Research Paper

Utilizing waste heat from data centers with adsorptive heat transformation – Heat exchanger design and choice of adsorbent

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ABSTRACT

The electricity and water consumption of data centers is growing on a global scale. A shift towards liquid cooled racks in combination with thermally driven cooling can help to reduce the electricity and water demand associated with the necessary heat rejection. To quantify the potential of adsorptive heat transformation devices in reducing the electricity and water demand, the prediction of thermal efficiency, heat flow rates and energy efficiency ratio is required. To this end, a numerical model is newly developed using basic adsorption heat exchanger theory. This model can predict the necessary performance indicators with respect to temperatures and volume flow rates, heat exchanger design and adsorbent. The full performance map of a market-available adsorption chiller (71 points) and own measurements are used for calibration and rigorous validation of the model. An average deviation (experiment vs. calculation) of 8.3 % in terms of thermal efficiency and 7.2 % in terms of heat flow rates is achieved, indicating a very good agreement for a wide range of temperatures. At a moderate liquid cooled rack outlet temperature of 50 °C, a heat rejection temperature of 26 °C and a cold aisle inlet temperature of 18 °C the cooling power of the silica gel reference chiller of 5.3 kW can be increased by 59 % to 8.5 kW at a partial energy efficiency ratio (pumps and control, no fans) of > 20 by assuming MIL-100(Fe) as adsorbent on a flat-tube lamella heat exchanger. The model can be used in subsequent annual system simulations to quantify the savings in electrical power and water consumption, which strongly depend on the ambient conditions.

1. Introduction

The worldwide data center electricity consumption is high and expected to increase [1,2]. Besides the high electricity consumption, Farfan and Lohrmann [2] point to a severe environmental impact next to electricity: depending on the climate and the electricity source there is also water consumption associated with cooling (liquid / spray cooling towers, indirect evaporative cooling) and associated with the electricity production [2]. Some countries, e. g. Germany, passed laws resulting in the legislative requirement of achieving a power usage effectiveness (PUE) value below 1.2 [3]. The PUE represents the ratio of the annual energy demand of the entire data center to the annual energy demand of the IT components. For German data centers the average PUE is around 1.5 and the energy demand for cooling accounts for around 25 % of the total energy demand [4]. Thus, there is a strong need to make data centres more energy efficient. In this paper the possibilities of thermally driven adsorptive heat transformation are assessed to quantify the energy saving potential of this technology.

Depending on the climate and the available heat rejection reservoirs (air, seawater, river) there are different cooling methods as outlined by Lei and Masanet [5]. With increasing rack power densities above 15 kW/rack that can be expected in the year 2025 according to recent surveys [6] and trends [7,8], free convective cooling with air as medium [9] is no longer possible. Instead, liquid cooling concepts as described for example by Hnayno et al. [10] will be needed, for further increasing rack power densities above 50 kW/rack even heat pipe technology as described by Wang et al. [11] might be required in the future.

In case of liquid cooling at temperatures between 40 °C and 60 °C usually around 70–80 % of the heat is transferred to the liquid and the

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remaining 20–30% are transferred to the surrounding air. Besides this, additional cooling for storage racks, network switches and other class A1 equipment (ASHRAE classification) in the temperature range between 20°C–30°C is necessary [12]. Using the heat transferred to the liquid, thermally driven cooling as shown in Fig. 1 is a solution that utilizes the waste heat between 40°C–60°C to cool down the class A1 equipment and other residual loads as demonstrated by Wilde et al. [13] and Ott et al. [14] for the German supercomputing cluster CooLMUC-2 and Gupta and Puri [15] for a data center in Toronto / Canada. In both cases, adsorption chillers with the silica gel-water working pair are applied.

As illustrated by Moayed Mohseni et al. [16], Mohammed et al. [17], Liu et al. [18], and Matemb Ma Ntep et al. [19] the development of novel adsorbents is quite dynamic. For heat transformation applications Liu et al. [18] identified 18 promising adsorbents out of more than 350 research papers based on their water uptake for 2–25°C low temperature inlet, 25–39°C medium temperature inlet, and 55–110°C high temperature inlet.

The selection of adsorbents based on a high water uptake at specific temperatures yields a potentially high efficiency. However, as basic considerations of Velte-Schäfer et al. [20] show, the cooling power depends on the driving temperature difference, the overall heat and mass transfer resistances of the adsorption heat exchanger and the evaporator, and the evaporation enthalpy of the working fluid. This aspect is hardly considered in the screening methods of recently published literature [16–18] on MOF selection for heat transformation applications.

Thus, the first aim of this work is to present a targeted adsorbent screening method that involves not only water uptake, but also driving temperature differences to identify promising adsorbents for data center cooling applications.

The second aim of this work is to quantify thermal efficiency, heat flow rates and energy efficiency ratio of adsorption chillers in data
centers with respect to temperatures and volume flow rates, heat exchanger design and adsorbent. To this end, a calibrated and validated model for a commercially available silica gel-water adsorption chiller is developed. Along with manufacturer data of the adsorption chiller with two adsorption modules operating in alternation, also own measurements of a single adsorption module supplement the manufacturers data. Based on this validated model, the equilibrium data can be changed to data of the most promising MOFs suited for cooling application in data centers.

In strongly seasonal climate zones with pronounced temperature differences between summer and winter the number of operating hours of data center cooling equipment besides the free convective cooling is limited to the summer months as illustrated by Ruch et al. [21] for the three different locations Munich, Dallas, and Ryad. On the other hand, there is a high energy demand for space heating in seasonal climate zones.

The required temperature level for space heating depends on the energy performance (insulation, heating system) of the building and the outdoor temperature [22,23] and lies between 25 °C and 65 °C, whereas the required temperature for domestic hot water heating is > 65 °C. For required heating system temperatures below the liquid cooled rack outlet temperatures (45–55 °C) the heat from data centers can be used directly for space heating. Adsorptive heat transformation as illustrated in Fig. 2 is a solution to lift medium temperature heat from the liquid cooled rack to a higher temperature as it is required for domestic hot water. Recently, Engelpracht et al. [24,25] have proven experimentally that the silica gel-water working pair is capable of transforming heat from 90 °C to 110 °C.

Switching an adsorption chiller between cooling mode and heat transformation mode is possible without changing anything besides the control algorithm. The modes can be switched by changing the external connections to heat sources and heat sinks. Thus, the heat transformation application could be an option for data center locations in strongly seasonal climate zones to increase the operating hours of a combined adsorption chiller/heat transformation device. Consequently, the third aim of this work is to explore the possibility of heat transformation application both experimentally and by simulation to quantify thermal efficiency, heat flow rates and energy efficiency ratio of this operation mode. Promising adsorbents for the heat transformation application are identified and evaluated with the calibrated and validated model analogously to the chiller application.

