



Norwegian University of  
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# Transitional season solar energy storage using physical sorption materials for low carbon district heating in cold climate

Lagring av solenergi i overgangsperioder ved  
hjelp av materialer basert på fysisk sorpsjon  
for lavkarbon fjernvarme i kaldt klima

**Ellen-Andrea Tilrum Myhre**

Master of Energy and Environmental Engineering

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Supervisor: Vojislav Novakovic, EPT

Co-supervisor: Yanjun Dai, Shanghai Jiao Tong University

Norwegian University of Science and Technology  
Department of Energy and Process Engineering



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**MASTER THESIS**

for

Student Ellen-Andrea Myhre

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**Transitional season solar energy storage using physical sorption materials  
for low carbon district heating in cold climate***Lagring av solenergi i overgangsperiodene ved hjelp av materialer basert på fysisk sorpsjon  
for lavkarbon fjernvarme i kaldt klima***Background and objective**

Modern buildings are expected to cover their energy need with own onsite production based on renewable sources. Harvesting of solar energy is one of possible solutions. However, the mismatch between available energy and demands make this a challenge. Seasonal heat storage is a method to postpone use of solar energy from periods of availability to periods with the demand. Sorption heat effect of desiccant materials, such as silica gel, molecular sieve, et al., can be used for seasonal solar energy storage without heat loss caused by temperature difference (like water pond storage). In summer, the solar radiation is strong and is used for desiccant regeneration; the regenerated desiccant has the ability for water vapour sorption, which will release a lot of heat. The heat can be used for space heating in wintertime through water sorption by desiccant unit.

The goal for this collaborative activity is to develop design methods for solar energy storage using physical sorption materials. The work is based on analyses of performances of the small prototype solar energy storage using physical sorption materials installed at the Green Energy Laboratory (GEL) of the Shanghai Jiao Tong University (SJTU), China. The study comprise laboratory measurements and simulations.

This collaborative assignment is realized as a part of the Joint Research Centre in Sustainable Energy of NTNU and SJTU. The necessary scientific and practical background for the work was developed through the project assignment accomplished at NTNU. The major part of the work on analysis and development of design methods will be performed during this Master thesis work accomplished at the GEL of SJTU.

**The following tasks are to be considered:**

1. Develop a detailed work plan to build a mathematical model of a plant applying physical sorption materials for transitional season solar energy storage.
2. Complete a critical analysis of existing models and simulation tools for physical sorption materials for transitional season solar energy storage. Discuss findings and propose the most suitable model(s) for further use.
3. Propose a mathematical model applying physical sorption materials for transitional season energy storage for a domestic household.
4. Propose appropriate design methods for systems using physical sorption materials for transitional season solar energy storage.
5. Make a draft proposal (6-8 pages) for a scientific paper based on the main results of the work performed in the master thesis.
6. Make proposal for necessary further work on the same topic.

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Within 14 days of receiving the written text on the master thesis, the candidate shall submit a research plan for his project to the department.

When the thesis is evaluated, emphasis is put on processing of the results, and that they are presented in tabular and/or graphic form in a clear manner, and that they are analyzed carefully.

The thesis should be formulated as a research report with summary both in English and Norwegian, conclusion, literature references, table of contents etc. During the preparation of the text, the candidate should make an effort to produce a well-structured and easily readable report. In order to ease the evaluation of the thesis, it is important that the cross-references are correct. In the making of the report, strong emphasis should be placed on both a thorough discussion of the results and an orderly presentation.

The candidate is requested to initiate and keep close contact with his/her academic supervisor(s) throughout the working period. The candidate must follow the rules and regulations of NTNU as well as passive directions given by the Department of Energy and Process Engineering.

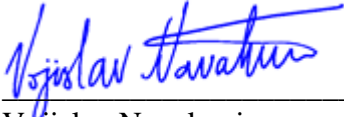
Risk assessment of the candidate's work shall be carried out according to the department's procedures. The risk assessment must be documented and included as part of the final report. Events related to the candidate's work adversely affecting the health, safety or security, must be documented and included as part of the final report. If the documentation on risk assessment represents a large number of pages, the full version is to be submitted electronically to the supervisor and an excerpt is included in the report.

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- Work to be done in lab (Water power lab, Fluids engineering lab, Thermal engineering lab)
- Field work

Department of Energy and Process Engineering, 15. February 2018



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Vojislav Novakovic  
Academic Supervisor

Research Advisor:

Prof. Yanjun DAI, Shanghai Jiao Tong University, e-mail: yjdai@sjtu.edu.cn



# Preface

This master's thesis was written as part of the master's degree in Energy and Environmental Engineering at The Norwegian University of Science and Technology (NTNU). The thesis is part of a collaboration between the Joint Research Center in Sustainable Energy at NTNU and Shanghai Jiao Tong University (SJTU). The work was carried out at SJTU in Shanghai during the spring semester of 2018.

The main objective of the thesis is to design a mathematical model of a seasonal sorption energy storage for a domestic household located in Beijing, China, and further evaluate and test it for a theoretical case located in Trondheim, Norway.

I would like to thank my supervisor at NTNU, Professor Vojislav Novakovic, for his patience and effort to stay positive despite the challenges that comes with communicating across different continents. Additionally, I would like to extend my gratitude to my co-supervisor at SJTU, Professor Yanjun Dai, for his expertise and guidance on the topic along with his effort to include me in the educational community at SJTU. I further wish to thank both Prof. Novakovic and Prof. Dai for making my stay in Shanghai, China, possible. The exchange has challenged me and taught me a lot about both thermal energy storage and myself.

Lastly, I wish to honor my father, Bent Myhre, for his endless help and motivation throughout both this thesis and my entire master degree. I would have never come this far without the valuable support I have received from him and the rest of my family during my years at NTNU.



Ellen-Andrea Tilrum Myhre

Harstad, 2018





# Abstract

The aim of the work with this thesis has been to acquire enough knowledge about solar energy and thermochemical storage systems to build a mathematical model of a theoretical system located in Beijing, China, and further test the model for a second theoretical case located in Trondheim, Norway.

A basic presentation of the solar potential and solar energy technology is included to introduce an understanding of the sun's potential as renewable source and how it can be utilized. A simplified method of dimensioning a solar collector system along with a description of the available stationary solar collectors is also presented.

While thermochemical energy storage is the main focus of this thesis, it also includes an explanation of the two other types of thermal energy storage; sensible and latent energy storage. The focus is further limited to thermochemical energy storage using sorption, with a special focus on seasonable storage. The purpose of the storage is to improve the general efficiency of the solar collector system by limiting the disparity between supply and demand. As thermochemical energy storage has a higher energy capacity and requires lower volumes, it has through literature studied been regarded as the best alternative between the available options of thermal energy storage. However, the technology is currently only in its research phase and will require more effort to optimize the working materials and components before it can be realized commercially.

The goal of the assignment is met by presenting a complete mathematical model for a 100 m<sup>2</sup> domestic household in Beijing, China. The model of the seasonal sorption storage using silica gel is functioning for the dimensioning parameters of the Beijing climate, and is successfully tested for the case located in Trondheim. Through simulations, the importance of correct water content, relative humidity and charging temperature became apparent. Results show that the model is able to store 0.089 kW/kg water while meeting the outlet temperature requirement of 35°C, but that the energy demand and water content heavily affects the necessary volume and size.



# Sammendrag

Målet med arbeidet med denne oppgaven har vært å tilegne seg nok kunnskap om solenergi og termokjemiske lagringssystemer for å kunne bygge en matematisk modell av et teoretisk system plassert i Beijing i Kina, og videre teste modellen for en ny plassering i Trondheim i Norge.

En grunnleggende presentasjon av solenergiens potensial og teknologi for solenergi er inkludert for å introdusere en forståelse av solens potensial som fornybar energikilde og hvordan den kan utnyttes. En forenklet metode for å dimensjonere et solfangersystem i tillegg til en beskrivelse av de tilgjengelige stasjonære solfangerne er også presentert.

Mens termokjemisk energilagring er hovedfokuset i denne oppgaven, inkluderer den også en forklaring av de to andre typene termisk energilagring, følbar og latent energilagring. Fokuset er videre begrenset til termokjemisk energilagring med adsorpsjon med et spesielt fokus på sesonglagring. Meningen med energilagringen er å forbedre den generelle effektiviteten for solfangersystemet ved å minimere forskjellen mellom tilførsel og behov. Siden termokjemisk energilagring har en høyere energikapasitet og trenger mindre volum, har det gjennom litteratur blitt regnet som det beste alternativet av de tilgjengelige metodene innen termisk energilagring. Teknologien er likevel i forskningsfasen og behøver mer innsats for å optimere materialer og komponenter før det kan tas i bruk kommersielt.

Målet med oppgaven er møtt ved å presentere en komplett matematisk modell for en 100 m<sup>2</sup> privat husholdning i Beijing. Modellen for sesonglagring av solenergi med silica gel fungerer for de dimensjonerende parameterne for Beijings klima, og har lyktes i å teste for et tilfelle plassert i Trondheim. Gjennom simuleringer har viktigheten med å velge korrekt vanninnhold, relativ fuktighet og ladetemperatur åpenbart seg. Resultatene viser at modellen er egnet til å lagre 0,089 kW/kg vann mens kravet til utgangstemperatur på 35°C blir møtt. Men det er også tydelig at energibehovet og vanninnholdet sterkt påvirker det nødvendige volumet og størrelsen på lagringsenheten.



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# Nomenclature

$A$	Area [ $\text{m}^2$ ]	$RH$	Relative humidity [-]
$B$	Atmospheric pressure [Pa]	$SF$	Solar fraction [-]
$C_P$	Specific heat capacity for constant pressure [ $\text{kJ}/\text{kgK}$ ]	$T$	Temperature [ $^{\circ}\text{C}$ ] or [K]
$f_a$	Correctional factor, azimuth [-]	$W$	Water content [kg water/kg desiccant]
$f_H$	Correctional factor, tilt [-]	$Y$	Humidity ratio [-]
$H$	Solar irradiation [ $\text{Wh}/\text{m}^2$ ]	<b>Greek</b>	
$h$	Enthalpy [ $\text{kJ}/\text{kg}$ ]	$\eta$	Efficiency [-]
$\dot{m}$	Mass flow [ $\text{kg}/\text{s}$ ]	$\bar{\eta}$	Average efficiency [-]
$P_s$	Saturation pressure [Pa]	$\psi$	Tilt of solar collector [ $^{\circ}$ ]
$Q$	Energy [kW]	$\theta$	Mean yearly temperature [ $^{\circ}\text{C}$ ]

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## Subscripts

$a$	Air	$HEX$	Heat exchanger
$ads$	Adsorption	$in$	Inlet
$amb$	Ambient	$max$	Maximum
$avg$	Average	$out$	Outlet
$B$	Beijing	$opt$	Optimal
$coll$	Solar collector	$prac$	Practical
$cond$	Condenser	$SH$	Space heating
$DH$	Dehumidification	$silica$	Silica gel
$eq$	Equilibrium	$T$	Trondheim
$evap$	Evaporator	$w$	Water
$fl$	Floor	$ym$	Yearly mean
$g$	Gas		

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## Abbreviations

$TES$	Thermal energy storage	$PCM$	Phase change material
$SHS$	Sensible heat storage	$nZEB$	Nearly zero energy building
$LHS$	Latent heat storage		

# 1. Introduction

## 1.1 Objective

The objective for this master thesis is to develop a mathematical model for a seasonal solar energy storage using physical sorption materials. The model should be dimensioned to function in Beijing, China. It should further be tested to determine whether such a storage could function in Norway, based on the set points of the system located in Beijing. The work should further include a critical analysis of existing models for transitional seasonal storage.

The thesis is part of a collaboration between the Joint Research Centre in Sustainable Energy of Norwegian University of Science and Technology, Norway, and Shanghai Jiao Tong University, China.

## 1.2 Delimitations

The author spent four months at Shanghai Jiao Tong University and the Green Energy Lab. In collaboration with Professor Yanjun Dai and his students, the author gained new and important knowledge in order to write this thesis. As there does not exist a prototype for reasonable energy storage nor silica gel at the Green Energy Lab it was necessary to build a completely theoretical model without the possibility of verifying it. Since China does not have standards for energy demands or energy efficient buildings in general, the author was forced to rely greatly on the information obtained through meetings with Prof. Dai. This information is gathered and presented in Appendix A in order to give a wholesome impression of the Chinese standards.

## 1.3 Outline

Chapter 2 gives a general introduction of the potential of the sun as an energy source along with a brief introduction to the available solar collector technology. This chapter also includes a description of how to dimension a solar collector system dependent on a given location. Further, information about thermal energy storage is included. Chapter 3 describes the thermochemical energy storage and its options in detail. A literature research regarding thermochemical energy storage is also enclosed focusing specifically on materials and long-term research. The research done at SJTU is described separately in chapter 4.

Chapter 5 describes the working principle of the seasonal sorption energy storage and the actual dimensioning of the solar collector systems located in Beijing, China and Trondheim, Norway. It also includes a description of the given system that should be used in the dimensioning of the sorption storage system. Chapter 6 describes the mathematical model in detail with equations for both charging and discharging. Chapter 7 presents the results for the mathematical model built for Beijing, China and Trondheim, Norway. A detailed calculation is included as well as a summary of the system parameters and variables.

The conclusion in chapter 8 summarizes the work and highlights the important conclusion of the work. The last chapter presents suggestions for further work based on the conclusions of the previous chapter.

## 1.4 Introduction

Through a report written by the Intergovernmental Panel (IPCC) [1], it is revealed that humans are, with 95% certainty, responsible for the global warming the world is experiencing. The same report confirms that the influence is still increasing. If the trend is allowed to continue, the climate will be disrupted to the point of no return as we risk causing severe and irreversible impacts on the ecosystems on earth. The parties of the United Nations Framework Convention on Climate Change (UNFCCC) has therefore taken action in order to prevent this from happening. This is done through the Paris Agreement of 2015, which aims to keep the global temperature rise below 2°C compared to pre-industrial levels in the respective countries [2]. Research has proven a direct relation between the global average temperature and the amount of greenhouse gas in the atmosphere. Therefore, by reducing the current concentration of carbon dioxide, CO<sub>2</sub>, and limiting the emission of greenhouse gasses into the atmosphere, they hope to reduce the increase in the global average temperature. This is an agreement and goal close to every country in the world (174 countries by July 2018 [3]) has committed to work towards.

According to the European Union (EU), the building sector will be forced to take measures in order to reach the goals of the Paris Agreement [4]. Buildings are in general responsible for 40% of the total energy consumption, and 36% of the CO<sub>2</sub> emission in the EU [5]. Reducing the energy consumption would in other words make a significant difference. Therefore, the concept of nearly zero energy building (nZEB) was introduced. The goal is that every public building will be nZEB by the end of 2018, and that nearly every building will within the category by the end of 2020. In a nearly zero energy building, the energy demand is

very low or nearly zero and the demand should be covered by renewable energy [4]. In this way, it is possible for the building to compensate for its own greenhouse gas emission throughout its lifetime. The energy demand can be reduced by increasing the insulation in exterior walls, floors and roofs along by installing well-insulated windows. By utilizing solar collectors and panels for energy, it is possible to produce the heat and electricity on site. Through solutions for energy storage the efficiency of solar heat systems can be increased by prolonging the usage of solar energy.

## 2. Solar energy and storage

### 2.1 Solar potential

On average  $1367 \text{ W/m}^2$  solar radiation hits the top of the earth's atmosphere perpendicularly. This measure is known as the solar constant [6]. At clear conditions a total of  $1000 \text{ W/m}^2$  radiation will reach the earth's surface, while the remaining is most likely reflected back into the universe [6, 7]. The International Energy Agency [6] estimate that about 885 million terawatt hours (TWh) solar power reach the earth's surface in a year. That equals 5700 times the total energy consumed in 2016 (155 000 TWh [8]). The sun is additionally the source of wind, falling water, tide and geothermal heat, which all can contribute to producing energy leaving it with great potential for energy production.

The solar energy available is affected by the season, weather and localization of the receiver. Equator is located on the highest curve facing the sun, and the sun hits the equator close to perpendicularly. Therefore, countries close to equator will receive more sun compared to countries further away. Norway for example, which is located far north on the globe, will receive less solar energy than a country closer to equator. Throughout the country the yearly solar irradiation of Norway, measured on a horizontal surface, varies between 600-900 kWh/m<sup>2</sup>·year [7]. China, on the other hand, located in the northeastern part of East Asia and about half the distance from equator compared to Norway, receives between 1050-1750 kWh/m<sup>2</sup>·year [9].

### 2.2 Solar radiation

Solar radiation is split into three types: beam, diffuse and reflected radiation. Beam radiation is direct, unobstructed radiation from the sun. This type of radiation is high on a

cloudless and sunny day, while a day of overcast will have little beam radiation. Beam radiation is often characterized by its ability to create shadows, and is commonly experiences as “sunshine” [6, 10].

Diffuse radiation, however, is the complete opposite of beam radiation as it is radiation diffused by the atmosphere. Therefore, while the beam radiation is low during overcast days, the diffuse radiation will be at its highest. Thus, will it be low on a clear day. This radiation is often experienced as “daylight” [6, 10].

The last type of radiation is the reflected radiation. This is the radiation that fails to reach the surface in question at first, but is later reflected off of the surroundings onto the surface. The surrounding’s ability to reflect the radiation varies greatly. While green grass reflects about 20%, fresh snow reflect a total of 80% [10]. As seen in Figure 2.1 the beams come in at an angle, which makes it challenging to reflect off the surroundings onto a plane horizontal surface. In this instance, a tilted surface would be more fitting. The reflected radiation will more easily be able to reach the surface as it is reflected, as illustrated in Figure 2.2.

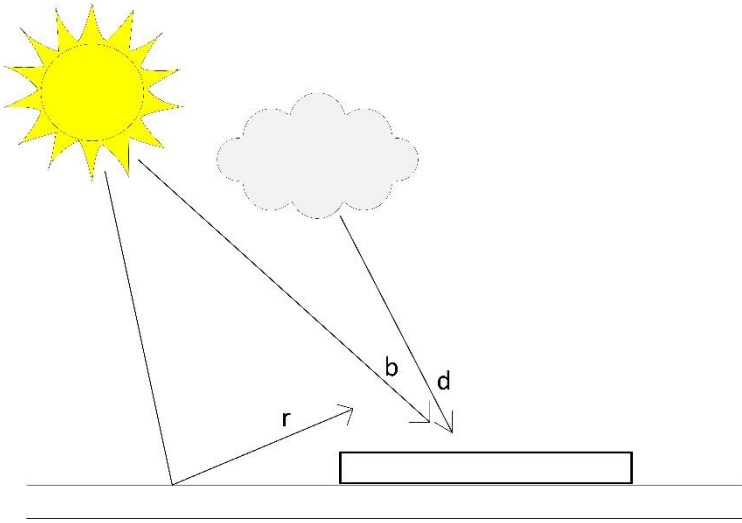


Figure 2.1: Sketch of global radiation for a horizontal collector.

