Long-life vibration-free 4.5 K sorption cooler for space applications

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A breadboard 4.5 K helium sorption cooler for use in vibration-sensitive space missions was developed and successfully tested. This type of cooler has no moving parts and is, therefore, essentially vibration-free. The absence of moving parts also simplifies scaling down of the cooler to small sizes, and it contributes to achieving a very long lifetime. In addition, the cooler operates with limited dc’s so that hardly any electromagnetic interference is generated. This cooler is a favorite option for future missions such as ESA’s Darwin mission, a space interferometer in which the sensitive optics and detectors can hardly accept any vibration. The system design consists of a hydrogen stage cooling from 80 to 14.5 K and a helium stage establishing 5 mW at 4.5 K. Both stages use microporous activated carbon as the adsorption material. The two cooler stages need about 3.5 W of total input power and are heat sunk at two passive radiators at temperatures of about 50 and 80 K—radiators which are constructed at the cold side of the spacecraft. We developed, built, and tested a demonstrator of the helium cooler. This demonstrator has four sorption compressor cells in two compressor stages. Test experiments on this cooler showed that it performs within all specifications imposed by ESA. The cooler delivered 4.5 mW at 4.5 K with a long-term temperature stability of 1 mK and an input power of 1.96 W. So far, the cooler has operated continuously for a period of 2.5 months and has not shown any sign of performance degradation.

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I. INTRODUCTION

Many detectors in scientific instruments for space missions rely on cryogenic cooling to very low temperatures to achieve their required performances. A typical example of such a cryogenic mission is the Infrared Space Observatory (ISO), developed by ESA and operated between 1995 and 1998. ISO was based on a 0.6 m telescope and a cryostat containing 2200 l of superfluid helium, which cooled different instruments between 1.8 and 10 K.

So far, to reach temperatures below 60 K in space, usually cryostats with stored cryogens such as liquid helium have been used. Stored cryogens, however, result in an intrinsically limited lifetime and/or a very high system mass. Therefore, the development of compact, long-life, and low-temperature cryocoolers is highly demanded for use in space. Different low-temperature cryocooler chains are currently under development to fulfill this demand. These coolers, however, all apply mechanical compressors, which generate mechanical and electromagnetic interferences. Such interferences will be more and more unacceptable in a number of future scientific space missions, in which increasingly sophisticated optical instruments and detectors are used to fulfill the scientific aim for higher spatial resolution and sensitivity. An example of such a future activity is ESA’s Darwin mission, a space interferometer consisting of a few free flying telescopes. Because of the sensitive optics involved in Darwin, hardly any vibration can be tolerated.

This article presents test results on a new type of vibration-free and long-life cryocooler for use in such space applications. The cooler was developed in recent years at the University of Twente and consists of a 4.5 K sorption cooler, which is directly heat sunk on passive radiators that operate around 50 K. This cooler concept has no moving parts and is, therefore, virtually vibration-free. The absence of moving parts also simplifies scaling down of the cooler to small sizes, and it contributes to achieving a very long lifetime. In addition, the cooler operates with limited dc’s so that hardly any electromagnetic interference is generated. The sorption cooler consists of an adsorption compressor that drives a Joule-Thomson based cold stage. The completely reversible physical adsorption and desorption of the working gas on activated carbon is the driving mechanism within the sorption compressor. The test results in this article demonstrate

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for the first time the feasibility of this novel cooling technology in an advanced breadboard model of the cooler. This technology can become an enabling factor for future scientific space missions that require long-term low-temperature and vibration-free operation.

At the start of our research, different cryocooler options and sorption cooler architectures were compared, and a system design was proposed consisting of a two-stage hydrogen-helium 4.5 K sorption cooler. This system design was further analyzed and a breadboard version of the helium sorption cooler stage was designed, built, and tested. This article presents the resulting cooler, and the most important test results. In the next section, the system design of the cooler is presented. It includes a brief explanation of the sorption technology; for more detailed background information the reader is referred to the literature. Section III describes the essential cooler components. In Sec. IV, simulation results of a dynamic model are discussed, which further clarify the operation of the sorption cooler. Section V presents the test setup with the breadboard cooler. Test results are presented and discussed in Sec. VI. A more general discussion of the developed cooler is given in Sec. VII.

