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Waste heat driven silica gel/water adsorption cooling in trigeneration

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Abstract
A low-grade waste heat driven solid/vapour adsorption chiller has been successfully designed and tested. A simple model was developed to aid the design and predict the performances. The system comprised two identical sorption reactors operating out of phase in order to ensure continuous cold production. One sorption reactor consisted of six commercially available automotive plate/fin heat exchangers in which silica gel grains were accommodated between the fins. The system was tested as to the power delivered at 12 °C and the power density. The average cooling power was 3.6 kW. This is only 72% of the design value and can be largely attributed to the lower heat transfer fluid flow rate through the sorbent reactor. The thermal efficiency, COP, was 0.62 and the power density was 17 kW/m$^3$ for the system as a whole. Higher power densities are possible. At present, the adsorption chiller is integrated in a prototype trigeneration system, which is tested at CRF’s Eco-building in Turin.

Keywords: Adsorption refrigeration; Energy saving; Heat pumps; Silica gel; Trigeneration; Waste heat

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## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>Heat exchanging surface area ($m^2$)</td>
</tr>
<tr>
<td>$COP$</td>
<td>Coefficient of performance, thermal efficiency</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Heat capacity at constant pressure ($J/mol\cdot K$)</td>
</tr>
<tr>
<td>$\Delta H$</td>
<td>Latent heat of ad-/desorption ($J/mol$)</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>$m_{sorb}$</td>
<td>Sorbent mass (kg)</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure (bar)</td>
</tr>
<tr>
<td>$P$</td>
<td>Power transferred (W)</td>
</tr>
<tr>
<td>$Q, q$</td>
<td>Heat quantity ($J$)</td>
</tr>
<tr>
<td>$R$</td>
<td>Universal gas constant ($8.3145 \text{ J/mol}\cdot \text{K}$)</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$\Delta S$</td>
<td>Entropy change upon ad-/desorption ($J/mol\cdot K$)</td>
</tr>
<tr>
<td>$SCP$</td>
<td>Specific cooling power ($W/kg$ dry sorbent)</td>
</tr>
<tr>
<td>$SWS$</td>
<td>Selective water sorbent</td>
</tr>
<tr>
<td>$t_{cycle}$</td>
<td>Duration of the thermodynamic cycle (s)</td>
</tr>
<tr>
<td>$T$</td>
<td>Absolute temperature (K)</td>
</tr>
<tr>
<td>$UA$</td>
<td>Overall heat transfer coefficient ($W/m^2\cdot K$)</td>
</tr>
<tr>
<td>$W$</td>
<td>Overall heat transfer (W/K)</td>
</tr>
<tr>
<td>$W$</td>
<td>Water content in adsorbent (w-%)</td>
</tr>
</tbody>
</table>

### Subscripts

- $evap$ Evaporator
- $htf$ Heat transfer fluid
- $H$ Concerning high temperature source, i.e. hot-water circuit or waste heat
- $L$ Concerning low temperature source, i.e. chilled-water circuit
- $M$ Concerning middle temperature source, i.e. cooling-water circuit
- $in$ Associated with what is going into the system
- $out$ Associated with what is coming out of the system
- $sorb$ Sorbent

## 1. Introduction

The EU Member States are facing a growing demand for comfort cooling in the residential and commercial area. Nowadays, electric vapour compression heat pumps are the most commonly used machinery in comfort cooling or refrigeration [1]. During energy consumption for cooling duty, greenhouse gases are released into the...
atmosphere. Moreover, common refrigerants in compression chillers, *i.e.* CFC’s, HCFC’s, and HFC’s, are notorious contributors to global warming; some of them have an ozone depleting effect as well. During production, servicing, and recycling part of these substance will end up in the environment. In the future, cooling demand, and hence energy consumption for climate control and refrigeration, will continue to grow. This intrinsically leads to a faster depletion of known fossil fuel reserves, more CO$_2$ emissions and a higher peak electricity demand [2].

Alternatives for vapour compression cooling are already in put into use or are under development. Promising technologies include heat driven liquid absorption refrigeration [3-5] and solid sorption heat pumps [5-8]. These types of cooling machines bring about a lower primary energy usage and lower CO$_2$ emissions per unit cold produced. The present sorption systems, however, are only suitable for large-scale industrial applications (> 100 kW). Research on heat driven solid sorption cooling technology concentrates on the development of cooling machines for smaller applications, such as for example comfort cooling and refrigeration in the build environment [9] and mobile air conditioning. Solid sorption heat pumps do not need any CFC, HCFC or HFC refrigerant, most of them make use of natural refrigerants. Examples of solid sorbent vapour combinations are silica gel/water [10-13], SWS/water [13-15], zeolite/water [16], and NH$_3$/carbon [17-19]. This technology does not have moving parts, is therefore silent, and does not require much servicing.

