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ANALYSIS OF THE OPERATION OF A COLD THERMAL ENERGY STORAGE SYSTEM COOPERATING WITH AN AIR CONDITIONING SYSTEM UNDER TEMPERATE CLIMATE CONDITIONS

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Abstract. Increased share of renewable energy sources and the dynamic nature of cooling demand in air-conditioning systems have intensified the need for effective thermal energy storage solutions, particularly under temperate climate conditions, where cooling loads vary significantly between day and night. The aim of this study was to investigate whether the integration of a cold thermal energy storage unit based on sorption technology with an air-conditioning system can improve system operation under variable cooling demand and fluctuating electricity availability. It was hypothesized that sorption-based cold storage can reduce peak electricity demand and enhance the flexibility of air-conditioning systems supplied with renewable energy. The study was conducted using a numerical and operational analysis of an air-conditioning system coupled with a sorption-based cold storage unit. System performance was evaluated under representative temperate climate conditions, considering variable cooling loads, intermittent renewable electricity generation, and time-dependent electricity prices. Key performance indicators included cooling energy coverage, storage utilization, and electricity consumption profiles. The results indicate that the application of cold thermal energy storage enables a significant shift in cooling production from daytime peak periods to nighttime or periods of high renewable energy availability. The sorption storage system demonstrated high volumetric energy density, allowing effective cold storage with reduced spatial requirements, and contributed to a measurable reduction in peak electrical power demand. The findings suggest that sorption-based cold thermal energy storage represents a rational and effective solution for improving the operational flexibility and energy efficiency of air-conditioning systems in temperate climates, particularly when integrated with variable renewable energy sources.

Keywords: sorption, thermal storage, cold thermal storage, air conditioning.

1. Introduction

In recent decades, a steady increase in the use of air conditioning systems has been observed in countries located in all temperate climate zones (Krajčík et al., 2023). This trend results from several concurrent factors, including rising average air temperatures, an increasing frequency of heat waves, and growing expectations regarding thermal comfort in residential, commercial, and public buildings. Air conditioning, which was previously considered necessary mainly in hot climate regions, is now becoming a common feature of buildings in temperate climates (Randazzo et al., 2020).

The growing penetration of air conditioning systems leads to a significant increase in electricity demand during summer periods, particularly during daytime hours when cooling loads reach their peak. At the same time, cooling demand exhibits strong diurnal variability, with substantially lower requirements during nighttime. This mismatch between cooling demand and electricity consumption profiles poses a challenge for the efficient and reliable operation of HVAC systems and power grids (Banaszak et al., 2025).

One approach to mitigating the adverse effects of increasing cooling demand is the application of thermal energy storage systems, particularly cold thermal energy storage (Grzebielec et al., 2014). These systems enable the temporal decoupling of cold production and cold utilization, thereby enhancing the operational flexibility of air conditioning installations (Wendołowicz et al., 2024).

Cold thermal energy storage systems can generally be classified into short-term and long-term storage solutions. Short-term cold storage systems are primarily designed to balance daily fluctuations in cooling demand and typically operate on a diurnal cycle, allowing cold to be stored during nighttime hours or periods of low system load and released during peak demand periods. In contrast, long-term cold storage systems are intended

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for seasonal operation, enabling cold to be accumulated during favorable ambient conditions and utilized during periods of increased cooling demand (Grzebielec & Szelągowski, 2019).

The second classification concerns the way in which cooling can be stored: in the form of sensible heat, latent heat, phase change or sorption heat. All such methods are used for cold and heat storage (Pakalka et al., 2025). Each has its advantages and disadvantages. However, what primarily distinguishes them is the energy storage density for cooling. In this comparison, sorption-based cooling storage proves to be the most favorable (Grzebielec & Szelągowski, 2020). For typical chilled water temperatures of 7/12 °C, the volume of a cooling storage system based on sorption is several tens of times smaller than that of a system using sensible heat (Chang et al., 2025).