II. SYSTEM DESIGN

Operation of a sorption compressor is based on the principle that large amounts of gas can be adsorbed on certain solids such as highly porous carbon. If a small pressure vessel is filled with an adsorption material and gas is adsorbed at a relatively low temperature and pressure, then a high pressure can be created inside the closed vessel by desorption, established by an increase of the temperature of the adsorption material. A controlled gas flow out of the vessel can be maintained at a high pressure by further increase of the temperature until most of the gas is desorbed. By nature, a sorption compressor is an intermittent system in which the sorption bed alternates between gas adsorption and desorption. By combining one or more sorption beds with passive check valves and buffer volumes, it is possible to produce a constant pressure difference which drives a continuous gas flow through the cold stage (the actual refrigerator) and thus achieve continuous refrigeration. In the cold stage, compressed gas coming out of the compressor flows through a counterflow heat exchanger to the cold side where Joule-Thomson expansion to low pressure occurs through a restriction. In this expansion, the gas cools and liquefies. After evaporation by the thermal load, the low-pressure vapor returns through the counterflow heat exchanger to the compressor unit.

Apart from physical adsorption of gases on materials such as activated carbon, some gases can also be chemically absorbed, such as H2 on metal hydrides, e.g., LaNiSn. Although this is a fundamentally different process that is not intrinsically reversible and, therefore, sensitive to degeneration, it can also be applied in sorption coolers. ESA’s Planck mission, which will be launched in the near future, contains a hydrogen absorption compressor operating from approximately 300 K.

Our proposed sorption cooler concept consists of a hydrogen stage cooling from 80 to 14.5 K and a helium stage cooling from 50 to 4.5 K with 5 mW of cooling power; see Fig. 1. Both stages use a microporous activated carbon monolith as the adsorption material in the compressor cells. The two cooler stages need only a few watts of input power and are heat sunk at two passive radiators that are constructed at the cold side of the spacecraft, at temperatures of approximately 50 and 80 K. Radiative cooling to temperatures below 80 K is possible in far-away positions such as the second Lagrange point (L2, used for missions such as Darwin), where the earth radiation is very limited. Precooling of the helium stage at 14.5 K by the hydrogen stage is needed to obtain sufficient enthalpy difference between the low- and high-pressure lines of the helium stage. Without precooling of the helium to well below the inversion temperature of 40 K, no net cooling effect can be obtained at 4.5 K using the Joule-Thomson (JT) effect.

This article focuses on the development and test of the helium sorption cooler. Compared to the hydrogen stage, the helium stage has the most challenging properties: the largest radiator area, limited gas adsorption at 50 K, very strict requirements on temperature stability at 4.5 K, high-efficiency counterflow heat exchangers, etc. Figure 2 shows a more detailed schematic of the helium sorption cooler. The compressor consists of two stages with each two sorption cells, which compress the gas from 1.3 to 4 bars in the first stage and from 4 to 16.5 bars in the second stage. The selected minimum and maximum cycling temperatures of the cells are given in Fig. 2, as well as the required input powers and resulting radiator areas. All mentioned operating parameters were selected as a result of a multidimensional optimization procedure, in which the total radiator area was minimized.
An advanced thermodynamic model of the sorption cooler with all relevant input and output parameters was used for this optimization procedure.

