Master Thesis

Storing sunlight
experimental investigation of a combined sensible and latent heat storage for concentrated solar power plants

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Storing Sunlight
Experimental Investigation of a Combined Sensible and Latent Heat Storage for Concentrated Solar Power Plants

A thesis
submitted in partial fulfilment
of the requirements for the

Master of Science
degree in
Mechanical Engineering

by
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Supervised by
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under the Auspices of
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Institute of Energy Technology
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Spring 2013
Dedicated to my parents
Abstract

This report presents the heat transfer modelling, design, construction and experimental investigation of a 40.29 KWhr_{th} laboratory scale combined latent and sensible heat storage for concentrated solar power. The combined storage consists of a 3.97 KWhr_{th} “latent” heat storage section, containing the eutectic alloy of Aluminium and Silicon, AlSi_{12}, encapsulated in stainless steel tubes, placed on top of a 36.32 KWhr_{th} “sensible” heat storage section, comprising of a packed bed of rocks. Adding a thin section of phase change material, comprising less than 7% of the total storage volume, on top of a sensible heat storage is observed to provide a highly stabilised outlet air temperature during the storage discharge cycle. Experiments show that, for a comparable range of air mass flow rates, the discharge outlet air temperature for the combined storage decreases by only 10-15 °C, in comparison to a 72-112 °C drop in temperature observed for a sensible only storage with the same volume, over the same time period. The combined storage concept can hence be used to provide heat at approximately constant temperatures over a significant time period for power generation or process heat applications, at little added cost, while maintaining the high thermodynamic efficiencies characteristic of thermally stratified single tank sensible heat storages.
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List of Symbols and Abbreviations

Variables

\( a \) \quad \text{interfacial surface area per unit volume, m}^2/\text{m}^3

\( \dot{m} \) \quad \text{mass flow rate of air, } \frac{\text{Kg}}{\text{s}}

\( d \) \quad \text{tube or rock diameter, m}

\( g \) \quad \text{edge to circumference gap, m}

\( h \) \quad \text{specific enthalpy, } \frac{\text{J}}{\text{Kg}}

\( n \) \quad \text{number, -}

\( r \) \quad \text{radial distance from centre of tank, m}

\( t \) \quad \text{time, s}

\( x \) \quad \text{axial position, measured from the top surface of the bottom perforated plate, m}

\( A \) \quad \text{cross sectional area of the storage tank, m}^2

\( C \) \quad \text{specific heat capacity, } \frac{\text{J}}{\text{Kg} \cdot \text{K}}

\( D \) \quad \text{internal diameter of storage tank, m}

\( G \) \quad \text{mass flow rate per unit area, } \frac{\text{Kg}}{\text{m}^2\text{s}}

\( H \) \quad \text{total heat / enthalpy, J}

\( L \) \quad \text{length, m}

\( M \) \quad \text{mass of storage material, Kg}

\( \text{Nu} \) \quad \text{Nusselt number, -}

\( P \) \quad \text{perimeter of storage tank, m}

\( \text{Pr} \) \quad \text{Prandtl number, -}

\( \text{Re} \) \quad \text{Reynolds number, -}

\( T \) \quad \text{temperature, K}

\( U \) \quad \text{overall heat transfer coefficient, } \frac{\text{W}}{\text{m}^2\text{K}}

\( \alpha \) \quad \text{heat transfer coefficient, } \frac{\text{W}}{\text{m}^2\text{K}}

\( \kappa \) \quad \text{thermal conductivity, } \frac{\text{W}}{\text{mK}}

\( \eta \) \quad \text{efficiency, -}

\( \epsilon \) \quad \text{void fraction, -}

Subscripts

\( c \) \quad \text{cold,}

\( \text{ch} \) \quad \text{charging cycle}

\( \text{dis} \) \quad \text{discharging cycle}

\( e \) \quad \text{edge
f fluid phase property
fus fusion
h hot
in inlet
j $j^{th}$ insulation layer from the tank surface
$\infty$ ambient conditions
out outlet
overall representative of complete (charging - discharging) cycle
r per row
ref a reference value
s solid phase property
th thermal
I based on the first law of thermodynamics
II based on the second law of thermodynamics
P constant pressure
L latent heat section
S sensible heat section

Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>TES</td>
<td>Thermal Energy Storage</td>
</tr>
<tr>
<td>CSP</td>
<td>Concentrated Solar Power</td>
</tr>
<tr>
<td>DSC</td>
<td>Differential Scanning Calorimetry</td>
</tr>
<tr>
<td>PCM</td>
<td>Phase Change Material</td>
</tr>
<tr>
<td>SCADA</td>
<td>Supervisory Control and Data Acquisition</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

Thermal energy storage is increasingly believed to be a promising means of storing intermittent solar energy in parabolic trough and central tower solar power plants, thereby increasing operational time, reliability and robustness of the power output from these plants (Hänchen et al., 2011). A parabolic trough concentrated solar power plant consisting of a single tank thermal storage is shown schematically in Figure 1.1. The storage is charged, during the day, by passing a fraction of the air heated up by the parabolic trough collectors from the top of the storage tank. The hot air transfers its heat to the rocks as it flows downward through the storage, and cold air comes out from the bottom. This process is referred to as the storage charging cycle. During the night, or during the storage discharging cycle, cold air is blown through the bottom of the storage, where it gets heated up as it travels upward, and hot air comes out from the top. This hot air can then be used for power generation or as process heat, enabling uninterrupted power output from the solar power plant.

1.1 Sensible Heat Storage in a Packed Bed of Rocks

Thermal energy storage systems storing heat as sensible energy in packed beds have been the subject of many theoretical and experimental studies. The theoretical basis for most of the analytical and numerical heat transfer models is the one dimensional, time dependent, two phase heat transfer model, first proposed by Schumann (1929). His analysis predicts the average solid and fluid temperatures over the storage cross section as a function of the axial position and time. A comprehensive summary of both one and two dimensional heat transfer models for packed beds is given by Beasley and Clark (1984). Experimental investigations into the cyclic behaviour of a packed bed of rocks used as a sensible heat storage are reported by Meier.

Figure 1.1: Schematic of a Parabolic Trough CSP Plant with a packed bed thermal energy storage. Image courtesy: Airlight Energy SA
Figure 1.2: A single tank sensible heat storage consisting of a packed bed of rocks. The storage is charged by introducing air from the top which cools down as it flows downward and discharged by blowing cold air from the bottom which heats up as it flows upwards. Temperature stratification over the storage length leads to high storage and discharge efficiencies.
et al. (1991). Hänchen et al. (2011) developed a heat transfer model based on the Schumann equations, additionally including the heat losses through the walls of the storage and validated their model using the data reported by Meier et al. (1991). Zanganeh et al. (2012) extended Hänchen’s model to include variable storage diameters, heat losses from the top and bottom covers and temperature dependent air and rock properties and validated their model using a 6.5MWth pilot scale rock bed storage.

An important functional aspect of a packed bed thermal storage is the presence of temperature stratification in the axial direction. This thermocline is shown schematically in Figure 1.2. The reversal of flow direction between the storage charging and discharging cycles in the presence of a temperature stratification makes the storage function similar, in principle, to a countercurrent heat exchanger, leading to a high second law efficiency by reducing the temperature difference through which heat transfer takes place. Thermal stratification, hence, allows overall thermal storage efficiencies of close to 95% (Zanganeh et al., 2012).