Implementation of both short-term and long-term cold thermal energy storage systems offers the potential to improve the energy efficiency of air conditioning systems, reduce peak electrical power demand, and facilitate the integration of renewable energy sources.

Sorption energy storage systems can be classified according to the working pairs employed. A summary of the potential configurations is presented in Figure 2. Each working pair exhibits favorable performance only within a specific temperature range. For thermal energy storage systems dedicated to air-conditioning applications, the water–silica gel working pair appears particularly promising (El-Ghetany et al., 2023).

2. Methodology

2.1. Weather parameters of the temperate climate zone

The temperate climate zone is characterized by moderate temperatures and distinct seasons. Average temperatures vary throughout the year, with warm summers and cool to cold winters. Precipitation occurs year-round and includes both rainfall and snowfall, with generally moderate volumes. Winds are usually moderate but may become stronger during storms. Humidity levels are also moderate, and cloud cover changes depending on the season and weather conditions (Wypych et al., 2024). Figure 1 presents typical summer temperature profiles for Warsaw, Poland, which is located in the temperate climate zone.

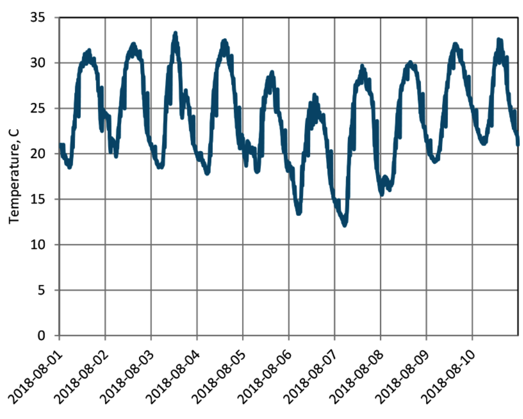

Figure 1. Typical summer outside temperature in Warsaw, Poland

From the perspective of air conditioning system design and operation, the transitional periods are of particular interest, with summer being the most critical. Summer conditions in the temperate climate zone are characterized by large diurnal temperature fluctuations, resulting in variable cooling loads. These day–night temperature differences significantly affect system sizing, control strategies, and energy efficiency, as well as indoor thermal comfort.

The objective of this study is to investigate the potential application of an adsorption-based cold energy storage system capable of storing energy at low temperatures and releasing it at higher temperatures. The entire process is intended to be driven solely by ambient outdoor air temperature, without the use of conventional mechanical cooling or external energy input.

2.2. Sorption energy storage systems

There are many energy storage methods based on sorption processes, which are presented in Figure 2. Due to the operating temperature parameters in the analyzed case, the water–silica gel working pair was selected for further analysis.

2.3. Silica gel – water equlibrium

To analyze water sorption on silica gel, a model based on the Dubinin–Polanyi adsorption potential was used. Adsorption equlibrium is described by Equation (1) (Sapienza et al., 2017):

$$w = w_0 \times \exp\left(\frac{-b \times F}{M_{H_2O}}\right), \tag{1}$$

where: $w\left[\frac{kg_{H_2O}}{kg_{abs}}\right]$ – equilibrium uptake;

$w_0\left[\frac{kg_{H_2O}}{kg_{abs}}\right] = 0.413$ – maximum equilibrium uptake;

$b\left[\frac{kg_{H_2O}}{kJ}\right] = 0.005536$ – fitting parameters determined from the experimental equilibrium data (Aristov et al., 2006); $F\left[\frac{J}{mol}\right] = -R \times T \times \ln\left(\frac{p_{H_2O}}{p_s}\right)$ – Dubinin-Polanyi potential; $M_{H_2O}\left[\frac{J}{mol}\right] = 18.01528$ – molar mass of water.