Gas buffers of 2 and 4 l are used at the intermediate- and low-pressure sides; see Fig. 2. These gas buffers have two important functions. In the first place, they facilitate the discontinuous and independent use of the individual compressor cells that are connected to the buffers: the buffers maintain the low and medium pressures during all phases of the compressor cycle while the sorption cells on average have to maintain the dc gas flow through the system. One of the attractive results is that the cycling of the cells of the first compressor stage does not need to be synchronized with respect to that of the second stage. The second function of the buffers is that it limits the system pressure to acceptable values when the system is not operational and stored at 300 K. At 300 K, virtually no helium gas is adsorbed on the activated carbon. If the buffers would not be present, then at 300 K, virtually no helium gas is adsorbed on the activated carbon. If the buffers would not be present, then at 300 K the system would be at a pressure above 100 bars. All parts of the system could be designed to stand such a high pressure, but for the compressor cells this is not attractive; it would result in very thick walls that would deteriorate the cooler performance dramatically due to the added thermal mass. Notice furthermore that no high-pressure buffer is used because that would significantly increase the required size of the low-pressure buffer. As a result, the two cells in the second stage do need to be synchronized by the control algorithm to provide a continuous high pressure and flow through the cold stage.

The required number of sorption cells is directly related to the size of these cells, the required mass flow through the cold stage and the cycling speed of the cells. A first-order relationship between these parameters can be expressed as

\[
\dot{m} = \frac{N m \Delta x_{\text{net}}}{t_{\text{cycle}}}
\]

where \( \dot{m} \) is the required mass flow, \( N \) is the number of sorption cells in one compressor stage, \( m \) is the sorber mass (proportional to the cell size), \( \Delta x_{\text{net}} \) is the net amount of gas that can be adsorbed and desorbed by a sorption cell (per gram of sorber material), and \( t_{\text{cycle}} \) is the cycling period of one cell. For a given fixed compressor cell size, a value of \( \Delta x_{\text{net}} \) and a required mass flow, the number of required cells is proportional to the cycle time of the cells. A minimum cycle time is set by the dynamic behavior within the sorption cells during heating and cooling; this behavior is illustrated in the simulation results in the next section. For the current design of the carbon in the cells (diameter=1.5 cm, length =15 cm, and mass=18.5 g), this minimum cycle time is about 5 min. Furthermore, \( \dot{m}=0.41 \, \text{mg/s} \) and \( \Delta x_{\text{net}} = 4.6 \, \text{mg/g} \) for stage 1 and 8.7 mg/g for stage 2. It follows that with two sorption cells for each compressor stage, Eq. (1) can easily be satisfied.

Equation (1) illustrates an important characteristic of sorption compressors: for a certain design of the sorption cycle and compressor cell (hence, a fixed \( \Delta x_{\text{net}} \) and \( m \)), an increase or decrease of the cooling power can be realized by a proportional increase or decrease of the number of sorption cells. In the current design, for instance, the cooling power can be increased to 10 mW by increasing the number of compressor cells from 2 \times 2 to 2 \times 4 cells thus facilitating a double mass flow through the cold stage.

Different alternatives for the passive radiator architecture were compared, including deployable radiator options. Figure 3 shows a side view of one of the favorite design options for use in missions such as Darwin. It consists of a fixed radiator in the shape of a truncated cone that is integrated in the spacecraft. This cone shape guarantees an optimal thermal view factor to space. A sun shield in combination with a V-groove radiator is used to shield the passive radiators from the warm side of the spacecraft. The thermal design of the spacecraft is such that there is a gradual temperature decrease from the sun side of the spacecraft to the cold optical bench that is located at the other (cold) side of the spacecraft. In this way, the passive radiators at 80 and 50 K can provide optimal passive cooling of the heat conduction losses through the mechanical supports that connect the cold optics to the warm parts of the spacecraft so that hardly any cooling is required to maintain the optics below 50 K. The sorption cooler cold stage is located inside the optical bench, to provide the required cooling power at 4.5 K to the detectors. The compressor cells are distributed over the
inside of the radiator surface, thus limiting thermal gradients over the radiator surface. The individual compressor cells are connected via two check valves to gas ring lines for the low, medium, and high pressures. Although the radiator area is relatively large, the radiator plates itself can be very thin and light because the heat flows through a 50 K radiator are very small (the radiated power per unit area scales with $T^4$).

III. DEVELOPED COOLER COMPONENTS

This section presents the most important building blocks of the sorption cooler: the sorption compressor cells, the check valves, the cold stage, and the control system.

