Doctoral Thesis

High-temperature thermal energy storage for concentrated solar power with air as heat transfer fluid

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HIGH-TEMPERATURE THERMAL ENERGY STORAGE FOR CONCENTRATED SOLAR POWER WITH AIR AS HEAT TRANSFER FLUID

A dissertation submitted to
ETH ZURICH

for the degree of
Doctor of Sciences

presented by
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2014
Abstract

Two types of thermal energy storage (TES) systems for concentrated solar power (CSP) and high-temperature air as heat transfer fluid are investigated. The first concept consists of a packed bed of rocks as storing material, where the air is fed through the top during charging and through the bottom during discharging. The direction of the flow creates a counter-current-like heat exchanger and helps to maintain a thermal stratification in the packed bed with the hottest region on the top. In order to overcome the practical limitations of the concept, the tank is immersed in the ground and has a truncated cone shape for exploiting the concept of lateral earth pressure at higher load bearing and for reducing the normal force on the walls during thermal expansion of the rocks by guiding them upwards. The tank walls are made of concrete with a thin layer of steel fiber reinforced ultra-high performance concrete with high mechanical stability on the inside and low density concrete to reduce the thermal losses on the outside. A 6.5 MWh<sub>th</sub> pilot-scale thermal storage unit is fabricated and experimentally demonstrated to generate thermoclines. A dynamic numerical heat transfer model is formulated for separate fluid and solid phases and variable thermo-physical properties in the range of 20-650 °C, and validated with experimental results. The effect of axial dispersion in the bed caused by conduction and radiation is investigated as well as the effect of variable material properties and variable tank diameter.

The model was applied in the scale-up design of two industrial TES systems. The first is a 100 MWh<sub>th</sub> TES unit for a 3 MW<sub>th</sub> CSP plant in Ait Baha, Morocco. Simulation results showed that for a charging temperature of 640 °C, the overall thermal losses remain below 3.5% of the input energy, the outflow temperature during the discharging remains above 550 °C, and the overall efficiency reaches 89% as the TES operation approaches steady cyclic behavior. The second is an array of two industrial-
scale thermal storage units, each of 7.2 GWh\textsubscript{th} capacity, for a 26 MW\textsubscript{el} round-the-clock concentrated solar power plant during multiple 8hr-charging/16hr-discharging cycles, yielding 96% overall efficiency and outflow temperatures above 590 °C during the whole discharging period, for a charging temperature of 650 °C.

Further, studies are carried out that can serve as a guideline for designing TES units. The effect of initial charging of the TES before commissioning of the power plant, as well as storage parameters such as the tank diameter to height ratio for cylindrical tanks, the cone angle for conical tanks, the particle diameter and the insulation thickness on the TES performance were investigated.

The second concept is a novel approach proposed for stabilizing the outflow temperature of a packed bed of rocks during discharging. Its concept is based on the combination of sensible and latent heat by adding a relatively small amount of phase change material (PCM) to the top of the bed. Transient simulations solving the energy conservation equations show that the outflow temperature during discharging can be kept constant around the PCM melting temperature, eliminating the inherent temperature drop of TES based on sensible heat only. A PCM volume of only 1.33% of the total storage volume is sufficient to accomplish stabilization, corresponding to 4.4% of the total thermal energy stored as latent heat.

Finally, an experimental campaign was carried out with a 42.4 kWh\textsubscript{th} lab-scale TES unit to investigate the concept of combined sensible and latent heat thermal energy storage. The combined rocks and PCM storage was compared to the rocks only storage and showed to stabilize the outflow temperature for about 90 minutes, 20 minutes of which it was higher than the rocks only outflow temperature. The experimental results were used to validate the numerical model.
Zusammenfassung

Tank, wie Wärmespeicher dieser Art typischerweise gebaut werden, verglichen.

Das validierte Modell wurde eingesetzt um zwei industrielle thermische Wärmespeicher zu dimensionieren. Der erste Speicher, mit einer Kapazität von 100 MWh\textsubscript{th}, wird für eine 3 MW\textsubscript{th} Solaranlage in Ait Baha, Marokko gebaut. Laut Modell betragen die Wärmeverluste für eine Ladetemperatur von 640 °C weniger als 3.5% der Eingangsenergie, die Strömungsausgangstemperatur während dem Entladen bleibt über 550 °C und die Gesamteffizienz des Speichers beträgt 89%. Das zweite dimensionierte Speichersystem besteht aus zwei Einheiten mit einer Kapazität von je 7.2 GWh\textsubscript{th} für eine 26 MW\textsubscript{el} Solaranlage zur kontinuierlichen 24 h Stromproduktion mit 8-stündiger Lade- und 16-stündiger Entladephasen. Laut Modellresultaten liegt die Gesamteffizienz mit einer Ladetemperatur von 650 °C bei 96% und die Ausgangstemperatur bleibt über 590 °C während der gesamten Entladezeit.


Das zweite untersuchte Konzept ist ein neuartiger Ansatz zur Stabilisierung der Ausgangstemperatur während der Entladephase. Es basiert auf der Kombinierung von fühlbarer und latenter Wärme durch das Hinzufügen einer relativ dünnen Schicht an Phasenwechselmaterial (PCM für Phase Change Material) am oberen Ende des Steinbetts. Laut Simulationsresultaten kann die Ausgangstemperatur während der Entladephase um die Schmelztemperatur des PCM stabilisiert und somit Temperaturabfälle, die für Steinbettwärmespeicher typisch sind, eliminiert werden. Ein PCM-Volumen von nur 1.33% des totalen Speichervolumens ist ausreichend, um die Stabilisierung zu erhalten, was einem Anteil von 4.4% latenter Wärme an der gesamten gespeicherten Energie entspricht.
Letztendlich wurde eine experimentelle Kampagne mit einem 42.4 kWh Speicher ausgeführt, um das Konzept des kombinierten Speichers zu untersuchen. Die Ausgangstemperatur des kombinierten Speichers wurde mit der des Steinbettspeichers ohne Phasenwechselmaterial verglichen und zeigte eine Stabilisierung für ca. 90 Minuten, wobei für die letzten 20 Minuten die Ausgangstemperatur höher war als die des Steinbettspeichers ohne Phasenwechselmaterial. Die experimentellen Resultate wurden genutzt um das numerische Modell zu validieren.
Nella presente tesi vengono esaminati due tipi di accumulatori di energia termica per applicazioni di solare ad alta temperatura utilizzanti aria come fluido termico. Il primo concetto consiste in un letto impaccato di sassi nel quale questi fungono da materiale di accumulo termico. Durante la fase di carica, l’aria calda viene fatta fluire attraverso il letto dall’alto verso il basso, mentre durante la scarica la direzione di flusso è invertita. Questa particolarità nella direzione del flusso crea un meccanismo simile ad uno scambiatore di calore controcorrente e aiuta a conservare una stratificazione termica nell’accumulatore dove la regione più calda si trova in alto. Per superare le limitazioni pratiche del concetto, il serbatoio è innanzitutto interrato, in modo tale da sfruttare la pressione passiva della terra per stabilizzare le pareti. Inoltre, il serbatoio è realizzato con la geometria di un tronco di cono al fine ridurre le forze che i sassi esercitano sulle pareti in conseguenza della loro espansione termica. Infatti, mediante tale configurazione, la controforza, avendo una componente verticale, guida i sassi verso l’alto. Le pareti sono costruite nella parte più interna da uno sottile strato di calcestruzzo ad alta stabilità meccanica rinforzato con fibre di acciaio, seguito da un secondo strato di calcestruzzo poroso a bassa densità e conduttività termica che funge da isolante. Un prototipo con una capacità di 6.5 MWh termici è stato costruito e testato. Un modello numerico del trasferimento di calore è stato sviluppato con le fasi fluido e solido separate e tenendo conto delle proprietà termo-fisiche variabili. Tale modello è stato in seguito validato con i risultati sperimentali. Gli effetti della dispersione assiale nel letto impaccato a causa di conduzione e irraggiamento, nonché delle proprietà variabili e della forma conica sono stati studiati.

Il modello è stato quindi usato per dimensionare due impianti di accumulo di scala industriale. Il primo è un accumulatore di 100 MWh
termici per un impianto solare con una potenza di 3 MW termici situato ad Ait Baha, Marocco. Per una temperatura di carica di 640 °C, le simulazioni indicano perdite termiche di meno del 3.5% dell’energia di carica, un’efficienza di 89% e una temperatura di uscita durante la fase di scarica che rimane superiore a 550 °C. Il secondo impianto consiste invece in due accumulatori da 7.2 GWh termici ciascuno, per un impianto solare da 26 MW_el capace di produrre elettricità ininterrottamente, con 8 ore di carico e 16 ore di scarico. Per una temperatura di carica di 650 °C, le simulazioni indicano un’efficienza del 96% ed una temperatura di uscita durante la fase di scarica che rimane al di sopra di 590 °C.

Sono inoltre stati effettuati studi destinati a guidare la progettazione di accumulatori di energia termica di questo tipo. L’effetto di diversi parametri, come il rapporto fra diametro ed altezza del serbatoio (nel caso di serbatoio cilindrico), l’angolo del cono (nel caso di serbatoio conico), il diametro dei sassi e lo spessore dell’isolamento sul rendimento dell’accumulatore è stato investigato.

Il secondo concetto è un nuovo approccio per stabilizzare la temperatura di uscita dell’accumulatore durante la fase di scarica. Esso è basato sulla combinazione di calore sensibile e latente tramite l’aggiunta di uno strato relativamente sottile di materiale a cambiamento di fase (PCM, per Phase Change Material) a di sopra del letto impaccato di sassi. Le simulazioni in regime transiente indicano che è possibile stabilizzare la temperatura di uscita durante la fase di scarica attorno al punto di fusione del PCM ed evitare in tal modo le cadute di temperatura intrinseche ai sistemi di accumulo di calore basati solo sul calore sensibile. Un volume di PCM pari a solo 1.33% del volume totale dell’accumulatore è sufficiente per raggiungere la stabilizzazione desiderata, corrispondente al 4.4% dell’energia immagazzinata sotto forma di calore latente.

È stata infine condotta una campagna sperimentale con un accumulatore da 42.4 kWh termici per studiare la fattibilità del concetto di accumulatore basato sul concetto ibrido sopra introdotto. I risultati sperimentali con il PCM aggiunto hanno dimostrato una stabilizzazione della temperatura di
uscita per 90 minuti, e per 20 minuti la temperatura è stata superiore al caso senza il PCM. Un modello numerico è stato validato con i risultati della campagna.
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# Nomenclature

**Latin characters**

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<tr>
<th>Symbol</th>
<th>Definition</th>
<th>Unit</th>
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<tr>
<td>$A$</td>
<td>Surface area</td>
<td>$[m^2]$</td>
</tr>
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<td>$a$</td>
<td>Surface area per unit volume</td>
<td>$[m^2/m^3]$</td>
</tr>
<tr>
<td>$C$</td>
<td>Heat capacity</td>
<td>$[J/kgK]$</td>
</tr>
<tr>
<td>$C_p$</td>
<td>Heat capacity at constant pressure</td>
<td>$[J/kgK]$</td>
</tr>
<tr>
<td>$D$</td>
<td>Diameter</td>
<td>$[m]$</td>
</tr>
<tr>
<td>$d$</td>
<td>Particle or tube diameter</td>
<td>$[m]$</td>
</tr>
<tr>
<td>$E$</td>
<td>Energy</td>
<td>$[J]$</td>
</tr>
<tr>
<td>$e$</td>
<td>Internal energy</td>
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<td>$f$</td>
<td>Fraction</td>
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<td>$f_{cs}$</td>
<td>Compressive stress limit of concrete</td>
<td>$[N/m^2]$</td>
</tr>
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<td>$f_s$</td>
<td>Safety factor</td>
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<td>$G$</td>
<td>Mass flow rate per unit cross section</td>
<td>$[kg/m^2s]$</td>
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<td>$g$</td>
<td>Gravitational constant, 9.81</td>
<td>$[m/s^2]$</td>
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<td>$H$</td>
<td>Total specific enthalpy of fluid</td>
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<tr>
<td>$h$</td>
<td>Specific enthalpy of fluid</td>
<td>$[J/kg]$</td>
</tr>
<tr>
<td>$h_{fus}$</td>
<td>Heat of fusion</td>
<td>$[J/kg]$</td>
</tr>
<tr>
<td>$h_p$</td>
<td>Particle convective heat transfer coefficient</td>
<td>$[W/m^2K]$</td>
</tr>
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<td>$h_{rs}$</td>
<td>Solid to solid radiative heat transfer coefficient</td>
<td>$[W/m^2K]$</td>
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<td>$h_{rv}$</td>
<td>Void to void radiative heat transfer coefficient</td>
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<td>$h_v$</td>
<td>Volumetric convective heat transfer coefficient</td>
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<td>$k$</td>
<td>Thermal conductivity</td>
<td>$[W/mK]$</td>
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<td>$k_{eff}$</td>
<td>Stagnant bed effective thermal conductivity</td>
<td>$[W/mK]$</td>
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<td>Symbol</td>
<td>Definition</td>
<td>Unit</td>
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<tr>
<td>$k_{20}$</td>
<td>Thermal conductivity at ambient temperature</td>
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<td>$M$</td>
<td>Molecular weight</td>
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<tr>
<td>$m$</td>
<td>Mass</td>
<td>[kg]</td>
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<td></td>
<td>Bending moment per unit length</td>
<td>[Nm/m]</td>
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<td>$\dot{m}$</td>
<td>Mass flow rate</td>
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<td>$p$</td>
<td>Pressure</td>
<td>[Pa]</td>
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<tr>
<td>$Q$</td>
<td>Heat flux</td>
<td>[W]</td>
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<tr>
<td>$q$</td>
<td>Heat flux per surface area Load</td>
<td>[W/m²]</td>
</tr>
<tr>
<td>$R$</td>
<td>Specific gas constant</td>
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<tr>
<td></td>
<td>Thermal resistance</td>
<td>[K/W]</td>
</tr>
<tr>
<td>$r$</td>
<td>Radius</td>
<td>[m]</td>
</tr>
<tr>
<td>$S$</td>
<td>Section modulus per unit length</td>
<td>[m³/m]</td>
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<tr>
<td>$S_l$</td>
<td>Longitudinal pitch</td>
<td>[m]</td>
</tr>
<tr>
<td>$S_t$</td>
<td>Transverse pitch</td>
<td>[m]</td>
</tr>
<tr>
<td>$T$</td>
<td>Temperature</td>
<td>[K] or [°C]</td>
</tr>
<tr>
<td>$t$</td>
<td>Time</td>
<td>[s]</td>
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<td></td>
<td>Thickness</td>
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<tr>
<td>$u$</td>
<td>Interstitial velocity</td>
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<tr>
<td>$V$</td>
<td>Volume</td>
<td>[m³]</td>
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<td>$W$</td>
<td>Function of $\gamma$ in Pfeffer’s correlation</td>
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<td></td>
<td>Weight</td>
<td>[N]</td>
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<td>$x$</td>
<td>Axial coordinate</td>
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**Greek characters**

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<tr>
<td>$\alpha$</td>
<td>Thermal diffusivity</td>
<td>[m²/s]</td>
</tr>
<tr>
<td>$\beta$</td>
<td>Ratio of the average length between the centers of two neighboring solids to the mean particle diameter</td>
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</tr>
<tr>
<td>$\chi$</td>
<td>PCM quality (molten fraction)</td>
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<td>Unit</td>
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<td>$\varepsilon$</td>
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<td>$\epsilon$</td>
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<td>Efficiency</td>
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<td>$\kappa$</td>
<td>Ratio of solid to fluid thermal conductivity</td>
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<td>$\mu$</td>
<td>Dynamic viscosity</td>
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<td>$\nu$</td>
<td>Poisson ratio</td>
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<td>$\rho$</td>
<td>Density</td>
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<td>Stefan-Boltzmann constant, $5.6704 \times 10^{-8}$</td>
<td>[W/m$^2$K$^4$]</td>
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<td>$\phi$</td>
<td>Relative humidity</td>
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<td>$\psi$</td>
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<td>$\omega$</td>
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**Dimensionless groups**

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### Nomenclature

#### Subscripts

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<td>0</td>
<td>Conditions of the free flow</td>
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<td>∞</td>
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<td>a</td>
<td>Air</td>
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<td>Charging</td>
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<td>PCM</td>
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<td>th</td>
<td>Thermal</td>
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<tr>
<td>v</td>
<td>Volumetric</td>
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</tbody>
</table>
\( w \) Water

**Superscripts**

0 \( \) Stagnant flow
\( t \) Time level

**Abbreviations**

- **AA-CAES**: Advanced adiabatic compressed air energy storage
- **AISI**: American Iron and Steel Institute
- **CFD**: Computational fluid dynamics
- **CFL**: Courant-Friedrichs-Lewy number
- **CSP**: Concentrated solar power
- **DSC**: Differential scanning calorimeter
- **DNI**: Direct normal insolation
- **EMPA**: Swiss Federal Laboratories for Materials Science and Technology
- **ETH**: Swiss Federal Institute of Technology
- **GP**: Grid points
- **HTF**: Heat transfer fluid
- **LD**: Low density concrete
- **PCM**: Phase change material
- **RMSD**: Root mean square deviation
- **r.h.s.**: Right hand side
- **SEGS**: Solar energy generating systems
- **TES**: Thermal energy storage
- **UHPC**: Ultra-high performance concrete
1 Introduction

1.1 Solar Power and Thermal Energy Storage

Solar energy is by far the most abundant source of energy on earth. The sum of the total reserves of non-renewable sources of energy from nuclear (~3000 TW-years, uranium only [1]), oil (~320 TW-years), natural gas (~260 TW-years) and coal (~520 TW-years, Anthracite and Lignite) [2] is only about 23% of the incoming solar radiation on earth, per year\(^1\) (~17’720 TW-years per year). The sum of the total potential power of renewable sources of energy by wind (~72 TW [4]), hydro (~2 TW), geothermal (~160 TW direct use and ~1.5 TW through electricity production), biomass (~8 TW), ocean thermal energy conversion (~10 TW), wave (~2 TW) and tides (~0.1 TW) [5] is only about 1.4% of the incoming solar power. However, solar energy is diluted, scattered and intermittent. Its potential is therefore best exploited in areas with high and uninterrupted insolation.

Concentrated solar power (CSP) plants are employed since decades in such areas to harvest the energy from the sun. Their concept is based on concentrating the sunrays to heat up a fluid which is in turn used for electricity production in a power block. CSP plants offer an economically and technically feasible opportunity to incorporate storage solutions in the form of thermal energy storage (TES) to compensate the intermittent nature of solar energy by decoupling production from consumption. Due to the temporal shift between peak insolation hours and the peak electricity consumption period, storage systems have always been an integral part of CSP plants.

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\(^1\) Solar energy received by land only, after passing through atmosphere, assuming 50% losses by clouds. Raw data from [3].
Fig. 1.1 summarizes existing concepts for high-temperature thermal energy storage for concentrated solar power plants. Thermochemical storage is an emerging concept and is still in the early research and development phase [6] and latent heat storage, due to the high cost of phase change material (PCM) and technical difficulties, has not found yet widespread commercial use for industrial scale applications. Storage concepts based on sensible heat, on the other hand, are the most developed for use in CSP plants.

Several publications review existing thermal energy storage concepts for CSP plants, including a series of two papers by Gil et al. [7] and Medrano et al. [8] reviewing concepts, materials, modelling and case-studies of high-temperature TES for power generation; a review of TES for parabolic trough power plants by Herrmann and Kearney [9]; and a review of TES for CSP plants focusing on storage system design and the integration into the power plant by Kuravi et al. [10]. Commercialized CSP plants use mainly molten salts, oil, or steam as heat transfer fluid (HTF) and have incorporated various storage systems. Solar One (1982-1988, USA) the first large-scale solar tower plant, used a direct steam receiver and a one-tank thermocline TES with oil as HTF and rocks and sand as filler material [11]. In 1986, the storage was damaged by a fire and not repaired thereafter. The solar tower plants Solar Two (1995-1999, USA) and Solar Tres (2011-today, Spain) use molten salt as HTF as well as storage medium using two separate cold and hot tanks [8, 12]. The Andasol (2008-today, Spain) and Extresol (2010-today, Spain) parabolic trough power plants use thermal oil as HTF of the solar field and a similar molten salt cold and hot tank TES concept as the Solar Two and Solar Tres [13]. This is nowadays the most widely employed TES concept and is generally denoted the two-tank system. Solar Energy Generating Systems (SEGS) is a series of nine parabolic trough power plants (constructed between 1985 and 1991, USA) using thermal oil as heat transfer fluid. The first plant, SEGS I, used a two-tank storage system with oil. However, it was damaged by fire in 1999 and not replaced afterwards. Subsequently built SEGS plants do not have a TES incorporated and use natural gas as backup [9, 14, 15].
Fig. 1.1: High-temperature TES concepts for CSP applications.

In the Small Solar Power Systems Project (SSPS, constructed and tested in the 1980’s in Spain) two storage systems were investigated: a one-tank thermocline with oil as the storage medium and a one-tank thermocline with cast iron as filler material and oil as heat transfer fluid [16]. Other storage concepts experimentally investigated in CSP plants include steam storage [17, 18], and concrete solid media storage [19]. Although the two-tank molten salt TES is the most widely applied concept, it has the disadvantage of high costs as well as the problem of freezing of the salt ($T_{\text{melt}} \sim 140 \, ^\circ\text{C}$), making it necessary to incorporate a fossil fuel-fired backup system to avoid damages to the piping in the case of long absence of
solar insolation. Thermal oils, on the other hand, are flammable, hazardous in case of leakages, limited in their temperature range \((T_{\text{max}} \sim 400 \, ^{\circ}\text{C})\) and expensive as well. In power plants where the heat transfer fluid is not the same as the storage medium, such as the Andasol and Extresol projects, a heat exchanger is necessary between the solar field and the TES, which entails costs and energy penalties.

Air, on the other hand, is not extensively applied as HTF in CSP plants due to its relatively lower volumetric heat capacity and thermal conductivity, which in turn requires larger heat transfer areas and volumetric flow rates. Nevertheless, air has several intriguing advantages, such as: 1) no costs; 2) no relevant operating temperature constraints; 3) no HTF degradation; 4) possibility of direct heat exchange with storage material and hence eliminating complications and penalties associated with heat exchangers; and 5) non-corrosiveness and non-toxicity. Examples of CSP designs that use air as HTF include pressurized receivers for solar tower systems [20-24] and non-pressurized receivers for solar trough systems [25-27]. These concepts, however, are mostly in the development phase and are not yet commercialized. Consequently, accompanying TES solutions are also not commercially used yet. Storage concepts experimentally investigated for use with high-temperature air include a lab-scale packed bed of iron balls TES [28], a lab-scale packed bed of rocks thermocline [29], a pilot-scale regenerative thermal oxidizer-type thermocline with alumina porcelain ceramics as storage material [30], a lab-scale packed bed of \(\text{ZrO}_2\) pellets thermocline [31, 32] and a direct sand heat exchanger where the sand serves as heat transfer as well as storage medium [33]. Particularly, rocks from gravel have low cost and are locally available. Its main advantages are: 1) abundant and economical storing material; 2) applicability in a wide temperature range, with limiting temperatures given by the rock’s melting point; 3) direct heat transfer between working fluid and storage material; 4) no degradation or chemical instability; 5) no safety concerns; and 6) elimination of chemicals and corrosive materials. However, the experimental work on packed bed of rocks so far has been limited to small scales and mostly low temperatures \((T_{\text{max}} = 30-200 \, ^{\circ}\text{C})\) [34-
A review of the work on packed bed TES can be found in [47] and [10].

In this thesis, in contrast to the previous works, a pilot-scale packed bed of rocks thermocline thermal energy storage for use with high-temperature air \(T_{\text{max}} = 650 \, ^{\circ}\text{C}\) as heat transfer fluid is experimentally and numerically investigated. Its design is aimed at large-scale applications and addresses typical issues associated with this type of TES systems such as storage tank deformation and rock fracturing. The goal is to acquire insight and knowledge that can be used for commercialization and scale-up design of TES units of this type. The concept is based on exploiting the sensible heat of rocks.

Further, a main disadvantage of thermocline TES systems – namely outlet temperature variations during discharging – is addressed and a concept suggested and experimentally investigated to overcome this drawback. The proposed design places a relatively small layer of PCM on top of the packed bed to stabilize the outflow air temperature around the melting temperature of the PCM. Thus, the concept exploits the benefits of latent and sensible heat storage while alleviating their drawbacks when applied separately.

This thesis was carried out in the framework of a joint ETH-Airlight Energy SA project entitled “Thermische Speichersysteme”. The storage is developed to be coupled with the Airlight Energy SA trough collector technology that uses a cavity receiver with air as heat transfer fluid reaching temperatures of up to 650 °C [26, 48, 49]. The aim of the project was to obtain commercially exploitable knowledge for the design, construction and production of thermal energy storage systems in the framework of a dissertation at the Institute of Energy Technology at ETH Zurich.