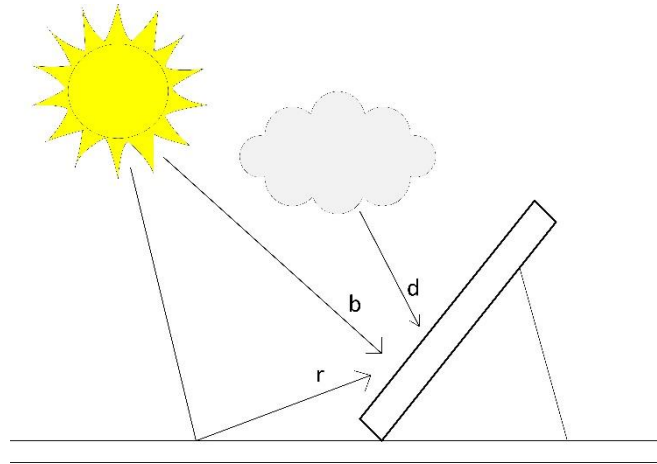


Figure 2.2: Sketch of global radiation for a tilted collector.

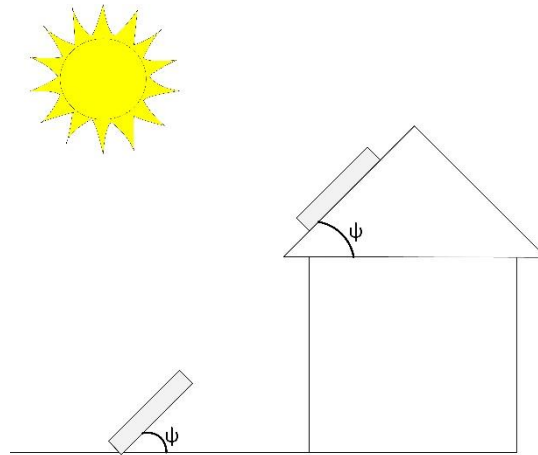
### 2.3 Orientation

The tilt of a panel ( $\psi$ ) is established in relation to the horizontal plane. Therefore, a horizontal panel has a tilt of  $0^\circ$ , while a vertical is tilted  $90^\circ$  [11].

The degree of tilt should be optimized for each panel, as the optimal tilt depends on the latitude and cloudiness of the location. All though, an angle of  $45^\circ$  is often used as a general value. To calculate the optimal tilt throughout a year, it is necessary to take the daily optimal angle and radiation into account [7]. If the panel, for whatever reason, is mounted at a different tilt than the optimal it is appropriate to use a correctional factor,  $f_H$ , based on how much it deviates.

$$\psi_{opt} = \frac{\sum_{n=1}^{n=365} (\psi_n \cdot H_n)}{\sum_{n=1}^{n=365} H_n} \quad (2.1)$$

$n$	=	day of the year	[-]
$\psi_n$	=	optimal angle at day n	[°]
$H_n$	=	average irradiation on day n	[kWh/(m <sup>2</sup> ·day)]



*Figure 2.3: Sketch to demonstrate the tilt of a panel.*

### 2.3.1 Azimuth

A panel's azimuth denote the angle between the direction of the panel and south. Directly south is considered the optimal azimuth angle as the sun is at its highest and most powerful here, giving south an azimuth angle of  $0^\circ$ . To account for the deviation in azimuth in calculations it is necessary to use a correctional factor,  $f_a$ , based on how much it deviates.

## 2.4 Solar collectors

As already established, the sun provides the earth with enough energy to cover the world's yearly energy demand. However, to do so, there is a need for sufficient technology. Solar panels enable an electricity production, while heat can be produced with solar collectors. Heat production is the only alternative covered in this report.

Solar energy collectors produce heat by collecting radiation energy and exchanging heat by heating a transport medium indirectly. This functions much like a traditional heat exchanger. The heat can in turn be utilized for both space heating and domestic hot water. While movable, sun-tracking collectors are available, only stationary collectors will be covered in this report.

### 2.4.1 Flat plate solar collectors

A flat plate solar collector has a thick casing with a dark glass plate on top. The radiation passes through the glazing and hits a blackened absorber plate located under some thin fluid



tubes filled with transport medium. The absorber is usually a thin copper or aluminum plate with a special coating to ensure high absorptivity and low reflectance. The absorbed energy is further transferred to the transport medium, for transportation to storage or to directly cover hot water or space heating demands. The glass plate enables a greenhouse effect within the box by admitting as much short-wave radiation as possible, without any discharge of thermal radiation. Both direct and diffuse radiation are collected.

An important feature of the collector is the insulation along the sides of the casing. This makes it possible for the collector to withstand high temperatures, and reduce conduction losses [7, 12]. The flat plate solar collector can either be mounted on existing roofs or walls, or be integrated in the façade. By integrating the panel, the need for façade materials like cladding and roof tiles will be reduced [7]. The robust composition of these panels gives them a long life time, and it is common to estimate around 30-50 years [7].

## 2.4.2 Evacuated tube collectors

Evacuated tube collectors are differently designed than the flat plate collector. These collectors consist of individual black evacuated glass pipes in a row, all connected to the same manifold. Every glass pipe is vacuum-sealed and contains a heat pipe that function as a highly efficient thermal conductor. This heat pipe is attached to an absorber plate much like in the flat plate collector. The design of the heat pipe can vary from single to double glass pipes, where double glass can increase the lifetime of the collector but increasing the cost. The glass pipe utilizes latent heat by phase change material, making it more suitable for poor weather conditions than a flat plate collector as it will not overheat nor freeze. A well-insulated evacuated solar collector is also very efficient, and can have up to 30% higher efficiency than a flat plate collector [7]. China is the biggest producer of this collector, and naturally therefore the biggest user.

## 2.5 Dimensioning solar collector

In order to dimension a solar collector it is necessary to identify the energy demand and solar irradiance available at the given location. Since the collected heat can be utilized for both domestic hot water and space heating, it is necessary to decide what and how much of the demand the solar heat should cover. The irradiance will vary depending on the tilt and azimuth

chosen, so it will be important to optimize this prior to the dimensioning. The necessary energy output can be then be calculated based on the demand, efficiency and irradiance, like seen in Equation 2.2.

$$Q_{out} = H_{opt,year} \cdot \overline{\eta_{coll}} \cdot f_A \cdot f_H \left[ \frac{kWh}{m^2} \right] \quad (2.2)$$

$H_{opt,year}$  = yearly irradiance at optimal angle [kWh/m<sup>2</sup>]

$\overline{\eta_{coll}}$  = average efficiency of solar collector [-]

$f_a$  = correctional factor for azimuth angle [-]

$f_H$  = correctional factor for tilt [-]

Further, in order to dimension the system it is important to calculate the necessary collector area. It is calculated based on the energy available and the solar fraction,  $SF$ . The solar fraction denotes how much of the total demand the solar collector should cover. In Norway, it will typically be 0.5-0.6 for domestic hot water alone, or 0.5 for domestic hot water and 0.1-0.3 for space heating in a combined system [7]. The necessary solar collector area is calculated through Equation 2.3.

$$A_{coll} = \frac{\sum Q_{demand} \cdot SF \cdot A_{fl}}{Q_{out}} [m^2] \quad (2.3)$$

$A_{coll}$  = necessary solar collector area [m<sup>2</sup>]

$A_{fl}$  = area of household [m<sup>2</sup>]

$Q_{demand}$  = heat demand from system [kWh]

$SF$  = solar fraction [-]

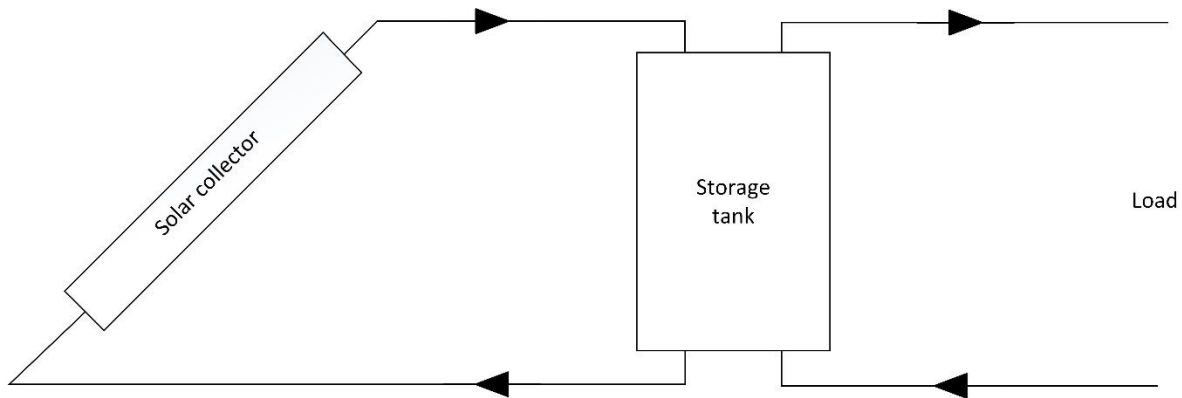
## 2.6 Thermal energy storage

The sun offers several opportunities as a renewable energy source, making it regarded as one of the most promising replacements for traditional energy sources [13]. The sun illuminates rooms through daylight, provides heat through irradiation and energy through solar collectors and panels. However, as the irradiation varies with the weather and season, it is difficult to rely on solar energy alone. This presents a major drawback as it leads to disparity between the supply and demand. One solution to this problem is to store the energy with thermal energy storage (TES) for when the supply is low and demand is high. It will be necessary in order to offer energy security for different types of renewable energy systems, in addition to improving the energy utilization efficiency in the systems.

By heating and cooling a storage medium, the thermal energy storage technology stock thermal energy so it can be available later on. Not only does this improve the general efficiency and reliability, it can also improve the economic aspect by prolonging the running time of the system [14]. Thermal energy storage solutions are divided into three types of energy storage; sensible, latent and thermochemical.

### 2.6.1 Sensible heat storage

Sensible heat storage (SHS) is the most mature technology out of the three types of thermal energy storage. This low-cost and reliable way of storing heat utilizes the temperature difference between the transport medium and storage medium without phase change. The storage medium can be both liquid and solid, with water, brick, rock and soil being the most common materials [13, 15, 16]. There are several versions of SHS, water tank storage and underground storage being two widely used technologies. The former is a common household item as it is the commercial way of storing hot water in buildings today. Here a heat transfer fluid indirectly heats water in a storage tank. The hot water is then stored until needed, either for heat exchange purposes or for domestic hot water. Underground storage is based on the same concept. But in this case the heat is stored in the ground, rather than in a water tank. The heat transfer fluid is pumped through pipe arrays in the ground, either to exchange or harvest heat from the ground depending on its purpose. This is an energy storage typically used in large-scale applications and can be used for both heat and cold [14]. Even though SHS is the simplest method of thermal energy storage, its low energy storage density require large storage capacities limiting the use of it in smaller projects [16, 17].



*Figure 2.4: A typical example of sensible heat storage using water tank storage.*

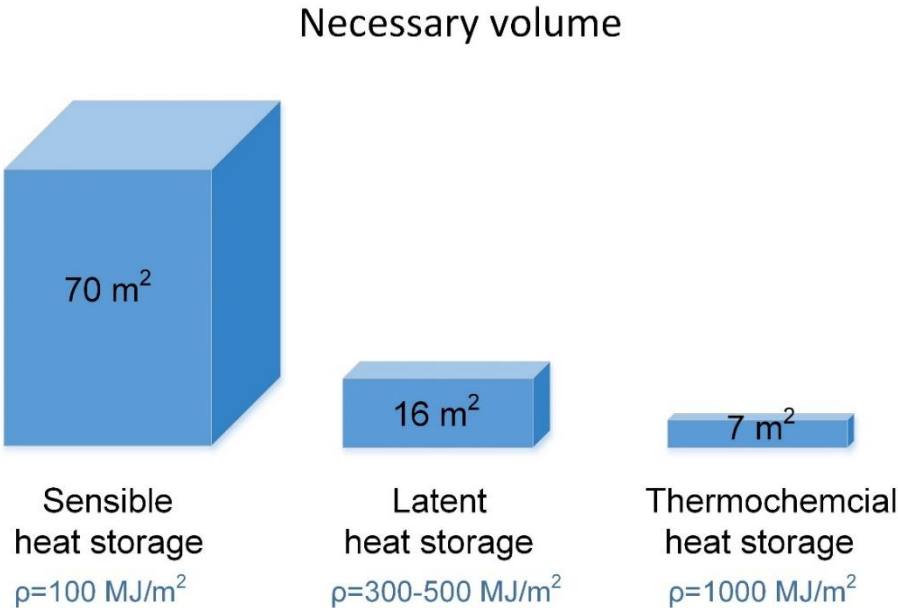
## 2.6.2 Latent heat storage

With latent heat storage (LHS) the heat is stored in a material by a nearly isothermal phase change. As soon as the temperature of the chosen phase change material (PCM) rises above its critical temperature it will change phase and simultaneously take up energy equivalent to its latent heat. When the temperature later decreases below this critical temperature, the material will release its latent heat and change phases into its initial state [14-16, 18]. The most familiar example of a PCM's ability to deliver thermal energy is the portable hand warmers that heats up when you bend the chip inside. The bag is filled with a supersaturated salt solution. As soon as the chip is pressed, the liquid will start to solidify and the process of crystallization will release heat [19].

In the case of the hand warmer, the phase change is liquid-solid. But the change in physical state can also occur in other forms, such as: solid-solid, solid-gas, liquid-gas and vice versa [20]. According to Tatsidjodoung, Le Pierrès and Luo [20] LHS has shown promise within residential applications. The main purpose of such systems in buildings is to decrease the temperature variation, especially due to solar radiation loads. This can be accomplished by inserting a suitable material into the building mass, for example the internal wall or ceiling. In this way, the excess heat from the room is extracted keeping the room at a steady temperature. The heat is stored within the wall until the temperature decreases below the critical point forcing the PCM to release the heat again.

Compared to sensible heat storage, latent heat storage has a much higher energy storage density. It will therefore be able to store more energy at a lower total storage capacity [13, 18]. All though it for residential applications can be stored in walls, a tank will still be considered

regarding storage of solar heat according to Tatsidjodoung, Le Pierrès and Luo [20]. However, there are some drawbacks regarding LHS with its dependency on temperature being one. Latent heat storage depends on the temperature staying above the phase change temperature after charging to avoid releasing the heat prematurely. This presents a limitation regarding seasonal storage, as the temperature might fluctuate over that period. Other disadvantages include a risk of subcooling and low thermal conductivity [13, 15, 20].



*Figure 2.5: Comparison of the volume necessary to meet an annual storage demand of an energy-efficient passive house (6480 MJ). Based on results from [20].*

### 3. Sorption energy storage

Chemical energy storage, in general, utilizes supply energy such as thermal energy, electrical energy or electromagnetic radiation [18]. In this report, thermal energy is the only supply considered, with emphasis on thermochemical storage with sorption. The general relation between chemical and thermochemical storage is described in Figure 3.1.

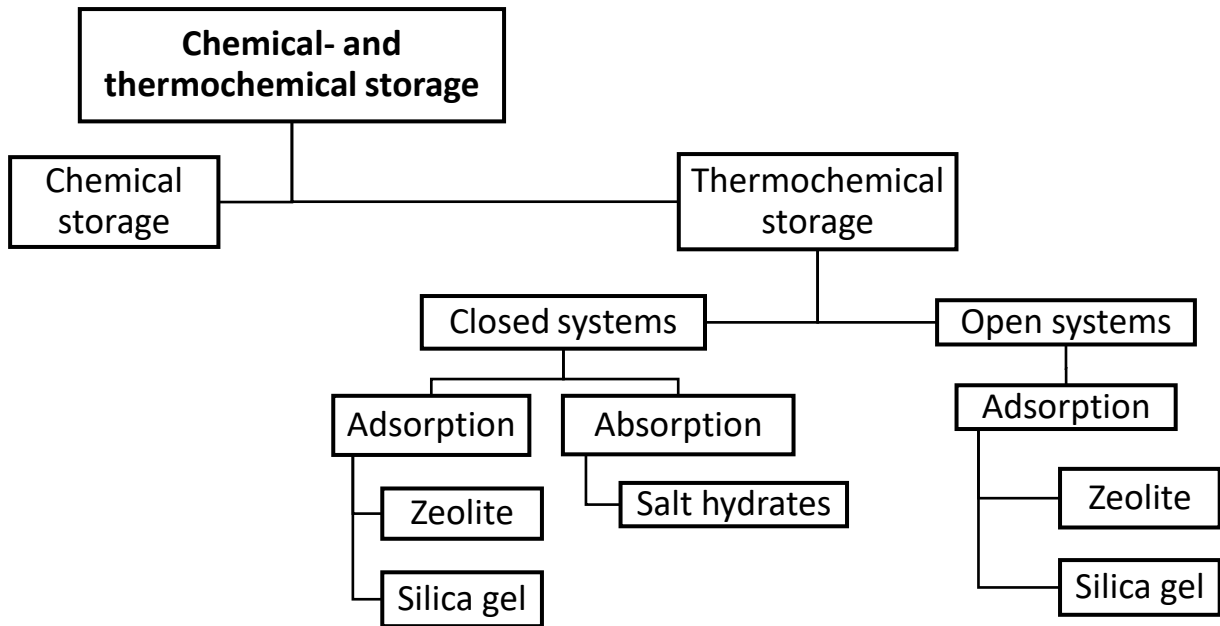


Figure 3.1: Classification of chemical and thermochemical processes for heat storage applications.

All types of thermochemical energy storage, i.e. with and without sorption, uses a reversible physico-chemical phenomena (chemical reaction) between materials to store energy in their chemical bonds as illustrated in Figure 3.2. When charging, energy is applied to a mixture of material A and B, breaking their binding and forcing them to separate. The two materials can then easily be stored individually until the energy is needed. To discharge, the materials are brought together and mixed, releasing the energy.