2. Materials and methods

2.1. Silica gel chiller and sorption module

The reference silica gel-water chiller is an eCoo10 type from Fahr-Enheit (Halle/Saale, Germany) that is available on the market. This silica gel chiller consists mainly of two sorption modules, hydraulic valves to connect the heat sinks and heat sources with the heat exchangers of the sorption modules as shown in Fig. 3, and a controller that performs the periodic switching of the hydraulic valves.

The sorption module consists of an adsorption heat exchanger and a combined evaporator-condenser assembled in a single vacuum chamber. These two heat exchangers are made of round tubes with lamellas, the adsorption heat exchanger being coated with silica gel granules glued with an epoxy resin on the metal surface. The main dimensions of the adsorption heat exchanger and the evaporator-condenser needed for the calculations of this study are listed in Table 1.

2.2. Sorption module test rig

The sorption module test rig at Fraunhofer ISE provides temperature step profiles to measure the thermal response of adsorption heat exchanger and evaporator-condenser as shown in Fig. 54 in Supplemental Note 2 schematically. Besides the temperatures at the inlet and outlet of the heat exchangers the volume flow rates of the two hydraulic circuits are measured. A pressure sensor is connected with the vacuum chamber of the sorption module as illustrated in Fig. 4.

A measurement is performed by setting the desired volume flow rates, inlet temperatures, and a half cycle time. Temperature swing cycles are performed until a state of periodic equilibrium is reached. Then
the data of 2–8 cycles is recorded for evaluation of efficiency and heat flow rates. The heat flow rates in the adsorption heat exchanger and the evaporator-condenser are calculated with equation (1) and equation (2), respectively. The data for the enthalpy $h_L$, the density $\rho_L$, and the specific heat capacity are calculated according to the IAPWS 97 formulation [26] with a script implemented by Holmgren [27].

$$\dot{Q}_{\text{e}}/c_L = \rho_L HTF (T_{i,\text{in}}) \cdot \Psi_{\text{HTF},L/C} \cdot (h(T_{i,\text{in}}) - h(T_{\text{e,final}}))$$ (1)

$$\dot{Q}_{\text{a}}/d = \rho_L HTF (T_{i,\text{in}}) \cdot \Psi_{\text{HTF},L/C} \cdot (h(T_{i,\text{in}}) - h(T_{\text{a,final}}))$$ (2)

The efficiency is calculated with equation (18). The amounts of heat $Q_e$, $Q_a$, $Q_c$, and $Q_d$ are obtained by integrating the heat flow rates over the respective half cycle. The hydraulic connections of the components with the heat sources and heat sinks are listed in Table 2.

2.3. Calculation of efficiency and heat flow rates

For a comprehensive performance evaluation of adsorption chiller and adsorption heat transformer including various temperatures and heat flows > 0 indicate connection with a heat sink, heat flows < 0 indicate the connection with a heat source.

Table 2
Connections of components with heat sinks and heat sources at low, medium, and high temperature levels for heat pump / chiller / storage application and heat transformation application. Heat flows > 0 indicate connection with a heat sink, heat flows < 0 indicate the connection with a heat source.

<table>
<thead>
<tr>
<th>Connections</th>
<th>Cooling</th>
<th>Heat transformation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adsorption ADHX</td>
<td>$Q_e &gt; 0$</td>
<td>$T_{\text{a,final}} - T_{\text{L}}$</td>
</tr>
<tr>
<td>Desorption ADHX</td>
<td>$Q_d &lt; 0$</td>
<td>$T_{\text{a,final}} - T_{\text{L}}$</td>
</tr>
<tr>
<td>$Q_c &gt; 0$</td>
<td>$T_{\text{a,final}} - T_{\text{L}}$</td>
<td>$T_{\text{a,final}} - T_{\text{L}}$</td>
</tr>
</tbody>
</table>

Temperature differences

Temperature lift $\Delta T_{\text{lift}}$ $T_{\text{a,final}} - T_{\text{L,final}} - T_{\text{L,final}} - T_{\text{L}}$

Temperature thrust $\Delta T_{\text{thrust}}$ $T_{\text{a,final}} - T_{\text{L,final}} - T_{\text{L,final}} - T_{\text{L}}$
mass flow rates a numerical model is required. Depending on the desired purpose of the model, different levels of detail are necessary. For their evaluation of silica gel water adsorption chillers in data centers Gupta and Puri [15] use an empirical model that is parametrized with efficiency and cooling power values from (formerly) commercially available adsorption chillers (InvenSor, Germany). The adsorption- compression cascade chiller investigated by Palomba et al. [28] is modelled with a multidimensional interpolation in TRNSYS that is fed with experimental data. Frey et al. [29] present an artificial neuronal network model of an adsorption chiller. These models have in common that they do not include any information about the components of the adsorption chiller.

The advantage of these models is the low computational cost – the prediction of efficiency and heat flow rates is very fast, since they do not require a system of differential equations to be solved. However, it is not possible to predict the behaviour of the adsorption chiller if the working pair is changed or temperatures and mass flow rates deviate from the data that the models are calibrated with.

On the other end of the scale there are rigorous physical, spatially resolved, transient models as presented by Mikhaila et al. [30] or Kowsari et al. [31]. These models allow for in-depth studies of geometric variants of the heat exchanger geometry but fail when it comes to simulation of an adsorption chiller within a system of other components and/or devices because the computational cost is too high.

In between these two poles there exist approaches in Modelica/ Dymola that still include differential equations but with spatial resolution only in the main fluid flow direction as presented by Bau et al. [32]. The operator splitting modelling approach presented by Gibelhaus et al. [33] can help to reduce the computational costs of such models further. However, the correct specification of a real heat exchanger geometry requires to enter more than 20 parameters and the computational cost is not negligible when it comes to annual simulations. Furthermore, the calibration and validation procedure of a transient model can be quite time-consuming if more than ten different measurements are involved. To overcome these limitations a simple and robust approach was recently presented by Velte-Schäfer et al. [20]. This approach is further developed in this paper to apply it to working pairs with a linear shape of the isotherm (e.g. silica gel-water), since the original approach as presented by Velte-Schäfer et al. [20] is limited to working pairs with a step-like isotherm shape. The development of an enhanced numerical model for a wider range of working pairs is described in the following.

