

## DEVELOPMENT OF A MICROMINIATURE SORPTION COOLER

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## ABSTRACT

Many low temperature electronic devices require only a small cooling power, in the range of 10 mW. For such devices we aim to realise a microcooler, for operating temperatures below 80 K.

A sorption compressor in combination with Joule Thomson expansion was selected for miniaturisation. The advantage of this system is the absence of moving parts, except for some check valves. This facilitates scaling down of the system, it minimises interferences, and it contributes to a long life time - all of major importance for space applications as well. In this paper system and scaling considerations are presented.

## 1. INTRODUCTION

Low temperatures provide an excellent operating environment for conventional and for superconducting electronics. It increases the speed of digital systems, it improves the signal to noise ratio and the bandwidth of analog devices and sensors, and it reduces aging. For superconducting electronics, it is an essential condition that the operating temperature is well below the critical temperature of the superconductor. There is a broad range of applications making use of cold electronics, such as computers, amplifiers, mixers, fast AD and DA converters, IR detectors and SQUID magnetometers<sup>1</sup>. Many of these systems need only a modest cooling power, as low as 10 mW or even lower. A range of cooling techniques is available for cooling such devices, but these cooling techniques are often largely oversized. For that reason the development of a *microcooler* would be of great use for low temperature electronics.

In order to lower mission costs of space flights, the reduction of weight, size and power requirements of spacecraft components is of course an important topic. Therefore, in line with the common trend of miniaturisation, the development of a microcooler is also an interesting subject in this field.

Walker<sup>1</sup> has given an overview of small scale cryocoolers. The most striking results in this field were obtained by Little who scaled down the conventional Joule Thomson cold stage to miniature size by patterning gas channels in glass layers using photolithographic techniques<sup>2</sup>. This cooler, however, has the disadvantage that high pressure gas is needed from a relatively large storage bottle. To our knowledge, the smallest closed cycle cooler realised so far is manufactured by Inframetics<sup>3</sup>. This company makes integral Stirling coolers of 0.3 kg weight, 9 cm maximum size, and 0.15 W cooling power at 80 K with an input of 3 W. A further reduction in size will be limited by the manufacturing techniques, and in this respect micromechanical techniques have been suggested by Walker as an alternative to build micro-scale cryocoolers<sup>4</sup>.

In the microcooling project at the University of Twente we aim to realise a small closed-cycle cooler with a cooling power in the range of 5 to 50 mW for operating temperatures below 80 K. The requirements are set for earth use, but the results might be very useful for space applications as well. Because micromechanical techniques will be an important tool for the realisation of some cooler-components, the project is carried out in co-operation with the micromechanics group of the University of Twente. The potential of micromechanical technologies for small coolers was treated elsewhere<sup>5</sup>.

A sorption compressor in combination with Joule Thomson expansion was selected for miniaturisation. In this paper, system and scaling aspects are treated. Firstly, the main requirements of the system are discussed. Secondly, arguments are given for the choice to miniaturise a sorption cooler. Different methods to reach 75 K with a sorption compressor are compared. Thirdly, a method is given to determine some of the major parameter settings of the system to get a reasonable efficiency. Fourthly, scaling aspects for the sorption compressor are treated.

## 2. REQUIREMENTS

The requirements for small scale cooling of low temperature electronics were chosen as follows:

1. *Working temperature: 60 - 80 K.* Our first aim is cooling of HTc superconducting devices, that operate around LN<sub>2</sub> temperature. The temperature of conventional electronics is not very critical. In general, it should not be lower than 40 K because of carrier freeze-out below that temperature. Therefore, LN<sub>2</sub> temperature is very suitable for many applications, also for hybrid electronics. For other applications, however, lower or higher temperatures may be required.

2. *Cooling power (net): 5 - 50 mW.* The net required cooling power is determined by three factors: (a) the power dissipated in the cold device, (b) the heat leakage from the environment and (c) the desired cool-down time. Each of these factors is discussed below.

(a) The power dissipated by superconducting electronics is very small, even a microprocessor with thousands of junctions on a substrate of 5 mm x 5 mm may dissipate less than a few milliwatts<sup>6</sup>. Dissipation of conventional electronics is more severe and, therefore, only small circuitry can be cooled. (b) The heat leakage is determined by conduction through the supporting structure and the wires, and by radiation. Conduction through the wires and radiation do not play a major role. As an example, one manganin wire between 80 K and 300 K with a length of 10 cm and a diameter of 100 μm gives a heat leakage of 0.2 mW. Further, radiation at 60 - 80 K can be reduced to roughly 1 mW via appropriate shielding. Note that the required *gross* cooling power may be much higher in order to deal with losses by conduction through the cooler itself, which puts severe restrictions on the design of the cooler. (c) The cool-

down time is determined by the combination of the heat capacity at the cold side and the cooling power. The specified net cooling power is for the stable end temperature, at higher temperatures usually a higher cooling power is available which decreases the cool-down time.