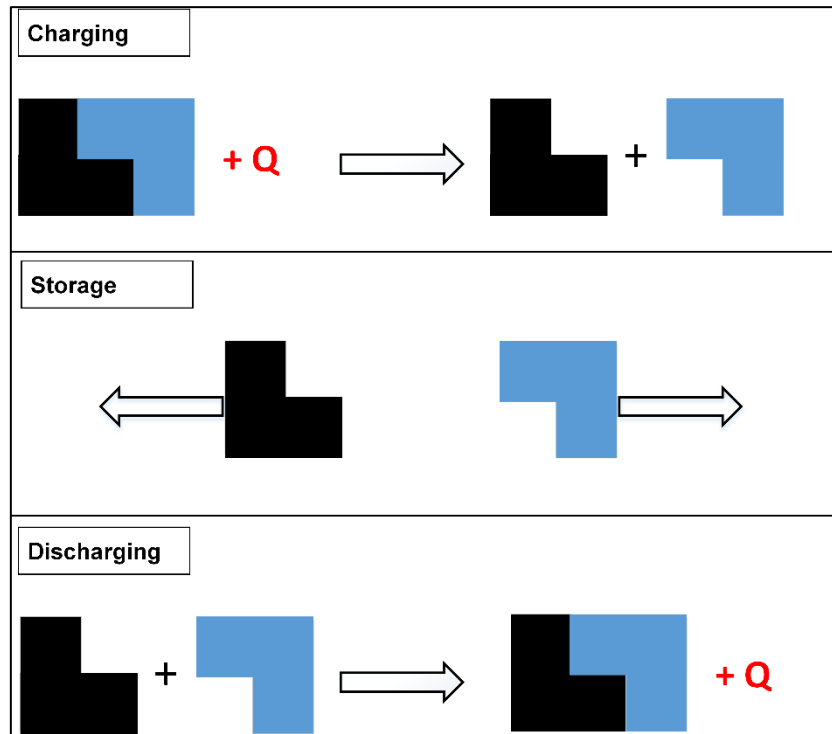


Figure 3.2: Illustration of how the thermochemical reactions occur.

Sorption is known as the phenomenon of capturing a gas or vapor (sorbate) by a sorbent existing in condensed state [18]. The sorbent is a substance and can be in either solid or liquid form [17]. Both thermo-physical and thermo-chemical aspects can be involved in sorption. In general, sorption is a unifying term including both adsorption and absorption. From the perspective of energy storage, absorption is the process where a liquid (absorbent) captures a gas or liquid. While adsorption refers to the process of binding gas or vapor to the surface of a solid or porous material. Since this operation rarely depends on temperature, the thermal energy can be stored with little to no energy loss, making it suitable for long-term energy storage [13, 15, 18].

The energy input to the thermochemical storage system is the desorption energy,  $Q_{des}$ . While the heat of adsorption energy,  $Q_{ads}$ , can be used for heating. The remaining energy is the heat of condensation,  $Q_{cond}$ , and evaporation,  $Q_{evap}$ , which can be used dependent on the temperature level [21].

Thermochemical energy storage with sorption, hereafter called sorption energy storage, can further be divided into open and closed systems.

### 3.1 Open system

Open sorption energy storage systems operate at atmospheric pressure and release its working fluid vapor into the surroundings. Because of this, water is the only medium utilized in these systems as it is nontoxic and harmless [16-18]. The open systems is an attractive solution due to its ability to store heat to meet peak load demands or energy redistribution requirements.

Charging of the system is typically fulfilled by hot air, heated by a solar collector, sent through a reactor filled with sorbent. By exchanging heat, the hot air desorbs the water from the adsorbent. The air then exits the reactor cooler and saturated, leaving the dehydrated adsorbent to store the heat. To discharge, moist air at medium to low temperature goes through the same reactor. The air then exchanges its moisture in exchange for the heat energy stored in the adsorbent. The air exits the reactor dry and heated, able to meet heat demands from space heating or domestic hot water [17, 21].

Open systems have a lower investment cost than closed systems since it require less components like condensers and evaporators, and can even provide better heat and mass transfer conditions. However, it is important to ensure air with a high enough moisture content and relative humidity to secure a good discharge rate and to avoid an additional humidifier [13].

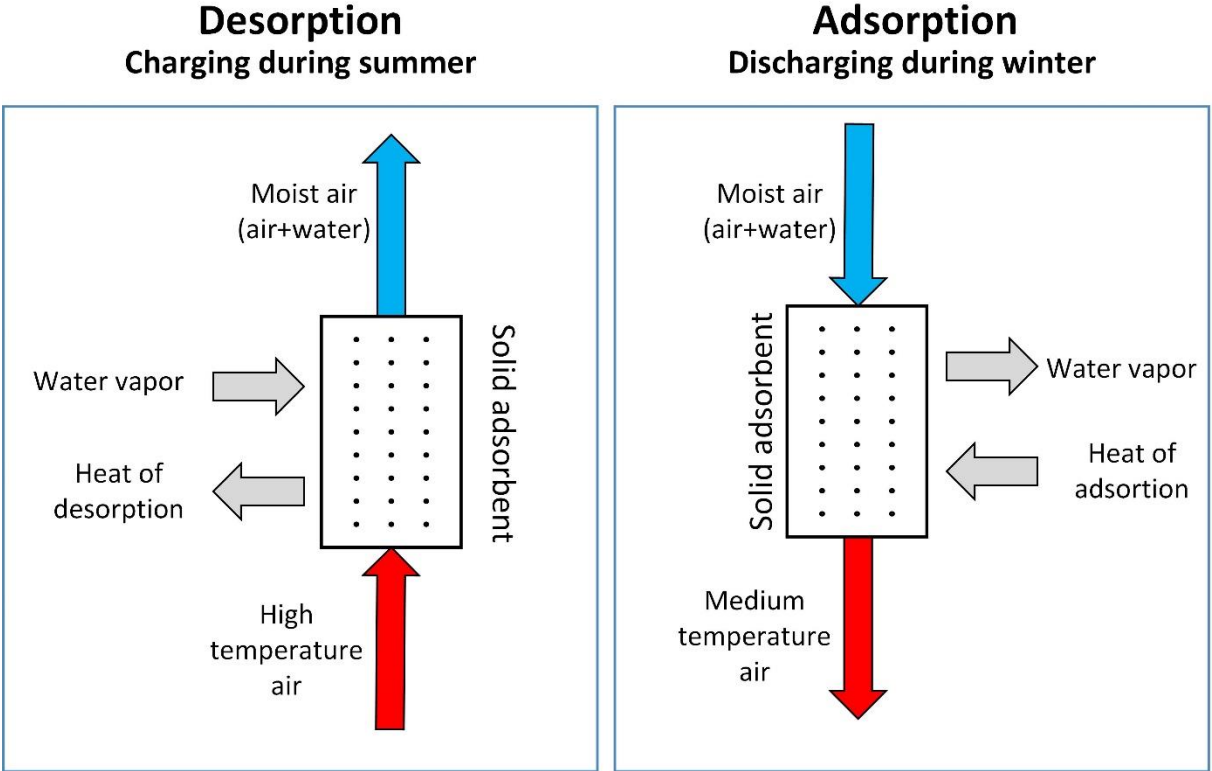


Figure 3.3: Graphical description of an open sorption storage system.



### 3.2 Closed system

The main difference between an open and closed thermochemical storage system is whether it is open to the surroundings or not. Since the closed system is not, there are a few restrictions regarding working fluid. Solar energy is however the prime source of energy for both systems. The heat stored in this system can also be utilized to cover both space heating and domestic hot water demands [17].

For charging, solar heat is applied indirectly to a storage tank filled with a sorbent. The sorbent starts to release its water vapor content in exchange for the heat applied. The vapor can then be led to a condenser where it is cooled and condensed for storage. The dehydrated adsorbent and condensed water vapor is stored separate until there is an energy demand. To discharge, the water vapor is evaporated at a low temperature and led back to the storage tank where the adsorbent releases its heat in favor of the vapor. The heat can then be exchanged with a transport medium to meet the chosen demand [17, 21]. The operating principle of the charge/discharge of an absorption energy storage is close to identical.

The challenge of the closed energy storage system is its need to transport and store the vapor separately from the adsorbent. The system, which presents very complex, also require advanced heat exchanger technologies in order to maintain the high energy density in the storage, along with more components like condensers and evaporators [13, 17].

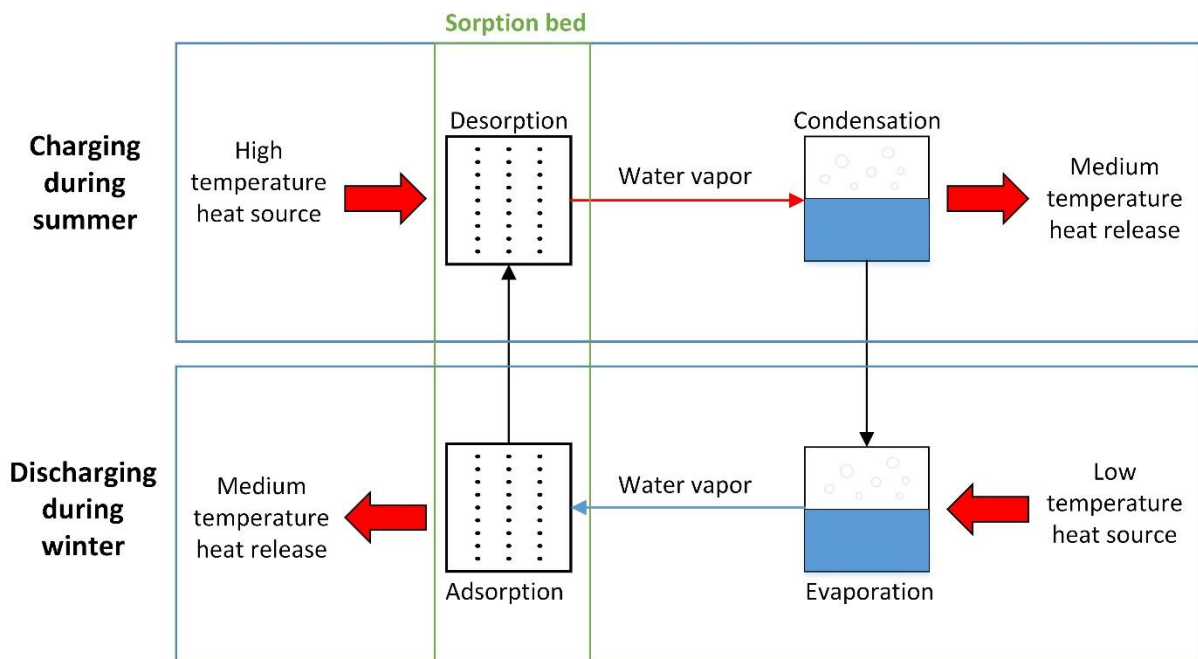


Figure 3.4: Operating principle of a closed adsorption system.

### 3.3 Challenges and selection criteria

Through research several challenges of sorption energy storage have been discovered, including the following [17, 18]:

- the choice of the best system: open or closed;
- the vessels/tanks design;
- risks regarding freezing of working material/fluid when working with water;
- maintaining vacuum to ensure proper operating efficiency in closed systems;
- pressure drop;
- cost;
- the storage density optimization regarding choice of materials.

The latter point is proven crucial by researchers as it plays a key role in all the aforementioned challenges for both open and closed systems. The energy storage system relies on the material to enable the system's best performance. The material is required to store energy without heat loss and to release energy immediately when a demand occurs. Other desired properties of storage materials used in thermochemical energy storage, include [13, 17, 18, 20, 22]:

- high affinity by the sorbate for the sorbent: which is important to achieve a usable power density, as it affects the rate of reaction;
- superior vaporization of the sorbate than the sorbent in absorption;
- high storage density (thermal energy density at operating temperature);
- high thermal conductivity and high heat transfer from the sorbate to the heat transfer fluid or heat exchangers;
- high uptake of water at operating temperature level;
- as low as possible desorption (charging) temperature: will result in higher solar collector efficiency;
- low regeneration time;
- able to evaporate at relatively low temperature, so one can increase number of possible heat sources;
- environmentally safe, non-toxic, low Global Warming Potential and Ozon Depletion Potential;
- non-corrosiveness of materials;
- ageing behavior: how many cycles it can survive;
- low material cost.

Still, despite their good theoretical potential, several thermochemical heat storage systems using materials that meets the criteria perform poorly in experiments. In Cot-Gores, Castell and Cabeza [23] the authors explain the reactant sorbent's low thermal conductivity as the reason behind the poor performance of the heat and mass transfer within the reactive bed. Tatsidjodoung et al. [20] describe suggestions from several authors about how one can overcome these limitations and thereby enhance the reactor's bed performance. These suggestions include improvements of processes such as mixing an additive with the sorbent for absorption, or the use of porous elaborated materials or natural expanded graphite for chemical reactions.

Along with the choice of materials comes the challenge of choosing the type of system and heat exchanger. The minimum temperature of discharge is decided by the minimum temperature required for the receiving technology, i.e. space heating or domestic hot water. If a high temperature lift is required during discharging, the achievable storage density decreases drastically. The heat exchanger, with the reactor in particular, require a high efficiency. This becomes even more important in cases of storage materials with low heat conductivity or reaction rate [18].

### 3.4 Materials researched

A material is evaluated based on how well it can perform for the individual thermochemical energy storage type and for how long it can store its energy. As explained earlier, there are two types of thermochemical energy storage using sorption; adsorption and absorption. There is also a difference between short- and long-term storage. Short-term storage will not be covered in this report, as the scope of the assignment revolves around long-term storage. There will also be put an emphasis on adsorption, rather than absorption due to the assignments choice of silica gel as working material.

The many studies on thermal storage with adsorption can be divided into mainly two groups; one for those who work with the identification of adsorption working pairs, and one for those who work with quantitative demonstration of the adsorption based thermal energy storage concept [24]. The latter type of studies focus on optimizing thermal energy storage with adsorption and its storage capacity. The former group focuses on the analysis and development of properties within the materials, in order to achieve a maximum energy density, capacity and

desorption temperature. Zeolite and mesoporous materials, including silica gel, have been studied as adsorbents in both focus groups. The most important property between the two popular adsorbents is their ability to contain water, which directly translates to how much water vapor can be extracted for energy storage. While both materials can contain between 25% and 38% water of its dry weight, a mixture of the two, Sizeo and SWS, have even higher values [16]. Water is normally used as the sorbate substance, especially in building applications, due to its minimal risk to the environment.

Zeolite and silica gel have according to Lizana et al. [16] been widely tested through several coordinated international programs and projects. Silica gel is often preferred due to its low cost, high affinity to water vapor and sorption capacity. However, according to N'Tsoukpoe et al. [18], AEE INTEC in Austria built a prototype for the framework of their HYDES project. This prototype made it possible to observe the performance of a sorption system combined with solar collectors for heating and domestic hot water working with silica gel. The experimental results did however present 20% less than expected from the theoretical predictions. In the follow-up project, MODESTORE [25], it was uncovered that silica gel is realistically only able to operate with water contents of 2-12%. This is drastically lower than the silica gel's maximum of 38%, which in turn hinders the actual energy capacity. The material storage density dropped to only 50 kWh/m<sup>3</sup>, which is 30% less than traditional sensible water storage [16, 18]. According to Henninger et al. in Tatsidjodoung et al. [20], the disadvantage of the working pair of silica gel/H<sub>2</sub>O is the low hydrophilic characteristic it has within the working window of adsorption, at 35°C and 1.2 kPa, and desorption, at 150° and 5.6 kPa. Yu et al. [13] further sum up the silica gel's shortcomings in that the water adsorption mainly occurs at too high relative pressures, which results in a low water exchange within a cycle, especially in closed systems. The last shortcoming mentioned is the silica gel's low temperature-lift during the adsorption process over a limited range of sorption capacity. In spite of this, silica gel/H<sub>2</sub>O is still the most widely used working pair due to its accessibility and low price.

Different types of zeolite has been thoroughly tested. N'Tsoukpoe et al. [18] presents several projects using zeolite. Among these, the German project Monosorp, which tested a potential open storage system for a single-family passive house. The desorption temperature did however prove to be very high, making the use of solar heat troublesome for this project. This has also proven to be a disadvantage of zeolite. In Tatsidjodoung et al. [20] zeolite is presented as more hydrophilic than silica gel, but that its high desorption temperature of more than 200°C, cause problems when installed in solar assisted applications.

Experiments done with the composite materials, i.e. materials made out of a combination of two or more materials, have on the other hand shown promise. By impregnating silica gel or mixing zeolite, it is apparently possible to design a material that will perform better than its origin. However, the research done on thermochemical heat storage with pure materials is marginal compared to sensible and latent heat storage materials, and the research done on composites is even more restricted [20].

Lizana et al. [16] advises that additional efforts should be put into research of optimizing operation conditions, storage cycles efficiency, material cost and system designs. The authors additionally state that there is currently no available material that satisfies all requirements for a viable deployment in building applications. This despite the technology's potentially high energy density and long-term storage ability.

### 3.5 Long-term storage research

For building applications, the research and development of thermochemical energy storage remain on an early stage, and there are as of now no commercial solution available. High cost, low heat transfer capacity and system energy lower than material energy density are the barriers and reason it has not come further. As mentioned earlier, most of the current research on materials are based on solar energy solutions [16].

Jaehnig et al. [25] built a prototype system within the framework of the EU-project MODESTORE and tested it in a laboratory. The prototype included a new compact design where all components were included in a single container, creating short distances between the adsorber and condenser/evaporator. Their test included approximately 200 kg of silica gel in a closed system. The authors reasoned their choice of working pair with silica gel and water's environment friendly properties, low cost and a low temperature demand possible to cover with a flat plate collector. The experiment did however end as previously discussed with a drastically low water content between 2-12%.

Deshmukh, Maiya and Murthy [26] performed a thermal analysis of a closed silica gel and water adsorption heat storage system. The system was designed to store 18 kWh heat and to deliver 3 kW for 6 hours for an average temperature lift of 25 K at any later time. The analysis showed that such a performance require 350 kg of silica gel with a heat transfer performance value of 400 W/K. The specific heating effect was about 190 kJ/kg. The system had an overall exergy loss of 0.146 kW with an exergetic efficiency of 73%.

Fernandes et al. [27] presents a dynamic model of a heat storage adsorption device operating with silica gel and water. A set of parametric tests were performed by applying the model to a solar thermal energy system. Similarly to the prototype of Jaehnig et al. [25], this system is compact. The adsorber tank is here located within the hot water tank, and the condenser right underneath in a second tank where also the water inlet is located. The authors conclude that high condenser lengths lead to higher savings of backup heating energy, independent of the secondary tank volume. The condenser is shaped as a round coil and therefore the length relates to the amount of coils and number of turns. The report further found that for a main tank volume of 300 L, the annual consumption of backup heating energy is reduced to a minimum. The annual savings value is in total increased to 16% after considering all conclusions. Nevertheless, the system in total needs to be studied and optimized in order to evaluate its real performance, as these were results from individual assessments.