Solid sorption technology can be deployed either stand-alone or integrated in a trigeneration system [2]. Such a system offers the possibility of saving 15 – 20 % primary energy for cooling, heating, and power demands, by utilisation of the waste
heat of a CHP system for the production of cold by means of solid sorption cooling technology.

In the present paper, the work on modelling, design, and testing of small-scale adsorption chiller prototype in our laboratories are described. The main scientific objective was the realisation of a 5 kW adsorption chiller with a power density of about 20 – 30 kW/m³, where 10 kW/m³ is state-of-the-art [20]. To achieve these targets new concepts for integration of a heat exchanger and solid sorbent material are required to achieve sufficient overall heat transfer. A simple model was developed to aid the design of the adsorption chiller test facility. After laboratory testing the adsorption chiller prototype was integrated in an experimental combined heat and power module trigeneration setup at CRF, Italy. Note that throughout this study all the temperatures mentioned are those of the external heat transfer fluids, unless mentioned otherwise.

2. Model development

2.1. Description of the thermodynamic cycle

Sorption cooling systems operate via a reversible sorption reaction between a solid sorbent and a vapour. This is an equilibrium reaction. The ‘counter reaction’ can be the alternating evaporation and condensation of refrigerant. The physical processes in a silica gel/water system can be represented as follows:

\[ Si-(OH)_x + H_2O(s) \leftrightarrow ^{k(T,p)} Si-(OH)_x(H_2O) + Q \]  
\[ H_2O(s) \leftrightarrow ^{k(T,p)} H_2O(l) + Q'' \]  

(1) 

(2)
The equilibria are dictated by the thermodynamics of the system and depend on local temperature and pressure. Adsorption of water by the silica gel and condensation of water are exothermal processes and generate heat. In a sorption cooling system, this heat is useless and is disposed of. Desorption and evaporation are endothermic, so heat is required to drive the equilibria in this direction. Desorption can be waste heat driven and is part of the regeneration phase. Heat required for evaporation is withdrawn from the environment, so causing the refrigeration effect. The extent of the refrigeration effect is determined by the amount of water circulating between evaporator and sorbent, and thus on the differential water loading of the silica gel, per unit time. The differential water loading of the silica gel depends on the intrinsic interaction of the refrigerant with the sorbent, the temperature, and pressure. When temperature and pressure inside the silica gel bed are known, the water loading in the sorbent ($W$) can be calculated:

$$
\ln\left(\frac{p}{p^*}\right) = \frac{\Delta S}{R} - \frac{\Delta H}{RT} = \left(a_n + a_1 W + a_2 W^2 + a_3 W^3\right) + \left(b_n + b_1 W + b_2 W^2 + b_3 W^3\right)
$$

(3)

The parameters $a_n$ and $b_n$ are characteristics of the sorbent refrigerant interaction and describe a set of isosteres, i.e. lines of constant refrigerant concentration, as a function of temperature, pressure, and refrigerant concentration (Figure 1).

In the ideal situation, in equilibrium the pressure inside the sorbent bed equals the pressure in either the evaporator or the condenser, and the sorbent bed has homogeneously adopted the temperature of the adjacent heat exchanger and heat transfer fluid. If it were possible in the sorbent reactors to alternately reach the cooling-water temperature ($T_M$) and the heating-water temperature ($T_H$) and corresponding evaporator, respectively condenser pressure, the thermodynamic cycle the silica gel/water system is subjected to can be represented by the dotted trapezium in Figure 1.

In horizontal direction ($L_{eq}$) the trapezium crosses several isosteres, therefore the
thermodynamic cycle predicts water flow from and to the silica gel. The more isosteres are crossed, the higher is the equilibrium differential water loading and, accordingly, the higher is the heat effect of that cycle:

\[ q = m_{\text{neq}} \int_{W_s}^{W_d} \Delta H(W) \, dW \]  \hspace{1cm} (4)

being

\[ \Delta H = - \left( b_0 + b_1 W + b_2 W^2 + b_3 W^3 \right) R \]  \hspace{1cm} (5)

and \( W_x, W_y \) the isosteres with maximum and minimum water content of the cycle.

