in excess of 0.4 mW (0.2 mW per switch).

A six liters unit remains within specifications even in the case the net heat lift reaches values around 20 µW.

4 Ultimate temperature

During the thermal summit meeting the operating temperature of the cooler was discussed. The detector system will be connected to the cooler via some thermal straps of finite conductance. Consequently the temperature gradient along these straps will set the operating temperature of the detectors. Then it would be desirable to lower as much as possible the operating temperature of the cooler.

In a well designed cooler the ultimate temperature is only limited by the pumping line between the sorption pump and the evaporator, i.e. the length, diameter and wall thickness of the tubes, and by the Kapitza resistance between the liquid and the external cold interface. Generally this temperature will lie between 250 and 300 mK. Lower temperature are extremely difficult to obtain since in this range to lower the temperature by 10% requires to lower the pressure P above the bath by about a factor of 4. If we assume that the applied thermal load on the evaporator (detectors, etc...) is negligible in comparison with the parasitic

load (heat conduction along pumping line) one can easily show that as a first approximation the pressure P is proportional to the charging pressure and inversely proportional to the tube diameter Ø. If the applied load is taken into account the diameter must be further increased.
But again neglecting the applied load and within a limited range for the tube length L (see further), the hold time can be shown to be proportional to the amount of gas, the tube length L and inversely proportional to the square of the pump tube diameter by the charging pressure. It could be set to any value by matching the above parameters. Yet there is obviously a balance to be found with the ultimate temperature.
The alternative is then to compensate these conflicting aspects by adjusting the tube length and amount of gas. However although not as simple as mention before the average power dissipated on the cold plate over one cycle (somewhat proportional to $HP.Ø^2/L$) is required in most case to be as small as possible.
Finally the condensation efficiency and consequently the hold time can be substantially degraded by an increase of the internal volume of the pumping line ($\approx L.Ø^2$). In fact for a given cooler without any applied load detailed calculations show that the ultimate temperature indeed does not depend on L, but that for large length the hold time is not anymore proportional to L and eventually decreases as L increases.
The bottom line is that the ultimate temperature, hold time and average power dissipated cannot be simultaneously optimized and a compromise has to be found.

With the present design to gain on this temperature by changing the dimensions of the pumping line would require some substantial increase in its diameter. This in turn will lead to a larger conductive load and a significant impact on the hold time and average load on the superfluid cryostat.
However from the experimental data we have acquired during the ESA TRP contract it seems clear either the Kapitza resistance model we use underestimate the ΔT, or the thermal paths within the evaporator are not efficient enough. The ultimate temperature of the cooler was found to be at 269 mK, when expected below 260 mK (with 10 µW applied load this ΔT raises to 20 mK – 270 mK expected, 290 mK measured).
We believe we can improve the internal design of the evaporator to try to gain on this temperature. It should be noted that the liquid behavior inside the porous material in terms of thermal conductance is not very well understood (also reported by many authors).
So for now we propose to keep the pumping line as is, and to improve the evaporator internal design.

5 Conclusion

Clearly a 6 liters unit will benefit both instruments, SPIRE and PACS, and will provide a substantial margin in the net heat lift available and/or hold time. Moreover calculations indicate that if all systems operate to specification this larger cooler will still beat the actual 4 liters cooler, originally sized on the basis of a 48 hours cycle.
The hold time under the nominal conditions is close to 3 days and thus compatible with the 24 hours cycle (download/upload of commands and data).
In any case a 6 liters unit is still compliant with the ESA specification of 5 mW average.

This decision has to be made in the light of the value of the overall power dissipation on the FIRST cryostat. We believe the coolers account for a fraction of this power. Slightly larger cooler units should have very little impact on this overall power yet a very substantial impact on the successful operation of both instruments, SPIRE and PACS.