A. Sorption compressor cells

Figure 4 shows a schematic picture of the inside parts of a sorption compressor cell. The compressor vessel is filled with an activated carbon monolith, which was especially optimized for use in this sorption cooler. Heating of the carbon is done with an electric heater that is located in the center of the carbon. The compressor vessel is on both ends supported by the heat sink. Between the surfaces of the vessel and the heat sink a narrow gap is present, which can act as a gas-gap heat switch. The gas pressure in the gas-gap heat switch can be varied to yield a relatively high or low thermal conduction between the vessel and the heat sink. In this way, the vessel can be thermally isolated from the heat sink during the heating phase of the cell and it can be thermally connected to the heat sink during the cooling phase of the cell. This pressure variation is carried out with the gas-gap actuator, a small piece of activated carbon that is connected with the gas gap and that can be varied in temperature.

Figure 5 shows the final design of a sorption compressor cell, together with a mounted check valve unit that contains the two check valves that interface to the compressor cell (see Fig. 2). The design of the compressor cell is identical for the two compressor stages, although they are operated at somewhat different conditions. The mass of the compressor cell is 420 g excluding the check valve unit and 500 g including it. The cell is fabricated entirely of stainless steel parts, except for the activated carbon, the heater, and the temperature sensor.

Fluidic interfacing of the compressor cell to the check valve unit is done via a Swagelok 1/8 in. VCR connection. Similar connections are applied to interface the check valve unit to the two pressure lines. The compressor cell is bolted to the heat sink plate (i.e., the radiator); this bolting provides a proper thermal and mechanical interface. The electrical interface connects three parts of a compressor cell: the heater, the gas-gap actuator, and the thermocouple.

B. Check valves

The developed check valve unit is shown in Fig. 5, integrated with the compressor cell. The unit contains the two check valves that connect the gas lines to the compressor cell (see Fig. 2). The all-metal internal design of the check valves is such that it fulfills the very strict operating requirements that are imposed by the system design of the cooler. Among others, these include cryogenic operation, low leakage flow in closed direction, low pressure drop in forward direction, low void volume, tolerant to contamination coming from the compressor cell, and reliable use for over 500 000 open-close cycles. A number of different tests on the check valve were carried out to verify that the design did meet these requirements. More details on the check valve design and test experiments were published elsewhere.

C. Cold stage

The design of the cold stage is given in Fig. 6. The total mass of the cold stage is 650 g; dimensions are given in the figure. The cold stage is constructed around two supporting frames, one at 50 K and one at 14.5 K, which are fixed together via a Kevlar wire system. This Kevlar wire system produces a completely rigid structure that is resistant to launch forces, while it limits conduction losses on the cold end to a very small value.

The first and the second counterflow heat exchangers are spiraled inside both supporting frames, and are also fixed via Kevlar wiring. The counterflow heat exchangers are constructed as tube in tube, with a 1/8 in. tube inside a 1/8 in. tube. Both heat exchangers are approximately 100 cm long.

The JT flow restriction is fabricated of sintered steel, and tuned accurately to the correct mass flow at 4.5 K. The copper evaporator has a fine copper matrix inside to guarantee a good thermal contact to the flow of liquid helium when it is being evaporated. The mass of the evaporator itself is very small, about 25 g.

FIG. 4. (Color online) Schematic picture of the inside parts of the compressor cell.

FIG. 5. Design of a sorption compressor cell with integrated check valve unit.
The radiator area required to cool the sorption compressor scales linearly with the required cooling power at 4.5 K. By minimizing the parasitic thermal loads at 4.5 K, virtually all cooling power is made available as net cooling, and the required gross cooling power can be kept at a minimum (and associated with that the required radiator area). Table I lists the calculated losses of the cold stage. These very small losses are achieved by (1) using a radiation shield at 14.5 K and (2) using a Kevlar wire support structure. Without using these two measures, the thermal losses would be several milliwatts.