1.2 Thesis Outline

Chapter 2 investigates a thermal energy storage concept based on exploiting the sensible heat of rocks. A numerical model was developed with variable fluid and solid properties considering thermal losses and the
specific geometry of the concept. A 6.5 MWh\textsubscript{th} prototype was constructed at the premises of Airlight Energy SA in Biasca, Switzerland and used to validate the model. The validated model was then used in the scale-up design of two TES systems of 100 MWh\textsubscript{th} and 7.2 GWh\textsubscript{th}, as well as to conduct parametric studies that can help for the design and optimization of future commercial TES units. Chapter 3 introduces a novel thermal energy storage concept based on the combination of sensible and latent heat to stabilize the TES outflow temperature during discharging. Numerical studies were conducted to show the feasibility of the concept. A 42.4 kWh\textsubscript{th} combined storage was then constructed and tested at the premises of Airlight Energy SA in Biasca, Switzerland and the results used to validate the numerical model. Chapter 4 summarizes the work presented in the thesis. Finally, in chapter 5, general suggestions and recommendations for future work are given.
2 Sensible Heat Energy Storage

2.1 Introduction

Thermal energy storage (TES) systems that can be used for concentrated solar power (CSP) plants include molten salt [9, 50], oil [11], steam [18], and concrete [19, 51] for sensible heat, phase change materials for latent heat [52], and reversible reactions for thermochemical storage [6, 53]. Of special interest is the use of a packed bed of rocks as sensible heat storing material and air as heat transfer fluid [3, 29]. Such a storage concept can be incorporated in solar power plants using air as working fluid [25, 26]. The characteristics of packed beds for thermal storage have been applied for food processing [54], catalytic reactions [55], waste heat recovery [32], and low-temperature seasonal [56] as well as high-temperature hourly solar energy storage [29]. Recently, rocks have also been suggested as filler material for one-tank thermocline storage systems using oil or molten salt as HTF in order to increase the energy density and help to create and maintain temperature stratification [46, 57, 58].

Pertinent modeling studies [34] are mostly based on the original analytic work by Schumann [59] and include 1D and 2D heat transfer models with

---


two separate phases [60], and several models developed for the temperature range 30 - 200 °C using air, water, or oil as working fluid [42, 61-63].

Fewer studies have dealt with the high-temperature range, including experimental tests and modeling of a lab-scale packed bed of rocks up to 550 °C [29, 64] and of ZrO$_2$ pellets up to 1000 °C [31, 32]. Recently, a packed bed of 150x150x150 mm$^3$ Al$_2$O$_3$ bricks with 9 MWh$_{th}$ storage capacity was tested using a gas burner for charging at 650 °C and discharging at 120 °C [30] and simulated using a thermo-mechanical model [65]. Most previous simulation models considered temperature-invariant properties of fluid and solid, which introduces uncertainties for wide-range temperature operation.

A schematic depicting the thermal energy storage and its operating principle is shown in Fig. 2.1. During charging, hot air enters the TES from the top, transfers heat to the rocks, and exits at the bottom. During discharging the flow is reversed: air enters from the bottom, is heated by the rocks, and exits at the top. The direction of the flow exploits buoyancy forces to create and maintain thermal stratification, with the hottest region at the top of the storage. In contrast to previous work, the storage design is aimed at large-scale applications by overcoming typical issues such as storage tank deformation and rock fracturing by using a truncated conical shape, multi-layer concrete walls, and subterranean placement of the tank.

In this chapter, the design, fabrication, and testing of a pilot-scale packed bed of rocks and its simulation by means of a dynamic two-phase heat transfer model are described. The storage tank geometry and practical design aspects are discussed for operating conditions relevant to large-scale industrial implementation and benefits of the truncated conical shape are elaborated. Temperature-dependent thermo-physical properties of the solid, as well as the axial void fraction distribution in the packed bed are experimentally measured, which have been neglected in most previous works. As it will be shown in the analysis that follows, the model incorporates variable thermo-physical properties for both fluid and solid phases and, hence, enables good matching between experimental and
numerical results. Further, in contrast to previous models, the interaction of the storage with its surrounding is accounted for by not only thermal losses through the lateral walls, but also through the lid as well as the bottom, which is significant due to the high temperatures involved. The validated model is further applied to optimize and scale-up the design of two thermal storage systems for industrial-size CSP plants and identify the major sources of irreversibility. Finally, studies are conducted that can help in the understanding and design of packed bed of rocks thermal energy storage systems.

2.2 Modeling

The model is based on the one-dimensional equation for the total enthalpy [66]. For the fluid phase this is

$$\frac{\partial (\rho H)}{\partial t} + \frac{\partial (\rho Hu)}{\partial x} = \frac{\partial p}{\partial t} + \frac{\partial (u\tau)}{\partial x} - \frac{\partial q}{\partial x} + \rho ug$$

(2.1)

The solid phase energy equation is obtained from Eq. (2.1) by setting $u=0$, with the assumption of incompressible solid phase
The above equations can be simplified for a quasi-one-dimensional plug-flow, following the methodology by Haselbacher [67] that consists of non-dimensionlization and scale-analysis in order to omit non-essential terms (here the viscous, gravity and pressure gradient terms) and integration over a control volume to average the simplified equations. A cell-centered finite-volume approach is used where the fluxes at the cell faces are approximated by a first-order approximation. The fluid phase energy equation becomes

\[
\rho_i \frac{\partial e_s}{\partial t} = -\frac{\partial q_s}{\partial x} \quad (2.2)
\]

The solid phase energy conservation equation becomes

\[
(1 - \varepsilon_i) \rho_s \frac{de_{s,i}}{dt} = h_{v,i} (T_{f,i} - T_{s,i}) + \frac{1}{V_i} \left[ k_{eff} \frac{dT_i}{dx} A_i - k_{eff} \frac{dT_{s,i}}{dx} A_{i-1} \right] \quad (2.4)
\]

The subscript \( i \) denotes the position in the grid ranging from 1 to \( N \). The time derivative in Eq. (2.3) and (2.4) is solved using the forward Euler method and the spatial derivative in the solid phase is approximated using a first-order forward difference.
Fluid:

\[
\begin{align*}
    h^{i+1}_i &= \frac{1}{\rho_i^i} \left[ \rho_i^i h_i^i + \Delta t \left[ -\frac{1}{\varepsilon_i^i V_i^i} \left( (\rho u h \varepsilon A)^i_i - (\rho u h \varepsilon A)^{i-1}_i \right) - \frac{h_{v,i}}{\varepsilon_i^i} (T_{f,i}^i - T_{s,i}^i) - \frac{A_{s,i}}{\varepsilon_i^i V_i^i} h_{\text{wall},i} (T_{f,i}^i - T_{\infty}) \right] \right] \\
    &= \frac{1}{\rho_i^i} \left[ \rho_i^i h_i^i + \Delta t \left[ -\frac{1}{\varepsilon_i^i V_i^i} \left( (\rho u h \varepsilon A)^i_i - (\rho u h \varepsilon A)^{i-1}_i \right) - \frac{h_{v,i}}{\varepsilon_i^i} (T_{f,i}^i - T_{s,i}^i) - \frac{A_{s,i}}{\varepsilon_i^i V_i^i} h_{\text{wall},i} (T_{f,i}^i - T_{\infty}) \right] \right]
\end{align*}
\]  

(2.5)

Solid:

\[
\begin{align*}
    e^{i+1}_{s,i} &= e_{s,i}^i + \Delta t \left[ \frac{h_{v,i}}{\rho_s (1 - \varepsilon_i)} (T_{f,i}^i - T_{s,i}^i) + \frac{1}{(1 - \varepsilon_i) \rho_s V_i} \left( k_{\text{eff},i} A_i \left( T_{s,i+1}^i - T_{s,i}^i \right) - k_{\text{eff},i-1} A_{i-1} \left( T_{s,i}^i - T_{s,i-1}^i \right) \right) \right]
\end{align*}
\]

(2.6)

The specific enthalpy of the fluid phase and the internal energy of the solid phase are defined as

\[
\begin{align*}
    h &= \int_{T_{\text{ref}}}^T C_{p,f} dT \\
    e_s &= \int_{T_{\text{ref}}}^T C_s dT
\end{align*}
\]

(2.7)  

(2.8)

Eq. (2.5) and (2.6) are solved explicitly for \( h^{i+1}_i \) and \( e^{i+1}_{s,i} \). Using correlations for the temperature-dependent fluid and solid heat capacity, \( C_{p,f} \) and \( C_s \) - that will be introduced in section 2.3 - the temperature is calculated at every time step and position by using a Newton-Raphson subroutine [68]. The air density is calculated using the equation of state (\( \rho = p / RT \)) with the pressure taken as the atmospheric pressure. Due to the small pressure variations in this application, this simplification has negligible effect on the results [69]. The fluid density in Eq. (2.5) is lagging in time, which means that it is taken out of the time derivative and the value at the previous time step is used. This simplification is justified due to the small contribution of the fluid enthalpy change with time to Eq. (2.3) [69]. The interstitial fluid
velocity, \( u \), is obtained from the mass flow rate with the assumption of homogeneous mass flow throughout the storage at any time step

\[
\dot{u}_i^t = \frac{\dot{m}_i^t}{\varepsilon_i A_i \rho_i^t} \tag{2.9}
\]

The overall wall heat transfer coefficient, \( h_{wall} \), considering the heat transfer from both phases to the surrounding wall, is considered in the fluid phase energy conservation equation. This is because obtaining separate wall heat transfer coefficients for the solid and fluid phase would require detailed modeling of the flow and the packed bed. Therefore, correlations from literature that have been developed for packed beds are applied. Further, an effective conductive heat transfer is considered in the solid phase that considers – in addition to the fluid and solid thermal conductivity – the radiation heat transfer in the packed bed. It is therefore more correctly called dispersion as labeled in Eq. (2.4). The following initial and boundary conditions apply: 1) the initial solid and fluid temperature distributions of the packed bed must be specified, but do not need to be uniform; 2) the inlet temperature and mass flow rate must be specified for all times, but do not need to be constant; 3) adiabatic conditions for the fluid phase at the outlet: \( h_{f,N+1} = h_{f,N} \); 4) adiabatic conditions for the solid phase at the inlet and outlet: \( T_{s,0} = T_{s,1} \) and \( T_{s,N+1} = T_{s,N} \).

For the first layer, the equations are modified to consider the conductive heat transfer through the cover. Under consideration of the boundary conditions introduced above, the equations become

\[
\begin{align*}
\dot{h}_{i}^{t+1} &= \frac{1}{\rho_i^t} \left[ \dot{h}_i^t + \Delta t \left[ -\frac{1}{\varepsilon_i V_i} \left[ (\rho u \varepsilon A)_i^t - (\rho u \varepsilon A)_{in}^t \right] - 
\frac{h_{v,1}}{\varepsilon_i} (T_{f,1} - T_{s,1}^t) - \frac{A_{c,1} h_{wall,1}}{\varepsilon_i V_i} (T_{f,1} - T_\infty) - \frac{h_{cover,cond} A_{cover}}{\varepsilon_i V_i} (T_{f,1} - T_{surface}) \right] \right]
\end{align*}
\tag{2.10}
\]
Solid:

\[ e_{s,1}^{t+1} = e_{s,1}^t + \Delta t \left[ \frac{h_{v,1}}{\rho_s (1 - \varepsilon_s)} (T_{f,1}^t - T_{s,1}^t) + \frac{1}{(1 - \varepsilon_s) \rho_s V_1} \left( k_{eff,1} A_1 \left( T_{s,2}^t - T_{s,1}^t \right) / \Delta x \right) \right] \]

(2.11)

The subscript in represents the data of the incoming air, which is defined as a boundary condition. A similar approach for the last layer yields

Fluid:

\[ h_{N}^{t+1} = \frac{1}{\rho_N} \left[ \rho_N^t h_N^t + \Delta t \left[ -\frac{1}{\varepsilon_N V_N} (\rho u e A)_N^t - (\rho u e A)_{N-1}^t \right] - \frac{h_{v,N} (T_{f,N}^t - T_{s,N}^t)}{\varepsilon_N V_N} A_{wall,N} (T_{f,N}^t - T_\infty) - \frac{h_{bottom,cond} A_{bottom}}{\varepsilon_N V_N} (T_{f,N}^t - T_{\infty,bottom}) \right] \]

(2.12)

Solid:

\[ e_{s,N}^{t+1} = e_{s,N}^t + \Delta t \left[ \frac{h_{v,N}}{\rho_s (1 - \varepsilon_N)} (T_{f,N}^t - T_{s,N}^t) - \frac{1}{(1 - \varepsilon_N) \rho_s V_N} \left( k_{eff,N-1} A_{N-1} \left( T_{s,N}^t - T_{s,N-1}^t \right) / \Delta x \right) \right] \]

(2.13)

Numerical stability is ensured by the more restrictive Courant-Friedrichs-Lewy condition [70] of the fluid and solid phases. According to the von Neumann stability analysis [71] on the simplified linear and periodic Eq. (2.5) and (2.6), with the source terms neglected, the criteria are 1) \( CFL = u \Delta t / \Delta x < 1 \) for the fluid phase and 2) \( \Delta t < \Delta x^2 (1 - \varepsilon) \rho c_s / 2k_{eff} \) for the solid phase. However, due to the presence of source terms and non-periodic boundary conditions, often unstable behavior is observed with the above criteria and it is necessary to reduce further the time step length to
ensure stability. Depending on the simulation, the CFL number needs to be multiplied by a factor of 0.3 to 0.5 for stable behavior.

In an internal report [69] the presented model has been compared to a similar model where the energy equation is based on the total internal energy instead of the total enthalpy [67]. After scale-analyses, the only difference between the two models is the consideration of internal energy of air in the time derivative of Eq. (2.3) instead of enthalpy as in the present model. Simulations were carried out with the two models for typical storage operating parameters and showed negligible difference in the results. This is despite the fact that the values for the heat capacity of air at constant pressure (used to calculate enthalpy) and at constant volume (used to calculate internal energy) differ by approximately 30%. This in turn shows that the time derivative of the fluid phase has little influence on the simulation results, and justifies having the air density lagging in time as explained above. As a consequence, the air enthalpy in the time derivation of the fluid phase is interchangeable with its internal energy. Further, the finite-volume approach has been compared to the method of finite differences. The difference between the two methods showed to be negligible.

2.3 Closure Relations

The following correlations and data are required to close the equations introduced in section 2.2. For air, temperature dependent properties as found in literature [72] are fitted to high order polynomials for facilitated use in the code.

2.3.1 Mass Flow Rate

If a velocity meter is used in the experimental setup – like the setup presented in section 2.4 – the mass flow rate is calculated using the air velocity, the pipe cross sectional area, and the density of moist air [73]:

\[ \rho_{\text{air, moist}} = \rho_{\text{air, dry}} \frac{1 - \omega}{(1 + R_w / R_a \omega)} \]

with \( R_w \) and \( R_a \) the specific gas constant of water and air respectively. The density of dry air is calculated
using the ideal gas law, \( \rho_{\text{air,dry}} = p / (R_a T) \), and the specific humidity using the partial pressure of water vapor: \( \omega = \frac{M_w p_w}{(M_a(p - p_w))} \), with \( M_w \) and \( M_a \) the molecular weight of water and air respectively, \( p \) the atmospheric pressure, and \( p_w \) the partial pressure of water vapor: \( p_w = \phi p_{\text{w,sat}} \). The relative humidity, \( \phi \), must be recorded during the experimental runs. The water vapor saturation pressure, \( p_{\text{w,sat}} \) is given by the extended Antoine equation [74]:

\[
\log_{10}\left(\frac{p_{\text{w,sat}}}{133.32}\right) = A - \frac{B}{T - C} \log_{10} T + DT + ET^2
\]

with the coefficients \( A \) to \( E \) given in Table 2.1 and \( T \) the ambient temperature in Kelvin.

<table>
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<tr>
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<th>A</th>
<th>B</th>
<th>C</th>
<th>D</th>
<th>E</th>
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</tr>
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</table>

2.3.2 Solid Properties

The rocks comprising the packed bed were excavated from the Rafzerfeld area in Zurich, Switzerland. The thermal properties of the rocks and concrete types used in the thermal energy storage were experimentally measured in two separate campaigns. In the first campaign, the temperature-dependent thermal conductivity up to 200 °C, and the density and the heat capacity at ambient temperature were measured in a geology lab in Zurich [75, 76]. The selected rocks in this campaign were identified as Helvetic Siliceous Limestone, Quartzite, Limestone, Calcareous Sandstone and Gabbro. The first four rock types are sedimentary rocks, while Gabbro is a plutonic rock. Two types of concrete were employed in the containing structures of the TES: Ultra-High Performance Concrete (UHPC) with high mechanical stability and thermal conductivity and Low Density Concrete (LD) with low thermal conductivity and lower mechanical stability [77] that were characterized as well. The five rock and two concrete types are shown in Fig. 2.2. Due to large variation of the heat capacity of rocks with temperature reported in the literature [78] and its importance on the simulation results, in a second campaign at the Swiss Federal Laboratories
for Materials Science and Technology, EMPA, the heat capacity of seven different rocks comprising the same packed bed was measured up to a temperature of 600 °C in a differential scanning calorimeter (DSC). These rocks are shown in Fig. 2.3.
Fig. 2.2: The five rock types and two concrete types characterized in the first (low-temperature) experimental campaign.
Fig. 2.3: The seven rock types characterized in the second (high-temperature) experimental campaign. The rocks in the second campaign were chosen from the same gravel as the first campaign.
Thermal Conductivity

The thermal conductivity of the solid material is measured using the hot-wire method at 4 different temperatures. The hot-wire method is a transient dynamic technique based on a linear heat source of infinite length and infinitesimal diameter and on the measurement of the temperature rise at a defined distance from the linear heat source embedded in the test material. As an electric current of fixed intensity flows through the wire, the thermal conductivity can be derived from the resulting temperature change over a known time interval [79]. The results are shown in Table 2.2, Fig. 2.4 and Fig. 2.5. The temperature dependent behavior of the thermal conductivity of rocks has been studied extensively [78, 80-83]. For most types of rocks the thermal conductivity decreases with temperature. Typically, the higher the thermal conductivity at ambient condition, $k_{20}$, the stronger the decrease with temperature. In contrast, for very small $k_{20}$, the thermal conductivity stays constant or even increases with temperature, which is the case for Gabbro and the low density concrete, whose characteristics are similar to rocks. The temperature dependent behavior of the measured conductivities agree well with literature data for sedimentary and plutonic rocks [82]. Several authors have suggested correlations for the temperature dependence of the thermal conductivity of rocks that are generally empirically derived.

Table 2.2: Thermal conductivity of the different rock and concrete types as experimentally measured, [W/mK].

<table>
<thead>
<tr>
<th>T [K]</th>
<th>Quartzite</th>
<th>Calcareous Sandstone</th>
<th>Helvetic Siliceous Limestone</th>
<th>Limestone</th>
<th>Gabbro</th>
</tr>
</thead>
<tbody>
<tr>
<td>298</td>
<td>5.39 ± 0.07</td>
<td>4.36 ± 0.15</td>
<td>3.60 ± 0.21</td>
<td>2.82 ± 0.06</td>
<td>2.05 ±0.08</td>
</tr>
<tr>
<td>348</td>
<td>4.05 ± 0.17</td>
<td>3.26 ± 0.24</td>
<td>3.32 ± 0.24</td>
<td>2.41 ± 0.07</td>
<td>1.99 ± 0.09</td>
</tr>
<tr>
<td>396</td>
<td>3.76 ± 0.13</td>
<td>3.16 ± 0.18</td>
<td>2.83 ± 0.16</td>
<td>2.20 ± 0.05</td>
<td>2.18 ± 0.06</td>
</tr>
<tr>
<td>446</td>
<td>3.37 ± 0.11</td>
<td>2.98 ± 0.09</td>
<td>2.72 ± 0.15</td>
<td>2.05 ± 0.03</td>
<td>2.05 ± 0.04</td>
</tr>
<tr>
<td></td>
<td>Ultra-High Performance Concrete</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>296</td>
<td>2.22 ±0.09</td>
<td></td>
<td></td>
<td>0.35 ± 0.06</td>
<td></td>
</tr>
<tr>
<td>353</td>
<td>2.10 ± 0.04</td>
<td></td>
<td></td>
<td>0.41 ± 0.06</td>
<td></td>
</tr>
<tr>
<td>391</td>
<td>2.07 ± 0.12</td>
<td></td>
<td></td>
<td>0.43 ± 0.05</td>
<td></td>
</tr>
<tr>
<td>449</td>
<td>1.94 ± 0.03</td>
<td></td>
<td></td>
<td>0.41 ± 0.04</td>
<td></td>
</tr>
</tbody>
</table>
correlations from experimental measurements and include coefficients that are determined from least-squares fit to measured data [78, 84, 85]. In this work, the correlations by Tikhomirov have been chosen, as they are the most elaborate and differentiate clearly between low conducting and high conducting rocks [78]

\[
k(T) = k_{20} - A(T - B)(k_{20} - C)\left[k_{20}(DT)^{E_{k_{20}}} + F\right]^{k_{20} - G} \quad k_{20} > 2 \text{ W/mK} \quad (2.14)
\]

\[
k(T) = k_{20} - A(T - B)(k_{20} - C) \quad k_{20} < 2 \text{ W/mK} \quad (2.15)
\]

with \( T \) in K. As it will be shown in section 2.4, the model introduced in section 2.2 showed to be insensitive to moderate variations in the thermal conductivity of rocks. Hence, the choice of the correlation does not have a crucial impact on the model results. Tikhomirov’s correlation is also used for the two concrete types [86, 87]. The values of the coefficients \( A \) to \( G \) are given in Table 2.3 for the different rock and concrete types. The extrapolations obtained with the correlation are also shown in Fig. 2.4 and Fig. 2.5.
Fig. 2.5: Thermal conductivity of the ultra-high performance (UHPC) and low density (LD) concrete as a function of temperature. Extrapolations obtained using Tikhomirov’s correlation. Markers indicate experimentally measured values and dashed lines extrapolations.

Table 2.3: Values for the coefficients of the correlations of Tikhomirov and Kelley.

<table>
<thead>
<tr>
<th>Tikhomirov, Eq. (2.14)</th>
<th>Quartzite</th>
<th>Calcareous Sandstone</th>
<th>Helvetic Siliceous Limestone</th>
<th>Limestone</th>
<th>UHPC</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k_{20}$ [W/mK]</td>
<td>5.39</td>
<td>4.36</td>
<td>3.6</td>
<td>2.82</td>
<td>2.22</td>
</tr>
<tr>
<td>$A$</td>
<td>1.10E-03</td>
<td>1.00E-03</td>
<td>1.00E-03</td>
<td>1.00E-03</td>
<td>9.00E-04</td>
</tr>
<tr>
<td>$B$</td>
<td>293</td>
<td>293</td>
<td>293</td>
<td>293</td>
<td>280</td>
</tr>
<tr>
<td>$C$</td>
<td>1.38</td>
<td>1.3</td>
<td>1.38</td>
<td>1.3</td>
<td>1.4</td>
</tr>
<tr>
<td>$D$</td>
<td>1.70E-03</td>
<td>1.65E-03</td>
<td>1.80E-03</td>
<td>1.70E-03</td>
<td>2.50E-03</td>
</tr>
<tr>
<td>$E$</td>
<td>0.25</td>
<td>0.28</td>
<td>0.28</td>
<td>0.5</td>
<td>0.3</td>
</tr>
<tr>
<td>$F$</td>
<td>1.28</td>
<td>1.2</td>
<td>1.4</td>
<td>1.4</td>
<td>1.15</td>
</tr>
<tr>
<td>$G$</td>
<td>0.65</td>
<td>0.6</td>
<td>0.6</td>
<td>0.5</td>
<td>0.75</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Tikhomirov, Eq. (2.15)</th>
<th>Gabbro</th>
<th>LD</th>
<th>Kelley, Eq. (2.17)</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k_{20}$ [W/mK]</td>
<td>2.05</td>
<td>0.35</td>
<td>$A$</td>
</tr>
<tr>
<td>$A$</td>
<td>2.00E-03</td>
<td>2.00E-03</td>
<td>$B$</td>
</tr>
<tr>
<td>$B$</td>
<td>293</td>
<td>250</td>
<td>$C$</td>
</tr>
<tr>
<td>$C$</td>
<td>2</td>
<td>0.6</td>
<td>$D$</td>
</tr>
</tbody>
</table>

$E$ (UHPC)
Heat Capacity

The heat capacity of the rocks was measured in two separate campaigns. In a first campaign, the heat capacity of the five different rock types introduced above were measured at low temperatures with a simple calorimeter and extrapolated to temperatures up to 650 °C using correlations in literature. Due to the importance of the temperature-dependent heat capacity of the rocks on the simulation results, a high-temperature campaign was later carried out on seven different rocks comprising the packed bed using a differential scanning calorimeter (DSC). The two campaigns are presented in the following.

Low-Temperature Campaign

In the first campaign, the heat capacity was measured using a simple calorimeter. Approximately 30 g of each specimen was ground, cooled to 0 °C, and dumped in the calorimeter containing 70 g water at ambient temperature. The heat capacity is calculated as

\[ C_s = f \frac{m_w C_w (T_w - T_{eq})}{m_s (T_{eq} - T_s)} \]  

(2.16)

where \( f \) is a calibration factor (\( f = 1.155 \)), \( T_w \) the initial water temperature and \( T_{eq} \) the final water temperature. The results are shown in Table 2.4.

Table 2.4: Heat capacity and density of the rocks as experimentally determined in the first campaign.

