Sorption compressor efficiencies of different designs

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Abstract. Sorption compressors do not have moving parts and are suitable for driving Joule-Thomson cryocoolers. This allows a complete cryogenic cooling system with the absence of moving parts, which is attractive especially for space applications, as well as for other applications. Sorption compressors are thermally driven, meaning, they operate in heating and cooling cycles. The main drawback of sorption compressors is their low efficiency, relative to mechanical compressors. The compressor efficiency is defined as the PV power divided by the heat which is supplied to the compressor, it is limited by the Carnot efficiency, and practically it is significantly lower. In addition, we define a thermal efficiency of a compressor, by the heat which is transferred to the adsorbent divided by the total heat which is supplied to the compressor. In the frame of our ongoing research on sorption applications we develop a numerical model for the heat and mass transfer in a sorption cell. The model allows investigating the performance of a variety of sorption cell configurations, including different geometries, dimensions, and materials. In the current paper we show preliminary results of sorption compressor efficiencies for different sorption cell designs. An investigation on the heater configuration, general dimensions, and different adsorbents is presented. The results show that a sorption cell configuration, which provides a maximum compressor efficiency, is not necessarily the cell configuration which provides the maximum thermal efficiency.

1. Introduction
Sorption compressors are thermally driven and have no moving parts, except for some check valves. Driving a Joule-Thomson (JT) cryocooler with a sorption cycle provides an entire cryogenic cooling system which is vibration-free and benefit from high reliability and long operating life. However, the main drawback of sorption compressors is their relatively low efficiency which is mainly governed by a significant temperature distribution in the sorption cells [1-4].

Our research aims for a high efficiency sorption compressor to drive JT cryocoolers in space applications. In a previous publication [5] we introduced a numerical heat transfer model of a sorption cell, where the heater can be of different configurations and several insulation methods can be incorporated. In the current paper we present the results of an advanced numerical model which incorporates the mass transfer and a gas gap heat switch insulator. This paper focuses on the first compression stage, which operates between 0.2 and 0.68 MPa, out of three stages.

2. Method

2.1. Sorption compressor cycle
A sorption compressor cycle consists of four phases; two heating phases and two cooling phases. The sorption cell cycle begins at a low pressure, low temperature and relatively high adsorption.

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concentration. A heating process in a closed volume (compressor’s valves are closed) bring the system to a higher pressure (“compression”). Further heating with a discharge valve open allowing outflow of the working gas, nitrogen in the current research, while maintaining a constant pressure in the cell, until reaching the maximum temperature of the cycle (“discharge”). A cooling process in a closed volume (compressor’s valve are closed again) decreases the pressure to its initial value (“expansion”). Further cooling with the intake valve open bring the system back to the initial state and completes the cycle, as nitrogen flows in and increases the adsorption concentration (“intake”).

2.2. Sorption cell configuration
The current research focuses on cylindrical cells filled with adsorbent pellets, cylindrical heaters, and a gas-gap heat switch (GGHS). The working fluid, nitrogen, in the sorption cells appears in two phases; adsorbed phase and gaseous phase. The gaseous nitrogen exists in the void volumes, which include gaps between adsorbent pellets, adsorbent macro pores, tubes, and check valves. While in previous work [5] we investigated different techniques for heating, cooling and insulating the cells, we currently present a study on a sorption cell configuration which complies with the schematic cross-section view in figure 1, where the different sections are indicated and \( r_{ij} \) is the radius of the heater. In the current research the adsorbent is a pelleted activated carbon adsorbent, which is produced by Chemviron-Carbon™.