3. *Closed cycle with long life-time.* We aim to realise a life-time of several years. This puts restrictions on the use of moving components, especially if friction is involved.

4. *Low electromagnetic and mechanical interference.* This is an important restriction if the cooler is being used in combination with HTc superconducting devices.

5. Other issues of interest are: *low heat rejection (i.e. high efficiency), low price.*

## 3. SORPTION COMPRESSOR + JT EXPANSION

Commonly used cryocoolers can be divided in recuperative (circulating gas flow: JT, Brayton) and regenerative (oscillating gas flow: Stirling, GM, pulse tube) cycles. Most of these cycles use the reversible expansion of an ideal gas, performing work on the environment and taking up heat at low temperature. Only Joule Thomson expansion is an irreversible process, performing *internal* work on a non-ideal gas. For microcooling one has to search for cooling principles which are particularly suitable if applied on micro-scale.

A regenerative cooling system may be a candidate if the pistons can be replaced by miniaturised membranes. In the micromechanical field much experience exists on these kind of membranes<sup>5</sup>. However, highly efficient regenerators have to be used

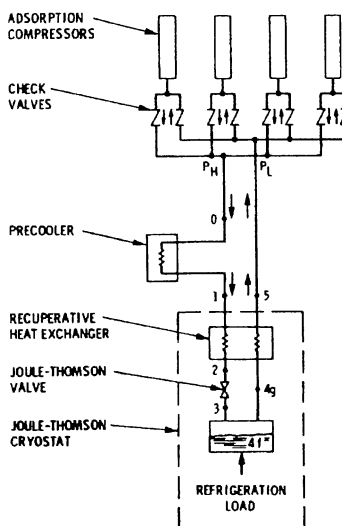


Figure 1a. Sorption cooler<sup>1</sup>

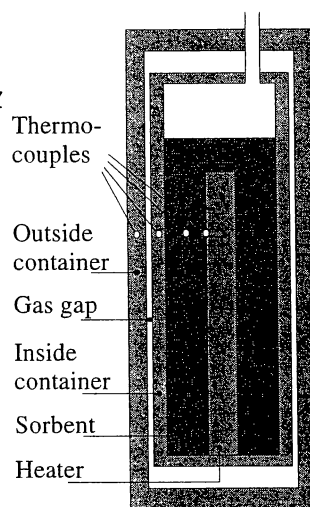


Figure 1b. Sorption compressor

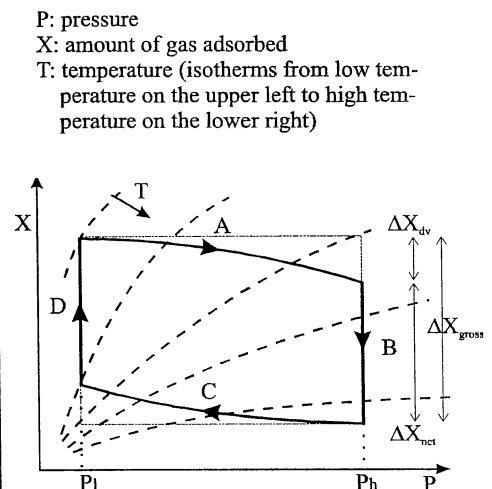


Figure 1c. Compressor cycle

because of the relatively low pressure differences that can be realised with membranes.

In our opinion a very good candidate for miniaturisation is a Joule-Thomson stage fed by a sorption compressor (see figure 1a). The system works as follows. It consists of a compressor unit, a counterflow heat exchanger, and a Joule Thomson expansion valve. Compressed gas coming out of the compressor unit is precooled to the precooling temperature after which it is fed into the recuperative heat exchanger.