<table>
<thead>
<tr>
<th></th>
<th>( C_p ) [J/kgK]</th>
<th>( \rho ) [kg/m³]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Quartzite</td>
<td>623 ± 12</td>
<td>2618 ± 1.68</td>
</tr>
<tr>
<td>Calcareous Sandstone</td>
<td>652 ± 31</td>
<td>2661 ± 1.71</td>
</tr>
<tr>
<td>Helvetic Siliceous Limestone</td>
<td>669 ± 15</td>
<td>2776 ± 2.95</td>
</tr>
<tr>
<td>Limestone</td>
<td>683 ± 15</td>
<td>2697 ± 1.66</td>
</tr>
<tr>
<td>Gabbro</td>
<td>643 ± 24</td>
<td>2911 ± 2.69</td>
</tr>
<tr>
<td>Average</td>
<td>654 ± 22</td>
<td>2732 ± 2.14</td>
</tr>
<tr>
<td>UHPC</td>
<td>500 ± 53</td>
<td>2500*</td>
</tr>
<tr>
<td>LD</td>
<td>663 ± 67</td>
<td>1500*</td>
</tr>
</tbody>
</table>

* value communicated by Airlight Energy SA.
Several studies have investigated the temperature dependence of the heat capacity of rocks [88-90]. Clauser [89] gives a summary of the different correlations between heat capacity and temperature that are available in literature. For the heat capacity – as for the thermal conductivity – these correlations typically include coefficients that depend on the rock type. Here, the correlation by Kelley is used [88]

$$C_s(T) = A(B + CT + D / T^2)$$  \hspace{1cm} (2.17)

with $T$ in °C. Due to close measured values for the different rock types and similar temperature related behavior of rocks [78], the average of the five rocks types is used to extrapolate the results up to 650 °C. The values of the coefficients of Eq. (2.17) are given in Table 2.3 and the results are plotted in Fig. 2.6. This data is used in the validation process presented in section 2.4. It is important to note that, due to lack of experimental data at elevated temperatures, the accuracy of Eq. (2.17) cannot be verified. Therefore a high-temperature experimental campaign was carried out that is presented in the next section.

**High-Temperature Campaign**

In order to overcome uncertainties related to extrapolation of the measured values to elevated temperatures, in a following experimental campaign carried out at the Swiss Federal Laboratories for Materials Science and Technology, EMPA, the heat capacity of the rocks was measured up to a temperature of 600 °C in a differential scanning calorimeter (DSC). Out of the rocks comprising the packed bed of Biasca, 7 rock types that demonstrated visual differences were selected for the measurements, denoted with S1 to S7 and shown in Fig. 2.3. The repeatability of the measured temperature dependent heat capacity was investigated by conducting two consecutive heating and cooling cycles.

---

*Experimental campaign carried out in the framework of a bachelor thesis by Claudia Walser, supervised by G. Zanganeh.*
Almost every rock showed a peak in heat capacity around 575 °C, which can be ascribed to the α-β-inversion of quartz that is present in sedimentary rocks such as the ones investigated here [91]. At atmospheric pressure, the heat required to complete this inversion is 20.2 Joule per gram of quartz [78]. For most of the rocks, the two heating runs and the two cooling runs overlapped respectively, proving the repeatability of the measurements. A shift in the peak was observed between the heating and cooling runs, which is ascribed to a different onset temperature of the quartz α-β-inversion for heating and cooling as well as the fast heating/cooling ramp of the DSC device of 15 K/min. The temperature-dependent heat capacity for the heating and cooling runs, averaged for the seven different rock types and the two runs, are shown in Fig. 2.6. Also shown is the data of the low-temperature campaign extrapolated using the correlation by Kelley. Comparison of the results shows that although the general temperature dependence is predicted more or less correctly by the correlation of Kelley, it fails to predict irregularities caused by the rocks’ composing materials.
The results of the high-temperature campaign are used in the simulations done in section 3.6.

**Density**

Archimedes’ principle was used to measure the density of the rocks [92]. Cubic samples with 2-4 cm edge length were saturated with water and immersed in a water recipient on a scale. The weight of the displaced water is then measured and used to calculate the density of the specimen with 

\[ \rho_s = \rho_w \frac{W_{\text{dry}}}{W_{\text{displaced water}}} \]

where \( W \) represents weight. The results are shown in Table 2.4. Their temperature-invariant average was used in the model for the validation process presented in section 2.4 [78]. Density measurements were repeated in the second experimental campaign at EMPA using the same principle. The results are shown in Table 2.5. The average density shows a difference of 3.6% compared to the first campaign. This data is used in the analysis of section 3.6.

**Particle Size Distribution, Mean Diameter and Sphericity**

In order to eliminate uncertainties regarding the mean particle size and sphericity, both of which are important parameters for the prediction of the TES performance as well as the calculation of the pressure drop, the packed bed of rocks, shown in Fig. 2.7a, was analyzed using photoanalysis software [93]. The results are shown in Fig. 2.7b and Table 2.6. Because this analysis was conducted towards the end of the thesis, these results are only considered in the simulations done in section 3.6. For the experimental validation presented in section 2.4, estimated values were used, as explained in the respective section.

<table>
<thead>
<tr>
<th>Rock Type</th>
<th>S1</th>
<th>S2</th>
<th>S3</th>
<th>S4</th>
<th>S5</th>
<th>S6</th>
<th>S7</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \rho ) [kg/m³]</td>
<td>2680</td>
<td>2630</td>
<td>2630</td>
<td>2780</td>
<td>2610</td>
<td>2450</td>
<td>2660</td>
<td>2635</td>
</tr>
</tbody>
</table>
Fig. 2.7: a) Photograph of the packed bed of rocks (bars are 1 m in length) and b) Particle size distribution of the rocks obtained using photoanalysis software [93].

Table 2.6: Properties of the packed bed analyzed using photoanalysis software on the photograph shown in Fig. 2.7a.

<table>
<thead>
<tr>
<th>Number of particles in image</th>
<th>$d_{\text{min}}$ [mm]</th>
<th>$d_{\text{max}}$ [mm]</th>
<th>$d_{\text{mean}}$ [mm]</th>
<th>Sphericity</th>
</tr>
</thead>
<tbody>
<tr>
<td>5095</td>
<td>1.1</td>
<td>74</td>
<td>32</td>
<td>0.713</td>
</tr>
</tbody>
</table>
2.3.3 Convective Heat Transfer Coefficient

There are several studies investigating the convective heat transfer coefficient in a packed bed [31, 35, 40, 94-96]. Some of these correlations that are relevant for conditions similar to the ones in this work are presented in Table 2.7. Most of these correlations are empirically derived from experimental data and are hence bound to the specific conditions in the experiment. An exception is the correlation by Pfeffer [97] that is derived from theoretical considerations. Löf and Hawley [40] and Chandra and Willits [41] concluded that varying the inlet air temperature has little or no effect on the coefficients, in the range investigated in their experiments (up to 120 °C). The correlation by Nsofor and Adebiyi [31], derived for cylindrical pellets, is valid for temperatures up to 1000 °C. However, their correlation is not directly dependent on temperature but considers the effect of temperature through the Reynolds and Prandtl numbers. The reported uncertainty is ±10-30%. Bradshaw et al. [96] investigated the convective heat transfer between air and nitrogen as heat transfer fluid and alumina and steel balls as solid particle in the range of 0-800 °C. They reported no significant dependence on temperature and based their correlation on the Reynolds number as well. Therefore, the correlations used in this thesis are primarily chosen according to their Reynolds number and particle diameter range, rather than their temperature range. The correlations shown in Table 2.7 are plotted in Fig. 2.8 in their valid range. For similar void fractions, the correlation by Pfeffer agrees well with the empirically derived correlation by Alanis. The advantage of the correlation by Pfeffer is that it is not bound to a specific void fraction and can hence be used for different packing configurations. However, it is valid only for low Reynolds numbers. A sensitivity analysis has been carried out to investigate the influence of different convective heat transfer correlations on the temperature profiles obtained for a simulation representing the experimental run presented in section 2.4. With the correlation by Alanis as reference, the average root mean square deviations of the temperature profiles obtained are 2 °C for
Pfeffer ($\varepsilon = 0.33$) and 0.16 °C for Coutier and Farber. Hence, the effect is not significant on the simulation results.

Table 2.7: A selection of correlations to calculate the fluid-solid convective heat transfer coefficient in the packed bed. The particle and volumetric convective heat transfer coefficients are related by $h_v = h_p \frac{6(1 - \varepsilon)}{d}$.

<table>
<thead>
<tr>
<th>Author</th>
<th>Correlation</th>
<th>Range and relevant data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coutier &amp; Farber [35]</td>
<td>$h_v = 700(G / d)^{0.76}$</td>
<td>100 &lt; Re$_0$ &lt; 350</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Bi $\sim 0.1$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>1.8 cm $&lt;$ d $&lt;$ 3 cm</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$T &lt; 100$ °C</td>
</tr>
<tr>
<td>Alanis [94]</td>
<td>$h_v = 824(G / d)^{0.92}$</td>
<td>10 &lt; Re$_0$ $&lt;$ 200</td>
</tr>
<tr>
<td></td>
<td></td>
<td>d = 5.1 cm and 2.8 cm</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\varepsilon = 0.42$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$18$ °C $&lt;$ T $&lt;$ $67$ °C</td>
</tr>
<tr>
<td>Pfeffer [97]</td>
<td>$W = 2 - 3\gamma + 3\gamma^5 - 2\gamma^6$</td>
<td>0.1 &lt; Re$_0$ $&lt;$ 70</td>
</tr>
<tr>
<td></td>
<td>$\gamma = (1 - \varepsilon)^{1/3}$</td>
<td></td>
</tr>
<tr>
<td></td>
<td>$h_p = 1.26 \left[ \frac{1 - (1 - \varepsilon)^{5/3}}{W} \right]^{1/3}$</td>
<td>($c_p G)^{1/3}$ (k / d)$^{2/3}$</td>
</tr>
</tbody>
</table>


### 2.3.4 Effective Thermal Conductivity of Packed Bed

The effective thermal conductivity in a packed bed has been extensively studied. A review of the previous work can be found in [98]. Because of the high temperature of charging and temperature gradient in the axial direction, the correlation of Kunii and Smith is applied, which considers the thermal conductivity of the solid and the fluid, as well as the radiative transfer [99, 100]

\[
k_{eff} = k_f \left[ \varepsilon \left(1 + \beta \frac{h_{rv}d}{k_f} \right) + \frac{\beta (1 - \varepsilon)}{1 + \frac{h_{rs}d}{k_f} + \gamma \left(\frac{k_f}{k_s}\right)} \right]
\]  

(2.18)

The void to void radiative heat transfer coefficient, \(h_{rv}\), is represented by

\[
h_{rv} = \left[0.1952 \left(1 + \frac{\varepsilon}{2(1 - \varepsilon)} \frac{1 - \varepsilon_s}{\epsilon_s} \right) \right] \left(\frac{T_f}{100}\right)^3
\]  

(2.19)

The solid surface to solid surface radiative heat transfer coefficient, \(h_{rs}\), is

\[
h_{rs} = 0.1952 \frac{\epsilon_s}{(2 - \epsilon_s)} \left(\frac{T_s}{100}\right)^3
\]  

(2.20)

The emissivity of the rocks, \(\epsilon_s\), is in the range of 0.83-0.9 and is taken here 0.85 [101]. The factor \(\beta\) in Eq. (2.18) is defined as the ratio of the average length between the centers of two neighboring solids to the mean particle diameter and ranges from 0.82 (close packing) to 1.0 (loose packing). Due to the random packing of the rocks, an average value of 0.9 is taken in the model. The factor \(\gamma\) is defined as the ratio of the length of solid affected by the thermal conductivity to the mean particle diameter; it has a value of \(2/3\). The factor \(\phi\) is a measure of the effective thickness of the fluid film adjacent to the contact surface of two solid particles
\[
\phi = \phi_2 + (\phi_1 - \phi_2) \frac{\varepsilon - \varepsilon_2}{\varepsilon_1 - \varepsilon_2}
\]  
(2.21)

where \(\phi_1\) and \(\phi_2\) represent the two extreme cases of the void fraction, \(\varepsilon_1 = 0.476\) and \(\varepsilon_2 = 0.26\), and are calculated as

\[
\phi_i = \frac{1}{2} \left( \frac{\kappa - 1}{\kappa} \right) \sin^2 \theta_i \ln \left( \kappa - (\kappa - 1) \right) - \frac{1}{\kappa} \left( \frac{\kappa - 1}{\kappa} \right) (1 - \cos \theta_i) - \frac{2}{3} \kappa
\]  
(2.22)

with \(\kappa = k_s / k_f\) and \(\sin^2 \theta_i = 1 / n_i\). \(n_1\) and \(n_2\) are the number of the contact points on a semispherical surface of one solid particle with \(n_1 = 1.5\) and \(n_2 = 4\sqrt{3}\).

### 2.3.5 Thermal Losses

#### Lateral Walls Heat Transfer Coefficient

For the lateral walls, the overall heat transfer coefficient is calculated as

\[
\frac{1}{h_{\text{wall}}} = \frac{1}{h_{\text{inside}}} + r_{\text{inside}} \sum_{j=1}^{n-1} \frac{1}{k_j} \ln \frac{r_{j+1}}{r_j}
\]  
(2.23)

The second term on the r.h.s. depends on the tank design and considers conduction through \(n\) layers of material between the packed bed and the surrounding ground. The wall-film heat transfer coefficient, \(h_{\text{inside}}\), is given as

\[
h_{\text{inside}} = h_{\text{conv}} + h_{\text{rad-cond}}
\]  
(2.24)

where \(h_{\text{conv}}\) accounts for convective heat transfer between the fluid and the lateral walls [55]

\[
h_{\text{conv}} = \frac{k_f}{d} \left( 3.22 \operatorname{Re}^{1/3} \operatorname{Pr}^{1/3} + 0.117 \operatorname{Re}^{0.8} \operatorname{Pr}^{0.4} \right)
\]  
(2.25)

and \(h_{\text{rad-cond}}\) accounts for radiation and conduction heat transfer between the packed bed and the walls and is obtained using correlations given by Ofuchi and Kunii [102] that are similar to the Eq. (2.18) to (2.22) introduced above.
\[ h_{rad-cond} = \frac{k_f}{d} \left( \frac{1}{1/A-1/B} \right) \]  

(2.26)

with

\[ A = \varepsilon \left( 2 + \frac{h_{rs} d}{k_f} \right) + \frac{1-\varepsilon}{1 + \frac{h_{rs} d}{3k_s} + \frac{k_f}{\phi}} \]  

(2.27)

and \( B = \frac{k_{eff}}{0.5k_f} \)

**Cover Heat Transfer Coefficient**

The heat loss through the lid is calculated by applying an energy balance on the cover at every time step

\[ Q_{coverloss} = Q_{cond} = Q_{conv} + Q_{rad} - Q_{solar} \]  

(2.28)

which states that the energy conducted from the storage to the lid surface must be equal to the energy loss by convection and radiation to the surrounding under consideration of incoming solar radiation during day time. The conductive term is

\[ Q_{cond} = h_{cover,cond} A_{cover} (T_{surface} - T_{f,1}) \]  

(2.29)

where \( h_{cover,cond} \) depends on the tank design and considers conduction through the different layers of material between the packed bed and the surrounding air [72]

\[ h_{cover,cond} = 1 / \sum_{j=1}^{n} \frac{t_j}{k_j} \]  

(2.30)

where \( t \) is the thickness of the respective layer.

\( T_{surface} \) in Eq. (2.29) is the surface temperature of the cover exposed to the ambient and \( T_{f,1} \) is the fluid temperature of the first layer in the packed bed. The convective term in Eq. (2.28) is calculated as

\[ Q_{conv} = h_{outside} A_{cover} (T_{surface} - T_{ambient}) \]  

(2.31)
$h_{outside}$ is the convective heat transfer at the lid outer surface under consideration of the wind speed and temperature of the ambient

$$h_{outside} = \frac{\overline{NuD}}{D_{cover}}$$

(2.32)

with [72]

$$\overline{Nu}_x = 0.332 Re^{1/2} Pr^{1/3} \quad \text{Re} < 10^5$$

(2.33)

$$\overline{Nu}_D = (0.037 Re^{4/5} - A) Pr^{1/3} \quad \text{Re} > 10^5$$

(2.34)

$$A = (0.037 Re_x^{4/5} - 0.664 Re_x^{1/2}) = 871 \quad Re_x^{critical} = 10^5$$

For $\text{Re} < 10^5$ $\overline{Nu}_x$ is integrated over the cover diameter to obtain the average value. The radiation loss from the cover is calculated as [101]

$$Q_{rad} = \varepsilon_{ground} \sigma A \left( T_{surface}^4 - T_{sky}^4 \right)$$

(2.35)

with $\varepsilon_{ground} = 0.95$, the ground emissivity (the experimental setup of section 2.4 was covered with soil during operation to reduce the thermal losses). The sky temperature, $T_{sky}$, is calculated as [103, 104]

$$T_{sky} = T_{ambient} \left[ 0.711 + 0.56 \left( \frac{(T_{dew\ point} - 273)/100}{1} \right) + 0.73 \left( \frac{(T_{dew\ point} - 273)/100}{1} \right)^{1/4} \right]$$

(2.36)

with $T_{ambient}$ and $T_{dew\ point}$ in Kelvin. The dew point temperature can be approximated by $T_{dew\ point} \approx T_{ambient} - (1 - \phi) / 5$ with $\phi$ the relative humidity [105]. $T_{ambient}$ and $\phi$ shall be taken from meteorological data from the desired location or measured during the experimental campaign. The incoming solar radiation is considered during daytime and is calculated as

$$Q_{solar} = q_{solar} A_{cover}$$

(2.37)

with $q_{solar} = 1000 \, \text{W/m}^2$ [3].

In the above equations the only unknown is $T_{surface}$, which is obtained at every time step by solving Eq. (2.28) using the Newton-Raphson method [68]. $T_{surface}$ is then used to calculate the cover losses.
**Bottom Heat Transfer Coefficient**

Conductive heat transfer through the layers at the bottom has to be considered, after which an ambient temperature can be assumed. This is done in a similar way as in Eq. (2.29), modified for the bottom layers and dimensions.

### 2.3.6 Pressure Drop

The pressure drop is calculated using the equation given by Ergun [106]

\[ \Delta p = \frac{HG^2}{\rho_f \bar{d}_p} \left( A \frac{(1-\epsilon)^2}{\epsilon^3 \psi^2} \frac{\mu}{Gd_p} + B \frac{1-\epsilon}{\epsilon^3 \psi} \right) \]

(2.38)

A and B are study dependent constants and \( \psi \) considers the sphericity of the rocks with unity for a perfect sphere [107]. Published data by Macdonald [108] for randomly shaped gravel with similar particle size and void fraction as in this study have been taken: \( A = 217 \), \( B = 1.83 \). Eq. (2.38) is solved for every layer after every time step considering the local flow conditions in order to calculate the pressure drop across the packed bed. However, due to the small values of pressure drop with respect to atmospheric pressure, the pressure does not need to be adjusted in the calculation of the air density [69].

### 2.3.7 Axial Void Fraction Distribution

The void fraction of the packed bed was measured in an experimental setup built for this purpose as shown in Fig. 2.9a. A cylindrical iron tank with 2 m inner diameter and 1.5 m height was filled with the same rocks as in the thermal storage. A steel tube with 40 mm outer diameter and same height as the tank was welded coaxially to the bottom of the tank to measure the water level that is filled in the packed bed during the measurements. A hole was bored at the bottom of the tube in order to assure equal water level as in the packed bed. The tank is filled in five steps and the water level in
the tube is measured after the end of every step using a Bosch DLR130K digital distance measurer (accuracy 1.5 mm). With the filled water volume known, the void fraction is calculated as

$$
\varepsilon = \frac{V_w}{A_{eff} \Delta H_w}
$$

where $V_w$ is the water volume filled in the tank and $A_{eff}$ the effective surface area of the tank, and $\Delta H_w$ is the difference in the water height between two consecutive measurements. The axial void fraction distribution showed a second order monotonic decrease with packed bed depth from 0.366 to 0.327 which corresponds to an 11% decrease in void fraction after which it approaches asymptotically a value of 0.325. The results of the measurement are shown in Fig. 2.9b. The reasons for the decrease are mainly the non-homogeneous and random distribution of size and shape of the rocks and the own-weight effect of the rocks [109]. Layers close to the bottom and top of the packed bed were omitted in order to exclude the effects of the bottom and the disturbance of the bed near the open area. Hence, the results are shown from around 5-10 rock diameters from both ends. The average bulk void fraction was 0.342. In a separate study, the influence of axially variable void fraction on the heat transfer and pressure drop in the packed bed was investigated via CFD simulations [110]. Non-uniform radial distribution or “the wall effect” can be neglected due to the large tank to rock diameter ratio of about 70 [29, 111]. Thus, in contrast to previous studies [63, 64], no bypass fraction of the mass flow has to be assumed.
Fig. 2.9: Axial void fraction measurement: a) Photograph of the setup used for the measurement, and b) Measured axial void fraction distribution in the packed bed (height=0 at top).
2.4 Experimental Validation

The model introduced in section 2.2 is used with the closing relations presented in section 2.3 to simulate the behavior of a 6.5 MWh\textsubscript{th} TES prototype constructed at the premises of Airlight Energy SA in Biasca, Switzerland. The pilot-scale storage design and experimental setup are shown schematically in Fig. 2.10 and depicted in Fig. 2.11. The tank is immersed in the ground and has a truncated cone shape for exploiting the effect of lateral earth pressure at higher load bearing and for reducing the normal force on the walls as well as between the rocks during their thermal expansion by guiding them upwards [112]. It has a dodecagon cross section, a total height of 4 m, and a radius of the inscribed circle that decreases from 2 m at the top to 1.25 m at the bottom at a wall inclination of 12°. The tank is made of concrete to avoid deformation caused by thermal expansion of the storing material. The 12 lateral wall segments consist of a 15-mm layer of ultra-high performance concrete on the packed bed side with high mechanical stability and thermal conductivity and a 235-mm layer of low density concrete with low thermal conductivity in order to reduce the thermal losses [77].

![Fig. 2.10: Schematic of the pilot-scale thermal storage configuration and experimental setup.](image-url)
Fig. 2.11: Photograph of the prototype constructed in Biasca a) during construction and b) during operation (the lid was later covered with soil to reduce cover losses).

The tank is filled to the height of 2.9 m with pebbles with an equivalent sphere diameter of 2-3 cm and an estimated sphericity of 0.6\(^5\). A 400 mm Foamglas® insulation is packed under the lid. The hot air enters through the inlet pipe from the top, flows through the packed bed, and is collected at the bottom after flowing across a metal grid perforated with 10 mm-dia. holes every 30 mm, which prevents congestion of the outlet pipe by the rocks and debris while ensuring low resistance to the flow. Charging from the top allows the exploitation of the buoyancy effect to create and maintain thermal stratification inside the packed bed, the hottest region being at the top. During discharging, the direction of the flow is reversed as cold air is circulated through the tank from the bottom.

As will be shown in section 2.5.1, an additional advantage of the conical shape of the storage tank vis-à-vis a cylindrical shape is the larger storing volume on top of the tank where the temperature is highest, leading to a higher volume-to-surface ratio and hence smaller losses from the lateral

\(^5\) Note that the photoanalysis presented in section 2.3.2 was not carried out yet at the time of the experimental validation. Hence, estimated values were used for the particle diameter and sphericity.
walls. The relatively smaller volume at the bottom where most of the energy is already extracted from the air results in lesser necessary storing material.

The temperature is recorded by K-type thermocouples, of which five are located inside the packed bed at different vertical positions ($T_1$ to $T_5$, $T_6$ is not covered with rocks) and at a distance of 10 cm from the lateral walls, as indicated in Fig. 2.10. Two thermocouples measure the temperature just above the packed bed on the tank axis ($T_{inlet}$) and in the outlet pipe ($T_{outlet}$). The air flow is fed with ambient air aspirated through air filters by a fan coupled to an electric motor and heated in a tubular 58 kW electric heater before entering the storage tank. Additional measurements include: ambient air temperature and relative humidity, flow velocity in the feeder pipe, and pressure drop in the air flow across the packed bed.

An experimental run consisting of a charging time of 110 hours was carried out in order to validate the numerical model. Real time inlet temperature and mass flow rates are considered. The average equivalent sphere diameter of the particles is taken as 2.5 cm. The maximum Biot number ($\text{Bi} = \frac{h_p r}{k_s}$) reached in the simulation is 0.09 (corresponding to $h_p = 15$ W/m$^2$K and $k_s = 2$ W/mK). The assumption of homogeneous intra-particle temperature distribution is hence satisfied. The average of fluid and solid temperatures numerically modeled (curves) and the experimentally measured temperatures (markers) are plotted in Fig. 2.12. The agreement is reasonably good. The root mean square deviations of the simulated and measured temperature profiles are shown in Table 2.8. Slight over-prediction of the temperature profiles by the model at the beginning of the charging phase is due to radial temperature distribution in the tank vs. the average temperature obtained by the 1D model as well as the assumption of plug-flow in the model. This effect becomes less pronounced towards the end of the charging phase and tank bottom due to more homogeneous temperature as well as flow distribution in the tank. The discrepancy between the modeled and measured temperature $T_1$ is due to near wall effects and the presence of the perforated grid in the vicinity of the thermocouple.
The thermal capacity of the storage, defined as the energy stored in the packed bed when charged from ambient temperature to be isothermally at the inlet temperature of 650 °C, is 6.5 MWh\textsubscript{th}. A total 3.1 MWh\textsubscript{th} was fed to the storage, of which 86% is stored in the packed bed while 10.2% is lost through the lateral walls and 3.1% through the cover. The energy lost through the bottom and the energy flowing out of the storage from the bottom sums up to 0.7% of the input energy. The high losses are correlated to poor insulation on the lateral walls and the relatively small volume-to-surface ratio of the tank. In order to investigate the sensitivity of the simulations on the measurement uncertainties of the thermal properties of the rocks and concrete types, the simulation was repeated with the mean standard deviation of the measurements, as indicated in Table 2.2 and Table 2.4, added to the reference values. The percentage mean deviations are 3.2% for the thermal conductivity of the rocks, 3.7% and 11.7% for the thermal conductivity of the ultra-high performance concrete and low density concrete respectively, 0.5% for the rocks’ density and 3% for the rocks’ heat capacity. The root mean square deviations of the temperature profiles obtained with respect to the temperature profiles obtained using the

Fig. 2.12: Comparison of numerically calculated (curves) and experimentally measured (markers) temperatures for the 6.5 MWh\textsubscript{th} pilot-scale thermal storage unit.
reference values are shown in Table 2.8. For the heat capacity the results are slightly affected, while the effect for the other properties is negligible. After several charging and discharging tests, neither the rocks nor the concrete wall segments showed any visible damage or signs of deterioration. No dust carriage from the packed bed out of the tank was observed during the tests.