Note that the system cannot be in equilibrium, whereas a pressure difference is required to obtain the necessary refrigerant vapour flow between the different components and a temperature difference is required to enable heat transfer.

Now, during evaporation heat is withdrawn from the refrigerant buffer and thus the temperature inside the evaporator will decrease. On the other hand, due to the heat of condensation the temperature inside the condenser will increase. The temperature effects mainly depend on the rate of evaporation/condensation and the thermal mass of the evaporator/condenser. Because of these temperature effects, the pressure inside the evaporator and condenser will change with respect to the ‘isothermal equilibrium’. The pressure limits in the sorbent bed will change accordingly. In Figure 1 this is represented by a vertical shift of the equilibrium pressure in the evaporator to \( P'_{\text{evap}} \) and in the condenser to \( P'_{\text{cond}} \). Accordingly, the two equilibrium corner-points of the trapezium at \( T_L \) and \( T_H \) are shifted to the positions denoted with an ‘\( \times \)’. In the case there is a non-zero pressure drop in the sorbent bed, the vertical shift of these points is larger and will be located below and above the pressures inside the evaporator and condenser, respectively. In reality, there will also be resistance to heat transfer between the heat transfer fluid and heat exchanger and between the heat exchanger and sorbent bed. This
results in a temperature difference between the sorbent bed and the heat transfer fluid. Moreover, due to a low intrinsic heat conductivity of the silica gel, the temperature in the sorbent bed will not be homogeneous and the average bed temperature will differ from that of silica gel near the heat exchanger. Both temperature effects are represented in Figure 1 by a horizontal shift of the positions marked ‘×’ to the positions indicated by black dots. The now originating trapezium crosses less isosteres: due to temperature effects in the evaporator, condenser and sorbent bed and due to the resistance to heat transfer, the real differential water loading \( L_{\text{real}} \) is reduced with respect to the ideal equilibrium case. The deviation between the equilibrium cycle and the real cycle was calculated via an iterative process starting from experimental data as model input parameters, such as measured heat transfer fluid flows, temperature effects, and calculated UA values (paragraphs 2.2 and 2.3).

2.2. *Experimental data used as model input parameters*

The key component for the adsorption chiller is the sorbent bed heat exchanger. As a building block a 1.4 kg compact aluminium automotive plate/fin type heat exchanger was chosen, through which the heat transfer medium is conducted in three passes (Figure 2). The internal pressure drop was negligible up to water flows of 4.5 L/min but increased exponentially above 5 L/min. A standard flow rate of 4 L/min was adopted \( u_{\text{max}} = 0.2 \, \text{m/s} \). The inner plate distance was 8 mm. Fins were stacked between the plates with a fin pitch of 3.5 mm. The specific surface area of the heat exchanger was estimated at 950 m\(^2\)/m\(^3\) including fin surface and 175 m\(^2\)/m\(^3\) without. With outer-dimensions of 200 × 265 × 65 mm, the total volume of the heat exchanger amounted to 3.5 dm\(^3\). The heat exchanger was filled in between the fins with about 1.45 kg of dry
micro porous silica gel (Ineos, type Sorbsil A) grains of 0.2 – 1.0 mm in diameter, resulting in a sorbent to metal mass ratio of slightly above unity. The grain size distribution was chosen such as to have the highest silica gel loading possible with an acceptably low pressure drop in the sorbent bed. The heat exchanger was enveloped with fine-wired metal gauze to retain the silica gel grains.

In a separate measurement [10], the average cooling performance of a silica gel filled heat exchanger was determined at 990 W under favourable test conditions and a heat transfer fluid flow rate of 4 L/min. This corresponds to 343 W/kg dry sorbent and about 144 kW/m³ of continuous cooling at ~ 20 °C. For a filled heat exchanger of this type, the largest resistance for heat transfer is located within the sorbent layer. As a result, a thermal front will travel through the sorbent layer affecting the local sorbent water loading. As follows, the reaction front is moving away from heat source/sink during the process of heating or cooling of the sorbent. In addition, the amount of water present in the sorbent will have effect on the thermal conductivity of the sorbent bed.