The calculation of the heat flow rates is based on effective heat and mass transfer resistances as shown in Fig. 5. With a finite mass flow rate the heat transfer fluid in the components, the outlet temperature of the adsorption heat exchanger is larger than the inlet temperature $T_{\text{out,a}} > T_{\text{in,a}}$ and the outlet temperature of the evaporator is lower than the inlet temperature $T_{\text{out,e}} < T_{\text{in,e}}$. It is obvious that lower mass flow rates will reduce the driving temperature differences $\Delta T_{\text{dev,a/d}}$ and $\Delta T_{\text{dev,e/c}}$ for a given characteristic temperature difference, resulting in a lower thermal output. Similar relationships apply to the desorption half cycle.

To calculate the heat flow rates $Q_{s,e}$, $Q_{\text{ads}}$, and $Q_{s}$ the half cycle is split into a transient switching phase and a quasi-isothermal phase. The switching phase describes the short period when the inlet temperatures are switched from $T_{\text{M}}$ to $T_{\text{H}}$ in the adsorption heat exchanger and from $T_{\text{L}}$ to $T_{\text{E}}$ in the evaporator-condenser. In the quasi-isothermal phase, the temperature change of the heat transfer fluid of the components is much lower than it is in the switching phase. As a robust first-order approximation all calculations in the following are performed with the mean values of the temperatures in the quasi-isothermal phase, i.e. the equations are not time-dependent anymore. The heats in the quasi-isothermal phase are calculated then with equation (3) and (4) for ADHX and EC, respectively. $\Delta X$ is the loading spread, $M_{\text{ads}}$ the adsorbent dry mass, $\Delta h_{\text{ad}}$ the adsorption enthalpy, $\Delta h_{\text{dv}}$ the condensation/ evaporation enthalpy, $t_{\text{hr}}$ the half cycle time, $R_s$ and $R_e$ are the effective heat and mass transfer resistances of the ADHX and the EC, $\Delta T_{\text{dev,a/d}}$ and $\Delta T_{\text{dev,e/c}}$ are the driving temperature differences.

$$Q_{\text{ads}} = \Delta T_{\text{dev,a/d}} \cdot M_{\text{ads}} \cdot \Delta h_{\text{ad}}$$

$$Q_{\text{dv}} = \Delta T_{\text{dev,e/c}} \cdot M_{\text{dv}} \cdot \Delta h_{\text{dv}}$$

The half cycle energy balance of the components yields equation (5), (6), (7) and (8) listed in Table 3. In equations (5) to (8) the heats of the condensation/evaporation enthalpy $\Delta h_{\text{dv}}$ and $\Delta h_{\text{c}}$ can be calculated out of the component energy balances of sorption module measurements. It depends on the sorption module geometry and is in the range of 10 ± 4 W/k for the sorption module presented here.

$$Q_{\text{d,v}} = (\text{UA})_{\text{d,v}} \cdot (T_{\text{out,s}} - T_{\text{out,e}})$$

With given effective heat and mass transfer resistances $R_s$ and $R_e$, half cycle time $t_{\text{hr}}$, adsorbent mass $M_{\text{ads}}$, adsorption enthalpy $\Delta h_{\text{ad}}$, and condensation/evaporation enthalpy $\Delta h_{\text{dv}}$ the driving temperature differences $\Delta T_{\text{dev,a/d,e/c}}$ and the loading spread $\Delta X$ must be calculated.

To calculate the driving temperature differences $\Delta T_{\text{dev,a/d,e/c}}$ in the...
quasi-isothermal phase \( \epsilon \)-NTU relationships are used. Although the heat and mass transfer processes in ADHX and EC are transient, it is assumed as a first-order approximation that the temperature changes in the quasi-isothermal phase are so small that constant mean values for the temperatures and accordingly for the driving temperature changes in the quasi-isothermal phase, as a first-order approximation that the temperature changes in the quasi-isothermal phase are transient, it is assumed that the heat transfer fluid, such as the timespan of the highly transient switching phase is below 20% of the half cycle time of each of the components (adsorption heat exchanger and evaporator-condenser). It has to be checked if NTU\(_{a/d/e/c}\) < 20 and \( C_{\text{HTF}} \) < 0.2, to avoid domination of the transient switching phase.

The temperature effectiveness \( \epsilon \) of the ADHX is calculated with equation (11) assuming an ideal linear isotherm as described by Velte-Schäfer et al. [20].

\[
\epsilon = \frac{\text{NTU}}{1 + \text{NTU}} \tag{11}
\]

The \( \epsilon \)-NTU relationship in equation (12) is applied to the EC.

\[
\epsilon = 1 - \exp(-\text{NTU}) \tag{12}
\]

With the definition of the heat exchanger effectiveness according to Shah et al. [35] in equation (13) the driving temperature differences can be calculated with equation (14) for the ADHX and equation (15) for the EC.

\[
e_{a/d/e/c} = \frac{\text{T}_{\text{out,eq}(X,p)} - \text{T}_{\text{in,a/d}}}{\Delta \text{T}_{\text{drv,a/d}}} = \frac{\text{T}_{\text{in,a/d}} - \text{T}_{\text{in,a/d}}}{\Delta \text{T}_{\text{drv,a/d}}} = \frac{\text{T}_{\text{out,e/c}} - \text{T}_{\text{in,e/c}}}{\text{T}_{\text{in,e/c}}} \tag{13}
\]

\[
\Delta \text{T}_{\text{drv,a/d}} = \text{T}_{\text{out,eq}}(X,p) - \text{T}_{\text{in,a/d}} = \text{T}_{\text{in,a/d}}(X,p) - \text{T}_{\text{out,a/d}} = \frac{\text{T}_{\text{eq}}(X,p) - \text{T}_{\text{in,a/d}} + \text{T}_{\text{in,a/d}}}{2} \tag{14}
\]

\[
\Delta \text{T}_{\text{drv,e/c}} = \frac{\text{T}_{\text{out,e/c}} - \text{T}_{\text{in,e/c}}}{\ln \left( \frac{\text{T}_{\text{in,e/c}}}{\text{T}_{\text{in,p}}} \right)} \tag{15}
\]

To solve equations (3) to (15) a relationship between the equilibrium temperature \( T_{\text{eq}}(X,p) \) and saturation temperature \( T_{\text{sat}}(p) \) in the quasi-isothermal phase is required. With the definition of the characteristic temperature difference in equation (16) according to Laurens [36], the characteristic temperature difference of any equilibrium data set consisting of pressure \( p \), Temperature \( T \) and loading \( X \) can be calculated.