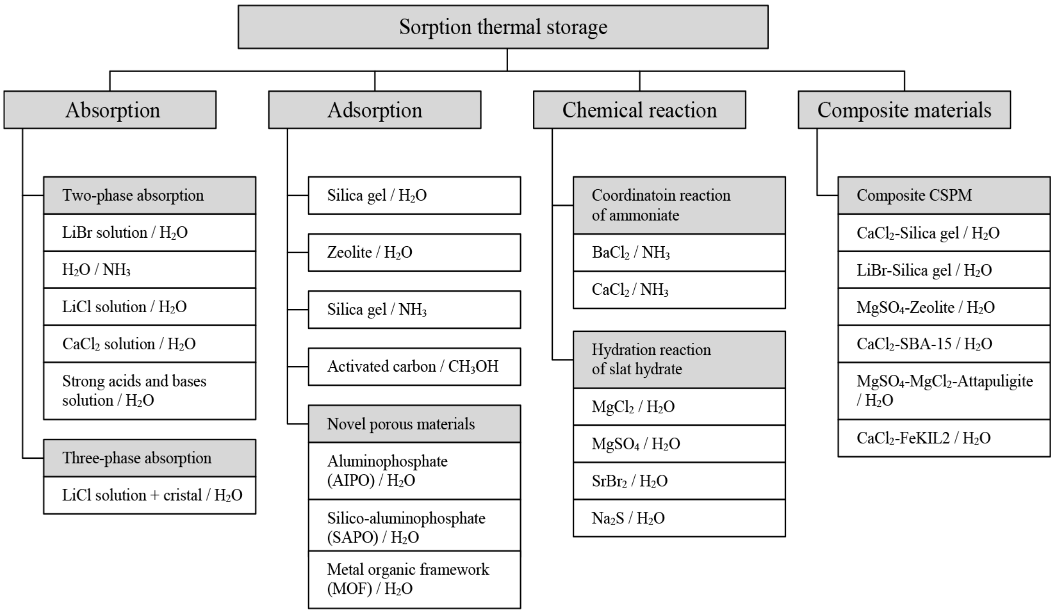

Figure 2. Sorption storage thermal working pairs

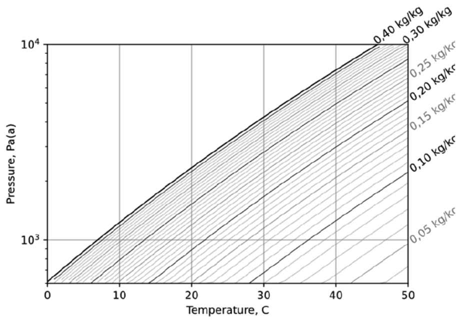

Figure 3. Water–silica gel equilibrium diagram

According to this equation, it is possible to create a p–T equilibrium diagram (Figure 3), which allows for rapid modeling and analysis of sorption systems.

2.4. Cold storage model

The subject of the study is a system in which a conventional refrigeration chiller (operating at 12/7 °C supply and return temperatures) is supported by an adsorption-based cold energy storage unit. The theoretical operation of the system is illustrated in Figures 4 and 5. It is referred to as theoretical because, within the scope of the initial modeling results, the supply water temperatures delivered to the adsorber acting as the storage medium were modified.

Figure 4 illustrates the cold energy storage charging stage. This operating condition occurs when the chiller produces more cooling capacity than required by the load side (the space air-conditioning system). As a