The compressor unit contains four sorption cells and several check valves to control the gas flows. Figure 1b gives a sketch of a sorption compressor<sup>7</sup>. Low and high pressures are generated by the cyclic ad- and desorption of a working gas on a sorption material, which is accomplished by cooling and heating of the sorption material. The gas can either be physically adsorbed onto or chemically absorbed into various solids. Heating occurs by an electrical heater, cooling is done with a (gas gap) heat-switch between the sorption cell and a cold source on the outside. A compressor cycle is schematically drawn in figure 1c. The compressor is heated during sections A and B, and cooled during C and D. During sections A and C both valves of the compressor are closed, and the compressor is in a regenerating phase. During sections B and D one of the valves is opened; the compressor generates a high pressure gas flow during B, and a low pressure gas flow into the compressor during D.

Except for some check valves, the cooler has no moving parts in it. This minimises electromagnetic and mechanical interferences, it contributes to a long lifetime (possibly decades), and it also facilitates a scaling down of the sorption compressor. Moreover, reasonable efficiencies can be obtained<sup>8</sup> that are not limited by scaling of the system. These are all important arguments to look at miniaturisation of this cooler-concept, and advantages for space applications as well. An additional advantage for space applications is that a sorption cooler might be powered directly by heat input from a solar concentrator or by waste-heat from a radioisotope power source.

Little has proved that miniaturisation of the counterflow heat-exchanger and JT-stage is possible<sup>3</sup>. Clogging of the JT-stage might form a problem for long term operation. However, micromechanical patterning techniques might facilitate a unique solution to this problem. One way can be to shape the nozzle in such a way that clogging is less likely<sup>9</sup>. Another way can be to incorporate a very small heater close to the JT-nozzle. Cloggings can then be melted by heating the nozzle locally for a short while.

A microminiature high pressure check-valve manifold will be a critical part of the system. Silicon appears to be a very suitable material for these valves; it can be shaped on micron-scale using micromechanical

techniques, and very strong constructions can be realised if properly designed<sup>5</sup>. In the micromechanical field many different kinds of active and passive valves have been realised so far<sup>10</sup>.

Very important for a sorption cooler is the choice of a suitable gas/sorber combination to reach the desired low temperature. Much research in the selection of gas/sorber combinations has been done at Jet Propulsion Laboratory, California. Actually, Vickers of JPL proposed in 1963 to use sorption compression for a JT refrigeration process. At JPL many gas/sorber combinations were analysed to determine the feasibility in JT refrigeration systems.

Four different options will be considered below: O<sub>2</sub>, N<sub>2</sub>, gas mixtures, and a new idea: the so called *Kleemenko* cycle in combination with sorption.

O<sub>2</sub> and N<sub>2</sub> appeared to be two suitable single gas components to reach 70 - 80 K by means of adsorption<sup>9,11</sup>. O<sub>2</sub> can chemically be adsorbed onto praseodymium cerium oxides (Pr<sub>1-n</sub>Ce<sub>n</sub>O<sub>x</sub>, PCO), originally discovered by Union Carbide as a means to chemically separate oxygen from air. The absorption reaction was found to be fully reversible in the range of pressures of interest and at temperatures below 750 °C<sup>12</sup>. A cooler was built that was based on this system. The cooler ran for 44000 hours without showing any degradation of the material, and without contamination.

N<sub>2</sub> can be used in combination with a physical adsorber, for instance activated charcoal. Such microporous materials have a very high density surface area. Physical adsorption of gases on charcoal is not limited to N<sub>2</sub>, other gases can also be adsorbed. Binding of the gas molecules is based on the relatively weak van der Waals forces between gas molecules and the sorption material. As a consequence, only significant amounts of gas can be adsorbed if the temperature of the adsorber is not much higher than the critical temperature of the gas. For instance, in order to bind significant amounts of N<sub>2</sub> gas, the temperature of the adsorber should be less than 200 K. This in contrast to chemical absorption, where the absorption temperature is dependent on the chemistry involved. For PCO the oxygen is absorbed and desorbed around 600 and 900 K respectively. This means that no precooling of the compressors is desired, in contrast to the physical adsorption of nitrogen that occurs at relatively low temperatures.

Another interesting option is the use of gas mixtures. Alfeev showed that the thermodynamic efficiency of a JT system can be increased dramatically by using some additives into the bulk working fluid<sup>13</sup>. The additive generally has a higher isothermal integral throttling effect and a higher boiling point than the working fluid. Mixtures have been optimised so that thermodynamic efficiencies were obtained close to the intrinsic Carnot efficiency. The feasibility of gas

mixtures compressed by a conventional oil-compressor in a closed cycle JT refrigerator is demonstrated at several institutes already. Mixed gases in combination with a sorption compressor might offer much potential for a simple microcooler. However, it is likely that demixing of the different gasses will occur due to the different affinities with the adsorber material. Apart from a reduction of the efficiency this will also lead to an instable cold end temperature, because of its sensitivity to the gas-composition. Therefore, more specific research is needed towards sorption of gas mixtures.