\[
\Delta T_{\text{ch}}(X,p) = T_{\text{eq}}(X,p) - T_{\text{sat}}(p) \tag{16}
\]

The mean value of the characteristic temperature difference over the loading range covered in the quasi-isothermal phase \( \Delta T_{\text{ch}} \) is calculated with equation (17) depending on the minimum loading \( X_{\text{min}} \) and maximum loading \( X_{\text{max}} \) reached. Since the minimum loading and the maximum loading achieved in the quasi-isothermal phase depend on the driving temperature differences \( \Delta T_{\text{drv,a/d,e/c}} \) the calculation of \( \Delta T_{\text{ch}} \) and \( \Delta X = X_{\text{max}} - X_{\text{min}} \) is an iterative procedure as shown in Supplemental Note 5.

\[
\Delta T_{\text{ch}} = \frac{1}{X_{\text{max}} - X_{\text{min}}} \int_{X_{\text{min}}}^{X_{\text{max}}} \Delta T_{\text{ch}}(X) \, dX \tag{17}
\]

The thermal efficiency and the mean heat flow rate are calculated with equation (18) and equation (19) as listed in Table 4. The electrical power consumption of pumps and fans is evaluated by relating the mean heat flow rate to the electrical power consumption as expressed with the energy efficiency ratio (EER) in equation (20). If it is not possible to assess the entire electrical power consumption, the EER will be called “partial” EER (PEER) in the following.

2.4. Calibration and validation procedure

The model is calibrated with experimental data in two steps. In the first step, the effective heat and mass transfer resistances \( R_{a/d/e/c} \) are calculated using the experimental data of the sorption module measurements. For this calculation the equations (10), (11), (12), and (13) are used together with the mean temperatures and mean module pressure in the quasi-isothermal phase of the half cycle. A typical example of this averaging procedure is shown in Fig. S9 and Fig. S10 in Supplemental Note 4.

In the second step, the values of \( R_{a/d/e/c} \) are used as starting values for the calibration of the model with the values for efficiency and heat flow rates calculated from experimental data. In this calibration step both the experimental data from sorption module measurements as well as the experimental data from the silica gel chiller are used. This second step yields the final values of \( R_{a/d/e/c} \) and the heat recovery factor \( f \text{HR} \).

<table>
<thead>
<tr>
<th>Table 3</th>
<th>Calculation of heat flow rates in adsorption, desorption, evaporation, and condensation depending on the application.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameters</td>
<td>Heat pump / chiller</td>
</tr>
<tr>
<td>( Q_a )</td>
<td>( Q_{a,e} + \Delta T_{\text{ch}} \cdot C_{\text{p,ads}} \cdot (1 - f_{\text{m}}) - Q_{\text{in,ads}} )</td>
</tr>
<tr>
<td>( Q_i )</td>
<td>( -Q_{\text{ch}} - \Delta T_{\text{ch}} \cdot C_{\text{p,ads}} \cdot (1 - f_{\text{m}}) - Q_{\text{in,ads}} )</td>
</tr>
<tr>
<td>( Q_x )</td>
<td>( -Q_{\text{ch}} + \Delta T_{\text{ch}} \cdot C_{\text{p,ads}} \cdot (1 - f_{\text{m}}) + Q_{\text{in,ads}} )</td>
</tr>
<tr>
<td>( Q_e )</td>
<td>( Q_{a,e} - \Delta T_{\text{ch}} \cdot C_{\text{p,ads}} \cdot (1 - f_{\text{m}}) + Q_{\text{in,ads}} )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 4</th>
<th>Calculation of efficiency and heat flow rates for different applications.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parameters</td>
<td>Chiller</td>
</tr>
<tr>
<td>COP</td>
<td>( \frac{\dot{Q}_e}{\dot{Q}_a} )</td>
</tr>
<tr>
<td>Mean heat flow rate</td>
<td>( \frac{2\dot{Q}_e}{\dot{Q}_a} )</td>
</tr>
<tr>
<td>EER</td>
<td>( \frac{\dot{Q}<em>e}{f</em>{\text{HR}} \dot{Q}_e} )</td>
</tr>
</tbody>
</table>
3. Results and discussion

3.1. Temperature boundary conditions

For both cooling application and heat transformation application the heat rejection temperature of the liquid cooled rack is assumed to be between 45 °C and 55 °C. The temperature for cooling the cold aisle inlet air is varied between 18 °C and 28 °C, and the heat rejection of the adsorption chiller is accordingly varied between 26 °C and 36 °C. From a purely equilibrium thermodynamics point of view, it is not possible to drive the adsorption chiller with 45 °C and achieve a cooling temperature of 18 °C if the heat rejection happens at 36 °C. Thus, these scenarios are excluded from the analysis. The resulting feasible scenarios are listed in Table 5.

In case of the heat transformation application the heat rejection to the environment is supposed to happen at T_{LT} being 10 °C (winter). The heat is assumed to be provided at 65 °C (inlet temperature of adsorption heat transformer) resulting in different temperature lifts as listed in Table 5.

The minimum temperature difference between inlet and outlet in the rack HTF circuit is set to 7 K.

3.2. Preselection of possible adsorbents

A larger loading spread ΔX leads to a higher efficiency as can be deduced from Eqs. (4), (7), and (18) and discussed in detail by Schnabel et al. [37] previously. Thus, an important criterion for the selection of possible adsorbents is a large loading spread at the temperature conditions presented in section 3.1. However, besides the efficiency also the thermal power is relevant in the application. Consequently, the preselection method should also include considerations on the thermal power - without the need for solving the whole system of equations presented in section 2.3.

If we transform Eq. (3) into Eq. (21), an elementary relationship becomes clear here: A higher thermal power requires a greater difference quotient ΔX/Δt, i.e. (Q_{thr} Q_{in}) ΔX/Δt. However, in the quasi-isothermal phase of the half-cycle according to equation (21), this difference quotient is only dependent on the effective thermal and mass transfer resistance of the ADHX R_{a/d}^{-1}, the driving temperature difference ΔT_{drv,a/d}, the adsorbent mass M_{sorb} and the adsorption enthalpy Δh_{ad}. If we exchange one adsorbent against the other without changing R_{a/d}^{-1} and M_{sorb} and if it is assumed that the change in adsorption enthalpy is negligible, the achievable difference quotient ΔX/Δt is only dependent on the driving temperature difference ΔT_{drv,a/d}:

$$\frac{ΔX}{Δt} = \frac{R_{a/d}^{-1}}{M_{sorb}} ΔT_{drv,a/d}$$

The plots of the equilibrium data in the form of the characteristic temperature difference curve and mean value ΔT_{thr} for silica gel, ΔT_{in} = 8 K, ΔT_{thr} = 24 K.