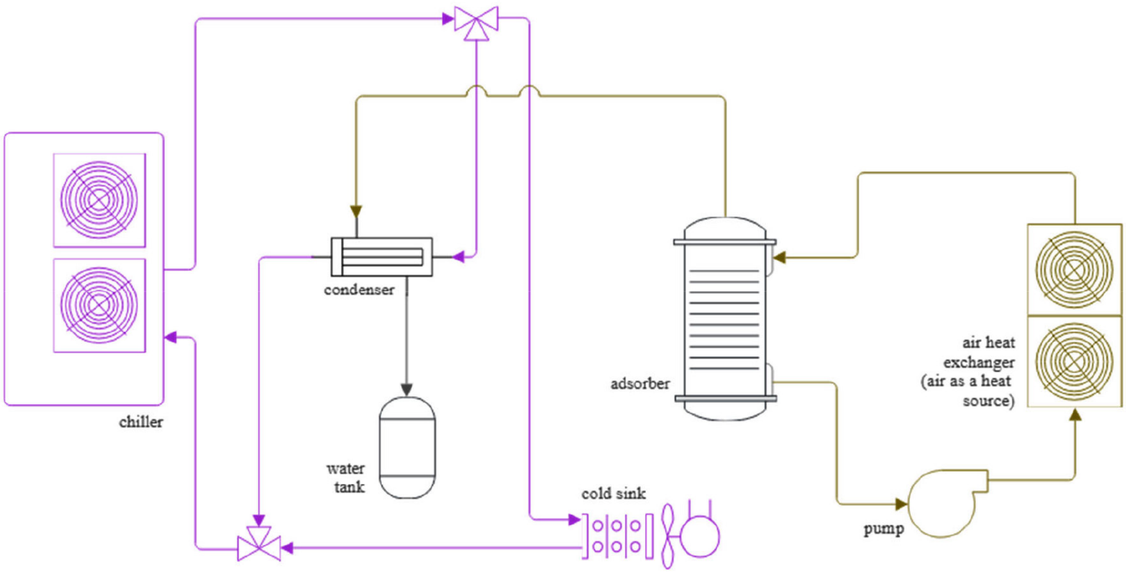

Figure 4. Charging process of cold storage

result, chilled water at a temperature of 7 °C is directed to the heat exchanger connecting the adsorber with the main chilled water network. In this mode of operation, the heat exchanger functions as a condenser. Due to low chilled water temperature, the pressure inside the adsorber is reduced, thereby forcing the desorption process. To intensify this process, and to satisfy the overall energy balance, heat must be supplied to the adsorber. For modeling purposes, it is assumed that water obtained via a heat exchanger (heater) operating with ambient outdoor air at a temperature of 30 °C, is delivered to the adsorber.

From a practical perspective, it should be noted that the actual cold energy storage medium is the water tank rather than the adsorber itself. The adsorber serves solely as a tool for charging and discharging the storage system.

Figure 5 presents the cold energy storage discharging process. This operating condition occurs when the chiller is unable to generate sufficient cooling capacity, and the additional cooling demand is therefore supplied from the storage system. In this mode, the heat exchanger connecting the storage unit with the refrigeration system operates as an evaporator. Consequently, the adsorber, which acts as the energy storage interface, must absorb the heat of evaporation. This is achieved by a water loop coupled with a dry cooler. Cooling the water from 30 °C to 24 °C enables the removal of the evaporation heat in the form of adsorption heat.

At this stage, it should be emphasized that the primary objective of modeling such a system is to determine the required capacity of the cold energy storage. This applies both to the sizing of the water storage tank -where oversizing may be acceptable – and, more importantly, to the correct sizing of the adsorber (bed size). Proper adsorber sizing is critical, as it governs the ability to maintain the required pressure levels. These pressure levels are directly linked to the condensation and evaporation temperatures, corresponding to the charging and discharging processes of the storage system.

Figure 6 presents the energy balance of the adsorber during the desorption process, corresponding to the charging of the cold energy storage system.

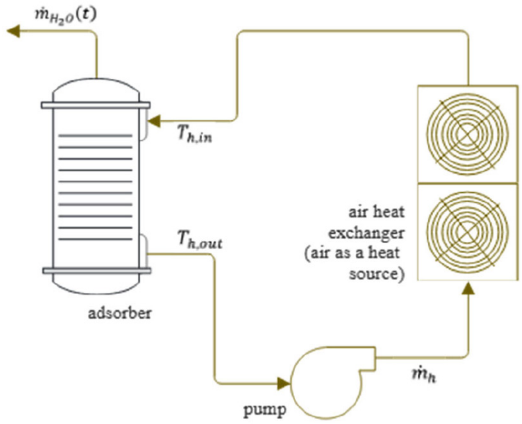

Figure 6. Heat supply scheme for the desorption process

The energy supplied to the adsorber by means of hot water is used to change the temperature of the adsorbent bed, the tube material, and to drive the desorption process itself. Assuming a uniform temperature of the adsorbent bed and the heat exchanger, the energy balance of the adsorber can be determined according to the Equation (2):

