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MODELS FOR SCALABLE HELIUM-CARBON SORPTION CRYOCOOLERS

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ABSTRACT

We have developed models for the rapid design of helium-based Joule-Thomson cryocoolers using charcoal-pumped sorption compressors. The models take as inputs the number of compressors, desired heat-lift, cold tip temperature, and available precooling temperature and provide design parameters, optimized for low power, as outputs. The output parameters are the compressor dimensions, required charcoal mass, cycle temperatures and pressures, heat exchanger dimensions and J-T orifice characteristics. We predict that a helium/carbon cryocooler with 18 K precooling can be as efficient as 60 W/W. These models, which run in MathCad and Microsoft Excel can be coupled to similar models for hydrogen sorption coolers to give designs for 2-stage vibration-free cryocoolers that provide cooling from ~50 K to 4 K. We will present a preliminary design for a two-stage vibration-free cryocooler that is being proposed to support a mid-infrared camera on NASA's Next Generation Space Telescope.

INTRODUCTION

A number of space missions currently in development are expected to use detectors that require cooling to less than 10 K. These missions include NASA's Next Generation Space Telesope (NGST), Interstellar Probe, and Terrestrial Planet Finder (TPF), as well as ESA's FIRST/Planck and Darwin. At present there are only limited options for cooling the detectors on these missions. Stored cryogens are planned for use on the Far Infrared and Submillimeter Space Telescope (FIRST) and NASA's Space Infrared Telescope Facility (SIRTF), but provide limited lifetime and contribute a substantial mass, from both the cryogen and its storage system, that must be launched into space. The Planck mission uses a combination of hydrogen-sorption cryocoolers (to reach 18 to 20 K) and helium-based Joule-Thomson (JT) mechanical cryocooler to reach 5 K.

Although mechanical cryocoolers can achieve the desired low temperatures of these types of missions, they have several disadvantages. The mechanical compressors must be vibration-isolated from the instruments; JT coolers allow this isolation to be made on the warm side of the spacecraft, but most other types of cryocoolers must have the compressor located near the cold-head, making vibration isolation substantially more difficult. Mechanical coolers are also not very scalable—the mass and volume of a 10 mW, 4 K cooler are not very different from those of a 300 mW cooler[1]. Because many of these space missions will require cooling power on the order of ~10 mW, there is substantial inefficiency in using mechanical coolers.

We are developing sorption cryocoolers that are inherently vibration-free and whose mass and power scale with the required cooling power. This paper will describe developments in the design models for 5-8 K low-power (10-100 mW) continuous coolers that use charcoal as the sorbent material and helium as the working gas. Periodic charcoal-different from those of a 300 mW cooler[1]. Because many of these space missions will require cooling power on the order of ~10 mW, there is substantial inefficiency in using mechanical coolers.

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BASIC PRINCIPLES

Charcoal-helium sorption coolers rely on the ability of charcoal to adsorb and retain substantial amounts helium at temperatures above the liquefaction temperature. The helium-charcoal system obeys a Clapeyron-like saturated vapor-pressure relationship (duband, dubinin)

$$\ln(P_s) = A + \frac{B}{T} \tag{1}$$

where A and B are constants and P_S is the effective saturated vapor pressure. At any given temperature and pressure, charcoal can adsorb some amount of helium onto its surface. The amount of helium adsorbed depends on the particular charcoal, and in the range that is useful for making sorption coolers can be described by the expression

$$C = \frac{M\alpha}{b}\exp(-\gamma x) \tag{2}$$

where

$$x = \frac{RT}{\beta}\ln(\frac{P_s}{P}). \tag{3}$$

M is the molecular weight of helium, R is the gas constant, T is the temperature, P the pressure, b specific volume of the adsorbed phase (taken to be the van der Waals volume), and α, β, and γ are experimentally determined parameters of the charcoal.

By changing the temperature, we can change the amount of helium stored in a compressor filled with charcoal. If we heat the charcoal, the helium is desrobed and can be driven through a JT expander to provide cooling. At the same time we must provide a charcoal-filled compressor depleted of helium and at a low temperature for the gas to adsorb onto after passing through the JT.

DESIGN MODEL

We have developed a pair of complementary design models. The first model is a simplified cooler system, in which the precooling temperature (typically also the low compressor temperature) and the desired cooling power are the input parameters. The model then runs a program in Visual Basic and Microsoft Excel to determine the optimal high and low pressures in the compressors and the high temperature in the compressors by iteratively evaluating a simplified design. These parameters are then used in the more detailed MathCad-based model. We used MathCad for model development because it offers the general functions of a spreadsheet, including programmability, but displays mathematical relations in a relatively easy to read format, much like standard textbook mathematical style. This feature makes debugging and modification of the model easier, particularly as the model becomes more complex.