![Fig. 6. Characteristic temperature difference curve and mean value ΔT_{thr} for silica gel, ΔT_{in} = 8 K, ΔT_{thr} = 24 K.](image-url)
In a first calibration step, the module measurements are used to evaluate the effective heat and mass transfer resistances as described more in detail in Supplemental Note 4. The analysis of measuring points with the temperature triple 10/27/75 °C and different half-cycle times (from 250 s to 1800 s) shows a clear dependence of the effective heat and mass transfer resistance in adsorption on the relative loading, as shown in Fig. 10. All other heat and mass transfer resistances show little or no such dependence. This observation is valid for almost all results of the 19 evaluated measurements as listed in Table S1 in Supplemental Note 2. The effective heat and mass transfer resistances in desorption, condensation, and evaporation lie within the respective error bars as shown in Fig. S11 in Supplemental Note 4 and no dependence on temperature or half-cycle time can be recognized that goes beyond the error bars.

The hypothesis behind the dependence of the effective heat and mass transfer resistance in adsorption on the relative loading is an indication that during adsorption the mass transfer resistance becomes more and more limiting. While heat transfer resistance should in general be independent of the water vapour pressure, especially macroscopic mass transfer e.g. between the coated fins can be highly sensitive to absolute pressures due to different slopes of the water vapour pressure curve at different pressures. This would mean that during adsorption, at low relative loading, the easily accessible adsorption sites are preferentially occupied. To increase the relative loading, the adsorption sites that are more difficult to access (longer path for mass transfer and thus higher pressure drop) must also be occupied and thus the effective mass transfer resistance increases. A quantification and differentiation between heat and mass transfer would need more specific insight and experiments. For the scope of this study a more or less empirical quantification and description of the combined heat and mass transfer, only differentiating between adsorption and desorption, is sufficient.

Besides the direct analysis of module measurements to determine the effective heat and mass transfer resistances, the device measurement data can be used as well. In this indirect method the effective heat and mass transfer resistances are adjusted such that the calculated data for COP and heat flow rates match with the measured data of the device. For the further parameterization of the model, all device measurements at 75 °C desorption temperature as listed in Table S2 in Supplemental Note 2 were used and comparative calculations were carried out at different temperatures.

Fig. 7. Characteristic temperature difference curve and mean value $\Delta T_{th}$ for Aluminium Fumarate, $\Delta T_{th} = 8 \, \text{K}$, $\Delta T_{ch} = 24 \, \text{K}$.

Fig. 8. Equilibrium loading spread $\Delta X_{eq}$ for different adsorbents and all “Cooling” scenarios listed in Table 5. The full list of results is available in Figure S1 in Supplemental Note 1.
desorption temperatures. Overall, no dependence of the effective heat and mass transfer resistance \( R_a \) on the desorption temperature was found in the comparison between calculation and device measurement or module measurement. The effective heat and mass transfer resistance of adsorption is described by a linear relationship according to equation (22) as a function of the relative loading difference.

\[
R_a(\Delta X_{\text{rel}}) = \frac{dR_a}{d\Delta X_{\text{rel}}} \cdot \Delta X_{\text{rel}} + R_{a,0}
\]  

(22)

The slope of equation (22) was found to be 1.66 K/kW for all measurements, whereas the y-axis intercept \( R_{a,0} \) varies with the temperatures \( T_N \) (pressure in the adsorption module) and \( T_M \) (adsorption heat exchanger inlet temperature). The values for \( R_a \) are shown in Fig. S14 in Supplemental Note 4.

The heat recovery factor \( f_{HR} \) is determined by comparing module measurements and device measurements using the values obtained from energy balance calculations as shown in Fig. S8 in Supplemental Note 3 as starting values. Since the effective heat and mass transfer resistances are the same for both data sets, the only difference is the heat recovery that happens in the device that consists of two modules. As the difference between module and device measurements in Fig. 11 shows, heat recovery works worse at low evaporator temperatures than at higher evaporator temperatures. While there are only slight differences in efficiency in the module measurements, there are very obvious in the device measurements.

On the one hand, this depends on the implemented device control, but on the other hand, it is also visible that the faster the dynamics of the heat and mass transfer processes take place, the better the heat recovery through delayed return switching works. At low evaporator temperatures and/or high adsorption temperatures, little or no adsorption or desorption occurs in the (theoretically ideally isosteric) switching phases. At high evaporator and low adsorption temperatures, adsorption and desorption processes already take place within the switching phase. In addition to the thermal capacity of the water in the pipes and other
components (heat exchanger metal), some of the sorptive heat can also be recovered with the delayed return switching. The resulting function for heat recovery factors is discussed in detail in Supplemental Note 4.

The calculation results for all 71 device measurements are compared with the experimental results in Fig. 12 in terms of efficiency and Fig. 13 in terms of heat flow rate of the evaporator (chiller application). The RMSD of the calculated efficiency is ± 0.032 and the RMSD of the heat flow rate is ± 0.45 kW resulting in CV values (RMSD relative to mean values of experimental efficiency and heat flow rate) of 8.3 % and 7.2 %. These values indicate a very good prediction quality of the calculation as it can be seen in Fig. 11 for the adsorption chiller case and in Fig. 14 for the heat transformation case.

3.4. Heat exchanger design of MOF adsorptive heat transformer

The performance evaluation of a MOF adsorptive heat transformer requires the choice of effective heat and mass transfer resistances of the ADHX $R_a$, $R_d$, and the evaporator-condenser $R_e$ and $R_c$, as well as the effective thermal masses $C_p$, $C_{p,tot}$, $C_s$ and $C_{p,tot,e}$. The concept of effective heat and mass transfer resistances allows for quick evaluation of experimental data sets on different levels (small, but representative heat exchanger piece, heat exchanger, adsorption module, and adsorption chiller) if inlet and outlet temperatures and the pressure are measured. Furthermore, this concept allows for separation of heat and mass transfer resistances according to Ammann et al. [38] and Velte et al. [39]. Thus, the first step of finding values for $R_a$, $R_d$, $R_e$, and $R_c$ is the calculation of the heat transfer resistance of different heat exchanger geometries (round tube lamella and flat tube lamella). This calculation is carried out in Supplemental Note 6 and visualized in Fig. S22.