The fluidic interface to the low- and high-pressure lines of the cold stage is by two 1/8 in. VCR connections. Thermal and mechanical interfaces to the cold stage exist at the bottom of the 50 K supporting structure and at the bottom of the 14.5 K supporting structure. In stationary operation, virtually no heat flows will be present via the mechanical interface at 50 K, and a small heat flow of about 10 mW will be present to the 14.5 K interface. In the integrated design of the helium/hydrogen sorption cooler architecture as described in Sec. II, the hydrogen cold stage will be integrated with the helium cold stage.

The current thermal-mechanical interface of the thermal load to the evaporator at 4.5 K has an area of approximately 1 cm². This area can easily be adapted, if necessary.

### D. Control system

The data acquisition and control system of the 4.5 K sorption cooler contains the following components:

1. power supplies for the different sensors in the system, as well as for the heaters that are located in the system;
2. a data acquisition system to sense the electrical signals from all sensors and from the currents and voltages that make up the heater powers;
3. a computing system (hardware+software) to calculate the required output (heater) signals that are needed to control the cooler, and (if necessary) to display and/or store the measured data; and
4. analog and digital output circuitry to power the different heaters in the system.

For the current breadboard cooler, a personal computer (PC)-based standard National Instruments data acquisition system was used to control the cooler. For a future flight cooler, a compact stand-alone controller can be designed which performs the same functions.

### IV. DYNAMIC MODELING RESULTS

This section presents modeling results of the thermal and fluidic dynamic behaviors of the sorption compressor. The results will be used to further explain the cycle and clarify aspects of the compressor operation that cannot be measured directly, such as thermal gradients that occur inside the sorber material during cycling. In addition, the results were used to determine the minimum required cycle time of the sorption compressor. The dynamic analysis was carried out using a sophisticated bond graph model,¹⁴ which contains constitutive relations for the relevant elements of the sorption compressor as well as a number of temperature-dependent material properties.

Figure 7 shows a plot of some important parameters of a compressor cell in the first compressor stage, during 1.5 cycle. This compressor cell is one of the two cells that is connected via check valves to the low- and medium-pressure buffers, see Fig. 2. The operating parameters (see caption) are chosen such that two of these compressor cells together will produce the required average mass flow of 0.41 mg/s, resulting in 5 mW of cooling power for the total system in Fig. 2. From top to bottom, the following plots are given in Fig. 7.
Parameters: actuator, pressures sor stage. From top to bottom: temperatures of the cell and the gas-gap.

FIG. 7. Simulation results for a single compressor cell in the first compres-sor stage. From top to bottom: temperatures of the cell and the gas-gap numbers given above the plots.

The different phases that can be distinguished in the cycle are discussed below; the numbering corresponds to the numbers given above the plots.

1. Heating and pressure buildup. The heating phase is concentrated in a relatively short period of time; for a fixed cycle time this leaves a maximum time interval for the cooling phases (4 and 5), which is the limiting factor for minimizing the cycle time. Clearly, during heating significant thermal gradients arise from the heater to the first few carbon rings—but these gradients level out very rapidly when the heater is switched off after phase 2.

2. Further heating and gas flow out of the cell into the medium-pressure buffer. Heating continues with the same input power, now desorbing gas at a constant pressure. The temperature increase is less progressive compared to phase 1 because part of the input power is now put in the heat of desorption used to free the gas from the carbon surface. Due to the rapid heating in a relatively short period of time, the mass flow out of the cell into the buffer is much larger (maximum of 3.2 mg/s) than the required average mass flow through the cold stage (0.41 mg/s).

3. Heater off and switch on of gas-gap heat switch. After switching off the heater, the temperature gradients in the carbon rapidly disappear.

4. Cooling down and depressurization of the cell. When the gas-gap heat switch starts to conduct, the cell rapidly cools down—reducing the pressure in the vessel. The heat flow through the gas gap is large in this phase. Also during cooling of the adsorption material, temperature gradients can be observed, but now in the opposite direction as compared to the heating phase.