$$\dot{m}_h c_w \left(T_{h,in} - T_{h,out}\right) = \left[m_{ads}\left(c_{ads} + w\left(T_{ads}\right)c_v\right) + m_{hx}c_{hx}\right]\left(\frac{dT_{ads}}{dt}\right) + H_{ads}\left(T_{ads}\right) \times m_{ads}\left(\frac{dw}{dt}\right), \quad (2)$$

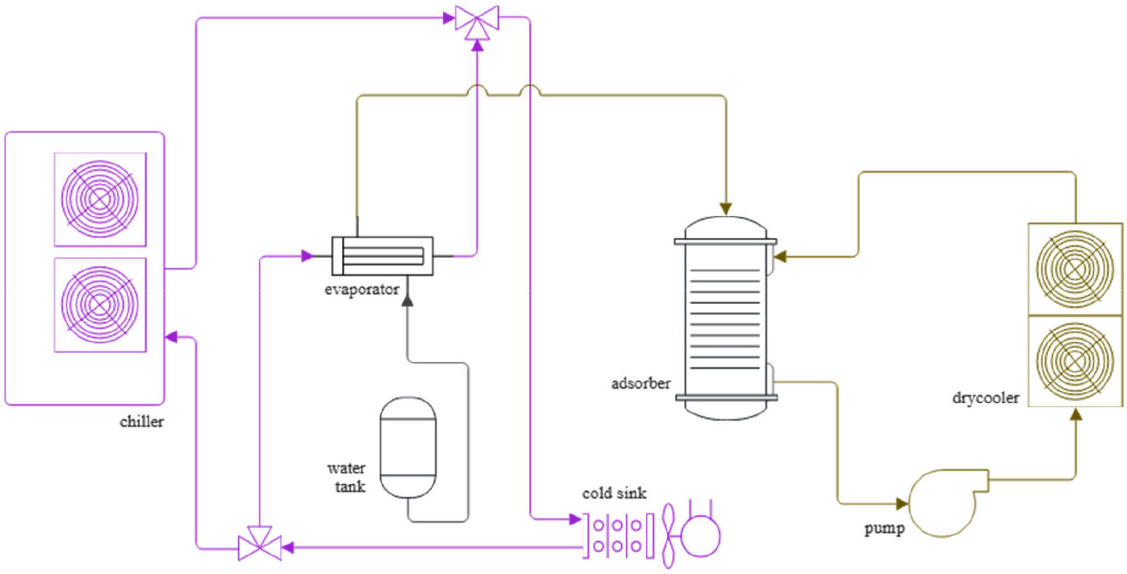

Figure 5. Discharging process of cold storage

where: $\dot{m}_h \left[\frac{\text{kg}}{\text{s}}\right]$ – mass flow rate of the water heating the adsorbent bed; $c_w \left[\frac{\text{kJ}}{\text{kg}\times\text{K}}\right]$ – specific heat capacity of water; $T_{h,in}\left[°\text{C}\right]$ – inlet temperature of the water entering the adsorber; $T_{h,out}\left[°\text{C}\right]$ – outlet temperature of the water entering the adsorber; $m_{ads}\left[\text{kg}\right]$ – mass of dry silica gel in the bed; $c_{ads}\left[\frac{\text{kJ}}{\text{kg}\times\text{K}}\right]$ – specific heat capacity of the silica gel; $w\left[\frac{\text{kg}_{\text{H}_2\text{O}}}{\text{kg}_{\text{abs}}}\right]$ – adsorption uptake; $c_v\left[\frac{\text{kJ}}{\text{kg}\times\text{K}}\right]$ – specific heat capacity of the adsorbed water; $m_{hx}\left[\text{kg}\right]$ – mass of the heat exchanger structural components surrounding the adsorbent bed; $c_{hx}\left[\frac{\text{kJ}}{\text{kg}\times\text{K}}\right]$ – specific heat capacity of the heat exchanger material (stainless steel); $T_{ads}\left[°\text{C}\right]$ – bed temperature; $H_{ads}\left[\frac{\text{kJ}}{\text{kg}}\right]$ – enthalpy of adsorption.