The ultimate performance of the model depends on the characteristics of the charcoal used in the compressors. The sorption properties and the void fraction are the main charcoal parameters. The model presently uses values from Duband, as they comprise a set of data that can be reasonably described by equations 1 and 2. Data from any other available charcoal can be readily substituted either as fitted equations, tabular values with a lookup procedure, or inserted by hand. The helium parameters (enthalpy, density, etc.) are obtained similarly from MathCad calls to Excel, which obtains the desired values from a GasPak dynamically linked library. The remainder of this section describes the subsections of the design model in the order in which they are evaluated.

Cooler Architecture

The basic design that we assume is that the cooler is composed of pairs of compressor elements, with the two elements connected via a bi-directional JT constriction and heat exchanger. A single pair of compressor elements would make up a periodic sorption cooler, in which the helium is transferred back and forth between the two compressors. In some circumstances the recycling time will be short compared to time constants at the cold head so that the cold head temperature will be sufficiently stable for the application and a single pair of compressors is all that is needed. Multiple pairs of compressors can be connected in parallel, with their heat exchangers and JT cold-heads tied together, and operated out of phase to smooth out the temperature fluctuations at the cold head. This arrangement also offers some redundancy in case of the failure of one of the compressor pairs or the plugging of one of the JT constrictions. Microfabricated valves could be used to simplify the plumbing of the system when they become available.

The compressor elements are located at the cold stage of the precooler, which should provide cooling to 12-20 K, and thermal conduction between them and the precooler is controlled by heat switches. The present design uses gas-gap heat switches, but mechanical suitable mechanical heat switches could be made from magnetostrictive materials. The gas from the hot, desorbing compressor is precooled back to the low compressor temperature before going to the cold counterflow heat exchanger. It may also be possible to improve the efficiency of the system by using the waste heat from the warm compressors to partially heat the cold compressors. Such a pairwise regeneration system could reduce the input power by 25 to 30%, which is less than the ~50% improvement possible in higher temperature systems because the heat of desorption is comparable to the heat capacity of the cooler and is not recovered in the regeneration process.

Compressors

Given the enthalpy ΔH removed by each unit mass of helium, determined by the compressor temperatures, cold-tip temperature and heat-exchanger efficiency, we can simply evaluate the required mass flow rate as the desired cooling power divided by ΔH. The cycle times for the compressors are determined by the user, and are typically chosen to be between 800 and 1200 s. Combining the cycle time and the mass flow rate gives the amount of helium that must be expelled from a single compressor during its desorption cycle. Additional gas is required to fill the dead volumes of the system and the void volume of the charcoal. Taking the total gas to be removed from the charcoal and relating it to the storage capacity of in Eq. 2, we can solve for the required charcoal mass.

To contain the charcoal and helium, we assume a cylindrical container with hemispherical endcaps and minimize the mass of constant-wall-thickness material to get the radius and length of the cylinder. The wall thickness of the container is then determined by the high pressure and safety margin required by the application (Burger). When determining the total volume to be enclosed we first solve the model with no heating element enclosed, and then re-solve with the volume of a wire heater of appropriate power. It is important to keep the mass of container material as small as possible to reduce the heat capacity of the components that must be cycled between the minimum and maximum temperatures of the compressor elements. The power to drive the compressors is determined by the energy required to heat the compressors and the gas in them from the low, adsorption temperature to the high, desorption pressure and the heat of desorption of the helium-charcoal (duband).

Gas-gap Heat Switch

Heat switches are required for coupling and uncoupling the compressor elements from the cold stage. It is desirable to have good heat switches to minimize the heat leak to the precooling system when the compressors are at their maximum temperature. Gas-gap heat switches have been demonstrated in space and are ideal for this temperature range. Magnetostrictive materials may also provide suitable heat switches if the mass of the solenoid required to actuate the material can be made small enough.

In the design for the NGST coolers described below, the heat switches are integrated into the cold-head for the precooling stage. The cold-head is made from a block of high thermal conductivity material that has been bored out to provide individual cylindrical wells approximately 200 μm larger in diameter than the outer shells of the compressor elements. A fine capillary tube connect each gas-gap volume to its own charcoal pump. Because the volume of the gap is very small, the size of the charcoal pump for the heat switch is determined more by manufacturing limits than by the charcoal properties. We have assumed a switching ratio of 300 based on information in the literature (duband).

Counterflow Heat Exchanger

The counterflow heat exchangers, located between the precooling stage and the low-temperature cold head, for the pairs of compressors are all coupled together so that they provide a common heat exchanger for all of the gas lines. The mass of the lines is very small (less than 1 g per pair), so the mass of the heat exchanger is dominated by the structure used to couple the lines together. The mass of this structure is assumed to be 50 g. The diameters of the lines are selected (Barron) to keep the Reynolds number below 2000 and keep the pressure drop to a user determined value (typically ~1%). The wall thickness is then determined by the high pressure in the lines, the elastic modulus of the material and the required safety factor of the application. In many cases the required wall thickness is less than can be economically manufactured, so a nominal minimum wall thickness of 50.8 μm is set by the model. Capillary tubing of ~150-200 μm and ~50-75 μm wall thickness can be readily obtained in stainless steel and other materials, such as cupronickel alloy.