Sensitivity analysis on number of grid points – In order to justify the number of grid points chosen, simulations are carried out for 25, 50, 100 and 200 grid points. For every case, the average root mean square of the deviations (RMSD) in the temperature profile is calculated with respect to the finest grid. The normalized results are plotted in Fig. 2.13. As a compromise between calculation time and accuracy, 100 grid points were chosen.

<table>
<thead>
<tr>
<th>Root Mean Square Deviation [K]</th>
<th>T_1</th>
<th>T_2</th>
<th>T_3</th>
<th>T_4</th>
<th>T_5</th>
<th>T_{outlet}</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference case vs. experimental values</td>
<td>16</td>
<td>4.4</td>
<td>8.3</td>
<td>7</td>
<td>15.9</td>
<td>3.3</td>
</tr>
<tr>
<td>Reference case vs. deviations of thermal conductivity of rocks considered</td>
<td>0.15</td>
<td>0.19</td>
<td>0.16</td>
<td>0.17</td>
<td>0.22</td>
<td>0.07</td>
</tr>
<tr>
<td>Reference case vs. deviations of thermal conductivity of concrete considered</td>
<td>0.21</td>
<td>0.44</td>
<td>0.68</td>
<td>0.72</td>
<td>0.58</td>
<td>0.07</td>
</tr>
<tr>
<td>Reference case vs. deviations of heat capacity of rocks considered</td>
<td>3.68</td>
<td>6.06</td>
<td>6.66</td>
<td>5.78</td>
<td>3.86</td>
<td>1.29</td>
</tr>
<tr>
<td>Reference case vs. deviations of density of rocks considered</td>
<td>0.64</td>
<td>1.04</td>
<td>1.12</td>
<td>0.97</td>
<td>0.65</td>
<td>0.23</td>
</tr>
</tbody>
</table>

Table 2.8: The root mean square deviation of the simulated temperature from the measured temperature for the thermocouples in the packed bed and in the outlet, and the reference case vs. measurement deviations considered.
Fig. 2.13: Normalized average RMSD of the temperature profile obtained with the indicated number of grid points with respect to the finest case (200 grid points).

2.5 Auxiliary Studies

The validated model is used to study the effect of the conical shape, the radiative heat transfer in the packed bed as well as variable solid and fluid properties.

2.5.1 Comparison of Cylindrical and Truncated Conical Tank

The model is used to compare the performance of a thermal storage unit with truncated conical shape versus the more commonly used cylindrical shape. Dimensions and operating conditions are shown in Table 2.9. Fig. 2.14 shows the temperature distribution and the energy stored per unit height after 8 h charging, as well as the pressure drop as a function of charging time for two tanks with equal volume. A smaller height and hence volume of the storing material is needed to extract the energy from the air flow through the conical storage tank compared to the cylindrical tank under the same operating conditions since more energy can be stored in the top of the tank because of the larger diameter. In addition, due to the larger
volume to surface ratio in the high-temperature region, the wall losses are smaller for the conical tank, as explained in section 2.4 (51.9 kWh for conical vs. 68.5 kWh for cylindrical). However, due to the larger cover area, the cover losses are larger for the conical tank (11.1 kWh for conical vs. 6.3 kWh for cylindrical). The pressure drop increases for both cases with time as the overall temperature inside the packed bed increases, causing the density to decrease and the dynamic viscosity to increase, which in turn increases the pressure drop according to Eq. (2.38). At the beginning of the charging phase, when the temperature is low in the whole packed bed, the pressure drop is larger for the conical tank because the cone-to-cylinder radius at the top is smaller than the cylinder-to-cone radius at the bottom. As the tank is charged, the pressure drop in the cylindrical tank exceeds that of the conical tank because of the larger average temperature in the cylindrical tank. A more thorough study on the effect of the cone angle on the storage performance for a 7.2 GWh_th scale-up TES unit is presented in section 2.7.4.

| Table 2.9: Dimensions and operating conditions of the storage units of Fig. 2.14. |
|-----------------------------------------|-------|-------|
|                                        | Cylindrical | Conical |
| \( r_{\text{top}} \) [m]              | 1.5   | 2     |
| \( r_{\text{bottom}} \) [m]           | 1.5   | 0.94  |
| \( H \) [m]                            | 3     |       |
| \( V \) [m³]                           | 21    |       |
| \( t_{\text{charging}} \) [h]         | 8     |       |
| \( \dot{m} \) [kg/s]                   | 0.4   |       |
| \( d_{\text{rocks}} \) [m]             | 0.03  |       |
| \( T_{\text{charging}} \) [°C]         |       | 650   |
Fig. 2.14: a) Temperature and energy distribution after 8 h charging, and b) Pressure drop as a function of charging time for a cylindrical storage tank versus a conical storage tank of equal volume. Dimensions and operating conditions given in Table 2.9.
2.5.2 Importance of Radiative Heat Transfer

As it can be seen in Fig. 2.15, radiative heat exchange is relevant due to the high temperatures involved. Hence, the radiation exchange between the particles as well as between the particles and the walls is considered. The former affects the effective thermal conductivity in the packed bed and the latter the thermal losses to the surroundings. The effect of radiation has been considered by employing Eq. (2.18) to (2.22). Fig. 2.16a shows the effective thermal conductivity of the packed bed and the contribution of conduction and radiation to the effective thermal conductivity, as a function of temperature. While conduction is superior at below 150 °C, radiation becomes predominant at higher temperatures. Fig. 2.16b shows the temperature profile in the packed bed after 8 h of charging for a conical tank according to Table 2.9 for three different cases: #1 $k_{\text{eff}}$ according to Eq. (2.18), #2 $k_{\text{eff}}$ according to Eq. (2.18) with radiation neglected ($h_{\text{rv}}=0$ and $h_{\text{rs}}=0$) and #3 $k_{\text{eff}}=0$, hence without axial conduction. As expected, the thermocline flattens with increasing effective thermal conductivity (from case #3 to #1). The effect of axial conduction when radiation is neglected is small (case #2 vs. #3), while considering radiation affects the temperature profile to a larger extent, especially in the high-temperature region.

Fig. 2.15: Photograph of the rocks during operation as taken from the top inlet.
The contribution of the radiative heat transfer to the thermal losses is not straightforward since the overall wall heat transfer coefficient, $h_{\text{wall}}$ (Eq. (2.23)), is also dependent on the convective heat transfer between the fluid and the wall, which in turn is dependent on the flow – and therefore – operating conditions. For case #1 investigated above, the heat transfer coefficient between the packed bed and the wall ($h_{\text{inside}}$, Eq. (2.24)) and the contributions of its composing terms (radiation, conduction and convection) are shown in Fig. 2.17 for the last time step as a function of tank height. Again, in the high-temperature region (top), the contribution of radiation is predominant and decreases towards the low-temperature region (bottom).

2.5.3 Comparison of Temperature-Dependent and Constant Properties

Fig. 2.18 shows a comparison of simulation results for constant and temperature dependent solid and fluid properties. All of the fluid and solid properties are evaluated at the average temperature of 335 °C for the case with constant properties. The most important parameter is the solid heat capacity, which affects the amount of energy that can be stored. Its value is under-predicted in the hot region and over-predicted in the cold region when constant properties are considered. Therefore, the temperature profile is slightly over-predicted for constant properties in the hot region, while the opposite effect is more pronounced in the cold region.
Fig. 2.16: a) Effective thermal conductivity of the packed bed and the contribution of conduction and radiation to the effective thermal conductivity as a function of temperature, and b) The temperature profile in a conical tank according to Table 2.9 for three cases: #1 \( k_{\text{eff}} \) according to Eq. (2.18), #2 \( k_{\text{eff}} \) according to Eq. (2.18) with radiation neglected and #3 without axial conduction \( (k_{\text{eff}} = 0) \).
Fig. 2.17: Heat transfer coefficient between the packed bed and the wall, $h_{\text{inside}}$, and the contributions of its composing terms for case #1 investigated above.
2.6 Scale-Up

The model is applied to design a 100 MWh\textsubscript{th} TES for a 3 MW\textsubscript{th} Airlight Energy pilot power plant in Ait Baha, Morocco, and an array of two 7.2 GWh\textsubscript{th} scale-up TES units for a hypothetical 26 MW\textsubscript{el} solar power plant.

*Grid Refinement Study* – A grid refinement study was conducted to determine the necessary number of grid points that ensure numerically reliable results for the scale-up simulations. Dimensions and operating conditions of the 7.2 GWh\textsubscript{th} TES as introduced in Table 2.13 in section 2.6.2 are used. 100, 200, 400 and 800 grid points were investigated.
Fig. 2.19 shows the temperature profile inside the TES for different times and different number of grid points during charging. The average RMSD of each case with respect to the finest grid (800 grid points) is shown in Table 2.10. Only the first 4 m of the TES are considered for the calculation of the RMSD values since, as seen in Fig. 2.19, the thermocline does not penetrate further after 8 h of charging. As a compromise between calculation time and accuracy, 400 grid points were chosen for the simulations of this and the next section (section 2.7), which results in a layer height of 6.25 cm. This layer height is respected if the height of the TES is modified in the studies that follow. The convective heat transfer correlation by Coutier and Farber [35] was chosen for the studies in this chapter as it is valid for a larger range of Reynolds numbers (see Table 2.7). All the other correlations are as introduced in section 2.3.

<table>
<thead>
<tr>
<th>Height [m]</th>
<th>Temperature [°C]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>t = 2h</td>
</tr>
<tr>
<td>1</td>
<td>t = 4h</td>
</tr>
<tr>
<td>2</td>
<td>t = 6h</td>
</tr>
<tr>
<td>3</td>
<td>t = 8h</td>
</tr>
<tr>
<td>4</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 2.19: The temperature profile in the TES for different times and number of grid points after one charging run.

Table 2.10: The average root mean square deviation for different numbers of grid points with respect to the finest grid (800 grid points).

<table>
<thead>
<tr>
<th>RMSD_{100-800} [K]</th>
<th>RMSD_{200-800} [K]</th>
<th>RMSD_{400-800} [K]</th>
</tr>
</thead>
<tbody>
<tr>
<td>44.3</td>
<td>22.8</td>
<td>8.6</td>
</tr>
</tbody>
</table>
The charging efficiency is defined as the ratio of the energy stored in the rocks at the end of the cycle to the net input energy

\[ \eta_{\text{Charging}} = \frac{E_{\text{Stored}}}{E_{\text{Net Input}}} \]  

(2.40)

The stored energy is calculated as

\[ E_{\text{Stored}} = \sum_{i=1}^{N} (1 - \varepsilon_i) \rho_i V_i \left( e_{x,i}^{\text{end}} - e_{x,i}^{\text{start}} \right) \]  

(2.41)

with \( t=0 \) representing the beginning of the charging phase and \( t=\text{end} \) its end, and \( i \) denoting the layer. The net input energy is

\[ E_{\text{Net Input}} = E_{\text{Input}} - E_{\text{Outflow}} \]  

(2.42)

with the input and outflow energy calculated as

\[ E_{\text{Input}} = \sum_{t=0}^{t_{\text{charging}}} h_{f,in}^i m_i \Delta t \]  

(2.43)

\[ E_{\text{Outflow}} = \sum_{t=0}^{t_{\text{charging}}} h_{f,N}^i m_i \Delta t \]  

(2.44)

The discharging efficiency is defined as the fraction of the recovered energy during the discharging phase to the stored energy

\[ \eta_{\text{Discharging}} = \frac{E_{\text{Recovered}}}{E_{\text{Stored}}} \]  

(2.45)

The recovered energy, \( E_{\text{Recovered}} \), is calculated similar to \( E_{\text{Net Input}} \), but for the discharging phase. The overall cycle efficiency of the TES is hence

\[ \eta_{\text{Overall}} = \eta_{\text{Charging}} \eta_{\text{Discharging}} \]  

(2.46)

The fractions of the cyclic thermal losses are defined as

\[ f_{\text{WallLoss}} = \frac{E_{\text{WallLoss,Cycle}}}{E_{\text{Net Input}}} \]  

(2.47)

\[ f_{\text{CoverLoss}} = \frac{E_{\text{CoverLoss,Cycle}}}{E_{\text{Net Input}}} \]  

(2.48)

\[ f_{\text{BottomLoss}} = \frac{E_{\text{BottomLoss,Cycle}}}{E_{\text{Net Input}}} \]  

(2.49)
and

\[ f_{\text{TotalLoss}} = f_{\text{WallLoss}} + f_{\text{CoverLoss}} + f_{\text{BottomLoss}} \]  \hspace{1cm} (2.50)

The fraction of the pumping energy is defined as

\[ f_{\text{Pump,Charging}} = \frac{E_{\text{Pump,Charging}}}{E_{\text{NetInput}}} \]  \hspace{1cm} (2.51)

\[ f_{\text{Pump,Discharging}} = \frac{E_{\text{Pump,Discharging}}}{E_{\text{Recovered}}} \]  \hspace{1cm} (2.52)

with the pumping energy calculated as

\[ E_{\text{Pump,Charging}} = \sum_{t=0}^{t_{\text{charging}}} \Delta p' \cdot m' \cdot \Delta t / (\rho \cdot \eta \cdot \eta_{\text{Rankine}}) \]  \hspace{1cm} (2.53)

considering the Rankine cycle efficiency to produce electricity. \( E_{\text{Pump,Discharging}} \) is calculated similar to \( E_{\text{Pump,Charging}} \), but for the discharging phase. The overall pumping losses are denoted as

\[ f_{\text{Pump}} = \frac{E_{\text{Pump,Charging}} + E_{\text{Pump,Discharging}}}{E_{\text{NetInput}}} \]  \hspace{1cm} (2.54)

The capacity ratio indicates to which extent the maximal theoretical capacity of the storage unit is being exploited and is defined as follows

\[ \sigma = \frac{E_{\text{stored,Tot}}}{E_{\text{stored}}} \]  \hspace{1cm} (2.55)

with \( E_{\text{stored}}^{\text{max}} \) being defined as the energy stored in the packed bed when charged from ambient temperature to isothermal conditions at the inlet flow temperature, and \( E_{\text{stored,Tot}} \) the total stored energy in the storage.

2.6.1 100 MWh\textsubscript{th} TES

The validated simulation model is applied in the design of an industrial-scale TES unit for a CSP plant located in Ait Baha, Morocco. The CSP plant delivers 3 MW of thermal energy to an organic Rankine cycle for electric power generation. Simulations are performed for the month of May, which has the highest monthly DNI. The dimensions and operating conditions of the storage unit are listed in Table 2.11. Thermo-physical
properties of the insulation material, the two concrete types and the surrounding soil are summarized in Table 2.12. In contrast to the experimental setup of section 2.4, insulation material is placed internally on the lateral walls to reduce the thermal losses. The discharging temperature of 280 °C is given by the organic Rankine cycle connected to the solar field⁶. The thermal capacity or $E_{\text{stored}}^{\text{max}}$, defined as the energy stored in the packed bed when charged from ambient temperature to isothermal conditions at the inlet flow temperature, is 100 MWh$_{\text{th}}$. The scheme and photograph of the TES unit is shown in Fig. 2.20. In an effort to achieve a plug-flow, the inlet tube’s outlet is partially blocked, lateral openings are introduced and an umbrella-like cap is placed between the packed bed and the lid, as seen in Fig. 2.20a. The cap, besides homogenizing the flow, will help reducing the radiative and convective heat transfer between the packed bed and the lid.

The cyclic energy flows and the final outflow temperature (top) at the end of discharging as a function of cycle number are plotted in Fig. 2.21. Due to the lack of insulation on the bottom and the elevated discharging temperature of 280 °C, the bottom losses are larger than the cover and wall losses. Since the temperatures at the top and bottom are not varying significantly with time, the cover and bottom losses remain nearly constant throughout the cycles, while the lateral wall losses increase due to the increase of average storage temperature. The sum of the thermal losses remains below 3.5% of the input energy. Although the outlet temperature at the end of the discharging phase drops to 420 °C for the first cycle, it increases with every cycle and remains above 550 °C as the storage approaches steady cyclic behavior. During the cyclic operation, the overall efficiency increases from 58% to 89%. The pressure drop is not calculated in this study.

Table 2.11: Dimension and operating conditions of the 100 MWh\textsubscript{th} TES in Ait Baha, Morocco.

<table>
<thead>
<tr>
<th>Dimensions</th>
<th>Operating Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>$r_{\text{top}}$ [m]</td>
<td>$t_{\text{charging}}$ [h] 10</td>
</tr>
<tr>
<td>$r_{\text{bottom}}$ [m]</td>
<td>$t_{\text{discharging}}$ [h] 4.5</td>
</tr>
<tr>
<td>$H$ [m]</td>
<td>$t_{\text{idle}}$ [h] 9.5</td>
</tr>
<tr>
<td>$d_{\text{rocks}}$ [m]</td>
<td>$\dot{m}_{\text{charging}}$ [kg/s] 1.716</td>
</tr>
<tr>
<td>$\varepsilon$ [-]</td>
<td>$\dot{m}_{\text{discharging}}$ [kg/s] 4.058</td>
</tr>
<tr>
<td>$\psi$ [-]</td>
<td>$T_{\text{charging}}$ [°C] 640</td>
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<td></td>
<td>$T_{\text{discharging}}$ [°C] 280</td>
</tr>
<tr>
<td></td>
<td>$T_{\infty}$ [°C] 20</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Insulation Thickness</th>
<th>Microtherm®/Foamglas®</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lid [m]/[m]</td>
<td>0.025/0.6</td>
</tr>
<tr>
<td>Lateral walls [m]/[m]</td>
<td>0.05/0.6</td>
</tr>
<tr>
<td>Bottom [m]/[m]</td>
<td>0/0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Concrete Thickness</th>
<th>UHPC/LD</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lid [m]/[m]</td>
<td>0.08/0</td>
</tr>
<tr>
<td>Lateral Walls [m]/[m]</td>
<td>0.03/0.85</td>
</tr>
<tr>
<td>Bottom [m]/[m]</td>
<td>0.02/0.48</td>
</tr>
</tbody>
</table>

Table 2.12: Thermo-physical properties of the insulation material and the concrete types used in the tank walls as well as the surrounding soil. For the specific heat and conductivity, which are modeled as being temperature-dependent, the ranges are listed.

<table>
<thead>
<tr>
<th></th>
<th>$\rho$ [kg/m\textsuperscript{3}]</th>
<th>$C(T)$ [J/kgK]</th>
<th>$k(T)$ [W/mK]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Microtherm*</td>
<td>250</td>
<td>690-1130</td>
<td>0.019-0.03</td>
</tr>
<tr>
<td>Foamglas*</td>
<td>120</td>
<td>840</td>
<td>0.05</td>
</tr>
<tr>
<td>UHPC</td>
<td>2500</td>
<td>500</td>
<td>1.9-2.2</td>
</tr>
<tr>
<td>LD</td>
<td>1500</td>
<td>663</td>
<td>0.35-0.4</td>
</tr>
<tr>
<td>Soil [72]</td>
<td>2050</td>
<td>1840</td>
<td>0.5</td>
</tr>
</tbody>
</table>

* Data taken from manufacturers’ data sheets
Fig. 2.20: a) Scheme and b) Photograph of the TES in Ait Baha, Morocco during construction.
The validated model is further applied in the design of an industrial-scale TES system for a hypothetical round-the-clock 26 MW\textsubscript{el} CSP plant located in Ait Baha, Morocco. Simulations are performed for the month of May, which has the highest monthly DNI for 8 hr of charging followed by 16 hr of discharging. Thus, 1/3 of the air mass flow rate coming from the solar field is directly fed to the heat exchanger for the steam-based Rankine cycle for electricity production and 2/3 of it is directed to the two thermal storage units, each with a thermal capacity of 7.2 GWh\textsubscript{th}. The thermal capacity is defined as the energy stored in the packed bed when charged from ambient temperature to isothermal conditions at the inlet flow temperature. Construction limitations determine the lid size to a maximal 20 m in radius, as calculated in Appendix A. The mass flow rate through every storage unit is chosen so that the total pumping energy required for a cycle stays below 2\% of the input energy of the storage unit per cycle. The charging temperature coming from the solar field is 650 °C and the

![Graph](image.png)

**Fig. 2.21:** Cyclic energy flows and final outflow temperature (top) at the end of discharging as a function of cycle number for the 100 MWh\textsubscript{th} TES. Dimensions and operating conditions are given in Table 2.11.

### 2.6.2 7.2 GWh\textsubscript{th} TES

The validated model is further applied in the design of an industrial-scale TES system for a hypothetical round-the-clock 26 MW\textsubscript{el} CSP plant located in Ait Baha, Morocco. Simulations are performed for the month of May, which has the highest monthly DNI for 8 hr of charging followed by 16 hr of discharging. Thus, 1/3 of the air mass flow rate coming from the solar field is directly fed to the heat exchanger for the steam-based Rankine cycle for electricity production and 2/3 of it is directed to the two thermal storage units, each with a thermal capacity of 7.2 GWh\textsubscript{th}. The thermal capacity is defined as the energy stored in the packed bed when charged from ambient temperature to isothermal conditions at the inlet flow temperature. Construction limitations determine the lid size to a maximal 20 m in radius, as calculated in Appendix A. The mass flow rate through every storage unit is chosen so that the total pumping energy required for a cycle stays below 2\% of the input energy of the storage unit per cycle. The charging temperature coming from the solar field is 650 °C and the
The discharging temperature coming from the power block is 150 °C [113]. The Biot number arrives at a maximum value of 0.09 (for $h_p = 8.1 \text{ W/m}^2\text{K}$ and $k_s = 1.35 \text{ W/mK}$). The dimensions and operating conditions of the storage unit are listed in Table 2.13. The cyclic energy flows and the final outflow temperature (top) at the end of discharging as a function of cycle number are plotted in Fig. 2.22. The duration of each cycle is equal to one day. Due to the small amount of thermal losses, the charging efficiency approaches 100% and hence the discharging and overall efficiency are almost equal throughout the cyclic operation. The cover and bottom losses remain constant throughout the cycles since the temperature at the top and bottom is not varying significantly with time while the lateral wall losses increase slightly due to the increase of average tank temperature, as seen in Fig. 2.23. The sum of the thermal losses remains below 0.5% of the input

<table>
<thead>
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<th>Dimensions</th>
<th>Operating Conditions</th>
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<tr>
<td>$r_{\text{top}}$ [m]</td>
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<td>8</td>
</tr>
<tr>
<td>$r_{\text{bottom}}$ [m]</td>
<td>$t_{\text{discharging}}$ [h]</td>
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<td>16</td>
<td>16</td>
</tr>
<tr>
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<td>$T_{\text{charging}}$ [$^\circ\text{C}$]</td>
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<td>25</td>
<td>650</td>
</tr>
<tr>
<td>$d$ [m]</td>
<td>$T_{\text{discharging}}$ [$^\circ\text{C}$]</td>
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<tr>
<td>0.03</td>
<td>150</td>
</tr>
<tr>
<td>$\varepsilon$ [-]</td>
<td>$T_{\infty}$ [$^\circ\text{C}$]</td>
</tr>
<tr>
<td>0.342</td>
<td>20</td>
</tr>
<tr>
<td>$\psi$ [-]</td>
<td>$m_{\text{total solar field}}$ [kg/s]</td>
</tr>
<tr>
<td>0.7</td>
<td>396</td>
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</tbody>
</table>

Table 2.13: Dimension and operating conditions of the 7.2 GWh$_{th}$ packed bed thermal energy storage.

<table>
<thead>
<tr>
<th>Insulation Thickness</th>
<th>Operating Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lid [m]/[m]</td>
<td>$m_{\text{per storage charging}}$ [kg/s]</td>
</tr>
<tr>
<td>0.6/0.5</td>
<td>132</td>
</tr>
<tr>
<td>Lateral walls [m]/[m]</td>
<td>$m_{\text{per storage discharging}}$ [kg/s]</td>
</tr>
<tr>
<td>0.3/0.5</td>
<td>66</td>
</tr>
<tr>
<td>Bottom [m]/[m]</td>
<td>Microtherm®/Foamglas®</td>
</tr>
<tr>
<td>0/0.4</td>
<td></td>
</tr>
</tbody>
</table>

Concrete Thickness | UHPC/LD |
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Lid [m]/[m]</td>
<td>$\eta_{\text{fan}}$</td>
</tr>
<tr>
<td>0.02/0.7</td>
<td>0.95</td>
</tr>
<tr>
<td>Lateral Walls [m]/[m]</td>
<td>$\eta_{\text{Rankine}}$</td>
</tr>
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<td>0.02/1</td>
<td>0.3</td>
</tr>
<tr>
<td>Bottom [m]/[m]</td>
<td>Number of storage units</td>
</tr>
<tr>
<td>0.02/0.48</td>
<td>2</td>
</tr>
</tbody>
</table>
energy. The pumping energy during charging is larger than during discharging due to the larger mass flow rate and increases with cycle number for both charging and discharging due to the increase in average tank temperature and the consequent increase in air viscosity. Although the outlet temperature at the end of the discharging phase drops below 400 °C for the first cycle, it increases significantly with every cycle and remains above 590 °C as the storage approaches steady cyclic behavior, which is crucial for the operation of the power block. During the cyclic operation, the discharging efficiency increases from 81% to 96%. The capacity ratio stays below 40% throughout the cyclic operation. As shown in Fig. 2.23, the thermocline inside the packed bed flattens and the temperature front moves downwards during the cyclic operation.