Michel, Neveu and Mazet [28] compared two 2D models to determine the optimal operating mode for thermochemical seasonal storage applied to house heating. The study disclosed that the heat transfer was the main limitation of the closed system. Although, it is suggested that an increase of thermal conductivity would strongly improve the reaction rate. As for the open system, the mass transfer proves to be main limitation, where an adjustment of the storage bed's permeability is suggested to improve the hydration rate. Both systems do however lead to close global performances for the chosen set of parameters. They do also present a higher average specific power than the target value of 0.3-0.7 W/kg for a reactor and energy density lower than 410 kWh/m<sup>3</sup>. The open system is concluded to be a promising way to implement thermochemical processes as a long-term heat storage as it requires a simpler construction and is therefore cheaper.

## 4. Research done at SJTU

While there is somewhat limited research done on the subject on thermal energy storage with sorption, some research has been performed at Shanghai Jiao-Tong University (SJTU).

Yu, Wang and Wang [13] published a review on sorption technologies aiming to provide an insight into the basic knowledge and current state of the art research in the field. Li et al. [15] proposed a dual-mode thermochemical sorption energy storage system using the working pair of expanded graphite/SrCl<sub>2</sub>-NH<sub>3</sub>. This system is able to produce heat at temperatures above

70°C even at ambient temperatures as low as -15°C. The authors wish to further improve the system performance, in spite of positive results regarding the technology being both feasible and effective.

Zhang, Wang and Li [29] successfully built a lab-scale open sorption prototype to experimentally investigate the possibility of storing low-temperature heat for space heating. By utilizing electricity in off-peak hours at night, energy is stored to be used during peak hours during the day. The experiment revealed that the relative humidity is a critical factor determining the output temperature, output power and energy storage density. In total, the prototype is able to fulfill the space heating requirements with a system efficiency of 84.5-96.9%. With a total volumetric storage density of 191 kWh/m<sup>3</sup>, air with temperature higher than 30°C can be provided over 7.1 h under the conditions of  $T_i=20^\circ\text{C}$ ,  $\text{RH}_i=80\%$  and  $q_m=17\text{ m}^3/\text{h}$ . The output energy then ranges between 56.7 W and 136.0 W.

## 5. Seasonal sorption energy storage

A theoretical system with thermochemical energy storage including sorption has been planned based on the assignment text and additional information provided by Professor Yanjun Dai at SJTU. A summary of this additional information can be found in Appendix A. The system is an open system working with silica gel and water. It should provide energy for space heating of a domestic household of 100 m<sup>2</sup> located in Beijing, China. The system should afterwards be tested to see if it can operate under a different climate in Trondheim, Norway. For the Trondheim case the Norwegian standards for passive house, NS 3700:2013 [30], applies. China, on the other hand, does not have building standards, so the values in question are provided by Prof. Dai based on his experience and expertise in the field, and can be found in Appendix A.2.

The main components of the system are:

- Vacuum flat plate solar collector
- Glycol/water to air heat exchanger
- Humidifier
- Dehumidifier
- Sorption storage tank
- Fan
- Dampers

A complete sketch can be seen in Figure 5.1.

### 5.1 Working principle

The system is supposed to aid the disparity between supply and demand by storing energy from a period of high supply and low demand, to a period of low supply and high demand.

#### 5.1.1 Charging

To charge the sorption storage, fresh outdoor air enters the duct. The air should be dry, i.e. with a relative humidity around 40% [31], and relatively warm. The air is then sent through the counter flow heat exchanger where the glycol/water solution heated by the solar collector



operates as the second flow. The hot medium from the solar collector heats the airflow to a set inlet temperature before it enters the sorption storage. The storage is filled with silica gel impregnated by water. When the air enters the storage, the silica gel adsorbs the heat and releases its water content, which the air absorbs. The air then exits the storage bed moister and cooler than upon entry. The exhaust air is discharged into the surroundings again, while the silica gel is stored dry.

### 5.1.2 Discharging

The aim of this storage is that the heat can be stored for a colder season when the solar energy potential is lower, but the demand is higher. To discharge, air is sent through a humidifier to ensure a high relative humidity at inlet. It is important to supply air with a sufficient temperature in order to reach the desired outlet temperature. The inlet air can therefore be a mixture of used indoor air and fresh outdoor air to avoid a second heat source for preheating. After being humidified, the air is sent through the storage bed where the dehydrated silica gel exchanges its heat for the moisture in the air. This way the energy storage is gradually discharged as the silica gel regains its moisture. The air exits the storage bed heated and dry, ready to be supplied for space heating.

## 5.2 Dimension of solar collector

In order to dimension the sorption storage it is necessary to dimension the solar collector system to calculate how much energy is available for storage. For this assignment, a vacuum flat plate solar collector was chosen according to Appendix A. As the solar energy potential is individual for each location and does not affect the sorption storage directly, it is not included in the mathematical model. The dimensions for both Beijing and Trondheim will therefore be included below, and summarized in Table 5.3.

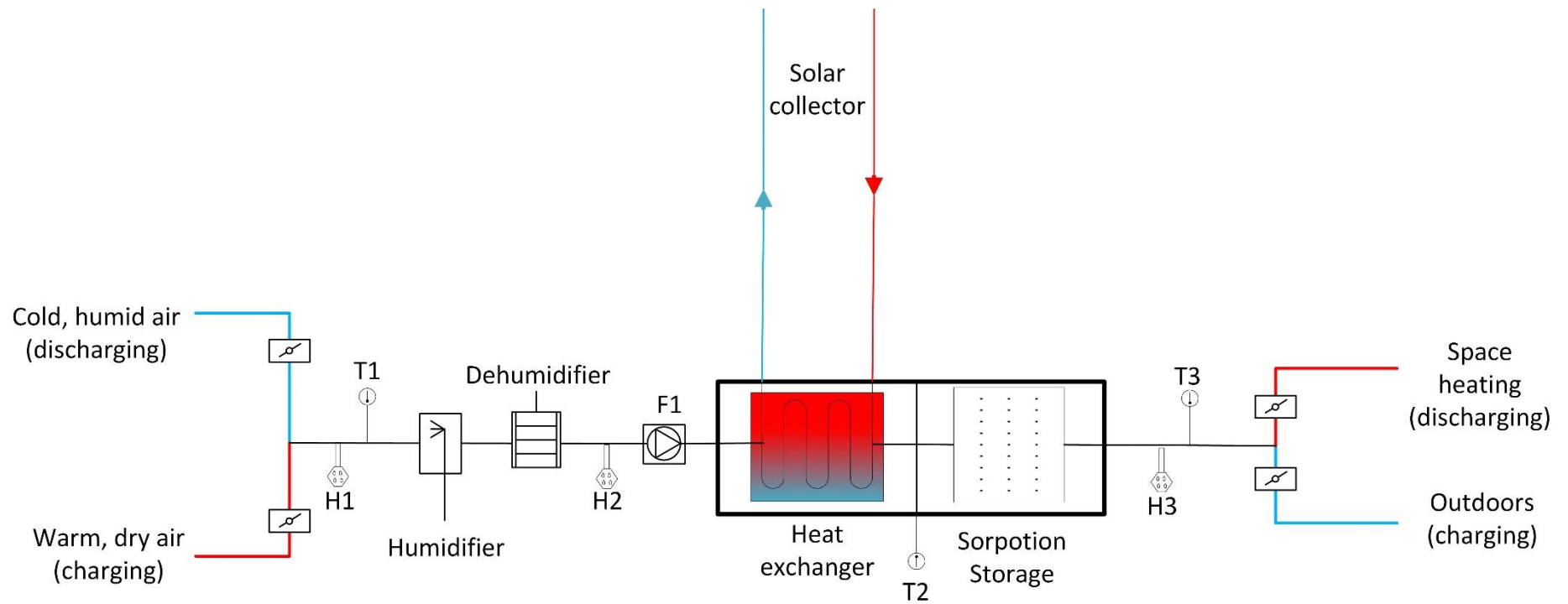


Figure 5.1: Sketch of the system consisting of a solar collector, heat exchanger and sorption storage.

## 5.2.1 Optimal tilt and irradiation

To be able to dimension a solar collector system, it is necessary to uncover how much irradiation is available for a collector at a given location. The European Commission Joint Research Center (JRC) [32] has a database where the daily average irradiation at the optimal tilt for each month,  $H_{opt,day}$ , is available for several places all over the world. Thus, through Equation 5.1 it is possible to calculate the average irradiation per month based on number of days in a month,  $H_{opt,month}$ . By summing each month, the total amount of irradiation through a year,  $H_{opt,year}$ , can be calculated.

$$H_{opt,month} = \frac{H_{opt,day} \cdot n}{1000} \left[ \frac{kWh}{m^2} \right] \quad (5.1)$$

### 5.2.1.1 Beijing

The results from the JRC database search for Beijing is listed in Table 5.1.

*Table 5.1: Average daily and monthly irradiation in addition to optimal tilt for Beijing, China.*

Month	$H_{opt,day}$ [Wh/m <sup>2</sup> /day]	Days per month	$H_{opt,month}$ [kWh/m <sup>2</sup> ]	$\Psi_{opt}$ [°]
<b>January</b>	4050	31	125.55	63
<b>February</b>	4800	28	134.4	56
<b>March</b>	5830	31	180.73	42
<b>April</b>	5460	30	163.8	25
<b>May</b>	5600	31	173.6	12
<b>June</b>	4910	30	147.3	6
<b>July</b>	4800	31	148.8	8
<b>August</b>	4830	31	149.73	18
<b>September</b>	4630	30	138.9	34
<b>October</b>	4520	31	140.12	49
<b>November</b>	4110	30	123.3	61
<b>December</b>	3830	31	118.73	66
<b>Year</b>	4780			38
<b>Sum (<math>H_{opt,year}</math>)</b>			<b>1744.96</b>	

From Table 5.1 one can find the suggested optimal tilt. But Equation 2.1 can further be simplified to Equation 5.2 in order to calculate the optimal tilt for a collector placed in Beijing throughout a whole year.

$$\psi_{opt,year} = \frac{\sum_{n=jan}^{n=dec} H_{opt,month} \cdot \psi_{opt,month}}{\sum_{n=jan}^{n=dec} H_{opt,month}} \quad [^\circ] \quad (5.2)$$

$$\psi_{opt,year} = \underline{35.26^\circ}$$

Through calculations the optimal tilt of Beijing is  $35.26^\circ$ , while the JRC Database suggests  $38^\circ$ . As it is not clear how the JRC determine its suggestion, the calculated value will be regarded as the correct one as it is the average throughout the months of the year.

### 5.2.1.2 Trondheim

The results from the JRC database search for Trondheim is listed in Table 5.2.

*Table 5.2: Average daily and monthly irradiation in addition to optimal tilt for Trondheim, Norway.*

Month	$H_{opt,day}$ [Wh/m <sup>2</sup> /day]	Days per month	$H_{opt,month}$ [kWh/m <sup>2</sup> ]	$\Psi_{opt}$ [°]
<b>January</b>	492	31	15.252	81
<b>February</b>	1700	28	47.6	74
<b>March</b>	3050	31	94.55	61
<b>April</b>	4250	30	127.5	45
<b>May</b>	4930	31	152.83	30
<b>June</b>	5160	30	154.8	21
<b>July</b>	4710	31	146.01	24
<b>August</b>	3890	31	120.59	35
<b>September</b>	2730	30	81.9	53
<b>October</b>	1590	31	49.29	67
<b>November</b>	724	30	21.72	78
<b>December</b>	267	31	8.277	83
<b>Year</b>	2800			44
<b>Sum (<math>H_{opt,year}</math>)</b>			<b>1020.319</b>	

As for Beijing, the optimal yearly tilt for a collector located in Trondheim can be calculated based on the values in Table 5.2 with Equation 5.2.

$$\psi_{opt,year} = \frac{\sum_{n=jan}^{n=dec} H_{opt,month} \cdot \psi_{opt,month}}{\sum_{n=jan}^{n=dec} H_{opt,month}} \quad [^{\circ}]$$

$$\psi_{opt,year} = \underline{41.15^{\circ}}$$

Through calculations the optimal tilt of Trondheim is  $41.15^{\circ}$ , while the JRC Database suggests  $44^{\circ}$ . As it is not clear how the JRC determine its suggestion, the calculated value will be regarded as the correct one as it is the average throughout the months of the year.

## 5.2.2 Energy

The estimation of energy demand is done differently for the two cases. However, both cases are theoretical and therefore both azimuth and tilt of the collector is assumed optimal, giving  $f_a=1$  and  $f_H=1$ . In a practical case, the efficiency of the solar collector would be stated by the manufacturer. For these theoretical systems, the efficiency was chosen in collaboration with Prof. Dai to a value of 0.5.

### 5.2.2.1 Beijing

As China lacks a standard for energy demand, Prof. Dai's has advised a general value. Based on the average Chinese energy demand, it was suggested that the energy demand should be  $50 \text{ W/m}^2$ , resulting in a total energy demand of  $438 \text{ kWh/m}^2$ .

Based on the yearly solar irradiance, efficiency and correctional factors, the available energy output of the solar collector can be calculated from Equation (2.2).

$$Q_{out,B} = 1744.96 \frac{kWh}{m^2} \cdot 0.5 \cdot 1 \cdot 1 = \underline{872.48 \frac{kWh}{m^2}}$$

### 5.2.2.2 Trondheim

The energy demand for space heating for the system located in Trondheim is dimensioned according to the Norwegian standard NS 3700:2013, table 3 [30]. Trondheim has a yearly mean temperature,  $\theta_{ym}$ , of 5.4°C [33] and should therefore according to the standard have a maximum energy demand for space heating according to Equation 5.3.

$$Q_{SH} = 15 + 5.4 \cdot \frac{(250 - A_{fl})}{100} + (2.1 + 0.59 \cdot \frac{(250 - A_{fl})}{100}) \cdot (6.3 - \theta_{ym}) , \quad \left[ \frac{kWh}{m^2 \cdot year} \right] \quad (5.3)$$

Where  $A_{fl}$  is the net area of the house. For a house of  $A_{fl} = 100 \text{ m}^2$  and  $\theta_{ym} = 5.4^\circ\text{C}$  the yearly energy demand for space heating will be the following:

$$\begin{aligned} Q_{SH} &= 15 + 5.4 \cdot \frac{(250 - 100)}{100} + (2.1 + 0.59 \cdot \frac{(250 - 100)}{100}) \cdot (6.3 - 5.4) \\ &= \underline{45.98 \text{ kWh/m}^2} \end{aligned}$$

Based on the yearly solar irradiance, efficiency and correctional factors, the available energy output of the solar collector can be calculated from Equation 2.2.

$$Q_{out,T} = 1020.32 \frac{kWh}{m^2} \cdot 0.5 \cdot 1 \cdot 1 = \underline{510.16 \frac{kWh}{m^2}}$$

### 5.2.3 Area

Based on the typical Norwegian values for solar fraction listen in Zijdemans [7] and information provided by Prof. Dai, the solar fraction for both cases is chosen to be 0.5 as this is a system for space heating alone. Through Equation 2.3, it is possible to calculate the necessary solar collector area based on the values of efficiency and energy output calculated earlier.

$$A_{coll} = \frac{Q_{SH} \cdot SF_{SH} \cdot A_{fl}}{Q_{out}} \quad (2.3)$$

#### 5.2.3.1 Beijing

With the respective values determined above, the necessary solar collector area for the Beijing household should be:

$$A_{coll} = \frac{438 \text{ kWh/m}^2 \cdot 0.5 \cdot 100 \text{ m}^2}{872.48 \text{ kWh/m}^2} = \underline{25.1 \text{ m}^2}$$

#### 5.2.3.2 Trondheim

With the respective values determined above, the necessary solar collector area for the Trondheim household should be:

$$A_{coll} = \frac{45.98 \text{ kWh/m}^2 \cdot 0.5 \cdot 100 \text{ m}^2}{510.16 \text{ kWh/m}^2} = \underline{4.51 \text{ m}^2}$$

### 5.2.4 Summary of dimensional data

A summary of the dimensional data computed in the earlier chapters is presented in Table 5.3. The biggest difference between the two cases is the energy demand for space heating. This in turn creates a huge gap between the solar collector area and energy output of the two systems.

Table 5.3: Dimensional data for the respective cases of Beijing and Trondheim

	<b>Beijing</b>	<b>Trondheim</b>
Average annual temperature, $\theta_{ym}$ [°C]	11.7	5.4
Optimal angle, $\Psi_{opt}$ [°]	35.26	41.15
Energy demand, $Q_{SH}$ [kWh/m <sup>2</sup> ]	438	45.98
Solar fraction, SF [-]	0.5	0.5
Yearly solar irradiance, $H_{opt, year}$ [kWh/m <sup>2</sup> ]	1744.96	1020.32
Efficiency of solar collector, $\eta_{coll}$ [-]	0.5	0.5
Energy output, $Q_{out}$ [kWh/m <sup>2</sup> ]	872.48	510.16
Solar collector area, $A_{coll}$ [m <sup>2</sup> ]	25.1	4.5
Actual energy output, $Q_{out, total}$ [kWh]	21,900	2299.13
Actual energy output, $Q_{out, total}$ [kW]	2.5	0.262

## 6. Mathematical model

In order to build the mathematical model of the sorption energy storage the software MATLAB was utilized. MATLAB is a platform which is optimized for solving engineering and scientific problems [34], making it suitable for the task of optimizing a mathematical model. Since this model is to be tested for several locations, it is especially appropriate to work with MATLAB. In that way the parameters can easily be changed, and a simulation takes shorter time than a practical calculation would.

By basing the mathematical model on the working principle described in chapter 5.1.1, it is necessary to determine the parameters of the system. As the objective of the assignment is to build a model that would function in Beijing primarily, the Beijing summer climate will dimension the parameters in question. The main parameters of the inlet air is the relative humidity and temperature. While the main parameter for the desiccant bed is its water content. Based on the information obtained from the dimensioning of the solar collector system, it is possible to dimension the heat exchanger and the airflow through it. The complete MATLAB script for charging and discharging can be found in Appendix B. The equations included in the script will be described in the following chapters in order to build a complete understanding of the model.



## 6.1 Inlet air

The temperature and humidity of the inlet air is important for both charging and discharging.

### 6.1.1 Charging

In order to charge the silica gel, the air must hold a temperature of 80°C according to Appendix A.4. It is impossible for the inlet air to reach such a high temperature on its own, but it is expected that the air holds a summer temperature when entering the heat exchanger. According to the working principle, the inlet air should be dry. Dry air is classified as air with relative humidity,  $RH$ , below 40% [31]. Warm summer air does however have the tendency of being rather humid. It is therefore necessary to install a dehumidifier in order to lower the relative humidity before the air is heated by the heat exchanger. Although, due to the added complexity of modelling a dehumidifier, it will only be included as a part of the system but not dimensioned.