2.3. A numerical heat and mass transfer model
A model which describes the thermodynamic processes of the sorption cell is required for the development and optimization of sorption compressors. In the current research, a one-dimensional numerical heat and mass transfer model is developed and implemented in MATLAB™ code. The heat transfer part of the model has been already validated against reported results [5]. The model is governed by energy and mass equations, where a uniform pressure distribution is assumed in the cell, due to slow flow processes. The differential energy equation for a one-dimensional conduction in cylindrical coordinates is:

\[
\rho \cdot c \cdot \frac{dT}{dt} = k \cdot \frac{1}{r} \cdot \frac{\partial}{\partial r} \left( r \frac{\partial T}{\partial r} \right) + q_{gen}
\]

where \( \rho \) is the density (kg/m³), \( c \) is the specific heat capacity (J/kg-K), \( T \) is the temperature (K), \( t \) is time (sec), \( k \) is the conductive heat transfer coefficient (W/m-K), \( r \) is the radius (m) and \( q_{gen} \) is an internal heat generation (W/m³).

The current model is based on an implicit finite differences method, where every section of the sorption cell is divided to elements. Each element is a lumped capacity and can generate or absorb heat. Therefore, the numerical energy equation for each element is:

\[
\rho_i^n \cdot c_i^n \cdot V_i \cdot \frac{T_i^{n+1} - T_i^n}{\Delta t} = \left( \hat{q}_{gen}(i) \cdot V_i + \sum_{j}^{m} \frac{T_j - T_i}{R_{ij}} \right)_n
\]

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{schematic_cross-section.png}
\caption{a Schematic cross-section view of the sorption cell.}
\end{figure}
where $\Delta t$ is the time step (sec), $n$ is the number of the time step, $m$ is the number of neighboring elements and $R_{ij}$ is the heat resistance between elements $i$ and $j$ (K/W).

The working fluid states are calculated by the Sips adsorption model for the adsorbed phase, and the Peng-Robinson equation of state for the gaseous phase. The adsorption isotherms of nitrogen and the selected activated carbon has been generated in previous work [6,7].

$$C_s = \left( a \cdot p \right)^{n_{ads}} - 1$$

where $C_s$ is the adsorption concentration, $C_{s,0}$ is the saturated adsorption concentration, $p$ is the pressure, (MPa), $a$ is the adsorption affinity (MPa$^{-1}$), and $n_{ads}$ is a dimensionless parameter that qualitatively characterizes the heterogeneity of the adsorbate-adsorbent system.

2.4. Sorption compressor parameters

To be able to compare the performance among different designs, a few parameters are defined. The average temperature of the adsorbent, $ATS$, is calculated at a given time as follows:

$$ATS = \frac{\sum T_i n_i \cdot V_i}{N_s}$$

where $V_s$ is the total adsorbent volume in the cell, $V_i$ is a volume of an element $i$, and $N_s$ is the number of the adsorbent elements. The efficiency of the sorption compressor, $Eff$, is defined as the ratio between the p-V power which the compressor generates and the invested power:

$$Eff = \frac{\dot{V} \cdot \Delta P}{\dot{Q}_{in}}$$

where $\dot{V}$ is volumetric flow rate of the working fluid (m$^3$/sec), $\Delta P$ is the pressure difference (Pa), and $\dot{Q}_{in}$ is the invested power in the cycle (W).

Two additional parameters are determined: volumetric power, $VP$ in (W/m$^3$), and specific power, $SP$ in (W/kg), which are the ratios between the p-V power and the system volume and mass, respectively. These parameters are calculated as follows:

$$VP = \frac{\dot{V} \cdot \Delta P}{V_{sys}}$$

$$SP = \frac{\dot{V} \cdot \Delta P}{m_{sys}}$$

where $V_{sys}$ is the total volume of the system (m$^3$) and $m_{sys}$ is the total mass of the system (kg). The flow rate provided by the compressor is:

$$\dot{m} = \frac{m_{e,discharge}}{t_{cyc}}$$

2.5. Test cases

In this paper we investigate the first stage, of a three-stage sorption compressor, with a low pressure which equals 0.2 MPa and a high pressure which equals 0.68 MPa. Three different vessels are tested, and their details are listed in table 1. All three vessels consist of a cylindrical electrical heater device, a stainless steel vessel, an adsorbent and a gas gap heat switch insulation. Table 2 details the different section properties. The vessel wall thickness is calculated by:

$$Vessel\ wall\ thickness = SF \cdot \frac{p_{high} \cdot r_{in, vessel}}{\sigma}$$

where $\sigma$ is the yield stress (Pa), $p_{high}$ is the maximum pressure allowed in the cell (Pa), $r_{in, vessel}$ is the inner radius of the vessel (m), and $SF$ is a dimensionless safety factor, which in the current research equals 4. According to practical aspects, if the calculated wall thickness is too small, a minimum wall thickness of 0.8 mm is chosen.