5. Further cooling down and gas flow from the low-pressure buffer into the cell. When the pressure in the cell drops below the pressure of the low-pressure buffer, the check valve opens and gas flows into the cell and is adsorbed in the carbon. The pressure in the low-pressure buffer initially reduces rapidly due to the large gas flow into the cell, until the pressure in the buffer slowly starts to rise again when the flow from the cold stage into the buffer is larger than the flow from the buffer into the cell. The heat of adsorption that is produced in this step slows down further cooling of the cell; this causes the small kink in the exponential shape of the reducing temperature of the cell at the beginning of this phase. The minimum cycle time is reached when most of the gas is readsorbed on the carbon, approximately 400 s.

6. Switch off gas-gap heat switch. At the end of the adsorption phase, the gas gap is switched off. When the conduction level of the heat switch is reduced below a certain level, heating of the cell can be started again.

The plot of the thermal powers illustrates how the heater power is provided to the system in a short period of time, and later “released” when the gas-gap heat switch is switched on: the heat flow through the gas gap peaks and then rapidly decreases. The “peaked” power through the gas gap is transferred to the heat sink, and in a delayed way transferred to the radiator. Finally, the radiator just continuously radiates the average power into space—no variation is visible anymore.

Figure 8 shows simulation results for an integrated first compressor stage consisting of two cells, connected to the low- and medium-pressure buffers. Relevant input parameters are the same as used for the single compressor cell. One additional output parameter is the total input power, which is now 2 \times 0.41 W = 0.82 W. Of particular interest is the pressure variation of the low-pressure buffer due to the cycling of the two cells, since this variation directly relates to the temperature of the (uncontrolled) evaporator. This variation equals about 12 mbars, which results in a cyclic temperature...
variation in the cold stage of about 12 mK—in the situation that no active temperature control is used for the evaporator in the cold stage.

Similar simulation results were made for the second compressor stage. Most aspects of the simulated behavior of the second stage are similar to that of the first stage. The resulting operating parameters are as follows: $T_{\text{sink}} = 52$ K, $t_{\text{cycle}} = 741$ s, $t_{\text{heating}} = 391$ s, $P_{\text{cell max}} = 10$ W, $P_{\text{cell average}} = 0.75$ W, and $P_{\text{2nd compr stage}} = 1.50$ W. Further details about the differences between the cycling of the first and second compressor stages will be discussed together with the test results in Sec. VI (see Fig. 10).

V. TEST SETUP WITH THE BREADBOARD COOLER

The test setup should provide the required operating conditions for the 4.5 K helium sorption cooler, so that it can be tested in the laboratory instead of a space environment. The test setup contains the following components (see Fig. 9):

1. a large vacuum bell jar;
2. a mechanical precooler that can provide the temperature levels of 50 K for the sorption compressor cells and 14.5 K for the precooling of the cold stage;
3. a 50 K thermal radiation shield around all components of the sorption cooler, to prevent 300 K radiation from reaching the cooler components; and
4. a helium gas handling system outside the bell jar, which is connected to the sorption cooler inside the bell jar; this gas handling system is required to pump down the sorption cooler and clean it from contaminant gases, and to fill it with helium gas.

VI. TEST RESULTS

After integration of the sorption cooler in the test setup, the following actions were carried out to prepare the system for the tests:

1. the sorption cooler was leak tested;
2. the carbon in the cells was baked out under vacuum at 250 °C;
3. the sorption cooler was filled with clean helium gas;
4. the 50 K radiation shield was placed, and the vacuum bell jar was closed and evacuated; and
5. the platforms with the sorption cells were cooled by the precooler, and temperature controlled to approximately 50 K; in addition, the cold stage precooling temperature of 14.5 K was provided by a temperature-controlled second stage of the precooler.

When the sorption compressors were started for the first time, it appeared that a check valve of a compressor cell in the first compressor stage was leaking in the reverse direction. We then decided to switch off this specific sorption cell but we left it connected to the system and to operate the cooler with three sorption cells: one in the first stage and two in the second stage. By a slight readjustment of the operating parameters of the single sorption cell in the first compressor stage, it appeared that the sorption cooler could be operated very well, with hardly any performance degradation.