Knowing the energy balance, it is possible to determine the instantaneous mass flow rate of water vapor released from the adsorbent bed.

$$\dot{m}_{\text{H}_2\text{O}}(t) = m_{ads}\left(\frac{dw}{dt}\right). \tag{3}$$

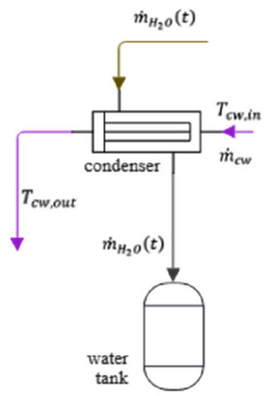

Figure 7. Energy balance of condenser

The released water vapor is directed to the main heat exchanger, which functions as a condenser. The condenser is cooled by chilled water from the chiller at 7 °C (shown in Figure 7). The energy balance of the condenser is as follows:

$$\dot{m}_{cw}c_{cw}\left(T_{cw,out} - T_{cw,in}\right) = \dot{m}_{\text{H}_2\text{O}}\left(T_a - T_{cond}\right) + \dot{m}_{\text{H}_2\text{O}} \times h_{fg}, \tag{4}$$

where: $\dot{m}_{cw}\left[\frac{\text{kg}}{\text{s}}\right]$ – mass flow rate of chilled water; $T_{cw,out}\left[°\text{C}\right]$ – outlet temperature of the water leaving the condenser; $T_{cw,in}\left[°\text{C}\right]$ – inlet temperature of the water entering the condenser; $T_{cond}\left[°\text{C}\right]$ – condensation temperature at the given pressure; $T_a\left[°\text{C}\right]$ – desorption process temperature; $h_{fg}\left[\frac{\text{kJ}}{\text{kg}}\right]$ – latent heat of condensation of water.

During the discharging of the storage system, the heat from the sorption process must be absorbed. Figure 8 presents the energy balance of this process.

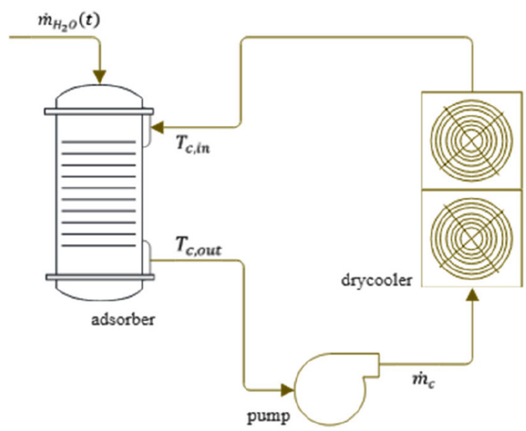

Figure 8. Adsorption heat dissipation process

Assuming that the temperature of the adsorbent bed and the heat exchanger remain constant, the energy balance can be expressed by the Equation (5):

$$\begin{aligned}&\dot{m}_c \times c_w\left(T_{c,out} - T_{c,in}\right) = \\ &\left[m_{ads}\left(c_{ads} + w\left(T_{ads}\right)c_v\right) + m_{hx} \times c_{hx}\right]\left(\frac{dT_{ads}}{dt}\right) + \\ &m_{ads}\left(\frac{dw}{dt}\right)c_v\left(T_{ads} - T_{evap}\right) + H_{ads} \times m_{ads}\left(\frac{dw}{dt}\right),\end{aligned} \tag{5}$$