### 6.1.2 Discharging

The discharging is expected to occur in a colder season where the solar energy supply is lower, but demand is higher. It is therefore reasonable to think that this will be outside the summer season, and the average temperature of the outdoor air can be low. To be able to reach the best performance possible for the sorption storage, it is necessary that the inlet air has a temperature of 20°C, or above, for discharging. This is to enable a sufficient temperature lift in order to reach the desired temperature for space heating. As the outdoor temperature can be quite low during the fall season, it is necessary to utilize used indoor air to accomplish a sufficient temperature. The fresh outdoor air and indoor air is therefore mixed through the dampers before entering the sorption storage. Due to its complexity, this is not included in the mathematical model and the temperature is just assumed at an appropriate level.

According to the working principle, the inlet air for discharging should be humid. In order to insure the humidity, a humidifier is installed before the sorption storage. The dimensioning and power demand from the humidifier is not included in the mathematical model, and the air is just assumed at a sufficient humidity.

## 6.2 Heat exchanger

The outdoor enters the heat exchanger and is indirectly heated by the heated glycol/water solution circulating in the solar collector loop. The air enters with at least 20°C and should according to Appendix A.4 exit the heat exchanger at 80°C. The heat transferred from the solar collector fluid is calculated through Equation 6.1 based on the available solar energy,  $Q_{out}$ . The heat exchanger efficiency is 90% according to Appendix A.4.

$$Q_{HEX} = Q_{out,total} \cdot \eta_{HEX} \quad [kW] \quad (6.1)$$

The airflow through the heat exchanger is calculated by altering the equation for heat transfer from Equation 6.2 to Equation 6.3.

$$Q_{HEX} = \dot{m}_{air} \cdot C_{P,air} \cdot \Delta T \quad [kW] \quad (6.2)$$

$$\dot{m}_{air} = \frac{Q_{HEX}}{C_{P,air} \cdot \Delta T} \quad \left[ \frac{kg}{s} \right] \quad (6.3)$$

## 6.3 Sorption storage bed

The sorption storage consist of the desiccant bed of silica gel and water.

### 6.3.1 Water content

The most important parameter in the desiccant bed is the water content,  $W$ . This highly affects the size of energy storage and temperature lift. According to Appendix A.4 this should be between 0.2-0.35 kg water/kg desiccant for charging, and as low as 0.05 kg water/kg desiccant for discharge. The adsorption heat of ordinary density silica gel is according to Dai, Wang and Zhang [35] calculated through the following, and is based on the water content for charging:

$$Q = \begin{cases} -13\,400 \cdot W + 3500, & W \leq 0.05 \\ -1400 \cdot W + 2950, & W > 0.05 \end{cases} \quad (6.4)$$

$Q$  = adsorption heat [kJ/kg water]

$W$  = water content of desiccant for charging [kg water/kg desiccant]

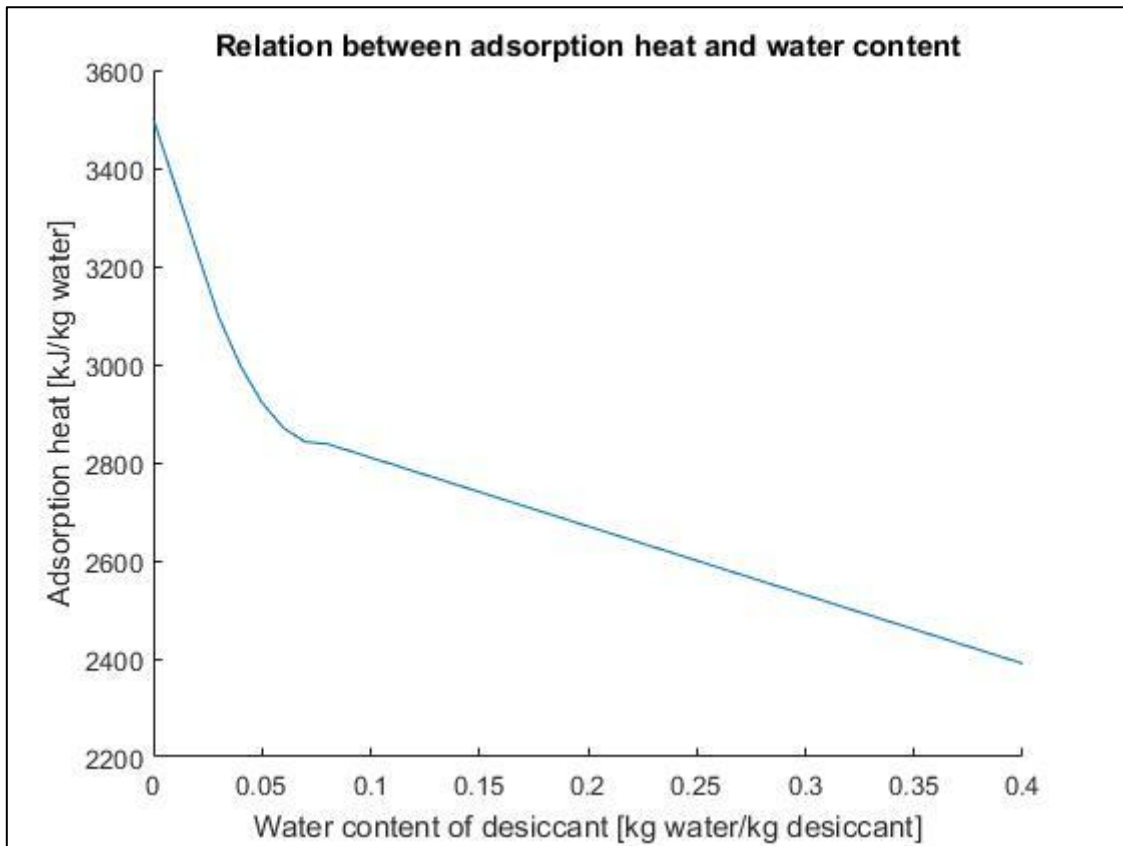


Figure 6.1: The relation between adsorption heat and water content of desiccant, from  $W=0$  to  $W=0.4$

### 6.3.2 Relative humidity

While the relative humidity for the inlet air is controlled, the relative humidity at equilibrium needs to be calculated. It is calculated based on the water content of the desiccant bed [35] for charging and discharging separately.

$$RH_{eq} = 0.0078 - 0.05759 \cdot W + 24.16554 \cdot W^2 - 124.78 \cdot W^3 + 204.226 \cdot W^4 \quad (6.5)$$

### 6.3.3 Saturation pressure

The saturation pressure of the water vapor inside the sorption storage during charging should be found for both the ambient temperature, and the inlet air temperature of the desiccant bed. It can be calculated through the following [35]:

$$\ln P_s = 23.1964 - \frac{3816.44}{T - 46.13} \quad [Pa] \quad (6.6)$$

In order to implement this in MATLAB, Equation 6.6 is altered to:

$$P_s = e^{23.1964 - \frac{3816.44}{T - 46.13}} \quad [Pa] \quad (6.7)$$

### 6.3.4 Humidity ratio

The humidity ratio at inlet and for equilibrium can be found through Equation 6.8 according to Dai, Wang, Zhang [35] using the saturation pressure calculated above. The saturation pressure at the ambient temperature is implemented in the humidity ratio of the inlet air, whilst the inlet temperature calculate the humidity ratio at equilibrium.

$$Y = \frac{0.622 \cdot RH \cdot P_s}{B - RH \cdot P_s} \quad [-] \quad (6.8)$$

Where  $B$  is the atmospheric pressure.

The relative humidity of the outlet air can further be found through Equation 6.11 based on the relations in Equation 6.9 and 6.10.

$$\Delta Y_{max} = Y_{in} - Y_{eq} \quad (6.9)$$

$$\Delta Y_{practical} = \eta_{DH} \cdot \Delta Y_{max} \quad (6.10)$$

$$Y_{out} = Y_{in} - \Delta Y_{practical} \quad (6.11)$$

### 6.3.5 Enthalpy

The sorption storage system is assumed optimal and therefore adiabatic, meaning no heat loss to the surroundings. Which leaves the enthalpy through the desiccant bed constant. As

the system utilizes air with water content, it is necessary to calculate the enthalpy for moist air. This is done by summing the enthalpy of dry air with the fraction of water present, like seen in Equation 6.14. The enthalpy of air and water is calculated through Equation 6.12 and 6.13 respectively.

$$h_a = C_{p,a} \cdot T_{in} \quad (6.12)$$

$$h_w = h_{gi} + C_{p,w} \cdot T_{in} \quad (6.13)$$

$$h_{in} = h_a + Y_{in} \cdot h_w \quad (6.14)$$

$h_a$  = specific enthalpy for air [kJ/kg]

$h_w$  = specific enthalpy for water [kJ/kg]

$C_{p,a}$  = specific heat capacity for air at constant pressure [kJ/kgK]

$C_{p,w}$  = specific heat capacity for water at constant pressure [kJ/kgK]

### 6.3.6 Temperature

The inlet temperature of the sorption storage is controlled by the heat exchanger. The temperature at the outlet is however controlled by the efficiency of the dehumidification of the desiccant bed. Based on the constant enthalpy it is possible to solve an energy balance in order to calculate the temperature out of the sorption storage.

$$h_{in} = h_{out}$$

$$h_{in} = C_{p,a} \cdot T_{out} \cdot Y_{out} \cdot h_g \cdot C_{p,w} \cdot T_{out}$$

$$T_{out} = \frac{h_{in} - h_g \cdot Y_{out}}{C_{p,a} + Y_{out} \cdot C_{p,w}} \quad (6.15)$$

### 6.3.7 Mass

The required mass of water and silica gel is based on the adsorption energy, water content and energy demand. The adsorption energy is solely based on the initial water content of the silica gel, and is determined through Equation 6.4. The mass of water is further calculated by the energy demand and adsorption energy available:

$$m_{water} = \frac{Q_{SH}}{Q_{ads}} \quad (6.16)$$

The mass of silica gel is calculated by dividing the mass of water with the difference in water content at charging and discharging.

$$m_{silica} = \frac{m_{water}}{W_{in} - W_{out}} \quad (6.17)$$

## 7. Results

### 7.1 Beijing

The mathematical model presented in chapter 6 has been applied to the 100 m<sup>2</sup> domestic household described in chapter 5. The necessary decisions regarding parameters have been based on the advices in Appendix A and results from test simulations of the model. They are thoroughly reasoned below.

#### 7.1.1 Charging

During charging the air is supposed to enter with a low relative humidity and high temperature in order to absorb the water content of the silica gel. The crucial parameter will be the ambient temperature at which the air enters the heat exchanger. The air will be discarded after exiting the sorption storage, so the outlet temperature is irrelevant, as it is not planned to utilize this air for anything.

##### 7.1.1.1 Inlet air

As the charging will occur during summer while there is supply of solar energy, it is reasonable to assume that the charging will happen during daytime. The average temperature during the summer months in Beijing varies from 20°C to 25.9°C with the average maximum temperature being 30°C [36, 37]. It is advised in Appendix A.4 that the inlet air temperature should be above 20°C for charging. However, through test simulations it was discovered that an ambient temperature of 20°C provides an unrealistic outlet temperature with several cases of close to and below zero. . It has therefore been necessary to raise the dimensioning ambient

temperature to  $T_{amb}=35^{\circ}\text{C}$ . The results from the simulation of  $T_{amb}=20^{\circ}\text{C}$  can be seen in Table 7.1, while a complete overview of the temperature simulations between  $20\text{-}35^{\circ}\text{C}$  can be seen in Appendix C.

The summer air of Beijing is far from dry as it is recorded with an average of 70% during the summer months [38, 39]. The dehumidifier will therefore be necessary in this case in order to lower the humidity.

### 7.1.1.2 Heat exchanger

As previously determined, the dimensioning ambient temperature for inlet air is  $35^{\circ}\text{C}$ . The available solar energy from the solar collector system in the Beijing case is as found in Table 5.3,  $Q_{out} = 2.5 \text{ kW}$ . The heat exchanger efficiency,  $\eta_{HEX} = 90\%$ . The heat transferred from the solar collector is therefore:

$$Q_{HEX} = 2.5 \text{ kW} \cdot 0.9 = \underline{2.25 \text{ kW}}$$

The heat exchanger airflow is calculated through Equation 6.3.

$$\dot{m}_{air} = \frac{2.25 \text{ kW}}{1.008 \text{ kJ/kgK} \cdot (353 \text{ K} - 308 \text{ K})} = \underline{0.0496 \text{ kg/s}}$$

### 7.1.1.3 Water content

The water content is recommended between 0.2 and 0.35 kg water/kg silica gel for charging. Simulations with these values did however not prove viable as it resulted in mostly unrealistic outlet temperatures. The results from the simulation of a water content of 0.2 kg/kg are presented in Table 7.2, and it can be seen that only low sorption storage efficiencies provide realistic values. The water content has therefore in collaboration with Prof. Dai been reduced to  $W=0.1 \text{ kg/kg}$ . The resulting outlet temperature for this parameter can be seen in Table 7.3.

Table 7.1: Outlet temperatures for  $W=0.1\text{kg/kg}$ ,  $T_{amb}=20^\circ\text{C}$  and all combinations of relative humidity in and sorption storage efficiency. Positive values are marked green and negative values are red.

nDH \ RH	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
0,0	80	80	80	80	80	80	80	80	80	80	80
0,1	66,01	66,5	66,99	67,48	67,97	68,45	68,94	69,43	69,92	70,41	70,9
0,2	52,24	53,2	54,16	55,13	56,09	57,06	58,02	58,99	59,96	60,92	61,89
0,3	38,67	40,1	41,52	42,95	44,38	45,81	47,24	48,67	50,1	51,54	52,97
0,4	25,32	27,19	29,06	30,94	32,82	34,7	36,58	38,47	40,36	42,25	44,15
0,5	12,17	14,47	16,78	19,09	21,41	23,73	26,05	28,38	30,72	33,06	35,41
0,6	-0,79	1,935	4,664	7,4	10,14	12,89	15,65	18,42	21,19	23,97	26,75
0,7	-13,5	-10,4	-7,28	-4,13	-0,97	2,196	5,373	8,561	11,76	14,97	18,18
0,8	-26,1	-22,6	-19,1	-15,5	-11,9	-8,37	-4,78	-1,18	2,432	6,059	9,699
0,9	-38,5	-34,6	-30,7	-26,7	-22,8	-18,8	-14,8	-10,8	-6,79	-2,76	1,298
1,0	-50,7	-46,4	-42,1	-37,8	-33,5	-29,1	-24,7	-20,3	-15,9	-11,5	-7,02

Table 7.2: Outlet temperatures for  $W=0.2$ ,  $T_{amb}=35^\circ\text{C}$  and all combinations of relative humidity in and sorption storage efficiency. Positive values are marked green and negative values are red.

nDH \ RH	0,0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
0,0	80	80	80	80	80	80	80	80	80	80	80
0,1	48,66	49,93	51,2	52,47	53,74	55	56,27	57,54	58,81	60,08	61,35
0,2	18,39	20,85	23,3	25,77	28,23	30,7	33,17	35,64	38,12	40,6	43,09
0,3	-10,9	-7,29	-3,72	-0,14	3,452	7,052	10,66	14,28	17,91	21,55	25,2
0,4	-39,1	-34,5	-29,9	-25,3	-20,6	-16	-11,3	-6,56	-1,84	2,906	7,671
0,5	-66,5	-60,9	-55,3	-49,7	-44	-38,4	-32,7	-26,9	-21,1	-15,3	-9,51
0,6	-92,9	-86,5	-80	-73,4	-66,8	-60,2	-53,5	-46,8	-40	-33,2	-26,3
0,7	-119	-111	-104	-96,5	-89	-81,4	-73,8	-66,2	-58,5	-50,7	-42,9
0,8	-143	-135	-127	-119	-111	-102	-93,7	-85,1	-76,5	-67,8	-59
0,9	-167	-159	-150	-141	-132	-122	-113	-104	-94,2	-84,6	-74,9
1,0	-191	-181	-172	-162	-152	-142	-132	-122	-111	-101	-90,5



Table 7.3: Outlet temperatures for  $W=0.1$ ,  $T_{amb} = 35^{\circ}\text{C}$  and all combinations of relative humidity in and sorption storage efficiency. Positive values are marked green and negative values are red.

nDH \ RH	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
0,0	80	80	80	80	80	80	80	80	80	80	80
0,1	66,01	67,2	68,38	69,57	70,75	71,94	73,12	74,31	75,49	76,67	77,86
0,2	52,24	54,57	56,91	59,25	61,6	63,95	66,3	68,65	71	73,36	75,72
0,3	38,67	42,13	45,59	49,06	52,54	56,03	59,52	63,03	66,54	70,06	73,59
0,4	25,32	29,86	34,41	38,98	43,57	48,18	52,8	57,44	62,1	66,77	71,47
0,5	12,17	17,76	23,37	29,02	34,69	40,4	46,13	51,89	57,68	63,5	69,34
0,6	-0,79	5,824	12,48	19,17	25,91	32,68	39,51	46,37	53,28	60,23	67,23
0,7	-13,5	-5,94	1,715	9,432	17,21	25,04	32,93	40,89	48,9	56,98	65,12
0,8	-26,1	-17,6	-8,91	-0,2	8,595	17,46	26,41	35,44	44,55	53,74	63,01
0,9	-38,5	-29	-19,4	-9,72	0,068	9,953	19,94	30,03	40,22	50,51	60,91
1,0	-50,7	-40,3	-29,8	-19,1	-8,38	2,507	13,51	24,64	35,9	47,29	58,81

The adsorption heat of the silica gel can then be calculated by Equation 6.4. Since  $W > 0.5$ , the second line of the equation is used.

$$Q_{ads} = -1400 \cdot 0.1 + 2950 = \frac{2810 \text{ kJ}}{\text{kg water}}$$

Considering a full year of use, the adsorption heat can be converted to W/kg water in order to use it in calculations with the energy demand:

$$Q_{ads} = \frac{2810 \text{ kJ/kg water}}{31536000 \text{ s}} = 0.0891 \frac{\text{W}}{\text{kg water}}$$

#### 7.1.1.4 Relative humidity and efficiency

As this is a theoretical system it has been necessary to decide the inlet relative humidity and efficiency of the dehumidification in the sorption storage. It is advised in the additional assignment text, Appendix A, that the relative humidity at inlet for charging should be between 40-60% despite the demand for dry air in the working principle. The efficiency of dehumidification is valid for both charging and discharging, and should according to Appendix A be between 60-80%. When reviewing Table 7.3 and Table 7.4 it is apparent that it comes

down to what temperature is best, as both tables show that the appropriate ranges overlap. As the summer average relative humidity in Beijing is 70%, the set point in this system is set to 50% to minimize the effect from the dehumidifier. The efficiency is set to 80% as it delivers appropriate temperature for both charging and discharging.