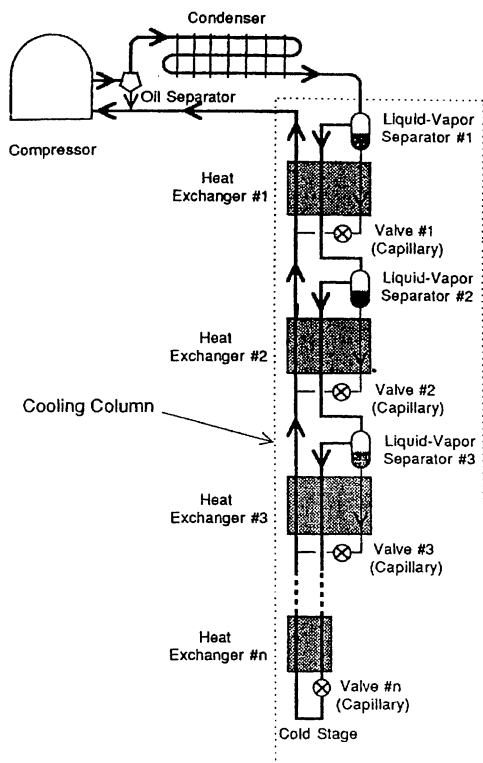


Figure 2. Sketch of the Kleemenko cycle<sup>15</sup>.

Some of the problems mentioned above might be overcome by using a sorption compressor with gas mixtures in combination with the *Kleemenko* cycle (see figure 2). This cycle was first proposed by Kleemenko<sup>14</sup> and is currently investigated by Little for cooling of electronic devices<sup>15</sup>. He uses a conventional oil-lubricated compressor where a gas mixture of light hydrocarbons with nitrogen or argon is compressed to roughly 15 bar. Each component of this gas mixture is then expanded separately from high to low temperature. This is made possible by trapping a liquefied fraction in a phase-separator, after which expansion takes place of this single fraction.

The Kleemenko cycle has some advantages relative to normal JT expansion when both are used in

combination with gas mixtures and a sorption compressor. Firstly, the cold end temperature is less sensitive for demixing in the sorption compressor, because at that position only the component with the lowest boiling temperature is expanded. This in contrast to a normal JT cooler where the complete mixture is expanded through one nozzle at the cold end. Secondly, the system is less susceptible to clogging because gas components with higher boiling temperatures (contamination!) are separated in the first stages.

Comparing the four options to reach 60 - 80 K, it is clear that more research is needed towards the adsorption of gas mixtures before this concept can be used for the development of an application such as a microcooler.

If the PCO/O<sub>2</sub> compressor is compared with the C/N<sub>2</sub> compressor, the PCO/O<sub>2</sub> combination is favourable. Both gases have to be precooled below 200 K in order to get a reasonable JT efficiency, but the required cooling power for this is very small in comparison with the required cooling power to precool the nitrogen compressors. Because the oxygen compressor does not have to be precooled at all (the waste-heat of this compressor can even be used to power an adsorption compressor to precool the O<sub>2</sub> gas), a total cooler from room temperature to 60 - 80 K using a PCO/O<sub>2</sub> compressor as a lower stage will be a couple of times more efficient than a cooler with a C/N<sub>2</sub> stage.

However, knowledge gained about a physical adsorption compressor with N<sub>2</sub> gas can easily be transferred to the physical adsorption of different gasses with corresponding different cold temperatures (such as ethane, methane, krypton, xenon, hydrogen, helium), whereas knowledge gained about a PCO/O<sub>2</sub> stage can probably not easily be transferred to physical sorption compressors. For that reason the charcoal/nitrogen compressor is selected for a first development of a miniaturised cold stage.

#### 4. PARAMETER STUDY

The obtained cooling power of a sorption stage relative to the input power is very dependent on the choice of the compressor temperatures and the gas pressures. In order to be able to determine the desired settings, a model is presented that gives some insight in the relationship between the various parameters. This model is partly based on the work of Chan<sup>16</sup>. It is extended with some extra assumptions; the influence of the dead volume in the sorber material is taken into consideration and a different parameter study is done.

Some essential assumptions are:

- The sorption cells are heated and cooled uniformly, so no temperature profiles exist inside the cells.
- The dead volume above the sorption material in the container plus the dead volume in the tubings between the sorption cell and the check valves is

neglected relative to the intrinsic dead volume of the sorption material itself.