Starting from the precooling temperature of around
50 K, several hours are needed to start the sorption cooler: first, the first compressor stage is started to pressurize the medium-pressure buffer, and then the second compressor stage is started to create the high pressure which eventually cools down the cold stage to 4.5 K. Figure 10 shows a plot of the extremely stable operation of the sorption cooler during two periods of half an hour, with a period of two weeks of continuous operation in between. Important resulting operating parameters are given in Table II, where they are compared with the specified values. In the plots a number of details in the operation can be observed, which will be discussed below. The different steps in the compressor cycling correspond to the simulation results in Figs. 7 and 8.

First-stage compressor temperature (cell 2). In order to cycle faster with the single cell of the first compressor stage (to compensate for the second nonoperating cell), the peak input power was taken as 9 W instead of the originally planned 5 W. This causes the faster temperature rise of the cell, as compared to the simulation results. Furthermore, the temperature reading in the first heating phase is too high and quite inaccurate, because the temperature sensor is positioned close to the heater. During the gas-flow phase this

<table>
<thead>
<tr>
<th>Specified</th>
<th>Measured</th>
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</thead>
<tbody>
<tr>
<td>Cooling capacity &gt;5 mW at 4.5 K</td>
<td>4.5 mW at 4.5 K</td>
</tr>
<tr>
<td>Temperature stability:</td>
<td></td>
</tr>
<tr>
<td>&lt;1 mK for a period of 1 h</td>
<td>1 mK for 1 h (controlled),</td>
</tr>
<tr>
<td>&lt;10 mK for 2 weeks</td>
<td>&lt;4 mK for 2 weeks (uncontrolled)</td>
</tr>
<tr>
<td>&lt;0.1 K over cooler lifetime</td>
<td></td>
</tr>
<tr>
<td>Exported vibrations &lt;1 μN/√Hz</td>
<td>≤1 μN/√Hz (analyzed)</td>
</tr>
<tr>
<td>Power consumption &lt;200 W</td>
<td>1.96 W (without electronics)</td>
</tr>
<tr>
<td>Lifetime &gt;5 years</td>
<td>expected, not measured</td>
</tr>
<tr>
<td>Cooler operable continuously during its lifetime</td>
<td>2.5 months tested with</td>
</tr>
<tr>
<td></td>
<td>17 days of uninterrupted operation</td>
</tr>
<tr>
<td>Cooler capable of 20 start-stop cycles (only compr. cell pressure buildup)</td>
<td>&gt;20 start-stop cycles</td>
</tr>
<tr>
<td>Cooler able to operate under any orientation</td>
<td>horizontal and tilted 45°</td>
</tr>
<tr>
<td>Cold radiator (T=40–80 K), maximum cooling power of 2.5 W at 50 K, or 9.1 m² radiator area</td>
<td>0.891 W at 47 K + 1.065 W at 50 K</td>
</tr>
<tr>
<td></td>
<td>or 4.2 m² + 3.7 m² = 7.9 m²</td>
</tr>
<tr>
<td>Cooler mass &lt;10 kg</td>
<td>8.3 kg (excl. platforms)</td>
</tr>
</tbody>
</table>
effect is compensated somewhat due to gas flow out of the cell along the sensor. For the rest, the behavior is similar to what was predicted in the simulations.

Second-stage compressor temperatures (cell 3 and 4). Because no high-pressure buffer is present (see Fig. 2), the second compressor stage operates in a slightly different way than the first compressor stage: there is always one cell delivering gas to the high-pressure line. The following steps in the cycle of one cell can be distinguished.

1. Heating of the cell with a relatively large input power, until a pressure is reached which is just below the actual high pressure (the high-pressure check valve remains closed).
2. A short period of waiting until the other cell has finished its gas-delivering phase.
3. Takeover of the delivery of gas from the other cell.
4. Delivery of high-pressure gas by controlled further heating of the cell.
5. Takeover of the delivery of gas by the other cell.
6. The heater is switched off and the heat-switch actuator is switched on. The behavior here is similar to that of the first-stage compressor.
7. The heat-switch actuator is switched off.