The relative humidity at equilibrium is calculated based on  $W = 0.1$  kg/kg, through Equation 6.5.

$$RH_{eq} = 0.0078 - 0.05759 \cdot 0.1 + 24.16554 \cdot 0.1^2 - 124.78 \cdot 0.1^3 + 204.226 \cdot 0.1^4 = \underline{0.1393 = 13.93\%}$$

### 7.1.1.5 Saturation pressure

The saturation pressure for  $T_{amb} = 35^\circ\text{C}$  is calculated according to 6.7:

$$P_{s,amb} = e^{23.1964 - \frac{3816.44}{308 - 46.13}} = \underline{5555.8 \text{ Pa}}$$

The saturation pressure for  $T_{in} = 80^\circ\text{C}$  is:

$$P_{s,in} = e^{23.1964 - \frac{3816.44}{353 - 46.13}} = \underline{47086.1 \text{ Pa}}$$

### 7.1.1.6 Humidity ratio

The humidity ratios are calculated based on the parameters calculated above. The humidity ratio for  $RH_{eq} = 13.93\%$  and  $P_{s,in} = 47086 \text{ Pa}$  is:

$$Y_{eq} = \frac{0.622 \cdot RH_{eq} \cdot P_{s,in}}{B - RH_{eq} \cdot P_{s,in}} = \frac{0.622 \cdot 0.1393 \cdot 5555.8}{101300 - 0.1393 \cdot 5555.8} = \underline{0.0431}$$

The humidity ratio for  $RH_{in} = 50\%$  and  $P_{s,amb} = 5555.8 \text{ Pa}$  is:

$$Y_{in} = \frac{0.622 \cdot RH_{in} \cdot P_{s,amb}}{B - RH_{in} \cdot P_{s,amb}} = \frac{0.622 \cdot 0.4 \cdot 4708.6}{101300 - 0.4 \cdot 4708.6} = \underline{0.014}$$

The humidity ratio out of the sorption storage,  $Y_{out}$ , is calculated through the following:

$$\begin{aligned}\Delta Y_{max} &= Y_{eq} - Y_{in} = 0.0431 - 0.014 = \underline{0.0291} \\ \Delta Y_{prac} &= \eta_{DH} \cdot \Delta Y_{max} = 0.8 \cdot 0.0291 = \underline{0.0233} \\ Y_{out} &= Y_{in} - \Delta Y_{prac} = 0.014 - 0.0233 = \underline{0.0373}\end{aligned}$$

### 7.1.1.7 Enthalpy

The enthalpies are calculated with the following specific heat capacity of air and water vapor along with the specific enthalpy of water vapor at  $T_{in}=80^\circ\text{C}$ :  $C_{p,a} = 1.006 \frac{\text{kJ}}{\text{kgK}}$ ,  $C_{p,w} = 1.84 \frac{\text{kJ}}{\text{kgK}}$  and  $h_g = 2643 \frac{\text{kJ}}{\text{kg}}$  [40]. The enthalpy of air:

$$h_a = C_{p,a} \cdot T_{in} = 1.006 \frac{\text{kJ}}{\text{kgK}} \cdot 353 \text{ K} = \underline{355.1 \frac{\text{kJ}}{\text{kg}}}$$

The enthalpy of water vapor:

$$h_w = h_{gi} + C_{p,w} \cdot T_{in} = 2643 \frac{\text{kJ}}{\text{kg}} + 1.84 \frac{\text{kJ}}{\text{kgK}} \cdot 353 \text{ K} = \underline{3292.5 \frac{\text{kJ}}{\text{kg}}}$$

The enthalpy of the moist air at the inlet:

$$h_{in} = h_a + Y_{in} \cdot h_w = 355.1 + 0.014 \cdot 3292.5 = \underline{401.1 \frac{\text{kJ}}{\text{kg}}}$$

### 7.1.1.8 Temperature

Through the calculations above it was possible to calculate the expected temperature out of the sorption storage with an efficiency of  $\eta_{DH}=80\%$ :

$$T_{out} = \frac{h_{in} - h_g \cdot Y_{out}}{C_{p,a} + Y_{out} \cdot C_{p,w}} = \frac{401.1 - 2643 \cdot 0.0373}{1.006 + 0.0373 \cdot 1.84} = \underline{281.61 \text{ K} = 8.6^\circ\text{C}}$$

## 7.1.2 Discharging

During discharging the air will be heated through the sorption storage so it can be utilized for space heating. The crucial parameter is the outlet temperature as it needs to be

sufficient for space heating. The air is therefore advised in Appendix A.4 to exit at a temperature between 35-45°C.

#### *7.1.2.1 Inlet air*

The discharging is expected to occur in a colder season where the solar energy supply is lower, but demand is higher. It is therefore reasonable to think that this will be outside the summer season, and the average temperature of the outdoor air in Beijing can be as low as 4°C as it is recorded in November [36, 37]. In order to deliver air with an inlet temperature of 20°C, mixing of fresh outdoor and indoor air will be necessary. According to Appendix A.3, a mixing ratio of one part outdoor air and five parts indoor air is the normal in China. This will not be considered further as the temperature is assumed 20°C at the inlet in the mathematical model, it is however necessary to take it into account in future work.

The average relative humidity in Beijing through the fall months are at an average of 60% [38, 39]. The humidifier will therefore need to raise the humidity, but as the mathematical model assumes sufficient humidity, it will not be considered further.

#### *7.1.2.2 Water content*

The water content of the silica gel at discharging is chosen to be 0.05 kg water/kg silica gel. This is based on the recommendations in Appendix A and has proven viable with good results. The outlet temperatures of the different combinations of efficiency and relative humidity is presented in Table 7.4, where the temperatures are marked according to their value above or below the lowest desired outlet temperature of 35°C.

Table 7.4: Outlet temperatures for  $W=0.05$ ,  $T_{amb}=5^{\circ}\text{C}$  and all combinations of relative humidity in and sorption storage efficiency. Values above  $35^{\circ}\text{C}$  are marked in green, and below are marked in red.

nDH \ RH	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
0,0	20	20	20	20	20	20	20	20	20	20	20
0,1	19,77	20,22	20,66	21,11	21,56	22	22,45	22,89	23,34	23,78	24,23
0,2	19,55	20,44	21,33	22,22	23,11	24,01	24,9	25,79	26,69	27,58	28,47
0,3	19,32	20,66	21,99	23,33	24,67	26,02	27,36	28,7	30,05	31,4	32,74
0,4	19,09	20,87	22,66	24,45	26,24	28,03	29,83	31,62	33,42	35,23	37,03
0,5	18,86	21,09	23,32	25,56	27,8	30,05	32,3	34,55	36,81	39,08	41,34
0,6	18,64	21,31	23,99	26,68	29,37	32,07	34,78	37,49	40,21	42,94	45,68
0,7	18,41	21,53	24,66	27,8	30,94	34,1	37,27	40,44	43,63	46,82	50,03
0,8	18,18	21,75	25,33	28,92	32,52	36,13	39,76	43,4	47,06	50,73	54,41
0,9	17,96	21,97	25,99	30,04	34,1	38,17	42,26	46,37	50,5	54,64	58,8
1,0	17,73	22,19	26,66	31,16	35,68	40,22	44,77	49,35	53,96	58,58	63,22

### 7.1.2.3 Relative humidity

The relative humidity at inlet is chosen based on the former decision of a dehumidification efficiency of 80%. Based on the results in Table 7.4 it is possible to choose between the advised values of a relative humidity between 60-80% in Appendix A.4. As the average relative humidity during fall in Beijing is 60%, the relative humidity set point is chosen at 70% to minimize the effect of the humidifier. This results in an outlet temperature that is within the desired temperature range of 35-45°C.

The relative humidity at equilibrium is:

$$RH_{eq} = 0.0078 - 0.05759 \cdot 0.05 + 24.16554 \cdot 0.05^2 - 124.78 \cdot 0.05^3 + 204.226 \cdot 0.05^4 = \underline{0.051} = 5.1\%$$

### 7.1.2.4 Saturation pressure

As the air is not additionally heated through a heat exchanger during discharging, only the saturation pressure for the inlet temperature is necessary. The saturation pressure for  $T_{in}=20^{\circ}\text{C}$  is:

$$P_{s,in} = e^{23.1964 - \frac{3816.44}{293 - 46.13}} = 2291.8 \text{ Pa}$$

### 7.1.2.5 Humidity ratio

The humidity ratios are calculated based on the parameters previously calculated. The humidity ratio for  $RH_{eq}=5.1\%$  and  $P_{s,in}=2291.8\text{Pa}$  is:

$$Y_{eq} = \frac{0.622 \cdot RH_{eq} \cdot P_{s,in}}{B - RH_{eq} \cdot P_{s,in}}$$

$$= \frac{0.622 \cdot 0.051 \cdot 2291.8}{101300 - 0.051 \cdot 2291.8} = \underline{0.00072}$$

The humidity ratio for  $RH_{eq}=70\%$  and  $P_{s,in}=2291.8\text{Pa}$  is:

$$Y_{in} = \frac{0.622 \cdot RH_{in} \cdot P_{s,in}}{B - RH_{in} \cdot P_{s,in}}$$

$$= \frac{0.622 \cdot 0.7 \cdot 2291.8}{101300 - 0.7 \cdot 2291.8} = \underline{0.01}$$

The humidity ratio out of the sorption storage,  $Y_{out}$ , is calculated through the following:

$$\Delta Y_{max} = Y_{in} - Y_{eq} = 0.01 - 0.000072 = \underline{0.0093}$$

$$\Delta Y_{prac} = \eta_{DH} \cdot \Delta Y_{max} = 0.8 \cdot 0.0093 = \underline{0.0074}$$

$$Y_{out} = Y_{in} - \Delta Y_{prac} = 0.0093 - 0.0074 = \underline{0.0026}$$

### 7.1.2.6 Enthalpy

The necessary enthalpies were calculated with following specific heat capacity of air and water vapor along with the specific enthalpy of water vapor of  $T_{in} = 20^\circ\text{C}$ :  $C_{p,a} = 1.006 \frac{\text{kJ}}{\text{kgK}}$ ,  $C_{p,w} = 1.84 \frac{\text{kJ}}{\text{kgK}}$  and  $h_g = 2565.3 \frac{\text{kJ}}{\text{kg}}$  [40].

$$h_a = C_{p,a} \cdot T_{in} = 1.006 \frac{\text{kJ}}{\text{kgK}} \cdot 293 \text{ K} = \underline{294.8 \frac{\text{kJ}}{\text{kg}}}$$

$$h_w = h_{gi} + C_{p,w} \cdot T_{in} = 2565 \frac{\text{kJ}}{\text{kg}} + 1.84 \frac{\text{kJ}}{\text{kgK}} \cdot 293 \text{ K} = \underline{3182.8 \frac{\text{kJ}}{\text{kg}}}$$

$$h_{in} = h_a + Y_{in} \cdot h_w = 294.8 + 0.01 \cdot 3182.8 = \underline{326.6 \frac{\text{kJ}}{\text{kg}}}$$

### 7.1.2.7 Temperature

Through the previous calculations it was possible to calculate the expected temperature out of the sorption storage with an efficiency of  $\eta_{DH} = 80\%$ :

$$T_{out} = \frac{h_{in} - h_g \cdot Y_{out}}{C_{p,a} + Y_{out} \cdot C_{p,w}} = \frac{325.83 - 2565.3 \cdot 0.0026}{1.006 + 0.0026 \cdot 1.84} = \underline{315.8 K = 43^\circ C}$$

### 7.1.3 Mass

The necessary mass of water in the sorption storage is calculated through Equation 6.16 based on the energy demand and adsorption heat from the sorption storage:

$$m_{water} = \frac{2.5 kW}{8.91 \cdot 10^{-5} kW/kg} = \underline{28056.9 kg}$$

The necessary mass of silica gel is then calculated based on the amount of water content:

$$m_{silica} = \frac{m_{water}}{W_{in} - W_{out}} = \frac{28057 kg}{0.1 - 0.05} = \underline{561138.7 kg}$$

### 7.1.4 Summary

Both the relative humidity and efficiency of dehumidification are realistic considering the available values from Zhang et al. [29], along with the recommendations from Prof. Dai. The relevance of choosing the correct relative humidity because of its impact on the outlet temperature as stated by Zhang et al. [29] is also quite apparent through the results in Table 7.2 and Table 7.3. The necessary inlet temperature is a little high compared to studies like Deshmukh et al. [26] and Fernandes et al. [27], but not to an extent where it is impossible. The water content is however critically low compared to the advised value and other studies. It is possible to consider the results of Jaehnig et al. [25], where the experimental results showed a water content between 2-12%. But as this is a theoretical model, the silica gel should have a potential water content up to 38% according to that same study. The low water content also results in a massive system regarding mass, potentially increasing the expenses of the system in total. Nevertheless, the theoretical model has viable results for its calculations and a good output temperature. And it is therefore considered a working model that can be further tested

for a new, separate location. A summary of the dimensioned parameters are presented in Table 7.5.

Table 7.5: A summary of the constants and set points for the Beijing case.

<b>Charging set point</b>			
$T_{in}$ [K / °C]	353 / 80	$Y_{in}$ [–]	0.014
$T_{amb}$ [K / °C]	308 / 35	$\Delta Y_{max}$ [–]	0.0291
$RH_{in}$ [–]	0.5	$\Delta Y_{prac}$ [–]	0.0233
$\eta_{DH}$ [–]	0.8	$Y_{out}$ [–]	0.0373
$W$ [kg/kg]	0.1	$C_{p,a}$ [kJ/kgK]	1.006
$\eta_{HEX}$ [–]	0.9	$C_{p,w}$ [kJ/kgK]	1.84
$Q_{ads}$ [W/kg water]	0.0891	$h_g$ [kJ/kg]	2643
$RH_{eq}$ [–]	0.1393	$h_a$ [kJ/kg]	355.1
$P_{s,amb}$ [Pa]	5555.8	$h_w$ [kJ/kg]	3292.5
$P_{s,in}$ [Pa]	47086.1	$h_{in}$ [kJ/kg]	401.1
$Y_{eq}$ [–]	0.0431	$T_{out}$ [K / °C]	281.61 / 8.61
<b>Discharging set point</b>			
$T_{in}$ [K / °C]	293 / 20	$\Delta Y_{max}$ [–]	0.0093
$T_{amb}$ [K / °C]	278 / 5	$\Delta Y_{prac}$ [–]	0.0074
$RH_{in}$ [–]	0.7	$Y_{out}$ [–]	0.0026
$\eta_{DH}$ [–]	0.8	$C_{p,a}$ [kJ/kgK]	1.006
$W$ [kg/kg]	0.05	$C_{p,w}$ [kJ/kgK]	1.84
$RH_{eq}$ [–]	0.051	$h_g$ [kJ/kg]	2565.3
$\eta_{DH}$ [–]	0.8	$h_a$ [kJ/kg]	294.8
$P_{s,in}$ [Pa]	2291.8	$h_w$ [kJ/kg]	3182.8
$Y_{eq}$ [–]	0.00072	$h_{in}$ [kJ/kg]	326.6
$Y_{in}$ [–]	0.01	$T_{out}$ [K / °C]	315.8 / 43
<b>Individual parameters</b>			
$Q_{HEX}$ [kW]	2.25		
$\dot{m}_{air}$ [kg/s]	0.0496		
$m_w$ [kg]	28056.9		
$m_{silica}$ [kg]	561138.7		



## 7.2 Trondheim

As the model itself is dimensioned for Beijing with set points for the parameters that are effected by the outdoor air, there are just some minor calculations necessary in order to make the model functional for Trondheim. Since the solar irradiation is different in Beijing and Trondheim, the heat exchanger will be dimensioned again. So will the mass of water and silica gel as the energy demand is different. The remaining parameters are assumed as listed in Table 7.5.

### 7.2.1 Heat exchanger

The temperature set point for the sorption storage system is already set, and will also apply for the Norwegian case. The summer temperature in Trondheim is recorded at an average of 13°C [41]. It will therefore be necessary to install an additional heat source to reach the set point temperature of  $T_{amb} = 35^{\circ}C$ .

The heat exchanger is dimensioned based on the set point temperature. With the solar collector output from Table 5.3 and the heat exchanger efficiency of 90%, the heat transferred from the solar collector fluid to the airflow is:

$$Q_{HEX} = 0.262 \cdot 0.9 = \underline{0.2358 \text{ kW}}$$

The heat exchanger airflow is calculated through Equation 6.3. The specific heat capacity of air is,  $C_{p,a} = 1.008 \text{ kJ/kgK}$  [40].

$$\dot{m}_{air} = \frac{0.2358 \text{ kW}}{1.008 \text{ kJ/kgK} \cdot (353 - 308 \text{ K})} = \underline{0.0052 \text{ kg/s}}$$

### 7.2.2 Relative humidity

As the summer climate in Norway is quite different than in China it is unrealistic to expect to reach the same average temperature. It is however easier to reach the set point of relative humidity at inlet. The average relative humidity of Trondheim during the summer is 79% [42, 43], so the dehumidifier will be necessary for this case as well.

Unlike Beijing, the relative humidity of Trondheim is quite stable and still high during fall. With an average of 81% [42, 43] throughout September, October and November, the humidifier will most likely not be necessary in this system.

### 7.2.3 Mass

As the energy demand in the Trondheim case is very low compared to Beijing, the mass of water and silica gel is assumed to be reduced as well. The necessary mass of water is calculated through Equation 6.16.

$$m_{water} = \frac{0.262 \text{ kW}}{8.91 \cdot 10^{-5} \frac{\text{kW}}{\text{kg water}}} = \underline{2940.37 \text{ kg}}$$

The necessary mass of silica gel is calculated based on the mass of water and the water content:

$$m_{silica} = \frac{2940.37 \text{ kg water}}{0.1 - 0.05 \frac{\text{kg water}}{\text{kg silica gel}}} = \underline{58807.35 \text{ kg}}$$

### 7.2.4 Summary

As expected, the Norwegian system presented a lower mass demand than the Chinese due to a lower energy demand. This will decrease the investment cost of the system. However, some concerns that should be taken into account include the mixing ratio of outdoor and indoor air during discharging. Norway has restricted laws regarding reused air in ventilation. The mixing ratio should therefore be investigated prior to further development. A solution could be to install a heat source that preheats the fresh air prior to the sorption storage, omitting the need for indoor air completely. The same heat source could be responsible for preheating the summer air if the temperature set point cannot be reached naturally. It is suggested that this heat source is renewable to keep with the aim of the project being environment friendly.

Table 7.6: A summary of the dimensioning parameters for the system located in Trondheim, Norway.

Variable	
$Q_{HEX}$ [kW]	0.2358
$\dot{m}_{air}$ [kg/s]	0.0052
$m_{water}$ [kg]	2940.37
$m_{silica}$ [kg]	58807.35

## 8. Conclusion

The work of this thesis has provided a presentation of the potential storage alternatives within the field of thermal energy storage of solar heat with an emphasis on thermal solar storage using sorption. The concept of solar energy and its potential is presented including a simplified dimensioning method. A short review of stationary solar collectors is also included.