Fig. 2.22: Cyclic energy flows and final outflow temperature (top) at the end of discharging as a function of cycle number for a 7.2 GWh \textsubscript{th} TES unit integrated in a 26 MW\textsubscript{el} CSP plant. Dimensions and operating conditions are given in Table 2.13.
In this section studies are carried out that can serve as a guideline for the design of thermal energy storage units. The effect of initial charging, storage parameters such as the tank diameter to height ratio for cylindrical tanks, the cone angle for conical tanks, the particle diameter and the insulation thickness on the TES performance are investigated. Further, equations are introduced to model the surrounding ground as a semi-infinite slab and its effect on the storage performance and thermal losses is investigated. Unless stated otherwise, the dimensions and operating conditions of the 7.2 GWh\textsubscript{th} TES introduced in section 2.6.2 are used.

**Fig. 2.23: Temperature profile in the packed bed at end of the charging and discharging phases for cycles 1, 10, 20 and 30.**

### 2.7 Guidelines for Design of TES Units

In this section studies are carried out that can serve as a guideline for the design of thermal energy storage units. The effect of initial charging, storage parameters such as the tank diameter to height ratio for cylindrical tanks, the cone angle for conical tanks, the particle diameter and the insulation thickness on the TES performance are investigated. Further, equations are introduced to model the surrounding ground as a semi-infinite slab and its effect on the storage performance and thermal losses is investigated. Unless stated otherwise, the dimensions and operating conditions of the 7.2 GWh\textsubscript{th} TES introduced in section 2.6.2 are used.
2.7.1 Initial Charging

To reduce the time required to reach steady cyclic behavior of the TES, a number of initial charging cycles can be carried out. This amounts to charging the storage without conducting the following discharging. Fig. 2.24 shows the final discharge temperature as a function of cycle number and the number of initial chargings. It can be seen that conducting initial charging cycles enables the storage to overcome the relatively poor performance during the first 15-20 cycles. Fig. 2.25 shows the thermal and pumping losses and the final outflow temperature of discharging and overall efficiency for the 20th cycle as a function of the number of initial charging cycles. The wall and pumping losses increase slightly with an increasing number of initial chargings due to the higher average temperature in the packed bed (see Fig. 2.26). The bottom and cover losses are not affected. The final discharge temperature and the overall efficiency of the storage increase with an increasing number of initial charging cycles. Comparing the final discharge temperature after 25 cycles for no and three initial charging cycles, the difference can be seen to be more than 40 °C.

![Fig. 2.24: The final outflow temperature of the storage during discharging as a function of cycle number for different number of initial chargings.](image)
Fig. 2.25: Thermal and pumping losses (a) and final discharge outflow temperature and overall efficiency (b) as a function of number of initial chargings for the 20th cycle.
Fig. 2.26 shows the temperature profile in the packed bed at the end of the 20\textsuperscript{th} charging period for different number of initial chargings. Since the pre-charged energy needs to be stored in addition to the cyclic energy fed to the TES, the thermocline is more advanced if initial charging cycles are conducted. Hence, for the same number of cycles, more storage material undergoes a temperature increase with increasing number of initial chargings. The results presented in the remainder of this paper were obtained with three initial charging cycles.

### 2.7.2 Surrounding Soil as Semi-Infinite Slab

The packed bed of rocks TES unit investigated in this thesis is immersed in the ground, with the lateral walls and the bottom exposed to the surrounding ground, while the lid is at ground level and hence interacts thermally with the surrounding air, the sun and meteorological conditions. The reasons for this configuration have been explained section 2.4. So far, the thermal losses from the lateral walls and bottom of the TES are calculated by assuming a fixed temperature of the surrounding soil at a

![Solid temperature profile in packed bed at the end of the 20\textsuperscript{th} charging period for different initial chargings. Initial charging causes the thermocline to be more advanced in the packed bed.](image-url)
preset distance from the storage tank (this method is called “Fixed Temperature” in the following). Here, the ground is modeled as a semi-infinite slab (called “Semi-Infinite Slab” in the following) in which the soil temperature is computed at every cycle, allowing its effect on the performance to be assessed. Due to the relatively large radius of the tank wall, resulting in a small curvature, the transient conduction is simplified as taking place across a plane wall. For simplicity, the axial conduction in the soil is neglected. The temperature in a semi-infinite slab can be estimated from [72]

\[
\frac{T_{\text{soil}}(r, t) - T_{\text{wall}}}{T_{\text{soil,init}} - T_{\text{wall}}} = \text{erf} \left( \frac{r}{2\sqrt{\alpha_{\text{equiv}} t}} \right)
\]

as a function of time and space when a wall temperature \( T_{\text{wall}} \) is imposed on the inside of the storage at \( t = 0 \). Here, the soil temperature 20 cm from the outside of the lateral walls is calculated, so \( \alpha_{\text{equiv}} \) is the equivalent diffusivity of the materials comprising the wall, as listed in Table 2.13, plus 20 cm of soil. \( T_{\text{soil,init}} \) is the initial soil temperature which is set equal to \( T_{\infty} \) at the beginning of the simulation. Equation (2.56) is solved for every layer of the domain separately considering the local temperature in the TES in order to obtain the axial temperature distribution in the soil. The equivalent conductivity is obtained as the average value for a multi-layer cylindrical wall

\[
k_{\text{equiv}} = \frac{\ln \left( \frac{r_{\text{out}}}{r_{\text{in}}} \right)}{\sum_{i=1}^{n-1} \ln \left( \frac{r_{i+1}}{r_{i}} \right) k_i}
\]

where the summation is over the materials comprising the wall, as listed in Table 2.13, plus 20 cm of soil, \( r_{\text{in}} \) is the internal radius of the tank, and \( r_{\text{out}} \) is equal to \( r_{\text{in}} \) plus the sum of the thicknesses of the materials comprising the wall plus 20 cm of soil. The equivalent diffusivity follows from

\[
\alpha_{\text{equiv}} = \frac{k_{\text{equiv}}}{C_{\text{equiv}} \rho_{\text{equiv}}}
\]
$C_{\text{equiv}}$ and $\rho_{\text{equiv}}$ are calculated as the volume-averaged values of the heat capacity and density of the materials comprising the lateral walls plus 20 cm of soil. $T_{\text{wall}}$ is obtained from

$$T_{\text{wall}} = T_f - \frac{Q_{\text{wall}}}{h_{\text{inside}} A_c}$$  \hspace{1cm} (2.59)

where $h_{\text{inside}}$ is obtained using Eq. (2.24), $T_f$ is the fluid temperature in the packed bed, $A_c$ is the circumferential area of the lateral walls, and $Q_{\text{wall}}$ is the heat flux through the lateral walls,

$$Q_{\text{wall}} = h_{\text{wall}} A_c \left( T_f - T_{\text{soil}}(r,t) \right)$$  \hspace{1cm} (2.60)

with $h_{\text{wall}}$ the overall wall heat transfer coefficient obtained with Eq. (2.23). Using the above relations, $T_{\text{wall}}$ is obtained from Eq. (2.59) and used together with $\alpha_{\text{equiv}}$ from Eq. (2.58) to calculate the time dependent soil temperature from Eq. (2.56). For the bottom, Eq. (2.57) and (2.58) are taken for flat walls and $T_{\text{wall}}$ is set to $T_{f,N}$. Fig. 2.27 shows the soil temperature 20 cm from the tank walls as a function of cycle number for four locations, where $x$ is the vertical position, with $x = 0$ at the top and $x = H$ at the bottom of the storage. Due to the temperature stratification in the packed bed, the increase in soil temperature depends on the vertical position. The soil temperature underneath the storage is higher than the soil temperature at $x/H=1$ and $x/H=0.5$ because of the worse insulation on the bottom (see Table 2.13). The maximum temperature increase after 60 cycles is 17 °C.

Fig. 2.28 shows the wall and bottom losses for the fixed-temperature and the semi-infinite slab models as a function of cycle number. Due to increasing soil temperature, the losses predicted by the semi-infinite slab model reduce with increasing cycle number compared to those predicted by the fixed-temperature model. However, the differences are only 0.005 and 0.0025 percent for the lateral wall and bottom losses, respectively. The conclusion is that assuming the soil temperature to be constant introduces small errors only and leads to conservative predictions of the losses and the overall efficiency.
Fig. 2.27: Increase in soil temperature with respect to the initial temperature, 20 cm from the tank walls as a function of cycle number. The temperature below the foundation increases faster than the temperature at $x/H=1$ and $x/H=0.5$ due to worse insulation on the bottom of the storage compared to the lateral walls.

Fig. 2.28: Fraction of the lateral wall losses and bottom losses of the inflow energy as a function of cycle number for the fixed-temperature and semi-infinite slab models. Due to increasing soil temperature, the losses for the semi-infinite slab model decrease compared to the fixed-temperature model.
2.7.3 Insulation Thickness

Lateral walls and bottom insulation – The insulation thickness was varied to quantify its influence on the performance of the storage as well as the surrounding soil temperature using the semi-infinite slab model. Therefore, the insulation thickness given in Table 2.13 was multiplied by an insulation thickness factor $f_{\text{insulation}}$ ranging from 0 to 1. For all the simulations that follow, the performance parameters are compared after the 15th cycle because they do not vary significantly in subsequent cycles. Fig. 2.29 shows the loss and pumping fractions and the final discharge outflow temperature and the overall efficiency as a function of $f_{\text{insulation}}$. It can be seen that the insulation can be reduced substantially without significant increases in the losses, decreases in outflow temperature, and overall efficiency. Because of the comparatively low temperatures at the bottom of the storage, even removing the bottom insulation does not cause a significant increase in the bottom losses. The pumping energy fraction decreases slightly when the insulation is removed because of the lower temperatures in the storage and the consequent reduced viscosity. The soil temperature 20 cm from the tank wall after 15 consecutive cycles is shown in Fig. 2.30 for three axial locations as well as underneath the storage as a function of the insulation thickness factor. Again, it can be seen that the insulation thickness can be reduced significantly without affecting the soil temperature.
Fig. 2.29: Thermal and pumping losses (a) and final discharge outflow temperature and overall efficiency (b) as a function of insulation thickness factor for the 15th cycle.
Fig. 2.30: Increase in soil temperature with respect to the initial temperature 20 cm from the lateral walls as a function of insulation thickness factor for the 15th cycle for different positions.

Cover insulation – The same study was carried out for the cover insulation with the insulation thickness factor for the lateral walls and the bottom set equal to 1. Fig. 2.31a shows the cover and pumping losses as a function of $f_{\text{insulation,cover}}$ after 15 consecutive cycles. Similar to the wall and bottom insulation, the cover insulation can be reduced significantly without a major influence on the losses. The wall and bottom losses are found not to be affected. The final discharge outflow temperature and the overall efficiency as a function of $f_{\text{insulation,cover}}$ are sown in Fig. 2.31b. As before, a significant reduction of the insulation does not have a strong effect on the performance.
Fig. 2.31: Cover and pumping losses (a) and final discharge outflow temperature and overall efficiency (b) as a function of cover insulation thickness factor for the 15th cycle.
2.7.4 TES Cone Angle

One of the design characteristics of the TES investigated in this work is its truncated conical shape that entails mechanical advantages and reduces the necessary storage height, as shown in section 2.5. Here the effect of the cone angle ranging from 0° (a cylindrical tank) to 30° on the performance parameters is investigated. The volume and height are kept constant for all cone angles by varying the radius at the bottom and top of the TES. The dimensions of the investigated cases are shown in Table 2.14. The rock diameter was reduced to 1.5 cm to ensure maximum Biot numbers that are small enough to satisfy the assumption of negligible intra-particle temperature gradients. Fig. 2.32 shows the solid temperature profile in the packed bed for different cone angles at the end of the 15th charging period. With increasing cone angle, the increased volume near the top of the tank stores a larger amount of the incoming energy. Hence, the axial temperature profile drops faster and the thermocline becomes less advanced. This causes the height and volume required to store a specified amount of thermal energy to be reduced with increasing cone angle. While at the end of the 15th cycle about 17 m of the height of a cylindrical tank undergoes a temperature increase, for a cone angle of 30° this height is reduced to about 13 m. Fig. 2.33a shows the thermal and pumping losses as a function of the cone angle for the 15th cycle. The cover losses increase with increasing cone angle due to the larger lid area. The opposite trend applies to the bottom losses. The lateral wall losses decrease slightly with increasing cone angle due to the larger volume-to-surface ratios near the hottest region at the top. Hence, the total thermal losses increase only slightly with increasing cone angle. Up to a cone angle of 20°, the decreasing velocity at the top, where

<table>
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<th>Cone angle [deg]</th>
<th>( r_{\text{top}} ) [m]</th>
<th>( r_{\text{bottom}} ) [m]</th>
<th>( H ) [m]</th>
</tr>
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<tr>
<td>0°</td>
<td>18.04</td>
<td>18.04</td>
<td>25</td>
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<tr>
<td>10°</td>
<td>20.20</td>
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<td>25</td>
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<td>20°</td>
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<tr>
<td>30°</td>
<td>24.77</td>
<td>10.33</td>
<td>25</td>
</tr>
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</table>
Fig. 2.32: Solid temperature in the packed bed for various cone angles at the end of the 15\(^{th}\) charging period.

The air viscosity is highest, outweighs the effect of the increasing velocity at the bottom and hence the overall pressure drop decreases compared to a cylindrical tank. For a cone angle of 30\(^{\circ}\), the increased velocities at the bottom caused by the smaller cross section outweigh the decreased velocities at the top and the overall pressure drop increases. Fig. 2.33b shows the final discharge outflow temperature and the overall efficiency as a function of cone angle for the 15\(^{th}\) cycle. The outflow temperature for larger cone angles drops to lower temperatures due to the larger lid area and the consequently larger cover losses. Therefore, the overall efficiency drops slightly with increasing cone angle. It is concluded that the cone angle should be determined for each specific design: the efficiency and final discharge outflow temperature decrease with increasing cone angle, but so do the necessary height and volume and therefore storage material as well as the excavation cost.
Fig. 2.33: Thermal and pumping losses (a) and final discharge outflow temperature and overall efficiency (b) as a function of cone angle for the 15th cycle.
2.7.5 Rock Diameter

To assess the effect of the rock diameter on the TES performance, simulations were carried out with rock diameters between 10 and 40 mm. The remaining parameters are as given in Table 2.13. The maximum Biot number remains below 0.2 even for the largest rock diameter. Fig. 2.34a shows the pumping losses as a function of rock diameter for the 15th cycle. The pumping loss fraction increases nine-fold from 0.25% for \( d = 40 \) mm to 2.25% for \( d = 10 \) mm according to Eq. (2.38). Fig. 2.34b shows the final discharge outflow temperature and the overall efficiency as a function of rock diameter for the 15th cycle. Both decrease with increasing rock diameter due to worse heat transfer between the fluid and the solid (See Table 2.7 for dependence of convective heat transfer coefficient on particle diameter). Fig. 2.35 depicts the solid temperature profile at the end of the 15th charging period as a function of the rock diameter. The thermocline becomes steeper with decreasing rock diameter, because of the improved heat transfer between the fluid and solid phases. Similar observations have been made by other authors [64]. It is concluded that the rock diameter should be optimized for high final outflow temperature and efficiency on the one hand and low pumping losses on the other.
Fig. 2.34: Thermal and pumping losses (a) and final outflow temperature of discharging and overall efficiency (b) as a function of particle diameter for the 15th cycle.
For TES units with a cylindrical shape, it is often unclear what diameter-to-height ratio performs best. With the model presented above, the effect of the diameter-to-height ratio on the TES performance can be investigated. As in the study on the cone angle, the rock diameter was reduced to 1.5 cm to ensure small Biot numbers. The $D/H$ ratios investigated and the corresponding storage height and diameters are shown in Table 2.15. All other parameters are the same as in Table 2.13. The total storage volume is equal for all cases. Fig. 2.36a shows the thermal and pumping losses as a function of $D/H$ for the 15th cycle. With increasing diameter to height ratio, the lateral wall losses decrease due to the increased ratio of storage volume to surface area of the lateral walls, while the cover and bottom losses increase due the larger lid and bottom areas. The total losses increase slightly with increasing $D/H$. The pumping losses increase strongly with decreasing $D/H$ due to decreased cross-sectional area and increased bed length, resulting in a nine-fold increase from $D/H = 2$ ($f_{pump} = 0.75\%$) to $D/H = 0.5$ ($f_{pump} = 6.75\%$). The final discharge outflow temperature and

![Solid temperature profile in packed bed after the 15th charging period for different particle diameters.](image)

**2.7.6 $D/H$ ratio for a Cylindrical Tank**
overall efficiency as a function of $D/H$ are shown in Fig. 2.36b. Both decrease with increasing $D/H$ due to the increasing cover losses as well as reduced convective heat transfer caused by the decreasing mass flow rate per unit cross section (see Table 2.7 for dependence of the convective heat transfer coefficient on mass flow rate per unit cross section). Hence, like the rock diameter, the tank diameter-to-height ratio needs to be chosen as a compromise between pumping loss and outflow temperature drop.

<table>
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<th>0.75</th>
<th>1</th>
<th>1.25</th>
<th>1.5</th>
<th>1.75</th>
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<td>$D$ [m]</td>
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<td>29.00</td>
<td>31.92</td>
<td>34.39</td>
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<td>38.47</td>
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<td>$H$ [m]</td>
<td>50.68</td>
<td>38.67</td>
<td>31.92</td>
<td>27.51</td>
<td>24.36</td>
<td>21.98</td>
<td>20.11</td>
</tr>
</tbody>
</table>
Fig. 2.36: Thermal and pumping losses (a) and final outflow temperature of discharging and overall efficiency (b) as a function of tank diameter to height ratio for the 15\textsuperscript{th} cycle.
2.8 Summary

The work in this chapter was aimed to obtain commercially exploitable knowledge for the scale-up design and fabrication of packed bed of rocks thermal energy storage systems. Hence, the experimental work and numerical modeling is developed to serve this purpose.

A dynamic unsteady numerical heat transfer model that solves the unsteady energy equations for the fluid and solid phases was developed. In contrast to pertinent models used in literature that are mainly an extended version of the original analytic work by Schumann [59], the model in this work is derived from the energy equation and accounts for thermal losses from the lateral walls, the lid and the bottom, variable fluid and solid properties – which were experimentally measured –, axially variable tank diameter and void fraction, and axial dispersion by conduction and radiation. The thermal losses through the lid – under consideration of meteorological conditions like solar insolation and wind – and through the bottom showed to be as significant as the lateral wall losses in their magnitude, which are generally the only source of irreversibility considered in literature. Due to the direct dependence of the thermal capacity of the storage on the void fraction, the experimentally measured axial void fraction distribution helps to create a more precise representation of the experimental setup.

The model was validated with experimental results from a 6.5 MWh\textsubscript{th} pilot-scale thermal energy storage. Its design and fabrication addresses typical issues for this type of storage systems such as tank deformation and rock fracture, by having a truncated conical shape, multi-layer concrete walls, and being immersed in the ground. The inclined walls reduce the force acting on them as well as among the rocks, by guiding the rocks upwards during thermal expansion. The ultra-high performance concrete on the inner side of the walls can withstand high temperatures and forces, and hence avoids deformation issues. The low density concrete on the outer side reduces the thermal losses. Subterranean placement of the tank helps exploiting the lateral earth pressure and hence increases its load bearing. No
systematic failure or fracturing of the rocks or the tank walls was observed after dismantling. Thus, these measures are essential for moving towards a commercially viable packed bed of rocks thermal energy storage system.

The validated model was applied in the design of two industrial-scale TES units. A 100 MWh$_{th}$ thermal storage for a 3 MW$_{th}$ concentrated solar power plant currently under construction in Ait Baha, Morocco, showed to achieve efficiencies of 89%, while an array of two 7.2 GWh$_{th}$ TES units for a theoretical 26 MW$_{el}$ round-the-clock CSP plant showed to achieve efficiencies of 96%.

Studies were carried out that can serve as a guideline for the design of packed bed TES units. It was shown that initial charging of the TES can increase the efficiency and decrease the temperature drop during discharging by more than 40 °C after 25 cycles. The disadvantage of initial charging is a more advanced thermocline in the packed bed. Using an unsteady one-dimensional conduction model, it was shown that the maximum temperature increase in the soil 20 cm from the storage is 17 °C after 60 cycles (days). The thermal losses were shown not to be affected significantly. An insulation thickness of 20 cm for the lid and 16 cm for the lateral walls was shown to be sufficient to ensure low thermal losses. It was found that the insulation at the bottom can be omitted without affecting the thermal losses. Increasing the tank cone angle lowered the final discharge outflow temperature but raised the thermocline, allowing smaller TES height. Decreasing the rock diameter resulted in a strong increase in pumping losses, but a decrease in the drop of the final outflow temperature. Finally, increasing the diameter-to-height ratio of a cylindrical tank decreased the pumping losses, but also caused the final discharge outflow temperature to drop and the thermal losses to increase. These studies indicate that there are no general design rules for packed-bed TES units. Each unit must be optimized for performance (efficiency, outflow temperature profile) and costs (thermal losses, pumping work, excavation, and material costs) given the requirements (such as capacity, charging and discharging duration, mass flow rate and temperature).
2.9 Conclusion

Based on the work presented in this chapter, the following conclusions are drawn:

- The design characteristics applied in the construction and fabrication of the pilot-scale TES used in the experimental setup showed to yield a solution that is adequate for industrial scale applications, since no systematic damage to the walls and rocks was observed after disassembly.
- Temperature-dependent thermo-physical properties of the fluid and solid, axial dispersion by radiation and conduction, and thermal losses from the lateral walls as well as the lid and the bottom need to be considered in the simulation of packed bed TES units.
- The quasi-one-dimensional model used in this chapter yields a fast and efficient design tool for optimizing large-scale TES units.
- Industrial scale thermocline packed bed TES units achieve overall efficiencies of up to 96%.
- The typical temperature drop during discharging of thermocline TES units can be drastically reduced by conducting initial charging runs on the storage before commissioning.
- For industrial scale TES units, the temporal variation of the surrounding ground temperature can be omitted in the design process without affecting performance parameters.
3 Combined Sensible and Latent Heat Energy Storage\textsuperscript{7,8}

3.1 Introduction\textsuperscript{9}

As shown in chapter 2, thermal energy storage in a packed bed of rocks is a promising solution for high-temperature applications with air as heat transfer fluid. However, an inherent disadvantage of the thermocline-type sensible heat storage is the drop of the outflow air temperature toward the end of the discharge period. Fig. 3.1 shows the outflow temperature of discharging as a function of time for the 7.2 GWh\textsubscript{th} TES unit investigated in section 2.6.2 for the 30\textsuperscript{th} cycle, when the storage unit has already reached its steady behavior. As can be seen in the graph, the outflow temperature drops by more than 50 °C during the discharging phase. Such a large variation might be: 1) unacceptable, if the downstream application involves a thermal or thermochemical reaction taking place at a certain temperature; or, 2) unfavorable, if the downstream application is a Rankine or Brayton cycle that will not be able to operate at its design point throughout the discharging phase [114, 115]. Another disadvantage of the thermocline TES is associated to the low capacity ratio, defined as the ratio of heat stored to the

\textsuperscript{7} Material from this chapter has been published in: Zanganeh G., Commerford M., Haselbacher A., Pedretti A., Steinfeld A., “Stabilization of the outflow temperature of a packed-bed thermal energy storage by combining rocks with phase change materials”, \textit{Applied Thermal Engineering}, 70(1), pp. 316-320, 2014.

\textsuperscript{8} Material from this chapter has been submitted for publication: Zanganeh G., Khanna R., Walser C., Pedretti A., Haselbacher A., Steinfeld A., “Experimental and numerical investigation of combined sensible-latent heat for thermal energy storage at 575 °C and above”, \textit{Solar Energy}, submitted 2014.

\textsuperscript{9} Preliminary studies on the concept carried out in the framework of a master thesis by M. Commerford, supervised by G. Zanganeh.
maximum theoretical thermal capacity, but this drawback can be tolerated for low-cost aggregate materials such as gravel.