Three different adsorption heat exchanger or adsorption chiller measurements from literature were deeply analysed to calculate effective heat and mass transfer resistances. The result is shown in Fig. 15 along with the values for the eCoo10 sorption module. The round tube-lamella heat exchanger designs (this work, Kummer et al. [40], Gkaniatsou et al. [41]) show values around 60 dm$^3$K/kW for $R_a$, which are in line with the upper range of the volume scaled heat transfer resistance $R_{Vht}$ of a round tube-lamella heat exchanger with a fin pitch of 3.6 mm as shown in Fig. S22 in Supplemental Note 6.

The adsorption heat exchanger measured by Kummer et al. [40] has a fin pitch of 3.0 mm, thus, a volume scaled heat transfer resistance $R_{Vht}$ of around 50...60 dm$^3$K/kW can be expected for a measurement that achieves a relative loading of 90 % (adsorption sites with long heat transfer paths have to be occupied). Since the volume scaled effective heat and mass transfer resistance shown in Fig. 15 is 60 dm$^3$K/kW, it can be deduced that the mass transfer resistance is negligible in this case and the process is completely limited by the heat transfer resistance. The values of Gkaniatsou et al. [41] are around 70...85 dm$^3$K/kW with large error bars because the adsorption chiller measurements are dominated by the pressure loss between evaporator/condenser and adsorption heat exchanger due to small cross-sections of the pipes or valves on the vapour side. Furthermore, the large fin pitch of 6 mm leads to volume scaled heat transfer resistances of > 60 dm$^3$K/kW – also here it can be deduced that the mass transfer resistance in the adsorption heat exchanger is negligible.

The results of Bendix et al. [42] indicate a strong mass transfer limitation in case of the coated adsorption heat exchanger – from small-scale samples and heat transfer resistance considerations in Fig. S22 in Supplemental Note 6 a volume scaled effective heat and mass transfer resistance in adsorption/desorption $R_{Vht}^{V}$ of around 14...18 dm$^3$K/kW.

Fig. 12. Calculated efficiency vs. measured efficiency for all 71 device measurements. The dashed lines indicate the measurement uncertainty of ± 0.025.

Fig. 13. Calculated heat flow rate vs. measured heat flow rate for all 71 device measurements.

Fig. 14. Calibration adsorption module heat transformation case.
would be expected. The value deduced from the adsorption heat exchanger measurements shown in Fig. 15 is \( > 40 \text{ dm}^3/\text{kW} \), pointing to an additional mass transfer resistance that was not observed on small-scale sample level.

The values for \( R_{V_a/d} \) shown in Fig. 15 for compact flat tube heat exchangers based on fibrous structures but also flat tube lamella heat exchangers coated with SAPO-34 prove that it is possible to design and manufacture high performance adsorption heat exchangers. However, these heat exchangers are coated with SAPO-34 by applying the partial support transformation technique as suggested by Bauer et al. [43]. In this special case the adsorbent is directly crystallized on the heat exchanger surface as detailed by Bauer et al. [43] and Wittstadt et al. [44] ensuring a low contact resistance between adsorbent and metal. However, this method is restricted to few zeo-type or zeolite adsorbents and there are no examples of MOF coated heat exchangers using a similar method.

In the following we will assume that the coating method for flat tube lamella heat exchangers can be improved to reach \( R_{V_a/d} \) around 35 \( \text{dm}^3/\text{kW} \) and that the improved adsorption module is equipped with a round tube-lamella evaporator-condenser. Thus, we will assume in the following performance evaluation that the volume scaled effective heat and mass transfer resistance \( R_{V_a/d} \) is 50 % lower than it is for the round tube-lamella silica gel adsorption heat exchanger as illustrated in Fig. 16. To avoid a limitation due to a too small evaporator-condenser, the effective heat and mass transfer resistances of the adsorption heat exchanger \( R_{V_a/d} \) and the evaporator-condenser \( R_{evap,cond} \) are balanced by adjusting the component volume as illustrated in Fig. 16. Additional data necessary for the model is listed in Table S7 in Supplemental Note 6.

3.5. Performance evaluation

3.5.1. Chiller application

In a first step, efficiency and cooling power are calculated for the silica gel chiller eCoo10 (reference) under the conditions as described in section 3.1. Heat rejection at medium temperature is at 26 °C, assuming a hybrid cooling tower with 30 °C inlet temperature and 26 °C outlet temperature. Heat can be removed from the liquid-cooled rack up to a maximum of 55 °C, whereby the required temperature spread between inlet and outlet temperature is 7 K. Gupta and Puri [15] state that the temperature of the rack has a strong influence on the computing power.

At the same time, however, the achievable loading spread and thus the efficiency and performance of the adsorption chiller increases with increasing desorption temperature. For this reason, the calculations are carried out for desorption temperatures (i.e. rack outlet temperatures) between 45 °C and 55 °C. The results of the calculations are shown in Fig. 17. The desorption temperature for silica gel has a major influence on the achievable capacity – when increasing from 45 °C to 55 °C, the heat flow rate \( \dot{Q}_L \) can be increased from 3.6 kW to 7.0 kW with approximately the same thermal efficiency of approx. 0.52.

The thermal efficiency of the silica gel chiller eCoo10 considered here can also be compared with measured data from a silica gel chiller by Pan et al. [45]. Pan et al. [45] present measurement data for the temperature triple 24/32/55 °C, which corresponds to a temperature lift of 8 K and a temperature thrust of 23 K. The temperature triple 18/26/50 °C calculated in our study can serve as a comparison. Pan et al. [45] give an efficiency of 0.48 with a specific cooling power (based on the adsorbent mass) of 93 W/kg. With the same efficiency, the silica gel chiller eCoo10 considered here achieves a specific cooling capacity of 246 W/kg, which reflects a considerably improved heat and mass transfer efficiency.
transfer characteristic.

To evaluate the potential of MIL-100(Fe) against the silica gel reference case, two calculations are performed: First, only the equilibrium data of silica gel is replaced against the equilibrium data of MIL-100(Fe). This results in an increase both in efficiency and heat flow rate \( \dot{Q}_L \) as illustrated in Fig. 17. As discussed in section 3.2, the increase in heat flow rate only depends on the driving temperature difference \( \Delta T_{drv} \) and not on the increase in equilibrium loading spread. Since MIL-100(Fe) has only a slightly higher driving temperature difference than silica gel, the increase in terms of heat flow rate is moderate.

Second, the equilibrium data of silica gel is replaced with the equilibrium data of MIL-100(Fe) and the effective heat and mass transfer resistances \( R \) and the overall thermal masses \( C_{p,\text{tot}} \) are changed according to Table S7 in Supplemental Note 6. Thus, these calculations are performed under the assumption of a different adsorption heat exchanger design as described in section 3.4. The results shown in Fig. 18 reveal the full potential of MIL-100(Fe) and a proper adsorption heat exchanger design. The heat flow rate \( \dot{Q}_L \) is increased between 41 % and 78 % compared to the silica gel reference scenarios.