The main purpose of seasonal solar heat storage is to meet the disparity between supply and demand present in today's traditional solar collector systems. By utilizing seasonal storage, solar heat can be stored from a season of low demand and high supply, to a season of high demand and low supply. Along with improving the efficiency of the solar collector system in total, such a solution should reduce the top load demand of heat pumps or electricity. Through a literature study it became apparent that thermochemical energy storage is the least researched out of the three thermal energy storage alternatives. Sensible heat is widely available through traditional hot water tanks, latent heat is becoming more and more commercially available. One of the challenges in thermochemical energy storage is finding the optimal working material to store energy, and this is one of the reasons it is still being researched. External literature and research has described the challenges of working materials, and revealed that currently no material fulfills the requirements when tested experimentally. The storage also faces challenges regarding technology as the materials are sensitive and require advanced heat exchangers and storage tanks in order to accomplish its full potential. Still, theoretically the thermochemical storage outperform the other alternatives regarding storage capacity and heat transfer. The chemical aspect of the technology enable energy storage for seasons without increasing the investment cost drastically.

A complete seasonal sorption energy storage with solar collector and heat exchanger is also presented. The system is dimensioned and constructed based on information acquired through the literature study and the collaborative work with Professor Dai at SJTU. The model has been based on a 100 m<sup>2</sup> domestic household located in Beijing, China. To charge the sorption storage, dry air heated by a solar collector is sent through a desiccant bed filled with silica gel. The silica gel then adsorbs the heat and releases its water content to the air which is discarded to the surroundings. The dehydrated silica gel is then stored alone until a heat demand is present. To discharge, humid medium temperature air is sent through the sorption storage releasing its water content while the silica releases its heat for the hydration. The heated air is then further utilized in space heating. The following values proved to be compatible in order to reach sufficient results for charging:  $RH_{in}= 50\%$ ,  $\eta_{DH}= 80\%$ ,  $W= 0.1 \text{ kg/kg}$ ,  $T_{amb}= 35^{\circ}\text{C}$  and  $T_{in}= 80^{\circ}\text{C}$ . The aim of discharge is to provide air at a sufficient outlet temperature for space heating. The system proved to function with the following values for discharge:  $RH_{in}= 70\%$ ,  $W= 0.05 \text{ kg/kg}$ ,  $T_{amb}= 5^{\circ}\text{C}$  and  $T_{in}= 20^{\circ}\text{C}$ , resulting in  $T_{out}= 43^{\circ}\text{C}$ . Based on these values, the sorption storage is able to store 0.089 W/kg water of adsorption heat. The adsorption heat will increase if the water content during charging,  $W$ , is increased. So in order to minimize the necessary volume of sorption storage, further work is required to optimize this parameter.

The importance of choosing the correct values has become apparent through the simulations as the calculated variables rely heavily on the constants. The water content of silica gel,  $W$ , is recognized as one of the most important parameter as it affects the energy storage and necessary mass of silica and water greatly. The relative humidity and inlet temperature also affect the performance of the system and especially the outlet temperature at discharge to some extent, and should therefore be chosen wisely.

The dimensioned system has also been tested under Norwegian conditions with a system located in Trondheim. The system theoretically proved to function here as well, even though the inlet temperature during charging is unrealistic for Norwegian conditions. An additional heat source should be considered for further work on the Trondheim system. As the energy demand for a passive house in Norway proved to be much lower than the assigned Chinese energy demand, the Norwegian system required much less material which is also economical beneficial.

## 9. Further work

Before proceeding to building a prototype of the system, the mathematical model should be further optimized. The current sorption storage is able to store a limited amount of energy due to the desiccants low water content. The low adsorption energy also affect the necessary mass of desiccant and the volume of the system. Further work optimizing the model should also include implementing a slightly lower charging temperature. This will make the system more applicable to other locations than hot climates like China as it will not specifically require an additional heater. As the efficiency of dehumidification is only chosen theoretically in this work, emphasis should be put into research regarding this value. This also applies to the heat exchanger and heat transfer that occurs.

Further, the method of space heating should be reconsidered if the system is to be applied in countries with cold fall and winters. A nearly zero energy house will potentially accept heating through air, but the outlet temperature and mass flow through the sorption storage and ventilation should be considered to ensure comfortable conditions for the residents.

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## Appendix A: Additional information

The following information regarding parameters are made by Prof. Yanjun Dai at Shanghai Jiao Tong University through meetings with the author throughout her stay in Shanghai. The parameters have been used as guideline for the choices made in the mathematical model based in Beijing. Prof. Dai also provided the additional information about the size of the system and chose the desiccant.

### A.1 The system

The system should be dimensioned for a 100 m<sup>2</sup> domestic household located in Beijing. The seasonal sorption storage should use silica gel and water as its working pair. The solar collector should be a vacuum flat plate solar collector. The system should further be tested in a Norwegian climate with a location such as Trondheim.

### A.2 Energy demand

As the Chinese do not have an energy standard for buildings, it in average is assumed an energy demand of 50 W/m<sup>2</sup> for all building types.

### A.3 Mixing air ratio

In China it would be reasonable to reuse indoor air by mixing one part fresh outdoor air with five parts indoor air in order to reach an appropriate temperature for space heating.

### A.4 Values

The following values are appropriate for a seasonal sorption storage system using silica gel based in China:

$SF_{SH}$ [-]	0.5
$\eta_{coll}$ [-]	0.5
$\eta_{HEX}$ [-]	0.9

$W_{charging} \left[ \frac{kg \text{ water}}{kg \text{ desiccant}} \right]$	0.2-0.25
$W_{discharging} \left[ \frac{kg \text{ water}}{kg \text{ desiccant}} \right]$	0.05
$T_{amb} [^{\circ}C]$	25-35
$T_{in} [^{\circ}C]$	80
$RH_{in,charging} [-]$	0.4-0.6
$RH_{in,discharging} [-]$	0.6-0.8
$\eta_{DH} [-]$	0.6-0.8

## Appendix B: Matlab scripts

### B.1 Heat exchanger

```
%- Solar heat to air HEX, charging -
clear all
%-- BEIJING --

Q_sol = 2.5; %Energy output from solar collector [kW]
n_hex = 0.9; %efficiency of HEX
T_a1 = 273+35; %temperature of air in [K]
T_a2 = 273+80; %temperature of air out [K]
dT_a = T_a2 - T_a1; %temperature difference in air in/out [K]
cp_air = 1.008; %specific heat capacity for air [kJ/kgK]
m_dot = Q_sol*n_hex / (cp_air*dT_a); %mass flow of air [kg/s]
```

### B.2 Charging

```
% - Charging script
clear all
e=exp(1);

%--Constants
W = 0.1; %water content of desiccant [kg water/kg
desiccant]
n_DH = 0.8; %dehumidification efficiency
RH_i = 0.5; %(ambient) relative humidity in
T_amb = 273+35; %ambient air temperature [K]
T_in = 273+80; %air temperature out of HEX [K]
B = 101.3*10^3; %Pa (1atm)
RH_eq = 0.0078-0.05759*W+24.1655*W^2-124.78*W^3+204.226*W^4;
%Equilibrium isotherm (Ideal relative humidity at exit)
Q_ads = (-1400*W+2950); %Adsorption heat for W>0.5
```

```

%--Saturation pressure
PS_a = e^(23.1964-(3816.44/(T_amb-46.13))); %saturation
pressure of water vapor [Pa]

PS_in = e^(23.1964-(3816.44/(T_in-46.13))); %saturation
pressure of water vapor [Pa]

%--Humidity ratio
Y_in = (0.622*RH_i*PS_a)/(B-RH_i*PS_a); %humidity ratio
in [kg moisture/kg dry air]

Y_eq = (0.622*RH_eq*PS_in)/(B-RH_eq*PS_in); %humidity ratio
at equilibrium [kg moisture/kg dry air]
dY_max = Y_in-Y_eq; %difference in
max. humidity ratio [kg moisture/kg dry air]
dY_prac = n_DH*dY_max; %difference in
realistic humidity ratio [kg moisture/kg dry air]
Y_out = Y_in-dY_prac; %humidity ratio
out [kg moisture/kg dry air]

%--Calculating enthalpy, assuming that dh=dH=0
Cp_a = 1.006; %specific heat capacity for air [kJ/kgK]
Cp_w = 1.84; %specific heat capacity for water vapor [kJ/kgK]
h_g = 2643; %specific enthalpy of water vapor at 80'C
[kJ/kg]
h_a = Cp_a*T_in; %specific enthalpy for air [kJ/kg]
h_sw = Cp_w*T_in+h_g; %specific enthalpy for water [kJ/kg]
h_in = h_a+Y_in*h_sw; %specific enthalpy for the moist air
[kJ/kg]

%--Calculating temperature out
T_out = (h_in-h_g*Y_out)/(Cp_a+Y_out*Cp_w); %Temperature out
of energy storage [K]

```

### B.3 Discharging

```

% - Discharging script
clear all
e=exp(1);

%--Constants
W = 0.05; %water content of desiccant [kg water/kg
desiccant]
n_DH = 0.80; %dehumidification efficiency
RH_i = 0.7; %(ambient) relative humidity in
T_amb = 273+5; %ambient air temperature [K]
T_in = 273+20; %air temperature out of HEX [K]
B = 101.3*10^3; %Pa (1atm)
RH_eq = 0.0078-0.05759*W+24.1655*W^2-124.78*W^3+204.226*W^4;
%Equilibrium isotherm (Ideal relative humidity)

```

```

%--Saturation pressure
PS_in = e^(23.1964-(3816.44/(T_in-46.13))); %saturation
pressure of water vapor [Pa]

%--Humidity ratio
Y_in = (0.622*RH_i*PS_in)/(B-RH_i*PS_in); %humidity
ratio in [kg moisture/kg dry air]
Y_eq = (0.622*RH_eq*PS_in)/(B-RH_eq*PS_in); %humidity ratio
at equilibrium [kg moisture/kg dry air]
dY_max = Y_in-Y_eq; %difference in
max. humidity ratio [kg moisture/kg dry air]
dY_prac = n_DH*dY_max; %difference in
realistic humidity ratio [kg moisture/kg dry air]
Y_out = Y_in-dY_prac; %humidity ratio out [kg moisture/kg
dry air]

%--Calculating enthalpy, assuming that dh=dH=0
Cp_a = 1.006; %specific heat capacity for air [kJ/kgK]
Cp_w = 1.84; %specific heat capacity for water vapor [kJ/kgK]
h_g = 2565.3; %specific enthalpy of water vapor at 35'C
[kJ/kg]
h_a = Cp_a*T_in; %specific enthalpy for air [kJ/kg]
h_sw = Cp_w*T_in+h_g; %specific enthalpy for water [kJ/kg]
h_in = h_a+Y_in*h_sw; %specific enthalpy for the moist air
[kJ/kg]

%--Calculating temperature out
T_out = (h_in-h_g*Y_out)/(Cp_a+Y_out*Cp_w); %Temperature out
of energy storage [K]

```

## B.4 Mass

```

% - Calculating the necessary mass of water and silica -
clear all

Q = 2.5 %demand for space heating [kW]
Q_ads = 2810; %the energy demand for space heating [kJ/kg]
q_ads = Q_ads/(60*60*24*365); %adsorption heat in kW/kg
m_w = Q/q_ads; %mass of water [kg]
w_in = 0.1; %water content in /charging [kg/kg]
w_o = 0.05; %water content out /discharge [kg/kg]
dw = w_in - w_o; %difference in water content in and out
[kg/kg]
m_s = m_w/dw; %mass of silica gel [kg]

```

# Appendix C: Matlab results

## C.1 Temperature variation

The result of the simulations with different inlet temperatures are presented in table below. Temperatures above 0°C are marked in green, while those below are marked in red.

Table C.1: Simulation result for  $W=0.1$  kg/kg and  $T_{amb}=20^{\circ}C$ .

nDH \ RH	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
0,0	80	80	80	80	80	80	80	80	80	80	80
0,1	66,01	66,5	66,99	67,48	67,97	68,45	68,94	69,43	69,92	70,41	70,9
0,2	52,24	53,2	54,16	55,13	56,09	57,06	58,02	58,99	59,96	60,92	61,89
0,3	38,67	40,1	41,52	42,95	44,38	45,81	47,24	48,67	50,1	51,54	52,97
0,4	25,32	27,19	29,06	30,94	32,82	34,7	36,58	38,47	40,36	42,25	44,15
0,5	12,17	14,47	16,78	19,09	21,41	23,73	26,05	28,38	30,72	33,06	35,41
0,6	-0,79	1,935	4,664	7,4	10,14	12,89	15,65	18,42	21,19	23,97	26,75
0,7	-13,5	-10,4	-7,28	-4,13	-0,97	2,196	5,373	8,561	11,76	14,97	18,18
0,8	-26,1	-22,6	-19,1	-15,5	-11,9	-8,37	-4,78	-1,18	2,432	6,059	9,699
0,9	-38,5	-34,6	-30,7	-26,7	-22,8	-18,8	-14,8	-10,8	-6,79	-2,76	1,298
1,0	-50,7	-46,4	-42,1	-37,8	-33,5	-29,1	-24,7	-20,3	-15,9	-11,5	-7,02

Table C.2: Simulation result for  $W=0.1$  kg/kg and  $T_{amb}=25^{\circ}C$ .

nDH \ RH	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
0,0	80	80	80	80	80	80	80	80	80	80	80
0,1	66,01	66,67	67,34	68	68,67	69,33	70	70,66	71,33	71,99	72,66
0,2	52,24	53,55	54,86	56,17	57,48	58,79	60,11	61,42	62,74	64,05	65,37
0,3	38,67	40,61	42,55	44,49	46,43	48,38	50,32	52,28	54,23	56,19	58,15
0,4	25,32	27,86	30,41	32,96	35,52	38,08	40,65	43,22	45,8	48,39	50,98
0,5	12,17	15,3	18,44	21,59	24,74	27,91	31,08	34,27	37,46	40,66	43,87
0,6	-0,79	2,915	6,629	10,36	14,1	17,85	21,62	25,4	29,19	33	36,82
0,7	-13,5	-9,29	-5,02	-0,73	3,582	7,91	12,26	16,62	21	25,4	29,82
0,8	-26,1	-21,3	-16,5	-11,7	-6,8	-1,92	2,994	7,93	12,89	17,87	22,88
0,9	-38,5	-33,2	-27,8	-22,5	-17,1	-11,6	-6,17	-0,67	4,853	10,41	16
1,0	-50,7	-44,9	-39	-33,1	-27,2	-21,2	-15,2	-9,19	-3,11	3,011	9,17

Table C.3: Simulation result for  $W=0.1$  kg/kg and  $T_{amb}=30^{\circ}C$ .

nDH \ RH	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
0,0	80	80	80	80	80	80	80	80	80	80	80
0,1	66,01	66,9	67,8	68,69	69,58	70,47	71,37	72,26	73,15	74,04	74,93
0,2	52,24	53,99	55,76	57,52	59,28	61,05	62,81	64,58	66,35	68,12	69,9
0,3	38,67	41,27	43,88	46,49	49,1	51,72	54,34	56,97	59,61	62,24	64,89
0,4	25,32	28,73	32,16	35,59	39,04	42,49	45,95	49,43	52,91	56,4	59,91
0,5	12,17	16,37	20,6	24,83	29,09	33,36	37,64	41,95	46,27	50,6	54,95
0,6	-0,79	4,188	9,185	14,21	19,25	24,32	29,41	34,53	39,67	44,84	50,03
0,7	-13,5	-7,83	-2,08	3,708	9,525	15,38	21,26	27,17	33,13	39,11	45,13
0,8	-26,1	-19,7	-13,2	-6,66	-0,09	6,522	13,18	19,88	26,63	33,42	40,26
0,9	-38,5	-31,4	-24,2	-16,9	-9,6	-2,24	5,177	12,65	20,18	27,77	35,41
1,0	-50,7	-42,9	-35	-27	-19	-10,9	-2,75	5,478	13,78	22,15	30,6

Table C.4: Simulation result for  $W=0.1$  kg/kg and  $T_{amb}=35^{\circ}C$ .

nDH \ RH	0	0,1	0,2	0,3	0,4	0,5	0,6	0,7	0,8	0,9	1,0
0,0	80	80	80	80	80	80	80	80	80	80	80
0,1	66,01	67,2	68,38	69,57	70,75	71,94	73,12	74,31	75,49	76,67	77,86
0,2	52,24	54,57	56,91	59,25	61,6	63,95	66,3	68,65	71	73,36	75,72
0,3	38,67	42,13	45,59	49,06	52,54	56,03	59,52	63,03	66,54	70,06	73,59
0,4	25,32	29,86	34,41	38,98	43,57	48,18	52,8	57,44	62,1	66,77	71,47
0,5	12,17	17,76	23,37	29,02	34,69	40,4	46,13	51,89	57,68	63,5	69,34
0,6	-0,79	5,824	12,48	19,17	25,91	32,68	39,51	46,37	53,28	60,23	67,23
0,7	-13,5	-5,94	1,715	9,432	17,21	25,04	32,93	40,89	48,9	56,98	65,12
0,8	-26,1	-17,6	-8,91	-0,2	8,595	17,46	26,41	35,44	44,55	53,74	63,01
0,9	-38,5	-29	-19,4	-9,72	0,068	9,953	19,94	30,03	40,22	50,51	60,91
1,0	-50,7	-40,3	-29,8	-19,1	-8,38	2,507	13,51	24,64	35,9	47,29	58,81

# Transitional season solar energy storage using physical sorption materials for low carbon district heating in cold climate

Ellen-Andrea Tiltrum Myhre

**KEYWORDS:** Sorption energy storage, Thermochemical energy storage, Silica gel, Seasonal solar energy storage

**ABSTRACT:**

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The purpose of energy storage is to improve the general efficiency of a solar collector system by limiting the disparity between supply and demand. As thermochemical energy storage has a higher energy capacity and requires lower volumes, it is in literature regarded as the best alternative amongst the available options of thermal energy storage.

In this study a mathematical model of a theoretical seasonal sorption energy storage located in Beijing, China, is dimensioned. It is not validated, but the resulting model is theoretically tested in the cold climate of Trondheim, Norway. The model of the seasonal sorption storage using silica gel is functioning. Through simulations, the importance of correct water content, relative humidity and charging temperature became apparent. Results show that the model is able to store 0.089 kW/kg water while meeting the outlet temperature requirement, but that the energy demand heavily affects the necessary volume and size.