Latent heat storage using phase change materials (PCM), on the other hand, has somewhat higher energy density and can store and release heat at constant temperature. It has been studied for various applications, such as heating and cooling of buildings, ice storage, outer space missions, refrigerated cargo transport, greenhouses, low and high-temperature solar energy systems, and thermal storage for nuclear and fossil-fuel fired power plants as well as industrial waste heat [116-123]. Several recent studies have reviewed latent heat storage with PCM: Zalba et al. [120] give a review on materials, heat transfer analysis and applications of latent heat storage and summarize the commercially available PCMs; Kenisarin and Mahkamov [124] review materials and heat transfer enhancement methods for low-temperature \((T < 200\, ^\circ C)\) solar energy storage; Farid et al. [125] review phase change materials and applications for low-temperature energy storage and point out typical problems associated with phase change materials; Nkwetta and Haghighat [13] review the literature for hot water tanks and transpired solar collectors using PCM for domestic use; Fernandes et al.
focus on inorganic salts for high-temperature latent heat storage and review heat transfer enhancement techniques; Agyeni et al. [127] review materials, heat transfer enhancement techniques, geometries and configurations for latent heat energy storage and summarize the numerical models and validations in literature for phase change problems; Kenisarin et al. [128] and Cárdenas and León [129] give an extensive summary of high-temperature latent heat storage materials, covering salts, salts compositions and metallic alloys; Dutil et al. [130] review mathematical models and simulations for phase change materials. The major part of the experimental work found in the literature concerns paraffin wax as PCM for low-temperature TES [15, 123, 131-133]. High-temperature experimental studies (\(T > 500 \, ^\circ C\)) include the work by Yagi and Akiyama that compare inorganic salts with metallic alloys [122], Jalalzadeh-Azar et al. that use a salt/ceramic composite (\(\text{Na}_2\text{SO}_4/\text{SiO}_2\)) [134], as well as a series of recent studies that use a metallic alloy of aluminum and silicon [135-137]. A suitable phase change material should exhibit the following properties [128]: 1) adequate melting temperature for the specific application; 2) high latent heat of fusion for decreasing the material amount; 3) high thermal conductivity for rapid heat transfer rate across the encapsulation and for homogeneous melting/solidification; 4) low supercooling\(^{10}\) and a small phase transition temperature range; 5) small volume expansion for reducing the risk of damaging the encapsulation; and 6) chemical stability, non-toxicity and compatibility with the encapsulation material.

The main disadvantages of latent heat energy storage systems are the high cost of the phase change material and its inability to deal with heat within a large temperature range. In fact, some studies comparing sensible and latent heat storage concluded that there is no significant improvement of the storage performance when sensible heat storage material is replaced by latent heat storage material [13, 134, 139]. Hence, several authors

\(^{10}\) Supercooling or subcooling is the drop of temperature of a liquid or gas below its solidification point without becoming a solid [138].
suggested a cascaded PCM configuration using PCM with different melting temperatures to improve the storage performance [52, 139-141], but at the expense of complicated designs and higher costs.

In this chapter, a concept is suggested and experimentally investigated where most of the energy is stored in a thermocline storage using readily available rocks as sensible heat storage material, while a relatively small layer of PCM – with a melting temperature around the inlet temperature of the TES – placed on top of the rocks helps stabilize the outflow temperature during the discharging. Hence, the concept combines the advantages of the sensible and latent heat storages while alleviating the critical issues incurred when using them separately. The design concept is shown schematically in Fig. 3.2. During charging, hot air enters the TES from the top, transfers heat to the PCM and rocks, and exits at the bottom. During discharging the flow is reversed: air enters from the bottom, is heated by the rocks and PCM, and exits at the top. The direction of the flow exploits buoyancy forces to create and maintain thermal stratification, with the hottest region at the top of the storage tank. As it will be shown in the numerical and experimental analysis that follows, the proposed combination of sensible and latent heat stabilizes the outflow air temperature around the PCM’s melting point. This TES concept is designed for incorporation with the CSP trough collector technology developed by Airlight Energy SA that can deliver air at up to 650 °C [26].

Previous work that has combined the concepts of sensible and latent heat energy storage includes a TES for direct steam generation that employs a sandwich concept, where sensible heat storage in concrete is used to heat water and superheat steam and a latent heat storage with NaNO$_3$ as PCM is used in between to evaporate the water [142], and an experimental investigation of a low-temperature TES using a container filled with encapsulated paraffin as PCM and hot water supplied by a solar flat plate collector that is also acting as the sensible heat storage medium [131]. However, the proposed combination of sensible and latent heat storage to stabilize the outflow temperature is novel.
3.2 Modeling

The heat transfer model is formulated for the two sections of the TES, namely sensible and latent heat sections, separately. The two domains are then connected by the conditions of the air flow at their interface, i.e. the outlet flow enthalpy of air from one domain is set as the inlet flow enthalpy for the other domain, e.g. $h_{f,\text{out,PCM}} = h_{f,\text{in,rocks}}$ during charging.

*Sensible heat section* — The sensible heat section of the TES, i.e., the packed bed of rocks, is simulated using the 1D dynamic finite-volume heat transfer model introduced in section 2.2. A brief outline is included here. The model considers separate fluid and solid phases with variable thermophysical properties and axial dispersion by conduction and radiation. The governing equations are,

**Fluid:**

$$
\varepsilon \rho_f \frac{d e_f}{d t} = \frac{A \varepsilon}{V} \left[ \left( u \rho_f h_f \right)_{in} - \left( u \rho_f h_f \right)_{out} \right] + h v (T_s - T_f) \quad \text{(3.1)}
$$
Solid:

\[
(1 - \varepsilon) \rho_s \frac{de_s}{dt} = h_v(T_f - T_s) + \frac{A}{V} \left[ k_{eff} \frac{dT_x}{dx} \bigg|_{out} - k_{eff} \frac{dT_x}{dx} \bigg|_{in} \right]
\]  

(3.2)

with \(A\) and \(V\) the tank cross section and volume for each layer, respectively. Other symbols are defined in the nomenclature. Physical properties and heat transfer correlations applied can be found in section 2.3.

**Latent heat section** — The latent heat section of the TES, i.e., the top layer containing encapsulated PCM, is modeled following the approach of Beasley et al. [133]. The model considers separate fluid and solid phases. PCM properties are assumed to be independent of temperature. The governing equations are:

Fluid:

\[
\varepsilon \rho_f \frac{de_f}{dt} = \frac{A\varepsilon}{V} \left[ (u \rho_f h_f) \bigg|_{in} - (u \rho_f h_f) \bigg|_{out} \right] + h_{v,eff,PCM}(T_{PCM} - T_f) + \frac{A\varepsilon}{V} \left[ k_{eff,PCM} \frac{dT_f}{dx} \bigg|_{out} - k_{eff,PCM} \frac{dT_f}{dx} \bigg|_{in} \right]
\]  

(3.3)

PCM:

Solid phase:

\[
(1 - \varepsilon)(\rho C)_{eff,solid} \frac{dT_{PCM}}{dt} = h_{v,eff,PCM}(T_f - T_{PCM})
\]  

(3.4)

Two-phase:

\[
(1 - \varepsilon)(\rho h)_{eff,fus} \frac{d\chi}{dt} = h_{v,eff,PCM}(T_f - T_{PCM})
\]  

(3.5)

Liquid phase:

\[
(1 - \varepsilon)(\rho C)_{eff,liquid} \frac{dT_{PCM}}{dt} = h_{v,eff,PCM}(T_f - T_{PCM})
\]  

(3.6)
where $\chi$ is the PCM’s quality that represents the molten fraction ranging from 0 to 1. For simplicity, the encapsulation material is not considered separately. Instead, its properties are incorporated into effective properties of the encapsulated PCM using the average volumetric heat capacity of the PCM and the encapsulation,

\[
(\rho C)_{\text{eff}} = \frac{\rho_{\text{PCM}} C_{\text{PCM}} V_{\text{PCM}} + \rho_{\text{enc}} C_{\text{enc}} V_{\text{enc}}}{V_{\text{total}}} \tag{3.7}
\]

When both phases are present (Eq. (3.5)), only the PCM properties are considered since the encapsulation does not contribute to the latent heat energy storage

\[
(\rho h)_{\text{eff, fus}} = \frac{\rho_{\text{PCM}} h_{\text{fus}} V_{\text{PCM}}}{V_{\text{total}}} \tag{3.8}
\]

The numerical solution method is analogous to that used for the sensible heat section.

### 3.3 Closure Relations

The convective heat transfer coefficient for the latent heat section is calculated using the correlation of Galloway and Sage [143], with parameters determined from data measured by Beasley and Clark [34],

\[
\text{Nu}_{\text{PCM}} = 2 + 2.03 \text{Re}_0^{1/2} \text{Pr}^{1/3} + 0.049 \text{Re}_0 \text{Pr}^{1/2} \tag{3.9}
\]

and adjusted to consider the intra-particle conduction [96] with

\[
h_{p,\text{eff, PCM}} = \frac{h_{p,\text{PCM}}}{1 + 0.25 \text{Bi}} \tag{3.10}
\]

For the sensible heat section the correlation by Pfeffer is used (See Table 2.7) and adjusted with Eq. (3.10) as well. The particle and volumetric convective heat transfer coefficients are related through $h_v = 6h_p(1-\varepsilon)/d$. The effective thermal conductivity of the latent heat section is given by [99]
\[
\text{Pe}_{\text{eff}} = \frac{\text{Pe}_0}{0.5\text{Pe}_0 + \frac{k_{0f}}{k_f}}
\]  

Grid refinement was carried out separately for the sensible and latent heat sections with grid spacing of 0.05 m and 0.01 m, respectively, giving good accuracy.

### 3.4 Simulation Results

**Model Validation** – The model of the sensible heat section was validated with experimental data obtained from a 6.5 MWh\textsubscript{th} pilot-scale thermal storage unit as presented in section 2.4. To test the implementation of the model of the latent heat section, the results of the cylindrical-storage experiment of Beasley et al. [133] using encapsulated paraffin wax are used. The relevant dimensions and properties are taken from Ref. [133] and listed in Table 3.1. Note that the high value of the specific heat of the PCM in the solid phase is ascribed to crystal transitions prior to melting [133]. The encapsulation material is not considered in the reference study. In Fig. 3.3,

<table>
<thead>
<tr>
<th>PCM Properties</th>
<th>Dimensions and Operating Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>$C_{\text{PCM, solid}}$ [J/kgK]</td>
<td>$D_{\text{tank}}$ [m] 0.208</td>
</tr>
<tr>
<td>$C_{\text{PCM, liquid}}$ [J/kgK]</td>
<td>$H_{\text{tank}}$ [m] 0.278</td>
</tr>
<tr>
<td>$\rho_{\text{PCM}}$ [kg/m$^3$]</td>
<td>$d_{\text{PCM}}$ [m] 0.02</td>
</tr>
<tr>
<td>$h_{\text{fus}}$ [kJ/kg]</td>
<td>$\epsilon$ [-] 0.369</td>
</tr>
<tr>
<td>$k_{\text{PCM}}$ [W/mK]</td>
<td>$m_{\text{charging}}$ [kg/s] 0.0328</td>
</tr>
<tr>
<td>$T_{\text{melt}}$ [°C]</td>
<td>$T_{\text{charging}}$ [°C] 59.5</td>
</tr>
</tbody>
</table>

* The high value of the specific heat of the PCM in the solid phase is ascribed to crystal transitions prior to melting [133].
the PCM temperature computed from Eqs. (3.4)-(3.6) is plotted as a function of time and compared with the model and experimental data of Ref. [133]. The agreement between the two models and with the measured data is reasonably good. Having validate the model, simulations were carried out for a TES baseline unit of cylindrical geometry containing a packed bed of rocks and PCM (Fig. 3.2). The unit is operated in consecutive cycles of 5 h-charging/5 h-discharging each. Dimensions and operating conditions are listed in Table 3.2. The charging temperature of 650 °C is the design outflow air temperature of Airlight Energy’s solar parabolic trough concentrator [26, 48, 49]. A solar power plant based on this concentrator will be operated such that the air fed to the TES is at the design outflow temperature of the concentrator. This is achieved by ramping the mass flow during the charging and discharging phases. To simulate the ramping during charging, we have chosen a sinusoidal variation of the mass flow rate
\[ \dot{m} = \frac{\dot{m}_{\text{charging, nom}}}{2} \left[ \sin \left( \frac{\pi t}{t_{\text{ramp}}} - \frac{\pi}{2} \right) + 1 \right] \]  
\text{for } 0 < t < t_{\text{ramp}}

\[ \dot{m} = \dot{m}_{\text{charging, nom}} \]  
\text{for } t_{\text{ramp}} < t < t_{\text{charging}} - t_{\text{ramp}}

\[ \dot{m} = \frac{\dot{m}_{\text{charging, nom}}}{2} \left[ \sin \left( \frac{\pi (t - (t_{\text{charging}} - t_{\text{ramp}}))}{t_{\text{ramp}}} + \frac{\pi}{2} \right) + 1 \right] \]  
\text{for } t_{\text{charging}} - t_{\text{ramp}} < t < t_{\text{charging}}

where the values of charging and ramp times are given in Table 3.2. Similar ramps are used for the discharging phase. The discharging temperature corresponds to the typical outflow temperature of a conventional power block [113].

Three PCM candidates are introduced in Table 3.3, that have melting temperatures around the charging temperature [117, 128]. The heat capacity, \( C_{\text{PCM}} \), is assumed not to vary from liquid to solid phase. The conductivity, \( k_{\text{PCM}} \), is taken as 1 W/mK for all three materials because of lack of detailed information. This value is plausible because \( k_{\text{PCM}} \) for salt mixtures ranges from 0.6 to 2 W/mK [117]. The precise value of \( k_{\text{PCM}} \) in this range was found not to affect the results significantly. Due to the low conductivity of the PCM and rocks, which results in large Biot numbers (\( \text{Bi}_{\text{max,PCM}} = 0.56, \text{Bi}_{\text{max,rocks}} = 0.25 \)), the convective heat transfer coefficient in both sections is adjusted to include the effects of the intra-particle conduction using Eq. (3.10). The encapsulation is considered to be made of AISI 316 stainless steel and to have a thickness of 1 mm. Relevant properties are shown in Table 3.4 [72]. PCM bed heights of 0.05, 0.1, and 0.2 m were investigated for each of the materials listed in Table 3.3. These values correspond to volumes of 0.67, 1.33, and 2.67% of the total volume of the TES. For each case, the height of the packed bed of rocks is reduced by the height of the packed bed of PCM to keep the total height constant.
### Table 3.2: Dimensions and operating conditions of the TES baseline unit.

<table>
<thead>
<tr>
<th>Dimensions and Rock Properties</th>
<th>Operating Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_{tank}$ [m]</td>
<td>$t_{charging}$ [h]</td>
</tr>
<tr>
<td>$H_{tank}$ [m]</td>
<td>$t_{discharging}$ [h]</td>
</tr>
<tr>
<td>$C_{rocks}$ [J/kgK]</td>
<td>$t_{ramp}$ [h]</td>
</tr>
<tr>
<td>$\rho_{rocks}$ [kg/m$^3$]</td>
<td>$T_{charging}$ [°C]</td>
</tr>
<tr>
<td>$d_{rocks}$ [m]</td>
<td>$T_{discharging}$ [°C]</td>
</tr>
<tr>
<td>$\varepsilon_{rocks}$ [-]</td>
<td>$m_{charging}$ [kg/s]</td>
</tr>
<tr>
<td></td>
<td>$m_{discharging}$ [kg/s]</td>
</tr>
</tbody>
</table>

### Table 3.3: PCM considered for the latent heat section of the TES [117, 128] and their properties. The numbers in parentheses represent the percentage weights in the mixture.

<table>
<thead>
<tr>
<th>Material #</th>
<th>1</th>
<th>2</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Material</td>
<td>NaF(27) + NaBr(73)</td>
<td>LiF(46) + NaF(44) + MgF$_2$(10)</td>
<td>LiF(52) + NaF(35) + CaF$_2$(13)</td>
</tr>
<tr>
<td>Composition</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$T_{melt}$ [°C]</td>
<td>642</td>
<td>632</td>
<td>615</td>
</tr>
<tr>
<td>$h_{fus}$ [kJ/kg]</td>
<td>360</td>
<td>858</td>
<td>640</td>
</tr>
<tr>
<td>$\rho_{PCM}$ [kg/m$^3$]</td>
<td>3033*</td>
<td>2610</td>
<td>2630</td>
</tr>
<tr>
<td>$C_{PCM}$ [J/kgK]</td>
<td>849*</td>
<td>1846*</td>
<td>1797*</td>
</tr>
<tr>
<td>$\varepsilon_{PCM}$ [-]</td>
<td>0.37</td>
<td>0.37</td>
<td>0.37</td>
</tr>
<tr>
<td>$d_{PCM}$ [m]</td>
<td>0.02</td>
<td>0.02</td>
<td>0.02</td>
</tr>
<tr>
<td>$k_{PCM}$ [W/mK]</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

* values calculated using properties of the constituents found in the literature [144, 145].

### Table 3.4: Properties of the encapsulation material [72].

<table>
<thead>
<tr>
<th>Encapsulation Material</th>
<th>$\rho_{enc}$ [kg/m$^3$]</th>
<th>$C_{enc}$ [J/kgK]</th>
<th>$k_{enc}$ [W/mK]</th>
<th>$t_{enc}$ [m]</th>
</tr>
</thead>
<tbody>
<tr>
<td>AISI 316</td>
<td>8238</td>
<td>535</td>
<td>18.8</td>
<td>0.001</td>
</tr>
</tbody>
</table>
As shown in chapter 2, the time to reach a steady cyclic performance of the TES can be reduced by initially charging the storage prior to the beginning of the cyclic operation. For the simulations reported here, the storage was pre-charged for 6.5 hours with a similar start-up and shut-down ramp as during the cycling. After 9 consecutive charging/discharging cycles, the outflow temperature profile reached a steady cyclic behavior for all cases investigated.

Fig. 3.4 shows the outflow air temperature as a function of time during discharging of the 9th cycle for three different PCM heights (0.05, 0.1, and 0.2 m) on top of the packed bed of rocks (indicated as “Rocks + Material #”). Curves are shown for the three PCMs of Table 3.3. Also shown is the curve for the packed bed of rocks without PCM (indicated as "Rocks only"). Due to the small heat of fusion and the high melting temperature of material #1, the outflow temperature was not stabilized for any of the investigated PCM heights, as it dropped drastically after 50% of the discharging time to temperatures below the “rocks only” case. Conversely, because of the comparatively low melting temperature of material #3, the outflow temperature stabilized at the very end of the discharging phase around the melting temperature and no significant advantage was observed with respect to the “Rocks only” case. Material #2, having a suitable melting temperature and heat of fusion, showed a stabilization of the outflow temperature around its melting point for the entire discharging period. While the temperature dropped slowly with time for a PCM height of 0.05 m, it stabilized for 0.1 m. Thus, increasing the PCM height to 0.2 m was not beneficial. After 4.5 h of discharging, the mass flow rate begins to decrease according to Eq. (3.12) and consequently the heat-extraction rate decreases. For the “Rocks only” and “Rocks + Material #1” case where all the PCM is already solidified, this causes the temperature drop rate to decrease. For materials #2 and #3 with a PCM height of 0.05 m, liquid PCM is still available but is not sufficient to stabilize the outflow temperature at the nominal mass flow rate. Here the decreasing heat-extraction rate causes the outflow temperature to increase and approach the melting temperature of the respective PCM. However, note that the extracted heat at the end of the
Fig. 3.4: Outflow air temperature as a function of time during discharging of the 9\textsuperscript{th} cycle for various PCM heights on top of the packed bed of rocks (“Rocks + Material #”), and for the packed bed of rocks without PCM (“Rocks only”). The three PCMs and their properties are listed in Table 3.3.

discharging is approaching zero. The thermal capacity of the entire bed for material #2, defined as the total energy stored in the bed when charged from ambient temperature to isothermal conditions at the inlet air flow temperature of 650 °C, was 54 MWh\textsubscript{th} for $H_{PCM} = 0.1$ m and $H_{rocks} = 7.4$m.

Fig. 3.5 shows the variation of the PCM quality or molten fraction as a function of the height of material #2 at the end of the 9\textsuperscript{th} charging period. For a PCM height of 0.2 m, only the first 9 cm were completely molten, which explains why increasing the PCM height beyond 0.1 m did not improve the TES performance. Further insight can be gained by considering the fractions of latent and sensible heats stored in the PCM compared to the total stored thermal energy. These fractions are defined by

$$f_{\text{latent,PCM}} = \frac{E_{\text{latent,PCM}}}{E_{\text{rocks}} + E_{\text{total,PCM}}}$$

(3.13)

$$f_{\text{sensible,PCM}} = \frac{E_{\text{sensible,PCM}}}{E_{\text{rocks}} + E_{\text{total,PCM}}}$$

(3.14)

where $E_{\text{latent,PCM}}$ and $E_{\text{sensible,PCM}}$ are the latent and sensible heats stored in
Combined Sensible and Latent Heat Energy Storage

Fig. 3.5: Variation of the PCM quality (molten fraction) as a function of the PCM height of material #2 at the end of the 9th charging period.

the PCM section, respectively, $E_{total,PCM}$ is the sum of the latent and sensible heats stored in the PCM section, and $E_{rocks}$ the total thermal energy stored in the rocks section.

Table 3.5 lists $f_{latent,PCM}$ and $f_{sensible,PCM}$ at the end of the 9th charging period for three PCM heights of material #2. While $f_{sensible,PCM}$ increased proportionally to the PCM volume fraction, $f_{latent,PCM}$ did not show such a behavior when increasing the height from 0.1 to 0.2 m, indicating that the PCM was not exploited as it was used as sensible heat storage only.

Table 3.5: Fraction of latent and sensible heats stored in the PCM section relative to the total stored thermal energy at the end of the 9th charging period for three PCM heights of material #2.

<table>
<thead>
<tr>
<th>$H_{PCM}$ [m]</th>
<th>$f_{latent,PCM}$ [%]</th>
<th>$f_{sensible,PCM}$ [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.05</td>
<td>2.38</td>
<td>3.37</td>
</tr>
<tr>
<td>0.1</td>
<td>4.35</td>
<td>6.51</td>
</tr>
<tr>
<td>0.2</td>
<td>4.71</td>
<td>12.65</td>
</tr>
</tbody>
</table>
The overall efficiency of the storage is defined as \( \eta_{\text{overall}} = \frac{E_{\text{recovered}}}{E_{\text{input}}} \) where \( E_{\text{input}} \) is the integrated input thermal power during charging and \( E_{\text{recovered}} \) is the integrated output thermal power during discharging as defined in section 2.6. Table 3.6 lists \( \eta_{\text{overall}} \) for each material and PCM height as well as for the “rocks only” case \((H_{\text{PCM}} = 0 \text{ m})\). It is seen that the addition of the PCM to the packed bed of rocks practically did not affect the overall efficiency.

Table 3.6: The overall efficiency of the 9th cycle for three PCM heights of the three materials of Table 3.3, and for the “rocks only” case \((H_{\text{PCM}} = 0 \text{ m})\).

<table>
<thead>
<tr>
<th>Material #</th>
<th>(H_{\text{PCM}} = 0 \text{ m})</th>
<th>(H_{\text{PCM}} = 0.05 \text{ m})</th>
<th>(H_{\text{PCM}} = 0.1 \text{ m})</th>
<th>(H_{\text{PCM}} = 0.2 \text{ m})</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>97.34%</td>
<td>97.40%</td>
<td>97.43%</td>
<td>97.44%</td>
</tr>
<tr>
<td>2</td>
<td>97.36%</td>
<td>97.30%</td>
<td>97.24%</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>97.33%</td>
<td>97.25%</td>
<td>96.93%</td>
<td></td>
</tr>
</tbody>
</table>

3.5 Experimental Campaign

In order to experimentally investigate the concept of combined sensible and latent heat storage, a 42.4 kWh lab-scale TES was built and tested. Similar to the concept shown in Fig. 3.2, PCM material was placed on top of a packed bed of rocks (“rocks + PCM”) which was later replaced by rocks (“rocks only”) to compare the two concepts.

3.5.1 Prototype Design

Choice of Phase Change and Encapsulation Material

Most of the previous experimental work for high-temperature latent heat storage involves the use of salt compositions as PCM [117, 134]. However, salts and salt mixtures, such as the ones used in the numerical investigation of section 3.4, have low thermal conductivity, large volume expansion and supercooling, are often corrosive towards the encapsulation material, and

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11 Experimental campaign carried out in the framework of a master thesis by R. Khanna, supervised by G. Zanganeh.
have demonstrated phase segregation and poor physical stability [7, 125, 129, 134, 146-148]. Therefore, thermal conductivity enhancement and adequate encapsulation techniques for this group of PCM have been the focus of several studies [149-152]. Metallic PCM in the form of metal alloys, on the other hand, are an attractive solution for high-temperature latent heat storage due to their relatively high thermal conductivity (up to two orders of magnitude higher than that of salts), as well as negligible supercooling and relatively small volume change during melting [135, 147]. However, metallic alloys have not been studied as extensively as salt mixtures. Birchenall and Riechman [147] were one of the firsts to investigate metallic alloys as PCM and compare them with salts. More recently, some authors have investigated a eutectic alloy containing 88% Aluminum and 12% Silicon by mass, frequently denoted as AlSi_{12}. Yagi and Akiyama [122] compared AlSi_{12} as well as AlSi_{26} and Pb to two inorganic salt PCMs and concluded that AlSi_{12} is the most suited PCM for latent heat storage among the materials investigated. He and Zhang [136] simulated and experimentally tested a shell-and-tube heat exchanger using AlSi_{12} as storage material. Wang et al. [137] investigated a high-temperature heater using AlSi_{12} as heat storage medium and compared it to a sensible heat storage using Fe_{3}O_{4}. Kotzé et al. [135, 153] investigated a high-temperature TES using AlSi_{12} as storage material and NaK as heat transfer fluid. Little data is available on AlSi_{12} cycling and corrosive behavior. Li et al. [154] report a 4% drop in the heat of fusion but effectively no change in the melting temperature of Al-Si alloys after 1200 cycles. Sun et al. [155] investigated another Aluminum alloy (Al-34%Mg-6%Zn) and reported a 11% change in the heat of fusion, a 3 °C variation in the melting temperature and negligible corrosion of stainless steel encapsulation (SS304L) after 1000 cycles. With a melting temperature of about 575 °C, it is also an attractive PCM candidate to be incorporated in the TES for Airlight Energy’s collector technology. The Airlight collectors can deliver air at 650 °C. However, in order to account for variations in DNI as well as thermal losses between the collectors and the storage, a phase transition temperature between 560 and 590 °C is found to be adequate.
AlSi\textsubscript{12} is commercial available due to widespread application in Aluminum welding and has significantly lower costs compared to salts and salt mixtures. As encapsulation material, stainless steel AISI 316 was chosen that is adequate for high-temperature applications and has high corrosion resistance. While spherical enclosures are usually used for the encapsulation of PCM, in order to minimize complexity and costs, tubular encapsulation is chosen in this work.