The dependency of the heat flow rates on the temperature conditions is further explored by increasing the low temperature (cold aisle inlet) and the medium temperature (heat rejection to the environment) as illustrated in Fig. 19. It can be assumed that the maximum acceptable cold aisle inlet temperatures will rise in the future due to improvements in the IT equipment. The electrical power consumption (fans, pumps) for the heat rejection can be decreased considerably if the medium temperature is increased. This study shows that the heat flow rates drop for both silica gel and MIL-100(Fe) around 3–3.5 kW if the medium temperature and the low temperature are increased by 10 K. Thus, in case of elevated cold aisle inlet temperatures and medium temperatures, adsorption chillers with a higher overall capacity have to be installed.

Besides the thermal efficiency and the heat flow rate the electric power consumption is decisive for an energetically and potentially economically viable application of an adsorption chiller. If a data center with around 70–80 % liquid cooling at 45–55 °C rack outlet temperature and 20–30 % air cooling at 18–25 °C cold aisle inlet temperature is considered, the reference case shown in Fig. 20 requires a pump for liquid cooling (P-HT), a dry heat rejection unit (RC-HT), an adiabatic heat rejection unit (RC-LT), and a pump P-LT.

While the heat rejection at RC-HT happens at sufficiently high temperature difference to the ambient that enables the application of a dry heat rejection unit, the air cooling at RC-LT requires adiabatic cooling.
and/or a compression chiller to achieve the required cold aisle inlet temperature. The adiabatic cooling at RC-LT requires water for evaporation and electrical energy to drive the fans. The closer the cold aisle inlet temperature and the wet bulb temperature of the ambient air, the higher the energy and water consumption will be. An adsorption chiller will make the adiabatic cooling and/or compression chiller at RC-LT obsolete as well as the free cooling at RC-HT as shown in Fig. 20 and listed in Table 6. However, the adsorption chiller will require free and/or adiabatic cooling at RC-MT at medium temperature level. On the one hand, the benefit of RC-MT over RC-LT is the reduction in water and electrical energy consumption due to the higher temperature difference to the wet (or dry) bulb temperature of the ambient air.

On the other hand, the RC-MT will have a higher water and electrical energy consumption than RC-HT. The exact calculation of the benefit of an adsorption chiller requires an annual simulation with realistic assumptions for water and electrical energy consumption of RC-HT, RC-MT, and RC-LT. This annual simulation is out of scope of our study. While it will remain an open question under which conditions RC-MT will consume less water and electrical energy than RC-HT and RC-LT in the reference case, the additional electrical energy consumption of the pumps P-LT1, P-MT, and P-HT1 required for the adsorption chiller can be quantified and used as an input for the subsequent annual simulation of the data center.

To this end, the performance maps of the pumps were used to calculate the electrical energy consumption depending on pressure loss and volume flow rate as detailed in Supplemental Note 7. Under the assumption of typical pressure losses and nominal mass flow rates (LT 2.9 m$^3$/h, MT 5 m$^3$/h, HT 2.5 m$^3$/h) the eCoo10 adsorption chiller has an electrical power consumption of approximately 450 W including three pumps and control. The electrical power was calculated for different mass flow rates (100%, 90%, 80%, and 70% of the nominal values). The results are shown in Fig. 21.

Fig. 20. Simplified flow chart of a liquid cooled data center with and without adsorption chiller (“Ads Chiller”). The component abbreviations are as follows: RC = heat rejection unit, P = pump, S = storage.

![Simplified flow chart](image)

Table 6
Component states (“x” indicates that the component is switched on or necessary) for reference case and adsorption chiller case.

<table>
<thead>
<tr>
<th>Component / Unit</th>
<th>Reference case</th>
<th>Adsorption chiller case</th>
</tr>
</thead>
<tbody>
<tr>
<td>P-HT1</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>P-HT2</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>S-HT</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>RC-HT</td>
<td>x</td>
<td></td>
</tr>
<tr>
<td>P-MT</td>
<td>x</td>
<td></td>
</tr>
<tr>
<td>RC-MT</td>
<td>x</td>
<td></td>
</tr>
<tr>
<td>P-LT1</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>P-LT2</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>S-LT</td>
<td>x</td>
<td>x</td>
</tr>
<tr>
<td>RC-LT</td>
<td>x</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 21. Thermal efficiency and (partial) energy efficiency ratio (pEER) vs. heat flow rate low temperature level (LT) for cooling application temperatures. The original points marked with large yellow symbols are taken from Fig. 17 and Fig. 18 (optimal half cycle times: 600 s silica gel, 700 s MIL-100(Fe)). The other results are calculated by decreasing the mass flow rate in LT and MT circuits from 100% to 70% as indicated by the arrows. The mass flow rate in the HT circuit is adapted to achieve a temperature difference of 7 K in all calculations.
efficiency ratio (pEER) as the ratio of heat flow rate $Q_h$ and electrical power $P_{el}$ increases between 100 % (silica gel) and 70 % (MIL-100(Fe)) with a decreasing mass flow rate. The calculated values for the electrical power are between 150 W and 370 W for points shown in Fig. 21. The experimental results of Velte et al. [46] proof that these are not just theoretical values. In the study of Velte et al. [46] the electrical power consumption of an adsorption heat pump was reduced from over 500 W to 180 W by implementing a control algorithm for the pumps. Thus, a reduction from 450 W at nominal mass flow rates to 150 W at lower mass flow rates is a realistic prediction.

The results in Fig. 21 show, that the pEER of an improved adsorption chiller with MIL-100(Fe) reaches values > 20 even at nominal mass flow rates in LT and MT circuits. A pEER > 20 can be achieved with the silica gel chiller only by decreasing the mass flow rates to 80 % of their nominal values which lowers the heat flow rate $Q_{LT}$ and the efficiency.