---

## 1. Introduction

Through a report written by the Intergovernmental Panel (IPCC) [1], it is revealed that humans are, with 95% certainty, responsible for the global warming the world is experiencing. The parties of the United Nations Framework Convention on Climate Change (UNFCCC) has therefore taken action in order to prevent this from evolving. This is done through the Paris Agreement of 2015, which aims to keep the global temperature rise below 2°C compared to pre-industrial

levels in the respective countries [2]. According to the European Union (EU), the building sector will be forced to take great measures in order to reach the goals of the Paris Agreement [4]. Buildings are in general responsible for 40% of the total energy consumption, and 36% of the CO<sub>2</sub> emission in the EU [5]. The concept of nearly zero energy building (nZEB) has therefore been introduced. By the end of 2018, the goal is that every public building will be nZEB. Further, by the end of 2020, nearly all buildings should be nZEB. In a nearly zero energy building, the energy

demand is very low or nearly zero and the demand should be covered by renewable energy [4]. In this way, it is possible for the building to compensate for its own greenhouse gas emission throughout its lifetime. The energy demand can be reduced by increasing the insulation in exterior walls, floors and roofs along with installing well-insulated windows. By utilizing solar collectors and panels for the energy, it is possible to produce the heat and electricity on site. The solar irradiation does however vary with the weather and season, making it difficult to rely on solar energy alone. This presents a major drawback as it leads to disparity between the supply and demand. One solution to this problem is to store the energy with thermal energy storage (TES) for when the supply is low and demand is high. A technology like this will be necessary in order to offer energy security for different types of renewable energy systems, in addition to improving the energy utilization efficiency in the systems.

## **2. Thermal energy storage**

By heating and cooling a storage medium, the thermal energy storage technology stock thermal energy so it can be available later on. Not only does this improve the general efficiency and reliability of a system, it can also improve the economic aspect by prolonging the running time of the system [14]. Thermal

energy storage solutions are divided into three types of energy storage; sensible, latent and thermochemical.

Sensible heat storage (SHS) is the most mature technology out of the three types of thermal energy storage. This low-cost and reliable way of storing heat utilizes the temperature difference between a transport medium and storage medium without phase change. The storage medium can be both liquid and solid, with water, brick, rock and soil being the most common materials [13, 15, 16]. There are several versions of SHS, where water tank storage and underground storage are two widely used technologies. The former is a common household item as it is the commercial way of storing hot water in buildings today. Here a heat transfer fluid indirectly heats water in a storage tank. The hot water is then stored until needed, either for heat exchange purposes or for domestic hot water. Even though SHS is the simplest method of thermal energy storage, its low energy storage density require large storage capacities limiting the use of it in smaller projects [16, 17].

With latent heat storage (LHS) the heat is stored in a material by a nearly isothermal phase change. As soon as the temperature of the chosen phase change material (PCM) rises above its critical temperature it will change phase and



simultaneously take up energy equivalent to its latent heat. When the temperature later decreases below this critical temperature, the material will release the latent heat and change into its initial state [14-16, 18]. The change in physical state can occur in several forms, such as: solid-solid, solid-gas, liquid-gas and vice versa [20]. According to Tatsidjodoung, Le Pierrès and Luo [20] LHS has shown promise within residential applications. The main purpose of such systems in buildings is to decrease the temperature variation, especially due to solar radiation loads. This can be accomplished by inserting a suitable material into the building mass, for example the internal wall or ceiling. This way, the excess heat from the room is extracted keeping the room at a steady temperature. The heat is stored within the wall until the temperature decreases below the critical point forcing the PCM to release the heat again. Compared to sensible heat storage, latent heat storage has a much higher energy storage density. It will therefore be able to store more energy at a lower total storage capacity [13, 18]. Although, to avoid releasing the heat prematurely the latent heat storage depends on the temperature staying above the phase change temperature after charging. This presents a limitation regarding seasonal storage, as the temperature might fluctuate over that period.

### 3. Sorption energy storage

Chemical energy storage, in general, utilizes supply energy such as thermal energy, electrical energy or electromagnetic radiation [18]. All types of thermochemical energy storage, i.e. with and without sorption, uses a reversible physico-chemical phenomena (chemical reaction) between materials to store energy in their chemical bonds, as illustrated in . When charging, energy is applied to a mixture of material A and B, breaking their binding and forcing them to separate. The two materials can then easily be stored individually until the energy is needed. To discharge, the materials are brought together and mixed, releasing the energy.

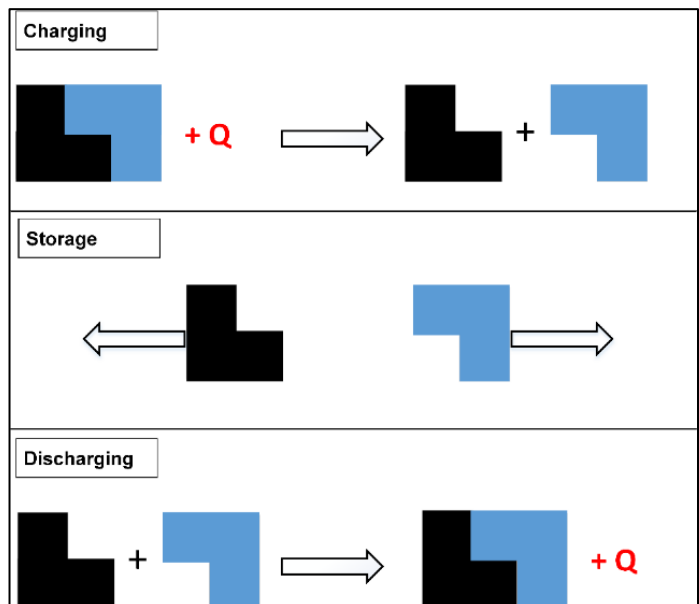


Figure 12: Illustration of how the thermochemical reaction occur.

Thermochemical energy storage with sorption, hereafter called sorption energy storage, can further be divided into open and closed systems.

Open sorption energy storage systems operate at atmospheric pressure and release its working fluid vapor into the surroundings. Because of this, water is the only medium utilized in these systems as it is nontoxic and harmless [8-10]. The open system is an attractive solution due to its ability to store heat to meet peak load demands or energy redistribution requirements. Charging of the system is typically fulfilled by hot air, heated by a solar collector, sent through a reactor filled with sorbent. By exchanging heat, the hot air desorbs the water from the adsorbent. The air then exits the reactor cooler and saturated, leaving the dehydrated adsorbent to store the heat. To discharge, moist air at medium to low temperature goes through the same reactor. The air then exchanges its moisture in exchange for the heat energy stored in the adsorbent. The air exits the reactor dry and heated, able to meet heat demands from space heating or domestic hot water [9, 12]. Open systems have a lower investment cost than closed systems since it require less components like condensers and evaporators, and can even provide better heat and mass transfer conditions. However, it is important to ensure air with a high enough moisture content and relative humidity to secure a good discharge rate and to avoid an additional humidifier [6].

The main difference between an open and closed thermochemical storage system is whether it is open to the surroundings or not. Since the closed system is not, there are a few restrictions regarding working fluid. Solar energy is however the prime source of energy for both systems. The heat stored in this system can also be utilized to cover both space heating and domestic hot water demands [17]. For charging, solar heat is applied indirectly to a storage tank filled with a sorbent. The sorbent starts to release its water vapor content in exchange for the heat applied. The vapor can then be led to a condenser where it is cooled and condensed for storage. The dehydrated adsorbent and condensed water vapor is stored separate until there is an energy demand. To discharge, the water vapor is evaporated at a low temperature and led back to the storage tank where the adsorbent releases its heat in favor of the vapor. The heat can then be exchanged with a transport medium to meet the chosen demand [9, 12]. The challenge of the closed energy storage system is its need to transport and store the vapor separately from the adsorbent.

#### **4. Seasonal sorption energy storage**

A theoretical system with thermochemical energy storage including sorption has been dimensioned. The system is an open system working with silica gel and water. It should provide energy for space heating of a

domestic house of 100 m<sup>2</sup> located in Beijing, China.

The main components of the system are:

- Vacuum flat plate solar collector
- Glycol/water to air heat exchanger
- Humidifier
- Dehumidifier
- Sorption storage tank
- Fan
- Dampers

In order to dimension the sorption storage it is necessary to dimension the solar collector system to calculate how much energy is available for storage. For this assignment, a vacuum flat plate solar collector was chosen. As the solar energy potential is individual for each location and does not affect the sorption storage directly,

it is not included in the mathematical model. The dimensions for both Beijing and Trondheim are therefore summarized in Table 7.

## 5. The mathematical model

In order to build the mathematical model of the sorption energy storage the software MATLAB was utilized. MATLAB is a platform which is optimized for solving engineering and scientific problems [13], making it suitable for the task of optimizing a mathematical model. Since this model is to be tested for several locations, it is especially appropriate to work with MATLAB. In that way the parameters can easily be changed, and a simulation will take shorter time than a practical calculation would.

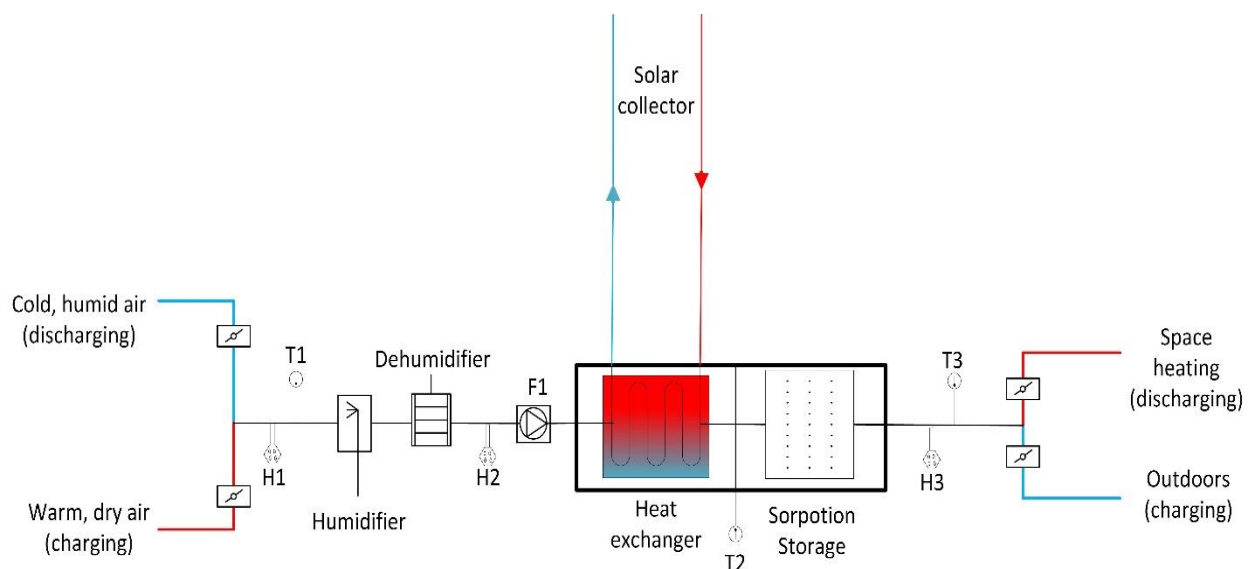


Figure 13: Sketch of the system consisting of a solar collector, heat exchanger and sorption storage.

Table 7 : Summarized values for solar collectors located in both Beijing and Trondheim.

	Beijing	Trondheim
Average annual temperature, $\theta_{ym}$ [°C]	11.7	5.4
Optimal angle, $\Psi_{opt}$ [°]	35.26	41.15
Energy demand, $Q_{SH}$ [kWh/m <sup>2</sup> ]	438	45.98
Solar fraction, SF [-]	0.5	0.5
Yearly solar irradiance, $H_{opt, year}$ [kWh/m <sup>2</sup> ]	1744.96	1020.32
Efficiency of solar collector, $\eta_{coll}$ [-]	0.5	0.5
Energy output, $Q_{out}$ [kWh/m <sup>2</sup> ]	872.48	510.16
Solar collector area, $A_{coll}$ [m <sup>2</sup> ]	25.1	4.5
Actual energy output, $Q_{out, total}$ [kWh]	21,900	2299.13
Actual energy output, $Q_{out, total}$ [kW]	2.5	0.262

## 6. Summary of the Beijing case

For this system of seasonal sorption storage the water content,  $W$ , relative humidity in,  $RH_{in}$ , and efficiency,  $\eta_{DH}$ , were decided based on test simulations and literature research. So were the temperature set points. The remaining parameters were calculated through a set of equations, mostly based on the equations of Dai, Wang and Zhang [35]. A full summary of set points and parameters can be found in Table 8.

Both the relative humidity and efficiency of dehumidification are realistic considering the available values from Zhang et al. [29]. The relevance of choosing the correct relative humidity because of its impact on the outlet temperature as stated by Zhang et al. [29] is also quite apparent

through the results. The necessary inlet temperature is a little high compared to studies like Deshmukh et al. [26] and Fernandes et al. [27], but not to an extent where it is impossible. The water content is however critically low compared to the average values and other studies. It is possible to consider the results of Jaehnig et al. [25], where the experimental results showed a water content between 2-12%. But as this is a theoretical model, the silica gel should have a potential water content up to 38% according to that same study. The low water content also results in a massive system regarding mass, increasing the expenses of the system in total. Nevertheless, the theoretical model has viable results for its calculations and a good output temperature. It is therefore considered a working model.

Table 8: A summary of the set points and parameters for the seasonal sorption energy storage.

<b>Charging set point</b>			
$T_{in}$ [K / °C]	353 / 80	$Y_{in}$ [–]	0.014
$T_{amb}$ [K / °C]	308 / 35	$\Delta Y_{max}$ [–]	0.0291
$RH_{in}$ [–]	0.5	$\Delta Y_{prac}$ [–]	0.0233
$\eta_{DH}$ [–]	0.8	$Y_{out}$ [–]	0.0373
$W$ [kg/kg]	0.1	$C_{p,a}$ [kJ/kgK]	1.006
$\eta_{HEX}$ [–]	0.9	$C_{p,w}$ [kJ/kgK]	1.84
$Q_{ads}$ [W/kg water]	0.0891	$h_g$ [kJ/kg]	2643
$RH_{eq}$ [–]	0.1393	$h_a$ [kJ/kg]	355.1
$P_{s,amb}$ [Pa]	5555.8	$h_w$ [kJ/kg]	3292.5
$P_{s,in}$ [Pa]	47086.1	$h_{in}$ [kJ/kg]	401.1
$Y_{eq}$ [–]	0.0431	$T_{out}$ [K / °C]	281.61 / 8.61
<b>Discharging set point</b>			
$T_{in}$ [K / °C]	293 / 20	$\Delta Y_{max}$ [–]	0.0093
$T_{amb}$ [K / °C]	278 / 5	$\Delta Y_{prac}$ [–]	0.0074
$RH_{in}$ [–]	0.7	$Y_{out}$ [–]	0.0026
$\eta_{DH}$ [–]	0.8	$C_{p,a}$ [kJ/kgK]	1.006
$W$ [kg/kg]	0.05	$C_{p,w}$ [kJ/kgK]	1.84
$RH_{eq}$ [–]	0.051	$h_g$ [kJ/kg]	2565.3
$\eta_{DH}$ [–]	0.8	$h_a$ [kJ/kg]	294.8
$P_{s,in}$ [Pa]	2291.8	$h_w$ [kJ/kg]	3182.8
$Y_{eq}$ [–]	0.00072	$h_{in}$ [kJ/kg]	326.6
$Y_{in}$ [–]	0.01	$T_{out}$ [K / °C]	315.8 / 43
<b>Individual parameters for Beijing</b>			
$Q_{HEX}$ [kW]	2.25		
$\dot{m}_{air}$ [kg/s]	0.0496		
$m_w$ [kg]	28056.9		
$m_{silica}$ [kg]	561138.7		

## 7. Summary for Trondheim case

As expected, the Norwegian system presented a lower mass demand than the Chinese due to a lower energy demand. This will decrease the investment cost of the system. However, some concerns should be taken regarding the mixing ratio of outdoor and indoor air during discharging. Norway has restricted laws regarding reused air in ventilation. The mixing ratio should therefore be investigated prior to further development. A solution could be to install a heat source that preheats the fresh air prior to the sorption storage, omitting the need for indoor air completely. The same heat source could be responsible for preheating the summer air if the temperature set point can not be reached naturally. It is suggested that this heat source is renewable to keep with the aim of the project being environment friendly.

*Table 9: Individual parameters for the Trondheim-case.*

### Individual parameters for Trondheim

$Q_{HEX}$ [kW]	0.2358
$\dot{m}_{air}$ [kg/s]	0.0052
$m_{water}$ [kg]	2940.37
$m_{silica}$ [kg]	58807.35

## 8. Conclusion

A complete seasonal sorption energy storage with solar collector and heat exchanger is presented. The model has been based on a 100 m<sup>2</sup> domestic household located in Beijing, China. To charge the sorption storage, dry air heated by a solar heat is sent through a desiccant bed filled with silica gel. The silica gel then adsorbs the heat and releases its water content to the air, which is discarded to the surroundings. The dehydrated silica gel is then stored alone until a heat demand is present. To discharge, humid air at medium temperature is sent through the sorption storage releasing its water content, while the silica releases its heat for the hydration. The heated air is then further utilized in space heating. The following values proved to be compatible in order to reach sufficient results for charging:  $RH_{in}=50\%$ ,  $\eta_{DH}=80\%$ ,  $W=0.1$  kg/kg,  $T_{amb}=35^{\circ}C$  and  $T_{in}=80^{\circ}C$ . The aim of discharge is to provide air at a sufficient outlet temperature for space heating. The system proved to function with the following values for discharge:  $RH_{in}=70\%$ ,  $W=0.05$  kg/kg,  $T_{amb}=5^{\circ}C$  and  $T_{in}=20^{\circ}C$ , resulting in  $T_{out}=43^{\circ}C$ . Based on these values, the sorption storage is able to store 0.089 W/kg water of adsorption heat. The adsorption heat will increase if the water content during charging,  $W$ , is increased. So in order to minimize the

necessary volume of sorption storage, further work is required to optimize this parameter.

The importance of choosing the correct values has become apparent through the simulations as the calculated variables rely heavily on the constants. The water content of silica gel,  $W$ , is recognized as one of the most important parameter as it affects the energy storage and necessary mass of silica and water greatly. The relative humidity and inlet temperature also affect the performance of the system, and especially the outlet temperature at discharge to some extent. It should therefore be chosen wisely.

The dimensioned system has also been tested under Norwegian conditions with a system located in Trondheim. The system theoretically proved to function here as well, even though the inlet temperature during charging is unrealistic for Norwegian conditions. An additional heat source should be considered for further work on the Trondheim system. As the energy demand for a passive house in Norway proved to be much lower than the assigned Chinese energy demand, the Norwegian system required much less material which is also economical beneficial.

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