Consequently, in contrast with an empty heat exchanger in vacuo, the overall heat transfer coefficient will not be constant. Under similar test conditions the $U_A$-value of selected empty heat exchanger in vacuo was 895 W/K, while that of a silica gel filled one was ranging between 250 and 350 W/K in the presence of water vapour.

The largest resistance to mass transfer is expected to be located within the packed bed. It was experimentally verified that, in most cases, the pressure loss in the sorbent bed is negligible (< 1 mbar). Therefore, it was considered fair to assume that, in general, the pressure inside the sorbent bed is homogeneous and equals the pressure inside either the evaporator or condenser.
2.3. **Model description**

The purpose of the system model was twofold. Firstly, and most importantly, by estimating cooling power and thermal efficiency, it should contribute to scaling and optimization of the solid sorption reactor design. Secondly, the model should provide an understanding in the system’s response on changes in external parameters, such as temperature levels, mass flow rates, and cycle time. It was not intended for the model to describe the behaviour of the adsorption chiller in much detail.

In the model, the adsorption chiller is considered as a black box in which the only parameters known are the temperatures and flow rates of the heat transfer fluid going into the system, as well as the cycle time that is imposed upon. If power is to be transferred, the temperature levels of the fluid flowing through the adsorption chiller cannot be constant. The overall power transferred can be calculated from the change in temperature, the mass flow rate, and heat capacity of the heat transfer fluid:

\[ P = h_{in} \cdot C_{pf} \cdot (T) \cdot (T_{in} - T_{w}) = h_{out} \cdot C_{pf} \cdot \Delta T_{pf} \]  

(6)

Secondly, the amount of power transferred depends on the driving force, i.e. the temperature difference between the two bodies power is interchanged between, and the efficiency of the heat exchanger, denoted by \( UA \):

\[ P = UA \cdot \Delta T \]  

(7)

The \( UA \)-values for the different heat exchangers can be calculated, either from manufacturer specifications or from experiments (Table 1). In Figure 1 the magnitude of the driving force at the start of each phase is visualized by the horizontal shift of the outermost corner points of the trapezium with respect to the positions in an ideal equilibrium. Note that, in time the driving force will change due to temperature changes in both the sorbent bed and the heat transfer medium.
Thirdly, the power generated (adsorption) or consumed (desorption) by the silica gel depends on the differential water loading and the cycle time:

\[ \text{P}_{\text{net}} = \frac{q}{2T_{\text{cycle}}} \]  

(8)

being \( q \) defined in Eq. (4)

Heat balance requirements states that the power transferred by the heat transfer medium Eq. (6) equals the power transferred by the heat exchanger Eq. (7), which in turn, equals the power transferred in the sorbent bed Eq. (8), corrected for sensible heat losses as a result of the thermal masses of the sorbent, refrigerant, heat transfer medium, and metal mass of the heat exchanger. This set of equations was solved iteratively following the steps described below:

1) Define input temperatures and flow rates for the heat transfer fluids and a fixed cycle time. The initial input temperatures define the ideal sorption cycle and initial conditions for each component. (evaporator, condenser, sorbent reactors)

2) Calculate the equilibrium pressures of the evaporator and the condenser.

3) Derive the equilibrium points of the adsorption and desorption process for the given temperatures and pressures using Eq. (3) and calculate the change in water loading the silica gel. (\( L_{\text{eq}} \) in Figure 1 for the initial calculation).

4) Calculate the thermal powers of each component using Eq. (8).

5) Calculate the new temperatures for each component using Eqs. (6) and (7).

6) Take the new temperatures of step 5 and repeat from step 2 till 5 until the temperature change between the nth and n-1 calculation is less than 0.001°C.

As a result, the model estimates the power transferred between each component and, with that, the corresponding local temperatures. Consequently, the deviancy of the thermodynamic cycle from the ideal equilibrium cycle is known. From the amount of
silica present, the heat transfer fluid flow rate, and the temperature drop in the chilled and heating-water circuit, the SCP and COP were estimated at a fixed cycle time:

\[
SCP = \frac{P_{\text{evap}}}{m_{\text{sorb}}} \tag{9}
\]

\[
COP = \frac{Q_L}{Q_H} \tag{10}
\]

Note that the model is not dynamic, meaning it only holds for none-transient conditions, i.e. constant power, under a set of imposed fixed input values. In real (batch-wise) operation temperature, pressure, and power will constantly change in time. In the model the SCP and COP were calculated from time-independent average values, in which the pressure in the sorbent bed heat exchanger was assumed equal to the pressure in either the evaporator or condenser. The latter assumption was experimentally verified and holds in practically all cases [10].