3.5.2. Heat transformation application

The results for the heat transformation application are shown in Fig. 22. Depending on the temperature boundary conditions the silica gel reference device can achieve maximum heat flow rates of 2.8 and 8 kW. However, the minimum driving temperature (rack outlet temperature) is 50 °C. Although from the equilibrium data we would expect both silica gel and CAU-10-H to work also at 10/45/65 °C it is not possible to reach 7 K temperature difference between rack outlet and inlet under these conditions. If the equilibrium data of silica gel is replaced with the equilibrium data of CAU-10-H the heat flow rate can be increased between 64 % and 118 % with a higher efficiency. Similar to the chiller application, this value can be further improved to 83–158 % as shown in Fig. 23 if we assume an improved adsorption module design as described in section 3.4. The thermal efficiency reaches values between 0.3 (silica gel) and 0.45 (CAU-10-H), which is a rather low value for temperature lifts between 10 K and 15 K. In other words, for lifting one unit of heat in the adsorption half cycle from 50 °C to 65 °C around three units of medium temperature heat from the liquid cooled rack are required for desorption and evaporation as illustrated in Fig. 24. If a compression heat pump is applied that is fed with the same heat flow rate from the liquid cooled rack at 50 °C, it will deliver 108 % of this heat flow rate at 65 °C, which is roughly three times more than it is for the adsorptive heat transformer. Of course, the electrical power consumption of the adsorptive heat transformer can be expected to be about one third of the compression heat pump as illustrated in Fig. 24.

It must be noted that the thermal efficiency of the adsorptive heat transformer strongly depends on the assumption for the heat recovery factor $\eta_{HR}$. Since the heat recovery factor was determined to a range around 0.2 for the adsorption chiller measurements as illustrated in Fig. S17 in Supplemental Note 4, this value was assumed to be valid also for the heat transformation application. However, this assumption has to be proven yet by making experiments or numerical simulations with a more detailed, transient model with heat exchangers being spatially resolved in heat transfer fluid flow direction.

4. Conclusions

In this study a numerical model for adsorption chillers and adsorption heat transformers based on algebraic equations is presented to predict heat flow rates, efficiency, and electrical power consumption of such devices. This model is used for an extensive performance evaluation of adsorption chillers and adsorption heat transformers for the application in data centers. Extensive calibration and validation of the model with measurements of a commercially available silica gel chiller and a silica gel adsorption module ensure the validity of the results. The prediction quality of the model is very high as indicated by the coefficient of variance (CV) of the efficiency being 8.3 % and the CV value of the heat flow rates being 7.2 % on average for 71 different measurements.

The numerical model itself is based on basic thermodynamic considerations using effective heat and mass transfer resistances to account for the heat and mass transfer processes in the adsorption heat exchanger and the evaporator-condenser. Starting with the calibrated and validated model for a silica gel chiller, literature data was used to...
determine the effective heat and mass transfer resistances of coated adsorption heat exchangers. This in-depth analysis of experimental results of other researchers showed that the common round-tube lamella heat exchanger design mainly suffers from heat transfer limitations. The volume scaled effective heat and mass transfer resistance can be reduced by using coated flat-tube lamella heat exchangers. Consequently, this adsorption heat exchanger design was selected as the basis for further calculations.

Besides changing the heat exchanger design, also the impact of different adsorbents on efficiency and heat flow rates was studied. In a pre-selection procedure, more than 40 different adsorbents were evaluated by calculating the equilibrium loading spread and the potential driving temperature differences. This analysis revealed that under the temperatures specified in this study many adsorbents have a higher equilibrium loading spread than silica gel, which potentially yields a higher efficiency. However, besides the efficiency also the heat flow rates are relevant for the application. Including this aspect, the metal organic framework MIL-100(Fe) was selected for further calculations of the cooling application. These calculations show that by changing the adsorbent an increase in cooling power between 15 % and 48 % (depending on the temperatures) can be achieved. This increase can be pushed to 41 % to 78 % if the adsorbent and the heat exchanger design are changed from silica gel to MIL-100(Fe) and from round-tube lamella to flat-tube lamella heat exchanger design. These results underline the necessity of a more holistic approach in improving adsorption chillers – changing the adsorbent alone can yield disappointing results if the heat exchanger design in terms of effective heat and mass transfer resistances are not improved as well. The thermal efficiency is around 0.53 in case of silica gel and around 0.6 in case of MIL-100(Fe) for all studied “Cooling” scenarios.

However, the cooling power strongly depends on the driving temperature (rack outlet temperature) – the calculations with silica gel show a drop from 7 kW to 3.6 kW if the driving temperature is decreased from 55 °C to 45 °C. The improved MIL-100(Fe) chiller shows a drop from 9.8 kW to 6.5 kW at the same temperatures. Installing an adsorption chiller in a data center to utilize the waste heat of the liquid cooled part to provide cooling to the air-cooled part can help to save water that is required in the reference case for the adiabatic cooling. It was shown that the mass flow rate in the hydraulic circuits of the adsorption chiller has a strong impact on the EER while the impact on the thermal efficiency is less pronounced. Of course, the cooling power decreases with decreasing mass flow rate in the hydraulic circuits. The savings in electrical power consumption of the whole system with and without adsorption chiller have to be quantified with an annual simulation of all components.

Besides the chiller application also the limits of the heat transformation application were explored both experimentally and with the numerical model. The results point to rather low thermal efficiencies between 0.35 and 0.4 for the silica gel reference case and the improved CAU-10-H adsorptive heat transformer. An improved adsorption module design and using CAU-10-H instead of silica gel can push the heating power 83–158 % for temperature lifts of 15–10 K. However, even if a heating power between 7 and 14 kW can be achieved with liquid cooled rack outlet temperatures between 50 °C and 55 °C it is questionable if a CAU-10-H heat transformer would be an energetically viable option compared to using a compression heat pump. On the one hand it was shown that the heat from the liquid cooled rack might be removed with an EER of 20 (including fans) relative to the heat transferred to the domestic hot water. This would be roughly one fifth of the electrical power consumption of a compression heat pump operating with an EER of 11. On the other hand, the heat transferred to the domestic hot water by the adsorptive heat transformer is only one third of the heat that is transferred by the compression heat pump. Also in this case the savings in electrical power consumption of the whole system with and without adsorptive heat transformer have to be quantified with an annual simulation of all components. The model presented here is robust and potentially simple enough to perform an annual simulation together with other components, paving the way for an economic analysis in the second step.

CRediT authorship contribution statement

Andreas Velte-Schäfer: Writing – original draft, Visualization, Validation, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. Christian Teich: Writing – review & editing, Visualization, Resources, Formal analysis, Data curation. Maximilian Stahlhut: Writing – review & editing, Visualization, Resources. Thomas May: Writing – review & editing, Resources, Data curation. Ralph Herrmann: Writing – review & editing, Resources. Thorsten Urban: Writing – review & editing, Supervision, Project administration, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.
Data availability

The source code for all calculations as well as the equilibrium data and the original experimental data for the calculations is available online via https://doi.org/10.24406/fordatis/333.

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Appendix A. Supplementary data

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References