In preliminary tests, a 6.5 cm stainless steel tube with 2 cm diameter was filled with molten AlSi\textsubscript{12} and closed under argon purge welding. The argon purge is to ensure that no air is captured inside the encapsulation that can lead to oxidation of the PCM and/or encapsulation during operation. The PCM filled encapsulation was placed in an oven while thermocouples on the outside and on two positions in the inside of the tube monitored the temperature distribution. Due to the high conductivity of the steel and the PCM, negligible temperature difference was observed across the tube and PCM, indicating very low Biot numbers and uniform melting. Fig. 3.6 shows the phase change regime during a preliminary test. The thermocouple reading accuracy was 1 K.

**Configuration**

The storage schematic and design is shown in Fig. 3.7. The structure consists of a stainless steel tank (AISI 304) of 1680 mm length, 400 mm outer diameter and 3 mm thickness. Perforated plates are welded at a distance of 200 mm from the bottom and 100 mm from the top of the tank for the purpose of creating a uniform air flow. The bottom perforated plate further supports the weight of the storage material. The rocks and encapsulated PCM are placed between the perforated plates. Steel sections with double cones are used to connect the top and bottom of the tank to the smaller delivery tubes. The tank dimensions, as well as the insulation materials, dimensions, and thermal conductivities are listed in Table 3.7. The insulation is thicker at the top than at the bottom to account for the temperature stratification.
Fig. 3.6: Wall and PCM temperatures of the tube used in the preliminary tests. Due to high conductivity of the PCM and encapsulation material, the curves mostly overlap, indicating very low Biot numbers and uniform melting.

Fig. 3.7: Scheme showing relevant dimensions (a) and a 3D rendering (b) of the 42.4 kWh\textsubscript{th} lab-scale prototype used in the experimental campaign.
Table 3.7: Tank and insulation data of the experimental combined storage. Thermal conductivity of the insulation taken from manufacturers’ data sheets.

<table>
<thead>
<tr>
<th>Height [m]</th>
<th>Insulation Thickness [m]</th>
<th>Thermal Conductivity [W/mK]</th>
</tr>
</thead>
<tbody>
<tr>
<td>$x &lt; 0.55$</td>
<td>0.02/0.04/0.1</td>
<td>$k_{\text{microtherm}}$ 0.026-0.038</td>
</tr>
<tr>
<td>$0.55 &lt; x &lt; 0.98$</td>
<td>0/0.06/0.1</td>
<td>$k_{\text{felt}}$ 0.046-0.078</td>
</tr>
<tr>
<td>$0.98 &lt; x &lt; 1.68$</td>
<td>0/0.05/0</td>
<td>$k_{\text{rockwool}}$ 0.038</td>
</tr>
</tbody>
</table>

**Sensible Heat Storage Section**

The sensible heat section of the storage consists of the rocks introduced in section 2.3.2. The average void fraction, $\varepsilon = \frac{V_{\text{water}}}{V_{\text{tank}}}$, was measured by filling the void space with water. The average temperature dependent heat capacity, measured in a differential scanning calorimeter and introduced in section 2.3.2, is used to calculate the temperature dependent internal energy of the rocks with $e_s (T + \Delta T) = e_s (T) + C_s (T) \Delta T$ with a resolution of $\Delta T = 5$ °C. Using the heating or cooling data has negligible effect on the values obtained for the internal energy. The results, with 0 °C as reference, together with the average heat capacity of the rocks as a function of temperature are shown in Fig. 3.8. A second order polynomial was fitted: $e_s = aT + bT^2$, with $T$ in °C, $e_s$ in J/kg, $a=747.0995$ and $b=0.2838$. Relevant dimensions and properties are summarized in Table 3.8.
Combined Sensible and Latent Heat Energy Storage

Fig. 3.8: Average internal energy ($T_{ref} = 0 \, ^\circ C$) and average heat capacity for the heating runs (see Fig. 2.6) as a function of temperature averaged for the seven rock types in the packed bed, measured using a differential scanning calorimeter.

Table 3.8: Dimensions and properties of the sensible and latent heat sections.

<table>
<thead>
<tr>
<th>Latent Heat Section</th>
<th>Sensible Heat Section</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H_{PCM}$ [m]</td>
<td>$H_{rocks}$ [m]</td>
</tr>
<tr>
<td>0.09</td>
<td>1.27/1.36 (“rocks only”)</td>
</tr>
<tr>
<td>$d_{enc}$ [m]</td>
<td>$d_{rocks}$ [m]</td>
</tr>
<tr>
<td>0.018</td>
<td>0.032</td>
</tr>
<tr>
<td>$\varepsilon_{PCM}$ [-]</td>
<td>$\varepsilon_{rocks}$ [-]</td>
</tr>
<tr>
<td>0.549</td>
<td>0.4</td>
</tr>
<tr>
<td>$\rho_{AlSi}$ [kg/m$^3$]</td>
<td>$k_{rocks}$ [W/mK]</td>
</tr>
<tr>
<td>2650</td>
<td>1-5</td>
</tr>
<tr>
<td>$\rho_{AlSi316}$ [kg/m$^3$]</td>
<td>$C_{rocks}$ [J/kgK]</td>
</tr>
<tr>
<td>7930</td>
<td>650-1150 (T-dependent)</td>
</tr>
<tr>
<td>$h_{fus}$ [kJ/kg]</td>
<td>$\rho_{rocks}$ [kg/m$^3$]</td>
</tr>
<tr>
<td>466</td>
<td>2635</td>
</tr>
<tr>
<td>$\Delta T_{melt}$ [K]</td>
<td>$E_{rocks}$ [kWh$_{th}$]</td>
</tr>
<tr>
<td>4</td>
<td>38.2/41 (“rocks only”)</td>
</tr>
<tr>
<td>$E_{PCM}$ [kWh$_{th}$]</td>
<td>$m_{rocks}$ [kg]</td>
</tr>
<tr>
<td>3</td>
<td>~245/262 (“rocks only”)</td>
</tr>
<tr>
<td>$E_{enc}$ [kWh$_{th}$]</td>
<td></td>
</tr>
<tr>
<td>1.2</td>
<td></td>
</tr>
</tbody>
</table>
Latent Heat Storage Section

The latent heat storage section is placed on top of the rocks. This section consists of AlSi$_{12}$, which acts as the PCM, encapsulated in stainless steel (AISI 316) tubes with 16 mm inner diameter, 1 mm wall thickness, and various lengths. The tubes were filled with molten PCM and closed by welding stainless steel caps to their ends under argon purge. They were arranged in 4 rows, each row containing 17 tubes aligned perpendicular to the adjacent row, as shown in Fig. 3.9. This arrangement gives a total latent heat section height of 90 mm with a calculated average void fraction of 0.549. The mass of PCM and encapsulation material as well as the total volume of the encapsulation tubes for each row is listed in Table 3.9.

Reported values of the heat of fusion of AlSi$_{12}$ varied from 460 to 560 kJ/kg [122, 136, 137, 153]. Therefore, the temperature-dependent heat capacity of the AlSi$_{12}$ was measured by DSC$^{12}$. Three consecutive heating and cooling runs with a rate of 15 K/min were carried out in order to ensure repeatability. The results showed negligible difference between the three heating and cooling cycles respectively, but a shift in the phase transition temperature range was observed between the heating and cooling runs. Fig. 3.10 shows the averaged results for the solid, two-phase, and liquid states. The melting onset was observed at 571 °C and 5 °C higher than the solidifying onset at 566 °C. In the solid state, the heat capacity increased

<table>
<thead>
<tr>
<th>Row 1 (top)</th>
<th>Row 2</th>
<th>Row 3</th>
<th>Row 4</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>$m_{PCM}$ [kg]</td>
<td>2.37</td>
<td>2.41</td>
<td>2.39</td>
<td>2.42</td>
</tr>
<tr>
<td>$m_{enc}$ [kg]</td>
<td>3.29</td>
<td>3.29</td>
<td>3.29</td>
<td>3.30</td>
</tr>
<tr>
<td>$V_{enc} \cdot 10^3$ [m$^3$]</td>
<td>1.237</td>
<td>1.238</td>
<td>1.237</td>
<td>1.241</td>
</tr>
</tbody>
</table>

$^{12}$Experimental campaign carried out in the framework of a bachelor thesis by Claudia Walser, supervised by G. Zanganeh.
from 950 J/kgK to 1190 J/kgK. In liquid state, the heat capacity was constant at around 1170 J/kgK. The heat of fusion, obtained by integration, was 466 kJ/kg. The results obtained from the DSC measurements indicate a melting/solidification temperature range of about 50 K, partially ascribed to the large heating/cooling rate of the DSC of 15 K/min. This assumption is supported by the congruent melting and solidification of the PCM in the range of 573-577 °C observed during the experimental campaign. A study by Arkar and Medved [156] on paraffin also shows that variations of the heating rate do not affect the area under the curve and hence the latent heat of fusion, but it can affect the onset and endset temperatures of the phase transition. Table 3.10 lists the heat capacities and thermal conductivities of PCM (AlSi_12) and encapsulation material (AISI 316). Data was taken from literature [72, 137] except the measured heat capacity of AlSi_12. The densities of AlSi_12 and AISI 316 were experimentally measured and are listed in Table 3.10.

The thermal capacity, defined as the total thermal energy stored when charged from ambient temperature to isothermal conditions at 650 °C, was
38.2 kWh\textsubscript{th} for the sensible heat storage section and 4.2 kWh\textsubscript{th} for the latent heat storage section (PCM 3 kWh\textsubscript{th} and encapsulation 1.2 kWh\textsubscript{th}), for a total of 42.4 kWh\textsubscript{th}.

Table 3.10: Heat capacity and thermal conductivity of PCM (AlSi\textsubscript{12}) and encapsulation material (AISI 316).

<table>
<thead>
<tr>
<th>Temperature Range</th>
<th>AlSi\textsubscript{12}</th>
<th>AISI 316</th>
</tr>
</thead>
<tbody>
<tr>
<td>573°C &gt; T</td>
<td>$C_{PCM}$ [J/kgK]* 1070</td>
<td>535</td>
</tr>
<tr>
<td>C\textsubscript{pseudo} = $h_{jua}$ / $\Delta T_{melt}$</td>
<td>1170</td>
<td>581</td>
</tr>
<tr>
<td>T &lt; 577°C</td>
<td>$k_{PCM}$ [W/mK][137] 160</td>
<td>590</td>
</tr>
<tr>
<td>T &gt; 577°C</td>
<td>$C_{enc}$ [J/kgK][72] 17.7</td>
<td>22</td>
</tr>
<tr>
<td></td>
<td>$k_{enc}$ [W/mK][72]  23.1</td>
<td></td>
</tr>
</tbody>
</table>

* experimentally measured.
Fig. 3.10: Heat capacity of AlSi$_{12}$ as a function of temperature for the solid, two-phase, and liquid states, measured using a differential scanning calorimeter, averaged for 3 heating and cooling runs.
**Experimental Setup**

A scheme and a photograph of the experimental setup are shown in Fig. 3.11 and Fig. 3.12, respectively. During charging, ambient air circulated by the blower is first heated to 600-700 °C using a 21-kW electric heater. The hot air enters the storage at the top, transfers heat to the storage material, exits from the bottom, and finally flows through the mass flow meter before exiting the circuit. During discharging, ambient air at $T_{\text{ambient}} \approx 25 ^\circ \text{C}$ flows in opposite direction, first through the mass flow meter, then into the storage at the bottom, recuperates the previously stored heat, and exits the circuit from the top. The blower frequency and heater power are controlled and the mass flow meter and thermocouple signals are recorded. In each PCM row, thermocouples are placed inside the center tube ($T_{\text{PCM}}$) and in the void space ($T_f$) to measure the PCM and air temperatures, respectively. Two thermocouples are placed on the inside of the tank wall at the height of the 2$^{\text{nd}}$ and 4$^{\text{th}}$ PCM row ($T_{\text{wall,2}}$ and $T_{\text{wall,4}}$). Further, the temperatures at the
inlet of the PCM section ($T_{\text{inlet}}$) as well as above the perforated plate ($T_{\text{plate,top}}$) and at the outlet of the heater ($T_{\text{Heater}}$) are monitored. In the sensible heat section, thermocouples are placed at the center of the packed bed at different heights ($T_{R}$). The positions of the thermocouples for the two sets of tests are summarized in Table 3.11.

Table 3.11: Distance of the thermocouples from the top of the storage domain for the two sets of tests (in mm).

<table>
<thead>
<tr>
<th>Designation</th>
<th>rocks + PCM</th>
<th>rocks only</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{inlet}}$</td>
<td>0</td>
<td>0</td>
</tr>
<tr>
<td>$T_{\text{PCM,1}}$, $T_{f,1}$</td>
<td>11</td>
<td>-</td>
</tr>
<tr>
<td>$T_{\text{PCM,2}}$, $T_{f,2}$</td>
<td>34</td>
<td>-</td>
</tr>
<tr>
<td>$T_{\text{PCM,3}}$, $T_{f,3}$</td>
<td>56</td>
<td>-</td>
</tr>
<tr>
<td>$T_{\text{PCM,4}}$, $T_{f,4}$</td>
<td>79</td>
<td>-</td>
</tr>
<tr>
<td>$T_{R,1}$</td>
<td>379</td>
<td>225</td>
</tr>
<tr>
<td>$T_{R,2}$</td>
<td>720</td>
<td>574</td>
</tr>
<tr>
<td>$T_{R,3}$</td>
<td>1045</td>
<td>865</td>
</tr>
<tr>
<td>$T_{R,4}$</td>
<td>1345</td>
<td>1335</td>
</tr>
</tbody>
</table>
Two possible configurations were investigated: the “rocks + PCM” setup as shown in Fig. 3.7, and the “rocks only” setup where the PCM is replaced by rocks, for the purpose of comparing the performance between the combined sensible/latent heat storage with the sensible heat storage only.

3.5.2 Experimental Results

The high temperatures involved and the consequent large thermal expansions of the connecting parts pose challenges in making an air tight circuit. While the storage tank was constructed in order to ensure no leakages, this was not possible for the connecting joints and valves that had to go through large temperature variations. Early in the experiments, leakages from the heater were observed that could not be overcome. Hence the mass flow meter was connected at the bottom for both charging and discharging runs. However, tests conducted after the experimental campaign showed the presence of leakages also at the bottom connection between the mass flow meter and the storage. Fig. 3.13 shows the relation between the mass flow before and after the bottom connection. These losses were quantified and accounted for in the model. The small tank to particle diameter ratio of about 12 will cause the void fraction to be larger close to the wall [109, 157, 158] which causes a lower flow resistance and hence higher fluid velocities there [111, 159, 160]. According to Schwartz and Smith [111] for tank to particle diameter ratios less than 30 the peak velocity close to the wall is 30 to 100% greater than the velocity at the center of the tube. Gross et al. [63] accounted for this effect by taking an effective heat transfer coefficient developed by Martin [161] that considers a bypass cross-sectional fraction close to the wall with a different void fraction and calculates an average heat transfer value. However, this approach can be used when the cross sectional average of the temperature is available. Nsofor [162] placed liners along the wall to reduce the bypass effect. Hänchen et al. [64] considered a bypass fraction of the flow and assumed that it bypasses the column without thermally interacting with the center of the packed bed. The same approach is used here with a bypass
fraction of 15% that yielded good matching for all cases.

Table 3.12 summarizes the experimental runs: runs #1 and #2 with the “rocks + PCM” setup and runs #3 and #4 with the “rocks only” setup. Runs #1 and #2 were carried out with similar mass flow rates and charging times as “rocks only” runs #3 and #4, respectively. However, due to failure of a heater element during run #4 direct comparison is only shown between run #1 and #3. For all runs, the discharging lasted until the storage was depleted. Before the charging process, it was ensured that the storage was uniformly at ambient temperature. For runs #1 and #3, the heater power was ramped up until 95% of maximum heater power at a rate of 5% every 3 minutes and kept unchanged for the rest of the charging period. For runs #2 and #4, the heater power was ramped up until 70% of maximum heater power at a rate of 5% every 5 minutes and kept unchanged for the rest of the period.

Table 3.12: Summary of the experimental runs.

<table>
<thead>
<tr>
<th>Run #</th>
<th>Setup</th>
<th>$\dot{m}_h$ [kg/s]</th>
<th>$\dot{m}_d$ [kg/s]</th>
<th>$t_{ch}$ [h]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>rocks + PCM</td>
<td>0.008-0.014</td>
<td>0.014</td>
<td>3:10</td>
</tr>
<tr>
<td>2</td>
<td>rocks + PCM</td>
<td>0.005-0.009</td>
<td>0.014</td>
<td>4:15</td>
</tr>
<tr>
<td>3</td>
<td>rocks only</td>
<td>0.008-0.014</td>
<td>0.014</td>
<td>3:10</td>
</tr>
<tr>
<td>4</td>
<td>rocks only</td>
<td>0.005-0.009</td>
<td>0.014</td>
<td>4:15</td>
</tr>
</tbody>
</table>

![Fig. 3.13: Relation between the measured mass flow before and after the bottom connection.](image)
charging period. The Reynolds number in the PCM section \( Re = \frac{\rho u_{\text{max}} d}{\mu} \), and the superficial Reynolds number in the rocks section \( Re_0 = \frac{\rho u_0 d}{\mu} = \frac{G d}{\mu} \), is in the range of 140-390 and 60-200, respectively. The Biot number range is 0.0004-0.0011 for the PCM and 0.04-0.24 for the rocks. Note that the low Biot number of the PCM is due to its high thermal conductivity. Due to \( Bi > 0.1 \) for the rocks, the effective convective heat transfer coefficient in the packed bed has to be adjusted in the model to account for intra-particle conduction like in section 3.3.

Fig. 3.14 compares the bottom and top temperature profiles obtained from the experiments for run #1 (“rocks + PCM” setup) and run #3 (“rocks only” setup). The bottom temperature profiles are similar for both cases indicating that the energy stored for both setups is comparable. Due to the presence of the melting regime for the “rocks + PCM” setup, the final top temperature of the charging period for this setup is about 10 °C lower than that for the “rocks only” setup. Although the outflow temperature during discharging drops faster initially for the “rocks + PCM” setup, a stabilization can be clearly observed. This in turn leads to the outflow temperature of the “rocks only” setup to drop below that of the “rocks +
“rocks + PCM” setup stabilizes the outflow temperature for about 90 minutes after which all the PCM is solidified and the temperature begins to drop. A qualitative comparison of Fig. 3.14 with the numerical results for “Rocks + Material #2” depicted in Fig. 3.4, shows good matching between the experimental and numerical studies for the relative temperature profile of the sensible heat only case with respect to the combined sensible and latent heat case.
3.6 Numerical Validation

3.6.1 Adaptation of the Numerical Model

The model introduced in section 3.2 had to be modified in several aspects in order to consider the specificities of the experimental setup. Fig. 3.15 shows the relevant heat fluxes of this section. The convective heat transfer in the packed bed of rocks is calculated using the correlation developed by Alanis [94], presented in section 2.3.3, obtained for similar particle size (\(d=2.1-5.8\) cm), void fraction (\(\varepsilon=0.421\)) and Reynolds numbers (10 < \(Re_0\) < 200) as in the experimental campaign, and is adjusted for intra-particle conduction using Eq. (3.10).

The convective heat transfer correlation for packed beds of spherical PCM (Eq. (3.9)) is replaced by the correlation developed by Zukauskas [163] for tubes in cross flow

\[
\text{Nu}_{\text{PCM}} = 0.51C_{\text{row}} \text{Re}^{0.5} \text{Pr}_f^{0.37} \left(\frac{\text{Pr}_f}{\text{Pr}_s}\right)^{0.25}
\]  

(3.15)

The above equation is valid for staggered tubes in cross flow and 100 < \(Re\) < 1000. The coefficient \(C_{\text{row}}\) accounts for row numbers less than 20 as in the present experimental setup. \(\text{Pr}_f\) is the fluid Prandtl number evaluated at the fluid temperature and \(\text{Pr}_s\) the fluid Prandtl number evaluated at the solid temperature. In contrast to packed beds where the

![Fig. 3.15: PCM section showing the relevant heat fluxes.](image-url)
Reynolds number is generally based on the superficial velocity, the Reynolds number in Eq. (3.15) is based on the maximum interstitial velocity (Re= \( \rho u_{\text{max}} d/\mu \))

\[
Re = \frac{\rho u_{\text{max}} d}{\mu}
\]

with \( S_T \) and \( S_D \) the transverse and longitudinal pitch and \( u_0 \) the superficial fluid velocity. The volumetric heat transfer coefficient is obtained by

\[
h_{\text{v,PCM}} = \frac{\text{Nu}_{\text{PCM}} k_f a_{\text{PCM}}}{d_{\text{enc}}}
\]

where \( a_{\text{PCM}} \) is the surface area per unit volume of PCM. The inlet temperature, \( T_{\text{inlet}} \), as shown in Fig. 3.15, is the temperature of the incoming air. Due to the high temperatures involved, the radiation exchange between the perforated plate and the first PCM row is considered

\[
Q_{\text{rad, plate}} = \frac{\sigma A_{\text{plate}} (1 - \varepsilon_{\text{plate}}) (T_{\text{plate, bottom}}^4 - T_{\text{PCM,1}}^4)}{1/\varepsilon_{\text{plate}} + 1/\varepsilon_{\text{enc}} - 1}
\]

where \( \varepsilon_{\text{plate}} \) is the void fraction of the perforated plate and \( T \) is in Kelvin. Due to the high conductivity of the PCM and encapsulation, the tube surface temperature is taken equal to the PCM temperature as justified by the low Biot numbers in the PCM section (Bi<0.001). A rough oxide layer was formed on the surface of the perforated plate and the encapsulation tubes after the preliminary tests. Therefore, emissivity values for oxidized stainless steel are considered. \( T_{\text{plate, bottom}} \) is obtained by applying an energy balance on the perforated plate. At any time step, the heat conducted through the plate, \( Q_{\text{cond, plate}} \), is equal to the heat radiated with the PCM, \( Q_{\text{rad, plate}} \), giving

\[
\frac{k_{\text{plate}} A_{\text{plate}} (1 - \varepsilon_{\text{plate}})}{t_{\text{plate}}} (T_{\text{plate, top}} - T_{\text{plate, bottom}}) = \frac{\sigma A_{\text{plate}} (1 - \varepsilon_{\text{plate}}) (T_{\text{plate, bottom}}^4 - T_{\text{PCM,1}}^4)}{1/\varepsilon_{\text{plate}} + 1/\varepsilon_{\text{enc}} - 1}
\]

with \( T_{\text{plate, top}} \) measured during the experiments, \( T_{\text{plate, bottom}} \) is thus obtained from Eq. (3.19) using the Newton-Raphson method [68].
is considered in the first PCM layer, a similar term is considered in the last PCM and first rocks layer accounting for the radiation exchange between the two parts

\[ Q_{\text{rad,interface}} = \frac{\sigma A_{\text{tank}} (1 - \varepsilon_{\text{PCM}}) \left( T_{\text{PCM}}^4 - T_{\text{rocks}}^4 \right)}{1/\varepsilon_{\text{enc}} + 1/\varepsilon_{\text{rocks}} - 1} \]  

(3.20)

The effective thermal conductivity considered in Eq. (3.3) is developed for packed beds and cannot be applied to the arrangement present in the experimental setup. Therefore, the conductive and radiative heat transfer between the PCM layers are accounted for separately and considered in the solid phase. The conductive heat transfer between the PCM layers is calculated by

\[ Q_{\text{cond,PCM}} = A_{\text{tank}} f_{\text{cont,PCM}} \left[ k_{\text{enc}} \frac{dT_{\text{PCM}}}{dx} \bigg|_{\text{out}} - k_{\text{enc}} \frac{dT_{\text{PCM}}}{dx} \bigg|_{\text{in}} \right] \]  

(3.21)

where \( f_{\text{cont,PCM}} \) is the fraction of the tank cross section where the tubes are in physical contact. The presence of the caps with a slightly larger diameter than the tubes, as seen in Fig. 3.9, causes the tubes to touch only at their ends. Hence, the contact fraction is much less than \( (1 - \varepsilon_{\text{PCM}}) \). The radiative heat transfer between the PCM layers is

\[ Q_{\text{rad,PCM},i} = \frac{\sigma A_{\text{tank}} (1 - \varepsilon_{\text{PCM}})}{2/\varepsilon_{\text{enc}} - 1} \left( T_{\text{PCM},i+1}^4 - 2T_{\text{PCM},i}^4 + T_{\text{PCM},i-1}^4 \right) \]  

(3.22)

\( i \) represents here row number, ranging from 1 to 4. For both Eq. (3.21) and (3.22), adiabatic conditions are applied at their boundary. For the “rocks only” setup the radiative exchange between the plate and the rocks is considered by Eq. (3.18), modified with the data for the packed bed of rocks. The interaction of the storage domain with the walls is accounted for in the sensible heat section by considering thermal losses through the lateral walls, as introduced in section 2.3.5, in the fluid phase equation

\[ Q_{\text{wall,rocks}} = h_{\text{wall}} A_c \left( T_\infty - T_f \right) \]  

(3.23)
The overall wall heat transfer coefficient, $h_{wall}$, considers the heat transfer between the packed bed and the wall with a convective term, as developed by Beek [55], and a conductive/radiative term, as developed by Ofuchi and Kunii [102], as well as the conductive heat transfer through the wall and insulation layers. For further details see section 2.3.5. The outer surface temperature of the insulation, $T_{\infty}$, is considered equal to ambient temperature and invariant with time as verified by thermal images taken of the storage when fully charged and shown in Fig. 3.16. In the latent heat section, the monitored wall temperature is used as source/sink term with conductive heat transfer to/from the tubes

$$Q_{wall,PCM} = \frac{k_{en} A_v f_{cont,wall}}{(r_{in} - t_{enc}) \ln\left(\frac{r_{in}}{r_{in} - t_{enc}}\right)} (T_{wall} - T_{PCM})$$

(3.24)

While the wall temperature was measured for the 2\textsuperscript{nd} and 4\textsuperscript{th} row, linear extrapolation and interpolation on the measured values was used to obtain the wall temperature for the 1\textsuperscript{st} and 3\textsuperscript{rd} row. $f_{cont,wall}$ is the fraction of the circumference that is in contact with the PCM tubes. The values of $f_{cont,wall}$

Fig. 3.16: Thermal image of the heater and storage during operation when fully charged, showing the surface of the storage to be at ambient temperatures due to adequate insulation.
and $f_{cont,PCM}$ were estimated visually from the experimental setup. All parameters used in the above equations are listed in Table 3.13.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
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<tr>
<td>$A_{plate}$ [m$^2$]</td>
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</tr>
<tr>
<td>$A_{tank}$ [m$^2$]</td>
<td>0.122</td>
</tr>
<tr>
<td>$t_{plate}$ [m]</td>
<td>0.02</td>
</tr>
<tr>
<td>$t_{enc}$ [m]</td>
<td>0.001</td>
</tr>
<tr>
<td>$k_{plate}$ [W/mK]</td>
<td>41-35</td>
</tr>
<tr>
<td>$\varepsilon_{plate}$ [-]</td>
<td>0.142</td>
</tr>
<tr>
<td>$\varepsilon_{enc}$ [-]</td>
<td>0.7</td>
</tr>
<tr>
<td>$\varepsilon_{rocks}$ [-]</td>
<td>0.7</td>
</tr>
<tr>
<td>$S_{T}$ [m]</td>
<td>0.0253</td>
</tr>
<tr>
<td>$S_{D}$ [m]</td>
<td>0.0253</td>
</tr>
<tr>
<td>$a_{PCM}$ [m$^2$/m$^3$]</td>
<td>103.45</td>
</tr>
</tbody>
</table>

* value known, measured or calculated. † value estimated.