High pressure. A slight fluctuation of the high pressure can be observed during the takeover between the two second-stage cells. This is caused by the fact that the software controller switches from one cell to the other.

Medium pressure. The fluctuation of the medium pressure is caused by the combined behavior of the first and the second compressor stages: the first stage blows gas into the buffer with a period of 289 s, and the second stage pumps gas out of the buffer with a period of 975 s. The two stages are clearly not synchronized. On average, the pressure is constant at approximately 4.2 bars.

Low pressure. The low pressure has, as expected, a fluctuation of approximately 12 mbars. The maximum pressure is about 1.24 bars, which corresponds to a liquid helium temperature of 4.44 K.

Cold-end temperature. The uncontrolled cold-end temperature fluctuates in proportion to the low pressure in the evaporators; the temperature variation is about 12 mK. In Fig. 11, the cold-end temperature is controlled to 4.5 K exactly, with a fluctuation of about ±1 mK. This temperature stability is obtained by a controlled evaporation of the excess liquid helium, which is not used for cooling a load.

Cooling power. The average cooling power is around 4.5 mW; it fluctuates between 4.4 and 4.6 mW. This fluctuation is inversely proportional to the low-pressure fluctuation: the temperature controller compensates for the varying boiling temperature. The measured cooling power of 4.5 mW is slightly lower than the expected cooling power of 5.0 mW, which is based on the calibrated mass flow through the flow restriction. We expect that a combination of several small measurement errors and residual conduction losses is causing this deviation.

FIG. 11. Thermal behavior of the cold end of the cooler while it is actively controlled to 4.500 K by means of a temperature sensor, PID controller, and heater. The low pressure, cold temperature, and heater power are depicted.

Resulting total input power and radiator area. The resulting total input power is 1.96 W, which corresponds to a radiator area of 7.9 m²—see Table II. The resulting radiator area of 7.9 m² is close to the predicted value of 8.3 m², and well within the specified area of 9.1 m².

VII. DISCUSSION

The sorption cooler architecture that was presented has several distinctive advantages for use in space applications, when compared to conventional mechanical cryocoolers.

- The system has no moving parts. Therefore, (1) it is vibration-free; (2) it has a long lifetime; and (3) it is scalable to small sizes and cooling powers.
- Because limited dc’s are used to power the compressor cells, the electromagnetic interference levels are very low (0.01 pT at 1 m distance).
- The cooler has a very small input power of a few watts.
- Separation of the compressor and the cold stage is possible, up to many meters if necessary.
- The compressor is operating at the cold side of the spacecraft so that no thermal or fluidic interfacing is required from the cold stage to a compressor in the warm service module.

Clearly, still a significant cold radiator area is the price that must be paid for these advantages. Minimization of this radiator area can be achieved by the following factors.
Minimization of the required cooling power at 4.5 K. This can be achieved by using as much as possible the precooling temperature and cooling power at 14.5 K for cooling away the radiative and conductive heat losses on the 4.5 K stage.

Increase of the cold-end temperature can also reduce the radiator area. An increase of the temperature from 4.5 to 7.5 K, for instance, reduces the radiator area with approximately a factor of 1.65 (for the same cooling power).

Further improvement of the activated carbon will result in a reduction of the radiator area. A new project is currently running in which another type of carbon monolith will be tested in the sorption cooler, which is expected to reduce the radiator area with about a factor of 1.5.

The system was tested with one sorption cell running in the first compressor stage. If both sorption cells are running it is expected that the cooler performance improves with about 10%. The single compressor cell operates “on the edge”; the cycling speed is so fast that the adsorption phase is not fully completed when the heat switch changes to the off state.

The vibration level of the sorption cooler is so low, compared to the vibration noise level in our laboratory, that it is impossible to detect it. Instead, an analysis was made, which was validated by experiments in which vibration levels were applied that could be detected. Our validated model shows that the vibration level of the sorption cooler is well below 0.1 μN/√Hz; details will be published elsewhere.

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