- A cylindrical container is chosen for the sorption cells. The aspect ratio and the construction is chosen such that the influence of the thermal mass of the end caps, heaters, thermocouples and other construction materials is very small relative to the thermal mass of the cylindrical container wall.
- The heat exchanger is considered to be ideal, and no losses exist due to radiation and conduction to the cold end.

The cooling power of a JT system is given by<sup>1</sup>

$$\dot{Q}_C = \dot{m}\Delta h \quad (1)$$

where  $\dot{m}$  is the mass flow and  $\Delta h$  the enthalpy difference between the high and low pressure gas at the warm end of the heat exchanger. The cooling energy during one cycle of a sorption cell can be expressed as:

$$Q_C = m_s \Delta X_{net} \Delta h \quad (2)$$

where  $m_s$  is the mass of the adsorber and  $\Delta X_{net}$  is the mass of the gas liberated from the sorption cell relative to the mass of the sorber material, corrected for the gas that remains in the dead volume of the sorber and that does not take part in the mass flow to the cold end (see figure 1c). Hence,

$$\Delta X_{net} = \Delta X_{gross} - \Delta X_{dv} \quad (3)$$

$$\Delta X_{dv} = \alpha \frac{\rho_{gas}}{\rho_s} \quad (4)$$

where  $\Delta X_{dv}$  is the gas mass that remains in the dead volume per mass sorber,  $\alpha$  is the fraction of the dead volume in the sorber material (for charcoal typically 40 - 50 %),  $\rho_{gas}$  is the high pressure density of the gas and  $\rho_s$  is the density of the sorber material.

The energy required to liberate the gas is determined by heat that is put in the total heat capacity, plus the desorption energy:

$$Q_{in} = (m_s c_{p,s} + m_c c_{p,c}) \Delta T + m_s \Delta X_{gross} \Delta U \quad (5)$$

where  $c_{p,s}$  and  $c_{p,c}$  stand for the heat capacity of the sorber and the container,  $m_c$  is the mass of the container,  $\Delta T$  is the temperature difference of the cycle and  $\Delta U$  is the desorption energy of the gas/sorber combination. The mass of the container is determined by the wall-thickness that is required to stand the high pressures. For a cylindrical configuration this thickness was found to be:

$$d = \frac{p_h R}{\sigma_{max}} \quad (6)$$

In this expression  $p_h$  is the high pressure,  $R$  is the radius of the cylinder and  $\sigma_{max}$  is the maximum tensile stress in the container material. With this expression the ratio of the container-sorber mass can be determined:

$$X_m = \frac{m_c}{m_s} = \frac{\rho_c}{\rho_s} \cdot \frac{2 p_h}{\sigma_{max}} \quad (7)$$

As can be seen, this ratio is independent of the

mass or size itself. Combining expressions 3,4,5 and 7, the input power follows as:

$$Q_{in} = m_s \left[ (X_m c_{p,c} + c_{p,s}) + \Delta X_{gross} \Delta U \right] \quad (8)$$

Now the coefficient of performance (COP) follows by dividing expressions 2 and 8:

$$COP = \frac{Q_c}{Q_{in}} = \frac{(\Delta X_{gross} - \alpha \frac{\rho_{gas}}{\rho_s}) \Delta h}{(X_m c_{p,c} + c_{p,s}) + \Delta X_{gross} \Delta U} \quad (9)$$

Notice that this COP is put together by the efficiency of the sorption process and by the intrinsic JT efficiency of the gas. A poor COP can be caused by one of these efficiencies, although both are related to each other.

In the next part the high and low pressures are denoted by  $p_h$  and  $p_l$ , and the high and low temperatures of the compressor are named  $T_h$  and  $T_l$ .  $\Delta X_{gross}$  in expression 9 is dependent on  $p_h$ ,  $p_l$ ,  $T_h$  and  $T_l$ ;  $\rho_{gas}$  is dependent on  $p_h$  and  $T_h$ ;  $\Delta h$  is dependent on  $p_h$ ,  $p_l$ , and the precooling temperature of the gas (which may or may not equal  $T_l$ ); and  $X_m$  is dependent on  $p_h$ . In principle the temperatures and pressures can now be varied to look for the behaviour of the COP. However,  $p_l$  will be taken constant because this value determines the cold end temperature which follows from the requirements. At first, also  $T_l$  will be taken constant, because this parameter determines the precooling temperature and, therefore, has much influence on the choice of the precooling stage. Now  $T_h$  and  $p_h$  can be varied to determine an optimum for the COP. The adsorption isotherms necessary to calculate  $\Delta X$  originate from experimental work, see also Huinink et al.<sup>7</sup>. As an illustration, figure 3 gives the COP for a C/N<sub>2</sub> stage, with  $p_l = 1$  bar and  $T_l = 180$  K. The observed behaviour can qualitatively be explained as follows.