where: $\dot{m}_c\left[\frac{\text{kg}}{\text{s}}\right]$ – mass flow rate of water cooling the adsorbent bed; $c_w\left[\frac{\text{kJ}}{\text{kg}\times\text{K}}\right]$ – specific heat capacity of water; $T_{c,in}\left[°\text{C}\right]$ – inlet temperature of the water entering the adsorber; $T_{c,out}\left[°\text{C}\right]$ – outlet temperature of the water leaving the adsorber; $w\left[\frac{kg_{H_2O}}{kg_{abs}}\right]$ – adsorption uptake; $T_{ads}\left[°C\right]$ – adsorption process temperature; $T_{evap}\left[°C\right]$ – evaporation temperature of water.

Similarly to the desorption process, in the case of adsorption, the mass flow rate of water flowing through the main heat exchanger (which now functions as an evaporator) is determined according to the Equation (6):

$$\dot{m}_{H_2O}\left(t\right)=m_{ads}\left(\frac{dw}{dt}\right). \tag{6}$$

Figure 9, in turn, presents the energy balance occurring during this process in the evaporator.

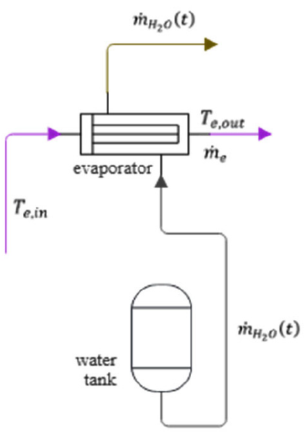

Figure 9. Energy balance of the evaporator

$$\dot{m}_e.c_w\left(T_{e,in}-T_{e,out}\right)=\dot{m}_{H_2O}\left(t\right)\times h_{fg}, \tag{7}$$

where: $\dot{m}_e\left[\frac{kg}{s}\right]$ – mass flow rate of chilled water; $T_{e,out}\left[°C\right]$ – temperature of the water leaving the evaporator; $T_{e,in}\left[°C\right]$ – temperature of the water entering the evaporator.

All equations were solved using the finite difference method, which is a suitable approach for preliminary energy balance calculations (Owczarek & Baryłka, 2019).

This process also provides information about the instantaneous cooling capacity recovered from the storage system.

3. Results

In the first step of the modeling, it was assumed that the phase change temperatures would be identical to the chilled water temperatures. This means that during the desorption process, the pressure would be maintained at the level corresponding to 12 °C, and during adsorption, it would correspond to the pressure at 7 °C. Assuming that air is to be used in both the sorption and desorption processes, the adsorbent bed temperature would vary between 24 °C and 30 °C (Figure 10). This range indicates that, if the bed operates only within these temperature saturation levels, the uptake of the bed would vary minimally, between 0.175 and 0.180, reflecting a relatively stable adsorption performance within this operating window. However, the above would only be possible if the heat exchangers were 100% efficient or if the process duration were infinitely long, which is practically impossible.

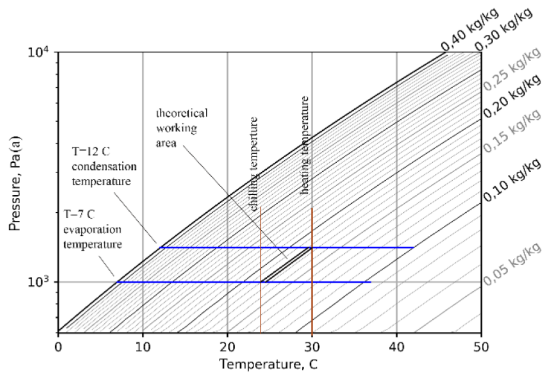

Figure 10. Temperature limitations of the original system

Therefore, it must be concluded that, under these assumptions, the construction of such a system is not feasible. The system requires an additional heat source to be used in the desorption process (charging the storage). For the calculations, the required mass of the sorbent was determined as a function of the temperature of the regenerating water, in order to store 1 kWh of cooling capacity. Results are shown in Figure 11.