Each vessel is examined with varying heater dimensions and different maximum allowed temperatures. The heaters consist of a wire with a thickness of 1 mm and a constant heat flux which equals 13.53
(kW/m²). Therefore, changing the dimensions of the heater changes its total heating power. All vessels are assumed to be made of stainless steel 316L, with a yield stress of 170 MPa. The ambient temperature is 290 K, the low temperature of the sorption cycle is 320 K, and the maximum allowed temperature varies between 600 and 900 K.

In order to investigate the temperature distribution versus time, two heaters are examined for Vessel 3: Heater 1 provides a maximum \( SP \) and \( VP \) and Heater 2 provides a maximum \( Eff \). The dimensions of these two heaters are listed in Table 4.

The numerical model allows investigating every parameter of the sorption cell and every operating condition. A packing density, \( PD \), is defined as the ratio between the adsorbent volume and empty vessel inner volume. The difference between the vessel inner volume and the adsorbent volume is the void volume, which has to be as small as possible. In the current research, a nominal \( PD \) equals 0.56, according to the existing adsorbent in the lab, and additional \( PD \) values are numerically examined. A GGHS is often use in sorption compressors, in order to switch the thermal resistance of the outer layer between heating and cooling phases. A thermal resistance ratio, \( TRR \), is defined as the ratio between the thermal resistances of the GGHS under the low pressure and the high pressure conditions. In the current research, a nominal \( TRR \) value is 38.

### Table 1: vessel dimensions.

|                      | Vessel-1 | Vessel-2 | Vessel-3 |
|----------------------|----------|----------|----------|
| Inner vessel radius (mm) | 10       | 15       | 20       |
| Vessel length (mm)    | 360      | 510      | 660      |
| Vessel volume (cc)    | 113      | 360      | 829      |
| Calculated vessel wall thickness (mm) | 0.39    | 0.58     | 0.78     |
| Actual vessel wall thickness (mm) | 0.8     | 0.8      | 0.8      |

### Table 2: material properties.

| Section: Material: | Adsorbent | Heater | Vessel |
|--------------------|-----------|--------|--------|
|                    | Chemviron | CrNi Steel | SS 316L |
| \( k \left( \frac{W}{m \cdot K} \right) \) | 0.2       | 16     | 20     |
| \( \rho \left( \frac{kg}{m^3} \right) \)  | 400       | 8000   | 8000   |
| \( c \left( \frac{J}{kg \cdot K} \right) \) | 1000      | 500    | 500    |

### Table 3: two heater options for vessel-3 configuration.

| Name | Radius (mm) | Total heating power (W) |
|------|-------------|-------------------------|
| Heater 1 | 16.5 | 925 |
| Heater 2 | 13.5 | 757 |
Table 4: optimal heater dimensions on Vessel-3.

| Configuration | Eff   | SP   | VP   | Mass flow rate |
|---------------|-------|------|------|----------------|
| Heater 1      | 2.70×10^{-4} | 0.116 | 130  | 1.91           |
| Heater 2      | 3.16×10^{-4} | 0.103 | 112.52 | 1.66           |

3. Results and discussion

Figure 2 shows the ATS and pressure for vessel-3, as a function of time, during a single cycle, with a maximum allowed temperature of 800 K, and for the two heaters which are listed in table 3. Heater 1 achieves higher ATS than heater 2, 680 K and 670 K, respectively, and its cycle duration is shorter: 770 sec, relative to 846 sec for Heater 2. Table 4 presents the compressor performances of Heater 1 and Heater 2 configurations. According to table 4, Heater 1 dimensions are optimal for the highest SP, VP, and \(m\), while Heater 2 is optimal for the highest Eff.