If the value of the high pressure is increased while the high temperature of the compressor remains constant, less gas will be liberated from the sorption cell. However, if the pressure does not become too high, this behaviour is more than compensated by an increase

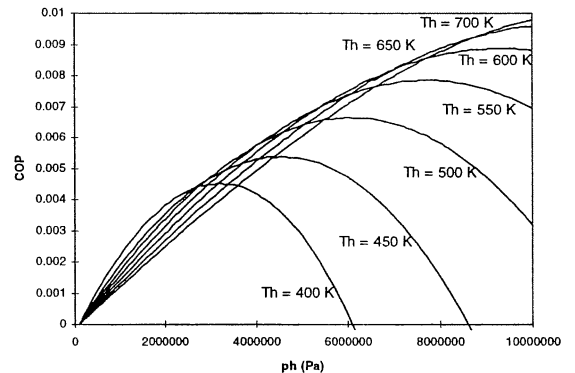


Figure 3. COP as a function of  $p_h$  for different  $T_h$ .

of the enthalpy difference causing the *COP* to rise. At higher pressures this is not the case anymore, and at a certain high pressure no gas is liberated from the compressor any more, causing the decrease of the *COP* to zero. This decrease is accelerated by the relatively large amount of gas that remains in the dead volume at higher pressures.

When increasing the high temperature of the compressor two things happen: At first more gas is liberated from the sorber at higher pressures. Secondly, the amount of gas stored in the dead volume decreases due to the decreasing density of the gas at higher temperatures. This causes the *COP* to rise, up to a certain value of  $T_h$ .

If more of these plots are calculated for decreasing values of the tensile strength of the container material, it is found that the *COP* decreases and that the optimum shifts to lower temperatures and pressures of the compressor. This is caused by the increase of the amount of heat stored in the thicker container wall, especially at higher pressures. This can be understood by looking at the number

$$X_{m,c,p,c} = \frac{2p_h \cdot \rho_c c_{p,c}}{\rho_s \cdot \sigma_{max}} \quad (10)$$

in expression 9. The part  $\rho_c c_{p,c} / \sigma_{max}$  is a measure of the amount of heat that is stored in the container of the sorption cells, it increases with increasing  $p_h$  and decreasing  $\sigma_{max}$ . It can be compared for different container materials. Much difference was found between different materials.

Another material parameter that can play an important role is the dead volume fraction  $\alpha$ .

The procedure to determine the *COP* can be followed for different precooling temperatures  $T_l$ , resulting in an optimum *COP*<sub>max</sub> at a certain value of  $p_h$  and  $T_h$ , for each value of the precooling temperature  $T_l$ . This can be used in the selection of a precooling stage, so that the properties of the total system match the needs of the user.

Table 1 gives some indicative values of the *COP*<sub>max</sub> for different values of  $T_l$ . The numbers for  $p_h$  and  $T_h$ , belonging to the optimum *COP*<sub>max</sub>, are also given.

Table 1. *COP*<sub>max</sub> for varying  $T_l$ ;  $p_h$  and  $T_h$  at *COP*<sub>max</sub>

$T_l$ (K)	<i>COP</i> <sub>max</sub>	$p_h$ (bar)	$T_h$ (K)
140	0.04	70	500
150	0.03	90	600
160	0.02	95	600
170	0.01	100*	650
180	0.01	100*	700**
190	0.005	100	700
200	0.0025	80	700

\* optimum at  $p_h > 100$  bar

\*\* optimum at  $T_h > 700$  K

If the precooling temperature  $T_l$  is larger than 200 K, only a very limited amount of  $N_2$  is adsorbed and the *COP* decreases rapidly. For that reason it is clear that a precooling stage is required if only a room temperature heat sink is available. Dependent on the environment, different precooling stages might be suitable.

## 5. PRECOOLING STAGES

A qualitative comparison is made between the following stages: Stirling, thermo-electric, radiator and another sorption stage. Table 2 gives some indicative numbers. The precooling temperature  $T_l$  of the  $C/N_2$  stage is varied between 140 and 200 K; the influence of this temperature on the *COP* of the first stage is calculated, as well as the total *COP* of both stages.