Table 1. Power Budget for a 6 K, 5 mW cooler with 18 K precooling

Item	**Power (mW)**
Charcoal Heat Capacity	25.4
Container Heat Capacity	81.8
Helium Heat Capacity	151.0
Heater Heat Capacity	7.0
Heat Leak in Gas-Gap	12.2
Desorption	128.8
Void Volume	22.2
Gas Gap power	3.0
Total Power	**431.4**

Table 2. NGST Mid-IR Cooler System Properties for Various 6 K Cooling Loads

Heat Lift At 6 K (W)	Charcoal Input Power (W) (at 18 K)	Charcoal Sys Mass (kg)	Hydride Input power (W) (at 270 K)	Hydride Sys. Mass (kg)	Total System Power (W)	Total System Mass (kg)	Passive Cooling requirements (W)	
							35 K	270 K
0.005	0.43	0.56	66.4	12	66.8	12.6	0.44	66.4
0.007	0.58	0.71	72.9	12.7	73.5	13.4	0.59	72.9
0.010	0.81	0.95	82.6	13.7	83.4	14.6	0.82	82.6
0.014	1.12	1.27	95.6	15.1	96.7	16.4	1.13	95.6
0.015	1.20	1.34	98.9	15.4	100.1	16.7	1.21	98.9

Summary of Expected Performance

As an example of the output from the pair of models, we assume a cooler with 5 mW of cooling power at 6 K and a precooling stage of 18 K, as can be provided by a hydrogen-sorption cooler. The high (desorption) temperature determined by the Excel model is 80 K, and the low and high pressures in the compressor elements are 0.23 MPa and 2.5 MPa. The input power required by the cooler is the sum of the power required to heat the charcoal, the helium, the charcoal and the heater, plus the power for desorption and the power for the gas-gap heat switches. The power to desorb gas into the lines is treated separately and added to the system power. All of these power requirements are summarized in Table 1. Table Summary gives a breakdown of the power input due to each component of the system, giving an efficiency of 86 W/W. The mass of this system will be 430 g. The efficiency might be improved to 60 W/W by a pairwise regeneration scheme for recycling waste heat from the compressors.

COOLER FOR NGST MID INFRARED CAMERA

A cooler system designed to provide base temperatures of 6 to 8 K will consist of two stages – a precooler to reach ~18 K and the final low-temperature stage. The precooler will require precooling of the working fluid to less than ~60 K, which will be provided by passive radiation. For the NGST ISIM cooler, the passive precooling temperature is expected to be 35-40 K. In this section, we present a design for a two-stage continuous sorption cooler system capable of achieving 6 K. The precooling stage is assumed to be a hydrogen sorption cooler, similar to the Planck cooler described elsewhere in these proceedings. The coldest stage is a charcoal sorption cooler, as described above.

The heat load on the 6 K stage is expected to be less than 15 mW, depending on the number of detector arrays used in the MIR focal plane and their characteristics. Each array will dissipate approximately 1 mW, with parasitic heat leak from lead wires (20/array, plus 16 for temperature monitoring and control) contributing an additional 0.03 mW/array. Heat leak through the supporting structure will also be small, assumed to be 0.16 mW independent of the number of arrays. We expect that ten 1024x1024 pixel arrays would require ~13.5 mW of heat lift at 6 K (including parasitic heat loads); the dependence of the cold-stage heat load on the number of detector arrays is shown in Figure Performance.

In order to provide 10 mW of heat lift, which would be sufficient for 7 detector arrays, the complete cooler system is expected to be 15 kg, requiring a total input power of 83 W, with a cooling efficiency of 8300 watts input power per watt of heat lift at 6 K. Table 2 shows the mass and power requirements of each of the stages, as well as the complete system, for a range of cooling powers at 6 K.

Figure 1 shows a semitransparent rendering of the precooler cold-head and charcoal sorption compressor system. The block for the cold-head is XxxYYxZZ cm. The compressor elements are inserted in the block and isolated from it by the conical standoffs on the ends. The standoffs also also serve to locate the compressor elements so that the walls of the compressor are concentric with and separated from the walls of the cold-head by approximately 100 μm. Each compressor has a gas input/output line and pair of heater leads penetrating the gas-gap and the pressure cylinder. There is a small sorption pump (located outside the cold-head) for connected to each gas-gap. The large tube penetrating the cold-head along the longer side is the gas supply/return for the hydrogen JT cooler system.

CONCLUSION

We have developed detailed models for the design of charcoal-helium based vibration-free Joule-Thomson cryocoolers. The simpler Excel based model allows us to quickly choose design parameters for an efficient cooler given the design constraints, and the MathCad model provides detailed calculations to determine sizing of the various parts of the system, including the charcoal mass, the compressor housing, the counterflow heat exchanger, and the gas-gap heat switch. The calculations can be done quickly and the model can be readily modified to match design changes as they arise. As we begin hardware development we will compare the predictions to the observed results and modify the model as necessary. We have used these models to develop a crycooler concept for a mid-infrared camera instrument on the Next Generation Space Telescope.

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