3. Design prototype adsorption chiller test facility

Figure 3 shows the prototype adsorption chiller as designed, assembled and tested in our laboratory. In brief, the test facility consisted of two identical sorption reactors interlinked with a collective evaporator and a collective condenser. External heating, cooling and chilled-water circuits were connected to mimic operating temperatures and flow rates of the trigenerator system. The basic layout of the test facility shown in Figure 4 includes two sorbent reactors. The sorption reactors must operate in counter-phase to enable a continuous cold production: one reactor is in the cooling-mode while the other is regenerating. At the end of the respective phases heat recovery between both reactors can be applied for short time after which the operation mode of both reactors is reversed. Pneumatically operated vacuum valves were installed to direct water vapour
flows from sorbent to condenser (valves 2 and 4) and from the evaporator back to the sorbent (valves 1 and 3). Within the vapour supply line, a demister was placed to prevent water droplets from entering the reactors. To close the refrigerant loop and to maintain the necessary pressure difference between condenser and evaporator, a U-tube was installed. Both the condenser and evaporator consisted of Vatherus plate/shell heat exchangers with a heat-exchanging surface of 3.1 m² and were designed for a 5 kW thermal capacity. The condenser heat exchanger had an overall heat transfer, or $UA$-value, of 1666 W/K at a cooling-water flow rate of 12 L/min. The evaporator was spray-assisted so as to enhance wetting of the heat exchanger surface area, and thus the total effective heat transfer. At a chilled-water flow rate of 12 L/min the evaporator heat exchanger has a $UA$-value of 1000 W/K. Previous experiments indicated a performance of 990 W of discontinuous cooling for one filled heat exchanger at a heat transfer fluid flow rate of 4 L/min and a minimum of heat losses, measured at favourable temperature conditions [10]. Therefore, from experiments it can be concluded that two assemblies of at least six heat exchangers are needed to obtain 5 kW of continuous cooling. However, it was shown that evaporation of refrigerant resulted in a shift in equilibrium temperature and pressure in the system; the more refrigerant is evaporating per unit time, the lower the temperature in the evaporator will become, and the less isosteres will be crossed in the thermodynamic cycle. Therefore, in the case of fixed evaporator and condenser heat exchangers, each additional heat exchanger will lead to a lower average SCP. This effect cannot be quantified by experiments on one heat exchanger, but was taken into consideration in the model. Moreover, in the model the inlet temperatures and flow rates of the heat transfer fluid flows could be set fixed at any level, independent of external factors. In
simulations the flow rate through each sorbent heat exchanger was set at 4 L/min, the flow rate through evaporator and condenser at 20 L/min. Model calculations on a system consisting of two sets of six heat exchangers and heat transfer fluid inlet temperatures of $T_H$ 90°C, $T_M$ 25°C, and $T_L$ 20°C predicted a performance of 5.2 kW at a cycle time of 10 minutes (COP 0.55). According to the model two assemblies of six heat exchangers will suffice: this is in fair agreement with experimental results.

Integrated in a combined heat and power module, in which the adsorption chiller is driven by low-grade heat from the jacket cooling-water of an internal combustion engine the inlet temperatures will be more in the order $T_H$ 85°C, $T_M$ 25°C, and $T_L$ 15°C. With these temperature levels the model predicted a performance of 4.5 kW at a 10-minute cycle (COP 0.50) and 4.0 kW at a 15-minute cycle (COP 0.55). Note that the temperature levels of the cooling ($T_M$) and chilled ($T_L$) water circuits of the sorption machine are dictated by the chilled and cooling water systems already present at the CRF site. The mentioned cooling water system is connected to an evaporative tower.

Regarding both experimental results and model predictions, a design containing two sorption reactor assemblies with six heat exchangers was chosen. To restrain the internal pressure drop in the reactor assembly as well as to prevent too large a temperature difference over the assembly, the sorbent heat exchangers were connected in parallel with regard to the heat transfer fluid (Figure 5). Since the intended flow rate per heat exchanger is 4 L/min, the total flow rate through the assembly should be 24 L/min. The sorbent heat exchanger assembly and condenser were connected in parallel with respect to the cooling-water, in order to achieve the lowest possible cooling-water temperature for both components.
The test facility was extensively instrumented with flow, temperature, and pressure measurement devices to monitor proper functioning of each component and to measure the thermal powers transferred by the heat-exchanging units in the cooling system.