The PCM showed a congruent phase trajectory of melting/solidification in the range of 573-577 °C, hence a melting temperature range $\Delta T_m$ of 4 K. Therefore, instead of the molten fraction in Eq. (3.5), the two-phase region is modelled with a pseudo-specific heat [133]

$$C_{pseudo} = h_{fus} / \Delta T_{melt}$$  \hspace{1cm} (3.25)

With the modifications introduced above, the governing equations become, for the packed bed of rocks

**Fluid:**

$$\varepsilon_{rocks} \rho_f V \frac{de_f}{dt} = A \varepsilon_{rocks} \left[ (u \rho_f h_f)_\text{in} - (u \rho_f h_f)_\text{out} \right] + h_v V (T_s - T_j) + Q_{wall,rocks}$$  \hspace{1cm} (3.26)

**Solid:**

$$\left( 1 - \varepsilon_{rocks} \right) \rho_s V \frac{de_s}{dt} = h_v V (T_f - T_s) + A \left[ k_{eff} \frac{dT_s}{dx}_\text{out} - k_{eff} \frac{dT_s}{dx}_\text{in} \right]$$  \hspace{1cm} (3.27)

and for the PCM section

**Fluid:**

$$\varepsilon_{PCM} \rho_f V \frac{de_f}{dt} = A \varepsilon_{PCM} \left[ (u \rho_f h_f)_\text{in} - (u \rho_f h_f)_\text{out} \right] + h_{v,PCM} V (T_{PCM} - T_f)$$
PCM:

\[
(1 - \varepsilon_{\text{PCM}})(\rho C)_{\text{eff}} V \frac{dT_{\text{PCM}}}{dt} = h_{v,\text{PCM}} V(T_f - T_{\text{PCM}}) + Q_{\text{cond,PCM}} + Q_{\text{rad,PCM}} + Q_{\text{wall,PCM}}
\]  

(3.29)

The effective volumetric heat capacity \((\rho C)_{\text{eff}}\) is calculated as the volumetric average of the product of the heat capacity and the mass of the PCM and encapsulation

\[
(\rho C)_{\text{eff}} = \frac{m_{\text{PCM}} C_{\text{PCM}} + m_{\text{enc}} C_{\text{enc}}}{V_{\text{enc}}}
\]  

(3.30)

\(V_{\text{enc}}\) is the total volume of the encapsulation tubes in a row, \(m\) and \(V\) are row specific and given in Table 3.9 and \(C\) is temperature specific and given in Table 3.10. Eq (3.29) is modified to account for \(Q_{\text{rad,plate}}\) in the first layer and for \(Q_{\text{rad,interface}}\) in the last layer by adding these terms to its r.h.s. Eq. (3.27) is modified for the first layer to account for \(Q_{\text{rad,interface}}\) by subtracting this term from its r.h.s. For the “rocks only” setup, \(Q_{\text{rad,plate}}\) is modified with data for rocks and added to the r.h.s. of Eq. (3.27) for the first layer. Other initial and boundary conditions are the same as introduced in section 2.2.

### 3.6.2 Comparison of Numerical and Experimental Results

Comparisons of experimentally measured and numerical calculated results for the four experimental runs (Table 3.12) are shown in Fig. 3.17 to Fig. 3.20. The PCM section is divided into four layers representing the four rows. The packed bed of rocks is divided in 64 and 68 layers for the “rocks + PCM” runs and “rocks only” runs, respectively. A grid size analysis showed that further refinement of the grid does not affect the numerical results. The numerical calculated temperatures for the packed bed of rocks are plotted for the average of the solid and fluid temperatures. \(T_{\text{inlet}}\) is recorded during the experimental runs and used as inlet temperature in the model.
Fig. 3.17: Experimentally measured (markers) and numerical calculated (curves) temperatures at various positions as a function of time during charging and discharging for the “rocks only” setup run #3. Also indicated is the mass flow rate of air.

Fig. 3.18: Experimentally measured (markers) and numerical calculated (curves) temperatures at various positions as a function of time during charging and discharging for the “rocks only” setup run #4. Also indicated is the mass flow rate of air.
The heat flows for the 4 experimental runs, calculated as the integral of the heat fluxes over the charging and discharging periods (as obtained by the simulation model) are summarized in Table 3.14. During discharging, the air flow is reversed and $T_{\text{inlet}}$ (position shown in Fig. 3.15) is actually recording the outlet temperature. Fig. 3.17 and Fig. 3.18 show the experimentally measured (markers) and numerical calculated (curves) temperatures at various positions as a function of time during charging and discharging for the “rocks only” setup runs #3 and #4 respectively. Also indicated is the mass flow rate of air. For the charging phase, the matching is reasonably good. For the discharging phase, the numerical values underestimate the experimental ones, especially on the top of the tank. This is attributed to the thermal inertia of the tank that is not accounted for in the model, as the thermal energy stored in the insulation during charging is released during discharging. This also explains why the discrepancy is higher at the top where the temperatures are higher. The thermal losses from the lateral walls, calculated by the simulation model, are 3% and 3.5% of the total input energy during charging from the incoming air flow ($E_{\text{input}}$) and the radiative heat transfer from the perforated plate ($E_{\text{rad,plate,\text{ch}}}$) for run #3 and #4, respectively. Due to the fast temperature drop of the simulation results during discharging, the net radiative heat transfer from the perforated plate to the packed bed, $E_{\text{rad,plate,\text{dis}}}$ > 0, as shown in Table 3.14.

Table 3.14: Heat flows (in kWh) for the 4 experimental runs (Table 3.12), calculated by the simulation model.

<table>
<thead>
<tr>
<th>Run #</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
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<tbody>
<tr>
<td>$E_{\text{input}}$</td>
<td>18.4</td>
<td>16.66</td>
<td>19.3</td>
<td>16.09</td>
</tr>
<tr>
<td>$E_{\text{rad,plate,\text{ch}}}$</td>
<td>0.95</td>
<td>1.2</td>
<td>1.01</td>
<td>1.15</td>
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<tr>
<td>$E_{\text{rad,plate,\text{dis}}}$</td>
<td>-0.14</td>
<td>-0.11</td>
<td>0.42</td>
<td>0.33</td>
</tr>
<tr>
<td>$E_{\text{wall,PCM,\text{ch}}}$</td>
<td>1.27</td>
<td>1.3</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$E_{\text{wall,PCM,\text{dis}}}$</td>
<td>0.43</td>
<td>0.67</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>$E_{\text{wall,rocks,\text{ch}}}$</td>
<td>-0.24</td>
<td>-0.29</td>
<td>-0.3</td>
<td>-0.36</td>
</tr>
<tr>
<td>$E_{\text{wall,rocks,\text{dis}}}$</td>
<td>-0.25</td>
<td>-0.21</td>
<td>-0.3</td>
<td>-0.25</td>
</tr>
</tbody>
</table>
Fig. 3.19a and Fig. 3.20a show the experimentally measured (markers) and numerical calculated (curves) temperatures of the packed bed and inlet (top) temperatures as a function of time during charging and discharging for run #1 and run #2, respectively. The matching in the packed bed of rocks is similar to the “rocks only” runs #3 and #4. However, the discrepancy of the top temperature during discharging is less pronounced here. This is attributed to the fact that the wall temperature is measured in the PCM section and used as boundary condition, as seen in Fig. 3.15 and explained in section 3.6.1. Fig. 3.19b and Fig. 3.20b show the comparison for the PCM and air temperatures of the 1\textsuperscript{st} and 3\textsuperscript{rd} row of the latent heat section. For the sake of clarity, the 2\textsuperscript{nd} and 4\textsuperscript{th} PCM row are not shown. The difference between numerical and experimental temperatures of the PCM and air for each row is attributed to the convective heat transfer correlation which was derived for staggered tubes with parallel axis, in contrast to the perpendicular axis of the experimental setup. In addition, thermocouples are exposed to radiative heat exchange, which can introduce an error in the reading when the surroundings are colder (during charging, the reading is lower than the actual air temperature) or hotter (vice versa during discharging). The faster solidification of the modelled PCM during the discharging compared to the experimental results is attributed to the lower incoming temperatures from the packed bed of rocks in the model. The thermal losses from the lateral walls in the sensible section, as obtained by the simulation model, are 2.5% and 2.8% of the total input energy for run #1 and #2, respectively. As shown in Table 3.14, due to the large thermal inertia of the insulation, the net integrated energy flow from the walls to the PCM during discharging, $E_{\text{wall},\text{PCM,dis}} > 0$. 
Fig. 3.19: Experimentally measured (markers) and numerical calculated (curves) temperatures as a function of time during charging and discharging for the “rocks + PCM” setup run #1: packed bed and inlet (top) temperatures as well as air mass flow rate (a), and PCM and air temperatures at 1st and 3rd row (b).
Fig. 3.20: Experimentally measured (markers) and numerical calculated (curves) temperatures as a function of time during charging and discharging for the “rocks + PCM” setup run #2: packed bed and inlet (top) temperatures as well as air mass flow rate (a), and PCM and air temperatures at 1\textsuperscript{st} and 3\textsuperscript{rd} row (b).
3.7 Summary

A concept was introduced for stabilizing the outflow temperature of a packed bed of rocks thermal energy storage during discharging by adding a relatively small amount of phase change material (PCM) to the top of the bed. First, numerical heat transfer simulations were used to investigate the concept. A 1D heat transfer model was formulated for the sensible and latent heat sections of the TES considering separate fluid, solid and molten phases. A PCM relative volume of 1.33% of the total storage volume was sufficient to achieve stabilization of the outflow air temperature around the PCM melting point, but shown to be strongly dependent on the PCM thermal properties. The overall efficiency showed to be not significantly affected by the addition of PCM.

After the feasibility of the concept was numerically demonstrated, an experimental campaign was carried out using a 42.4 kWh lab-scale TES to further investigate the concept of combined sensible and latent heat storage. The metallic alloy AlSi\textsubscript{12} was chosen as PCM due to its adequate melting temperature for integration with Airlight Energy’s trough collectors ($T_{\text{melt}} \approx 575 ^\circ \text{C}$), as well as high heat of fusion, high thermal conductivity and low thermal expansion. AISI 316 stainless steel tubes were used as encapsulation, filled with molten AlSi\textsubscript{12} and arranged in a mesh-like configuration. For the “rocks + PCM” setup, a latent heat section of 9 cm of encapsulated PCM was placed on top of 127 cm of rocks, which was later replaced with rocks for the “rocks only” setup. The combined storage stabilized the outflow temperature around the PCM’s melting temperature for 90 minutes, 20 minutes of which the temperature was higher than in the sensible heat only case of the “rocks only” setup. The numerical model was adapted to the specificities of the experimental setup and compared to the experimental results. Thermal inertia of the experimental setup and radial variation of void fraction due to the small tank to particle diameter ratio showed to affect the validation process. The combination of sensible and latent heat storage addresses the requirement of a constant-temperature stream of the heat transfer fluid during discharging.
3.8 Conclusion

Based on the work presented in this chapter, the following conclusions are drawn:

- By adding a relatively small layer of PCM on top of a packed bed of rocks thermocline storage, the outflow air temperature during discharging can be stabilized around the melting temperature of the PCM.
- Typical issues of the two concepts when applied separately are therefore alleviated.
- For lab-scale high-temperature packed bed TES prototypes, thermal inertia of the experimental setup and radial variation of the void fraction can affect the validation process.
- The relatively small amount of PCM required yields a solution that is economically attractive for high-temperature TES.
- The concept can be applied where the downstream application is sensitive to temperature variations, e.g. thermal or thermochemical reactions, or to help the turbine in a Rankine or Brayton cycle to operate at its design point throughout the discharging phase.
4 Summary

In this thesis, two concepts for high-temperature thermal energy storage (TES) with air as heat transfer fluid have been numerically modelled and experimentally investigated. The first concept is based on a packed bed of rocks enclosed in a concrete tank with the shape of a truncated cone that is immersed in the ground. A 6.5 MWh\textsubscript{th} pilot-scale TES unit was constructed and tested. The design and fabrication of the storage is aimed at industrial-scale applications by addressing typical issues of this kind of storage systems. The conical shape, multi-layer concrete walls and the subterranean placement of the TES help avoid damaging the tank and the rocks during cyclic operation. A dynamic numerical heat transfer model that solves the unsteady energy equations for the fluid and solid phases was developed and validated with experimental results from the 6.5 MWh\textsubscript{th} storage. Some of the main differences of the model to pertinent models in literature are: consideration of variable fluid and solid thermo-physical properties, where the temperature-dependent solid properties were experimentally measured; consideration of thermal losses through the lid and the bottom in addition to the lateral wall losses; and consideration of axially variable void fraction, which was experimentally measured as well. The validated model was used for the design and performance assessment of two industrial-scale TES systems of 100 MWh\textsubscript{th} and 7.2 GWh\textsubscript{th}, which achieved efficiencies of 89% and 96%, respectively. Further the model was used for studies that can serve as a guideline for the design of thermal energy storage units. Parameters such as initial charging of the TES unit, the tank diameter to height ratio for cylindrical tanks, the cone angle for conical tanks, the particle diameter, and the insulation thickness were investigated. These studies concluded that there is no unambiguous code of practice for the design of TES units and that every unit has to be optimized for performance from one side (efficiency, outflow temperature profile) and costs from the other (thermal
losses, pumping work, excavation and material costs). Further, the surrounding ground was modelled as a semi-infinite slab and its effect on the storage performance and thermal losses were studied.

The second concept was introduced to address one of the main disadvantages of one-tank thermocline TES systems, namely large outflow temperature variations during the discharging phase. It consists of combining sensible and latent heat storage by means of adding a small layer of phase change material (PCM) on top of the packed bed of rocks and hence stabilizing the outflow temperature during discharging around the melting temperature of the PCM. The numerical heat transfer model was extended for the latent heat section and simulations were carried out to investigate the concept. It was numerically shown that a PCM relative volume of 1.33\% of the total storage volume is sufficient to achieve stabilization of the outflow air temperature around the PCM’s melting point, but shown to be strongly dependent on the PCM’s thermal properties. The overall efficiency was shown not to be significantly affected by the addition of the PCM. Following the numerical study, an experimental campaign was carried out using a 42.4 kWh\textsubscript{th} lab-scale TES, with 9 cm-height of a metallic PCM (AlSi\textsubscript{12}) encapsulated in stainless steel tubes (AISI 316) as the latent heat section and 127 cm-height of sedimentary rocks as the sensible heat section. The latent heat section was later replaced by rocks for comparison of the two concepts. The numerical model was adapted to the specificities of the experimental setup and compared to the experimental results. Thermal inertia of the experimental setup and radial variation of void fraction due to the small tank to particle diameter ratio were shown to affect the validation process. The combined storage stabilized the outflow temperature for 90 minutes, 20 minutes of which the outflow temperature was higher than the rocks only setup.
5 Outlook

5.1 Sensible Heat Energy Storage

The validation in chapter 2 showed that the quasi-one-dimensional model used in this work is adequate for simulating large scale storage units, since border effects – typically present in lab-scale units – are negligible. The speed of the presented numerical scheme, compared to 2D models or commercial CFD software, makes it a powerful tool for designing and optimizing large-scale TES units for commercial applications. By substituting the explicit numerical scheme with an implicit one, the speed could be further increased. Year-round simulations using real DNI and solar field data would help understand the effect of variable charging duration, temperature and mass flow on the storage performance.

As done in this thesis, the thermo-physical properties of the rocks that will be used in a TES unit should be experimentally investigated before its design and used therein, since properties such as the density and thermal capacity have a major impact on the TES size and performance. It is recommended to conduct accelerated aging tests on the rocks by means of thermal cycling, since studies have shown that not all types of rocks are adequate for repeated, large variations in temperature [164]. It should be kept in mind, however, that due to the thermal stratification in the tank, once the TES has reached its steady behavior, no single rock will undergo cycles larger than 100-150 K, as evident from Fig. 2.23.

Although not observed in the experimental campaign, another practical concern for the commercialization of the storage concept is dust carriage from the TES into the solar field which can be reduced by a) controlled air velocities in the packed bed – by means of adequate design of tank diameter and air mass flow rates; b) rinsing the rocks before filling the TES; and c)
placing a filter at the outlet of the storage (with the penalty of increased pressure drop).

Although CFD simulations showed that the flow will homogenize shortly after entering the packed bed\(^{13}\), design considerations should be made to create a plug-flow from the inlet tube, like the ones presented in section 2.6.1 for the 100 MWh\(_{th}\) TES unit in Ait Baha, Morocco, or alternatively by other means such as multiple inlets.

A 5.9 MWh\(_{th}\) TES unit is currently under construction at the premises of Airlight Energy SA in Biasca, Switzerland, that will be equipped with more sophisticated measurement and peripheral instruments, as well as better insulation. A photograph and a schematic are shown in Fig. 5.1. Experimental results from this TES unit should be used to further investigate characteristics such as radial temperature distribution, pressure drop, thermal losses and thermocline behavior, as well as for further validation of the numerical model.

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\(^{13}\) Preliminary results of a semester project carried out by Francesco Andreoli, supervised by G. Zanganeh.
5.2 Combined Sensible and Latent Heat Energy Storage

The numerical model showed to require a much finer grid for the PCM section compared to the rock section in order to obtain numerically reliable results, which is attributed to switching to a different solid phase energy equation during the phase transition. This causes the numerical scheme to become relatively slow and lose its quality as a fast and efficient tool for system design simulations. It is therefore recommended to adapt an implicit scheme to overcome this drawback.

With a more efficient numeric scheme, it should be investigated whether variations in the charging duration, temperature, and mass flow, typically caused by variations in the DNI and intermittency of solar radiation, will have a larger negative effect on the performance of the combined storage compared to the sensible heat only storage. Further, it should be investigated whether using PCM materials with different melting temperatures at different heights of the TES can improve its performance, particularly when considering seasonally variable inlet temperatures.

An experimental campaign was carried out with a 42.4 kWh$_{th}$ lab-scale TES. The numerical model was adapted to the experimental setup and compared to its results. The small tank to diameter ratio made it necessary to consider a bypass fraction of the flow that does not thermally react with the centre of the packed bed, where the thermocouples were placed. Further, for the discharging phases, especially in the high-temperature regions, the model under-predicted the temperature profiles in the packed bed, which is attributed mainly to the energy stored in the insulation that is released during discharging, and is not accounted for in the model. Therefore, for future investigations of lab-scale storage units, it is recommended to expand the existing model to 2D in order to be able to account for radial variations in temperature, velocity and void fraction, and to include the thermal capacity of the insulation in the model.

The heat capacity and heat of fusion of the PCM used in the experimental campaign were measured in a differential scanning calorimeter surrounded by atmospheric air for three consecutive heating and
cooling runs and did not show any signs of oxidation or change in characteristics. Neither was any change in PCM behavior observed during the experimental campaign. Further, cyclic investigation by other authors showed small changes in the properties of this particular or similar phase change materials after 1000 cycles or more \cite{154, 155}. Nevertheless, concerns exist that the PCM oxidizes, corrodes the encapsulation material or reacts with it, or phase separation occurs during the course of the lifetime of a commercial TES unit. It is hence recommended to conduct a thorough study on the cyclic PCM behavior and interaction with the encapsulation material before continuing towards a large-scale or commercial-size storage of this kind. Also, it is recommended to optimize the encapsulation method and procedure to decrease labor costs and ensure no air presence in the encapsulation.

The thermal energy storage systems presented in this work have been developed for concentrated solar power with high-temperature air as heat transfer fluid. However, the nature of the presented concepts makes them suitable for any application with high-temperature air that requires an efficient energy storage system. One of these concepts is electricity storage in the form of compressed air denoted as Advanced Adiabatic Compressed Air Energy Storage (AA-CAES) \cite{165}. In AA-CAES plants, excessively produced electricity in the grid is stored in the form of compressed air using electrically driven compressors. During peak demand periods, electricity is produced by expanding the compressed air in a turbine and driving an electric generator. The air is heated up during compression and cooled down during expansion. In existing CAES plants this heat is generally removed after the compressor and discarded, and a gas burner is used to heat up the air before expansion in the turbine. The AA-CAES concept uses adiabatic compressors and a thermal energy storage to store and recuperate the heat and thereby improve the cycle efficiency while eliminating the CO$_2$ emissions.
Appendix A

Maximum Size for Flat Circular Armored Concrete Lid

Since the low tensile stress limit of concrete can be overcome by armoring, the size of the concrete lid for a given thickness and concrete type is limited by the compressive stress limit, $f_{cs}$. For a circular plate supported on its edge and unconstrained rotational degrees of freedom, the maximum bending moment in the plate center is calculated as [166]:

$$m_{max} = (3 + \nu) \frac{qr^2}{16}$$  \hspace{1cm} A.1

with $\nu$ the poisson ratio, $r$ the lid radius, and $q$ the load on the lid, calculated as $q=\tau \rho g$, with $\tau$ and $\rho$ the thickness and density of the concrete, respectively, and $g$ the gravitational constant. Further, the elastic section modulus is calculated as:

$$S = \frac{t^2}{6}$$  \hspace{1cm} A.2

The maximal stress is then $\sigma_{max} = m_{max} / S$ which has to be smaller than the compressive stress limit under consideration of a safety factor, hence $\sigma_{max} < f_{cs} / f_s$. For a standard concrete with $f_{cs}=40$ N/m$^2$, a thickness of 0.5 m, a density of 2500 kg/m$^3$ and a safety factor of $f_s=1.5$, the maximal radius for the concrete lid will be 20 m.
References


[38] Shitzer A., Levy M., Transient behavior of a rock-bed thermal storage system subjected to variable inlet air temperatures: Analysis


References


[98] van Antwerpen W., du Toit C.G., Rousseau P.G., A review of correlations to model the packing structure and effective thermal


**Curriculum Vitae**

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<td>2010-2014</td>
<td>PhD candidate at the Professorship of Renewable Energy Carriers, ETH Zurich; supervision: Prof. Aldo Steinfeld</td>
</tr>
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</tr>
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</table>

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Journal Publications


Conference Proceedings


**Patents**

Pedretti A., Zanganeh G., Pressurized energy storage system in which the heat accumulator is arranged in an overpressure zone. Patent application number: PCT/CH2013/000039, filed: 05.03.2012, Applicant: Airlight Energy IP SA.