Figure 3, 4 and 5 show the results of Vessel-1, Vessel-2, and Vessel-3, respectively. Each figure presents the (a) Eff, (b) SP, (c) VP and (d) \(m\), as a function of the heater dimension, \(r_H\), (see figure 2) at four maximum allowed temperatures: 600, 700, 800, and 900 K. \(r_H\) is the radius of the heater, and its minimum value represents a central rod heater (thus, it doesn’t reach a zeroth value).

For a given vessel, the highest allowed temperature provides the maximum efficiency, with a certain heater dimensions. Moreover, the heater radius which provides the maximum efficiency reduces with the increase of the maximum allowed temperature. The larger vessel provides higher efficiencies, relative to smaller vessel, due to the larger volume to outer surface ratio.

All the three parameters, SP, VP, and \(m\), show similar characteristics for all test cases. An increase in the heater dimension (larger \(r_H\)) increases all three parameters, however, increasing the maximum temperature isn’t necessarily beneficial. In contrast to the efficiency, smaller vessels provide higher SPs, VP s, and flow rates, and moreover, for a given vessel, the heater which provides the maximum efficiency isn’t the heater which provides the maximum SP, VP, and \(m\).

Figure 6 shows that all the parameters increase with the increase of the packing density. The results are presented for a minimum packing density of 0.56, which is the nominal value of the existing adsorbent, and up to 0.68, which is the value that we managed to achieve by compressing the adsorbent in a vessel. Finally, the influence of the TRR of the GGHS is presented. Figure 7 shows the improvement in SP and VP with the increase of TRR. In the current research, the nominal TRR equals 38, and an increase of TRR is defined by decreasing the thermal resistance of the GGHS at the “on” phase, meaning, during the cooling phase of the compressor. Therefore, Eff isn’t influenced by TRR and it isn’t presented in figure 7. One should notice that a higher TRR increases SP and VP, however, the benefit in increasing TRR is getting moderated, and there is most likely a maximum practical TRR which shall be obtained.

Figure 2: a complete cycle of (a) ATS and (b) pressure, as a function of time, for Vessel-3 operating between 320 K and 800 K, with two heaters options.
Figure 3: Vessel-1 (a) $Eff$, (b) $VP$, (c) $SP$, and (d) mass flow rate as a function of the heater dimensions, $r_H$, for maximum allowed temperatures of 600, 700, 800, and 900 K.

Figure 4: Vessel-2 (a) $Eff$, (b) $VP$, (c) $SP$, and (d) mass flow rate as a function of the heater dimensions, $r_H$, for maximum allowed temperatures of 600, 700, 800, and 900 K.
Figure 5: Vessel-3 (a) $Eff$, (b) $VP$, (c) $SP$, and (d) mass flow rate as a function of the heater dimensions, $r_H$, for maximum allowed temperatures of 600, 700, 800, and 900 K.

Figure 6: Vessel-3 (a) $Eff$, (b) $VP$, (c) $SP$, and (d) mass flow rate as a function of the heater dimensions, $r_H$, for different values of packing density, PD.
Figure 7: Vessel-3 (a) SP and (b) VP as a function of the heater dimensions, $r_H$, for different TRRs

4. Conclusions
Research on sorption compressors, for driving Joule-Thomson cryocoolers in space application, is conducted in our lab. One of the main research goals is to increase the sorption compressor efficiency, and therefore, several design approaches are investigated. Currently the investigation is numerical, however, an experimentation apparatus is already under construction, in order to validate the numerical model. The results which are presented in this paper shows the efficiency dependency on the heater dimensions, the maximum temperature of the cycle, the sorption cell volume, and the packing density. In addition to the efficiency, other parameters are also investigated: the specific power, $SP$, the volumetric power, $VP$, and the mass flow rate. The results show that the design for a maximum efficiency differs from the design for maximum $SP$, $VP$, and mass flow rate.

The research consists of an activated carbon which is available in our lab, and nitrogen as the working fluid. The numerical model which is developed, and soon will be validated, allows the investigation of many other working pairs, and for additional applications.

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