4. Results and discussion

4.1 Cycle time and heat recovery

To identify the optimum duration of the thermodynamic cycle, the SCP and COP of the adsorption chiller were determined experimentally under varying cycle times. First observation was, that the obtained heat transfer fluid flow rates through the heat exchanging components were divergent with respect to design-values, especially the flow rates through the sorption reactor heat exchanger assemblies were notably lower (Table 2). This certainly will have effect on the performance of the adsorption chiller.

The performance ratings were conducted at $T_H$ 85 °C, $T_M$ 25 °C, and $T_L$ 15 °C. A good compromise between cooling power and efficiency was found at a cycle time of about 15 minutes (Figure 6). Based on these results, a thermal cycle of 15 minutes was proposed consisting of the following steps: 7 minutes of heating followed by a 30-second interval, then 7 minutes of cooling and again a 30-second interval before repeating the whole cycle. At the start of each 30-second interval only the heat transfer fluid flows at the inlet of both reactors were switched; at the end of the intervals, also the flows at the exit were switched. This way, the hot water still present in the heat exchanger stack at the beginning of this interval, is prevented from entering the cooling-water circuit for half a minute, and, similarly, the relative cold cooling-water is prevented from entering the heating-water circuit. Yet, after 30 seconds, when the
temperature streams emerging from both heat exchanger stacks were levelled to some extent, connections to the ‘proper’ circuits were made. With heat recovery, less heat is short-circuited between the cooling and the hot water supply, and therefore the COP of the system benefits from it considerably. It was estimated that due to heat recovery, under these conditions and at a cycle time of 15 minutes, the COP of the system increased from 0.4 to 0.7. For comparison, the COP of a similar system, viz. SWS-1L/water, for water chilling driven by low temperature heat without heat recovery was 0.6 [15].

The SCP was about 250 W/kg dry silica gel. This is 73% of the performance measured with one heat exchanger building block [10]. As mentioned, the SCP will decrease as a result of lower heat transfer fluid flow rate through the reactors, a lower chilled-water loop temperature, higher heat losses compared to the isolated measurement in vacuo, and the average SCP decreases with each additional heat exchanger building block. Moreover, we could not obtain optimal wetting of the evaporator because of poor spray nozzle performance due to cavitations in the refrigerant pump, a result of the low pressure in the evaporator. In a new design, this problem should be addressed.

4.2 Thermal power and chilling temperature profiles

Evidently, in real applications it is important to maintain a constant chilled-water outlet temperature. As important, or even more so, is a constant cooling power. Figure 7 demonstrates the temporal fluctuation of the chilled-water outlet temperature and the instantaneous cooling power of the adsorption chiller. Over the thermal cycle, the outlet temperature of the chilled-water circuit was 8.0 ± 0.5 °C with an average of 7.8 °C, except for a one-minute spike directly after switching (the temperature of) the heat
transfer fluid flows through the sorption reactor assemblies. The maximum temperature of the chilled-water outlet, however, never exceeded 10 °C.

Overall, the chilling power of the evaporator remained stable at 3.5 ± 0.5 kW with an average of 3.5 kW. Switching caused the chilling power to collapse briefly to a minimum of about 2.1 kW. Next to the chilled-water power output, the power extracted from the heat source and the power rejected in the cooling-water circuit, did fluctuate. This may cause problems when integrated in a trigenerator system. Implementation of more reactor beds [12] possibly subjected to a regenerative thermal wave [19,21], the use of thermal buffers, a variable flow rate through the sorbent reactor assemblies, or an active regulation on refrigerant vapour flow, are possible solutions to reduce time-dependent power transfer and temperature fluctuations. The observed periodical change in the measured temperature of the chilled-water inlet in Figure 7 is fictitious and was caused by conduction effects through the metal tubing. The lower performance than the targeted value of 5 kW was caused by aforesaid factors. Some of these factors could be taken into account by changing model input parameters. By imposing experimentally obtained flow rates instead of design values and a chilled-water inlet temperature of 12 °C, the model predicted an average chilling power of 3.6 kW with a COP of 0.62. The model fairly well predicted the experimentally measured value.