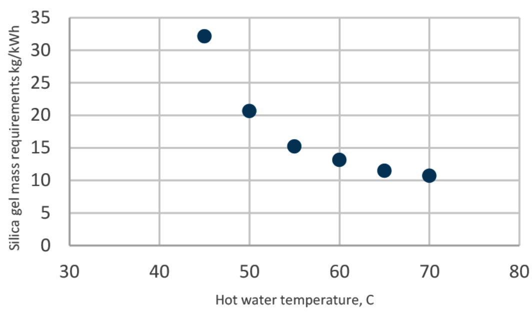

Figure 11. Silica gel mass requirements

It should be noted that, to store the same amount of cooling using the sensible heat of water (at 7/12 °C), approximately 172 kg of water is required. In the calculations, the minimum size of the water storage tank was also determined, and amounted to 1.45 dm^3 per 1 kWh of cooling capacity.

It was also decided to examine how lowering the temperature during the discharging process affects storage capacity. The results are presented in Figure 12. Because it was used drycooler for chilling the proces results are calculated according the ambient air temperature.

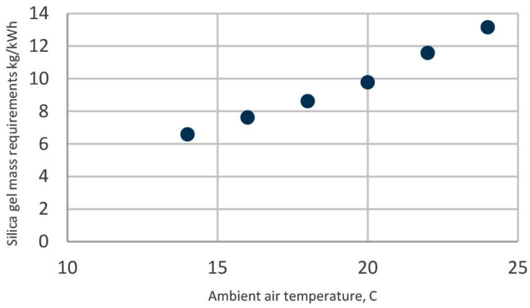

Figure 12. Silica gel mass requirements according to ambient air temperature

The results show that the lower the temperature of the air to which the heat of adsorption is released, the greater the storage capacity becomes – fewer kilograms of silica gel are needed to store 1 kWh of cooling.

4. Conclusions

This study presents the operating principle of an adsorption-based cold energy storage system working at typical chilled water temperatures used in HVAC systems, namely 7/12 °C. The first conclusion from the conducted work is that the system cannot operate without an additional heat source. This is due to the fact that, when relying solely on ambient air parameters in a temperate climate zone during the summer, desorption (i.e., charging the storage) is not feasible. Research indicates that the supply water temperature to the desorber must exceed 40 °C, while a rational and effective operation is achieved at temperatures above 55 °C. Therefore, for the system to be economically viable, it must utilize heat from renewable sources or waste heat.

The second conclusion is that the lower the ambient temperature, the greater the amount of cooling that can be stored in the system (or, more precisely, the greater the amount of cooling that can be extracted from it). However, a practical limitation arises in that periods of low ambient temperature typically coincide with reduced demand for cooling.

To address these challenges, several optimization strategies can be considered. Intelligent control and adaptive charging/discharging schedules can align storage operation with periods of heat availability and cooling demand. Thermal system design improvements, such as enhanced heat and mass transfer surfaces in the desorber and drycooler, can increase adsorption/desorption efficiency. Additionally, managing multiple adsorption beds with staggered cycles can help maintain a stable cooling output and better exploit favorable ambient conditions.

These constraints indicate that adsorption-based cold storage is viable primarily in applications with continuous cooling demand (e.g., data centers or industrial processes), systems integrated with stable high-temperature heat sources, or locations with favorable ambient conditions during the discharge phase. In conventional comfort cooling applications in temperate climates, the mismatch between cooling demand, heat availability, and ambient conditions significantly limits practical applicability. Nevertheless, the outlined optimization strategies offer potential pathways to improve the performance and economic viability of such systems.

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