*Stirling.* Small Stirling coolers with a cooling power of 1 - 10 W and a low temperature of 30 - 90 K have a *COP* of roughly 5 - 10 % of the Carnot efficiency<sup>1</sup>. In table 2 the *COP* is calculated for 10 % of Carnot. From the total *COP* of the two-stage cooler it appears that it is mostly efficient to take the precooling temperature as low as possible. Actually, the best thing to do is to skip the  $C/N_2$  stage and cool directly to 77 K with the Stirling cooler. The combination of these stages is not a good idea: the advantages of each cooler are cancelled out by the disadvantages of the other.

*Thermo-electric.* TE multistage coolers can reach temperatures of 150 K from room temperature<sup>17</sup>. Due to the absence of moving parts, TE coolers are also suitable for miniaturisation and do have the potential of

Table 2. Comparison between different precooling stages.

2 <sup>nd</sup> stage		1 <sup>st</sup> stage						
$C/N_2$		Stirling		Thermo Electric		radiator	sorption C/ethane	
$T_l$ (K)	<i>COP</i> <sub>max</sub>	<i>COP</i> <sub>Stirling</sub>	<i>COP</i> <sub>tot</sub>	<i>COP</i> <sub>TE</sub>	<i>COP</i> <sub>tot</sub>	Area (m <sup>2</sup> /mW)	<i>COP</i> <sub>ethane</sub>	<i>COP</i> <sub>tot</sub>
140	0.04	0.09	$3.2 \cdot 10^{-3}$	-	-	0.011	-	-
150	0.03	0.10	$2.7 \cdot 10^{-3}$	$1.0 \cdot 10^{-4}$	$2.9 \cdot 10^{-6}$	0.012	-	-
160	0.02	0.11	$1.9 \cdot 10^{-3}$	$7.5 \cdot 10^{-4}$	$1.5 \cdot 10^{-5}$	0.013	$1 \cdot 10^{-3}$	$2.0 \cdot 10^{-5}$
170	0.01	0.13	$1.2 \cdot 10^{-3}$	$2 \cdot 10^{-3}$	$2.0 \cdot 10^{-5}$	0.021	0.04	$3.8 \cdot 10^{-4}$
180	0.01	0.15	$1.2 \cdot 10^{-3}$	$5 \cdot 10^{-3}$	$5.0 \cdot 10^{-5}$	0.017	0.10	$9.0 \cdot 10^{-4}$
190	0.005	0.17	$7 \cdot 10^{-4}$	0.015	$7.4 \cdot 10^{-5}$	0.027	0.14	$6.1 \cdot 10^{-4}$
200	0.0025	0.20	$4 \cdot 10^{-4}$	0.020	$5.0 \cdot 10^{-5}$	0.044	-	-

a long life-time. However, the *COP* of these coolers is very low which make them unsuitable for a precooling stage of the compressor cells. On the other hand, TE coolers offer much potential for lowering the precooling temperature of the gas some tens of degrees below  $T_l$  of the sorption compressor. This can increase the JT efficiency of the gas with a factor of two or three, and costs only a fraction of the power that is put in the sorption compressor, resulting in an overall increase of the total *COP*.

*Radiator.* For space applications a passive radiator can be used as a precooling stage. Consider the cooling power of the radiator is given by  $Q_c = 0.3 \cdot Q_{c,max}$ , where  $Q_{c,max}$  is the theoretical maximum cooling power that can be obtained if the emissivity equals one<sup>1</sup>. Now the required radiation area per mW of cooling power at the cold end of the C/N<sub>2</sub> stage can be calculated for each precooling temperature. It can be shown that the smallest area is obtained if the radiator cools directly to 77 K, similar to the situation of the Stirling cooler. However, the difference is small, and if TE precooling of the gas is used, it will probably result in a smaller radiator when the C/N<sub>2</sub> stage is used. Because it is also easier to construct a radiator at 150 K than one at 77 K, this might be a good option for space applications.

*Sorption.* This is of course the most logical option, because the advantages of the sorption concept are used in both stages of the cooler. Sorber/gas combinations exist that have a reasonable *COP* for cooling to 160 K from room temperature, such as ethane or xenon on active charcoal. For the combination C/ethane some values of the *COP* are estimated for different values of the precooling temperature  $T_l$  (this can be done by varying  $p_l$ ). An optimum exists for the total *COP* of this cooler-combination. For Xenon it is expected that much better numbers can be found<sup>11</sup>. Also, improvement of factors can be obtained by the mentioned use of TE precooling of the gas. Another important improvement of a factor of two can be obtained by regeneration of the heat of the sorption cells: one cell can be heated halfway by the waste-heat of another cell<sup>18</sup>. On the other hand, if a very small cooling power around 10 mW is required, and a power supply is nearby, an input power of 10 W will not pose a problem for many applications.