4.3 Source temperature dependency

A parameter study of all inlet temperatures was performed to get hold of the system’s response on fluctuations of these variables in terms of SCP and COP. For that purpose, all parameters were fixed at realistic ‘standard’ values (Table 2) except for the inlet temperatures that were varied by turns. The (specific) thermal power transferred in the
evaporator, the refrigerant effect, and COP as function of the cooling-water inlet
temperature are shown in Figure 8 in the case of a chilled-water loop of 12 °C and of 20
°C. Although the model fairly well simulated the temperature dependencies, it slightly
underestimated the SCP and COP, especially at lower cooling-water temperatures. The
differences were larger at the highest evaporator temperature. However, regarding the
simplicity of the model and the intended accuracy, it can be said that it adequately
describes the behaviour of the test facility. In the examined temperature interval, the
SCP decreased almost linearly with increasing cooling-water temperature. The targeted
cooling power of 5 kW was not reached; at standard conditions (Table 2), the average
chilling power was 3.6 kW. This corresponds to a SCP of 208 W/kg dry silica gel.
Considering only the total volume of the heat exchangers in the system, i.e. 62 dm³, a
volumetric power density of 58 kW/m³ was calculated. However, the power density
only remains 17 kW/m³ for the system as a whole. With that, the target of a 20 – 30
kW/m³ power density comes near. The adsorption chiller was a laboratory prototype
and was set-up spaciously, leaving enough room to further compact the system.
Therefore, no technical problems are foreseen to comply with the targets for power
density, in near-future systems.

In Figure 9 the SCP and COP at various heating-water inlet temperatures are shown.
Again, the model predicted a lower performance than experimentally established, but
the trends were similar. Both the performance in terms of cooling power and efficiency
changed only little in the range of 70 – 90 °C driving temperature. Figure 10
summarizes the results when varying only the chilled-water inlet temperatures.
Obviously, the system performed better at a high chilled-water temperature. The SCP
increased most linearly with a chilled-water inlet temperature ranging from 6 – 20 °C. The COP was increasing according to a convex curve from 0.45 to 0.75.

5. Conclusions

We have experimentally demonstrated the performance of a two-bed silica gel/water adsorption chiller. A simple model was developed and successfully employed to aid the design of the test facility. The targeted average cooling power of 5 kW was only reached under favourable conditions. Under more realistic operating conditions, the cooling power was 3.6 kW, corresponding with a specific cooling power (SCP) of 208 W per kg dry sorbent. The main reason for the lagging performance can be attributed to the lower heat transfer fluid flow rates realised in the laboratory compared to its design values. The thermal efficiency, or coefficient of performance (COP), of the adsorption chiller increased considerably as a result of direct heat recovery, in which the sensible heat of the heat transfer fluid is utilized to lead of the cooling or heating of the neighbouring sorbent reactor upon switching.

With experimental input parameters, the model fairly well predicts the system’s performance in terms of average SCP and COP. The obtained volumetric power density of 17 kW/m$^3$ based on the system as a whole, was somewhat lower than the targeted 20 – 30 kW/m$^3$. However, since the laboratory test facility was spaciously built, there remains lots of room to improve the power density of future systems. For comparison, based on the total volume of the most important and dominating components of the test facility, the power density was 58 kW/m$^3$. 
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Tables

Table 1 UA values of sorption reactor heat exchanger, evaporator, and condenser

<table>
<thead>
<tr>
<th>Heat exchanger</th>
<th>UA [W/K]</th>
<th>Remarks</th>
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<tr>
<td>Sorbent reactor</td>
<td>250 – 350</td>
<td>From dynamic measurements at 4 L/min water</td>
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<td>Evaporator</td>
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<td>Manufacturer specifications at 12 L/min water</td>
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<tr>
<td>Condenser</td>
<td>1666</td>
<td>Manufacturer specifications at 12 L/min water</td>
</tr>
<tr>
<td>Water circuit</td>
<td>Flow rate [L/min] (design value)</td>
<td>Standard inlet temperature [°C]</td>
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<td>-------------------</td>
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<tr>
<td>Heating reactor(s)</td>
<td>16.2 (24.0)</td>
<td>87</td>
</tr>
<tr>
<td>Cooling reactor(s)</td>
<td>15.2 (24.0)</td>
<td>25</td>
</tr>
<tr>
<td>Cooling condenser</td>
<td>13.8 (12.0)</td>
<td>25</td>
</tr>
<tr>
<td>Chilling evaporator</td>
<td>11.0 (12.0)</td>
<td>12</td>
</tr>
</tbody>
</table>
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