## 6. SCALING

In this section some brief notes are given about scaling of the sorption compressor.

The *COP* in expression 9 is independent of size. Of course there will be limitations to this ideal behaviour if parasitic effects become important that were set aside in the modelling. For the miniature application that we have in mind it is important to search for these limits, and find out what factors can determine the minimum compressor size, given a certain required cooling power.

*General scaling behaviour.* The desired gas

pressures follow from the parameter study (section 4), and are chosen by arguments that are not related to the compressor size. This choice fixes  $\Delta h$  in expression 1. Given that choice, the massflow is directly proportional to the cooling power. If we now focus on part B of the sorption cycle where high pressure gas is flowing out of the sorption cell (see figure 1c), for the massflow can be written:

$$\dot{m} = \frac{m_s \Delta X_{net}}{t_B} \quad (11)$$

where  $t_B$  is the time that high pressure gas flows out of the cell. It follows that the massflow can be kept constant by scaling the flow-time  $t_B$  proportionally with the sorber mass  $m_s$ . The same follows for the total cycle time,  $t_{cycle} = 4t_B$ . This is caused by the synchronisation requirement: the four different periods of the four sorption cells should be synchronised in order to get a continuous gas-flow in time.

*Temperature-profiles.* The sorber material in the container has a finite heat conductivity. This will cause unwanted T-profiles to be developed if the cells are heated and cooled too fast. For a certain given mass-flow, this problem becomes more seriously for smaller sizes of the compressor (and, therefore, faster cycling). This behaviour was modelled, and it was concluded this will not be a serious problem for the parameters we have in mind.

*Dead volumes.* The dead volumes between the sorber material and the check valves need to be filled with high pressure gas each cycle. This gas does not take part in the cooling process. In order to reduce this influence, the amount of gas in this volume should be small compared to the amount of gas that takes part in the cooling cycle. This puts restrictions on the design of the check-valve manifold and connections to the compressor cells, relative to the size of the cells itself.

*Heat-switch thermal resistance in on-state.* An upper limit for this resistance is defined by the need to cool the compressor within half of  $t_{cycle}$  from  $T_h$  to  $T_L + \Delta T$ , where  $\Delta T$  is the temperature difference between the heat-sink temperature  $T_L$  and the lower compressor temperature (a larger  $\Delta T$  has a strong negative influence on the amount of gas that can be adsorbed and, therefore,  $\Delta T$  should be kept as small as possible!). This cooldown time is proportional to the RC behaviour of the thermal mass of the compressor and the thermal resistance to the heat-sink. Because for a given mass-flow the required cooldown time scales linearly with the thermal mass, the required thermal resistance of the heat-switch in the on-state is in first order *independent* of the compressor-size. If a gasgap heat-switch is used with a fixed heat conductivity per unit area, it follows that the required area of the heat-switch is proportional to the mass-flow and independent of the compressor size. This puts fundamental restrictions on the compressor size.

*Aspect ratio.* For a given mass in a cylindrical geometry, an increase of the aspect ratio has three influences. Firstly, it reduces the influence of the thermal mass of the end caps. Secondly, it increases the outside area of the sorption cell. This area might be used as a part of the gasgap heat-switch, and thereby influence the properties. Thirdly, it reduces the radial T-profiles.

*Speed of the heat-switch.* Going to very small sizes and cycling times (typically: minutes), the speed of the heat-switch may become a limiting factor, especially if a gasgap heat-switch with adsorption pump is used.

*Feasibility of the construction.* This is of course a topic of major importance. For instance, for a small diameter of the sorption cell, the wall-thickness becomes very thin. This thickness should fit within the possibilities of realisation (laser-welding for instance).

## 7. CONCLUSIONS

An interesting field exists in the combination of micromechanical techniques and small-scale cooling. A sorption compressor in combination with Joule Thomson expansion was found to be particularly suitable for miniaturisation. The combination of O<sub>2</sub> with chemisorption is a good candidate to reach about 70 K; also N<sub>2</sub> with physisorption can be used, although precooling of the compressor is required.

As a precooling stage another sorption cooler may be used, and for a small cooler a reasonable total efficiency can be obtained.

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