### 1.2 Latent Heat Storage

Storing thermal energy as latent heat of phase change, usually fusion, offers several advantages over the sensible only design. The higher energy storage density implies considerably lower amounts of material and space required. However, most latent heat storage systems suffer from serious drawbacks, due to low thermal conductivity, high cost of encapsulation and high volume expansion during melting causing leakages, to name a few. Latent heat storage systems have also been widely studied. Beasley et al. (1989) developed a computational model of the transient response of a packed bed of spheres containing a phase change material, and validated their model using a packed bed of spheres containing paraffin wax. Their model predicted an essentially constant outlet fluid temperature from the packed bed during the recovery (discharge) cycle. However, since it is necessary to melt most of the PCM during the storage charging cycle, such a configuration leads to a large amount of thermal energy, at temperatures below the melting point of the PCM, being lost. Also, latent heat storage with a single PCM does not allow for any significant thermal stratification leading to low energetic and exergetic efficiencies (Bjurström and Carlsson, 1985). Most of the previous research related to latent heat storage has been limited to melting temperatures below 100 °C. More recently, high temperature energy storage in phase change materials has gained some interest (Kenisarin, 2010), however very little literature is available on materials and methods for latent heat storage above 500 °C.

### 1.3 Combined Latent and Sensible Heat Storage

Although the single tank sensible heat storage proves to be highly efficient from a thermodynamic point of view (Zanganeh et al., 2012), a typical performance issue associated with storing energy as only sensible heat is the large temperature drop observed in the outlet fluid temperature over the course of the storage discharge cycle. The simulated fluid outlet temperature over a representative discharge cycle is shown in Figure 1.4. The results show that the outlet air temperature, even for highly optimised storage designs, drops by about 40-50 °C during the course of the discharge cycle. In order to address this issue and design a storage capable of providing a more constant outlet fluid temperature, while maintaining high thermodynamic efficiency, the novel concept of a combined latent and sensible heat storage was introduced.
Figure 1.3: Conceptual schematic of a combined sensible and latent heat storage. A thin “latent” section consisting of encapsulated phase change material is placed on top of a “sensible” heat storage section.

In this configuration, depicted in Figure 1.3, a thin layer of suitable phase change material is placed on top of the rocks. The longer, and usually less expensive, sensible section enables storage of all the heat present in the incoming fluid during the charging cycle and maintains the thermal stratification. Since the phase change material stores a large amount of energy within a small temperature range during phase transition, the latent section may be used to heat up the air to temperatures very close to the melting point of the phase change material used, maintaining a uniform outflow fluid temperature during the discharging cycle. This concept was investigated theoretically in a previous thesis (Commerford, 2012) where it was shown that stabilised air outlet temperatures may indeed be achieved throughout the discharge cycle if a suitable phase change material was chosen (Figure 1.4). The steadier discharge air temperatures hence obtained may improve the efficiency of plant components such as the power block and process heat reactors.

1.4 Objectives

In light of the theoretical findings, the current thesis was performed to experimentally establish this predicted temperature stabilisation using the combined storage concept. The thesis was performed in collaboration with Airlight Energy SA and the prototype constructed at the
company premises in Biasca, Switzerland. The detailed objectives of this master thesis were to,

- design and manufacture a laboratory scale combined energy storage,
- investigate its performance through experimental test runs, and
- experimentally show the stabilisation of air outlet temperature during the discharge cycle for the constructed combined storage.
Chapter 2

Heat Transfer Model

A mathematical model describing the behaviour of a physical system is an extremely powerful tool for design and analysis. It gives both quantitative and qualitative insight into the different processes and mechanisms which determine the performance of a given physical system. The impact of design changes on the performance of the system may quickly be determined if a reliable mathematical model for the system is available.

In order to describe the behaviour of the combined storage, a one dimensional, two phase, transient heat transfer model was developed as a part of this thesis. The model was used to simulate the behaviour of the lab scale storage to assist during its design. The model may now be validated using the experimental data obtained from the constructed combined storage prototype and then used to accurately predict the performance of storages of different sizes, thus helping in the design of industrial scale combined thermal storages of different capacities. This chapter describes the physical formulation of the heat transfer model, its numerical formulation and solution using implicit time stepping is formulated as MATLAB® code.

2.1 Governing Equations

Figure 2.1 schematically shows a combined sensible and latent heat storage. The storage consists of a latent section of length $L_L$, consisting of encapsulated phase change material, at the top followed by a sensible section, consisting of rocks, of length $L_S$. During the storage charging cycle, hot air, at temperature $T_h$, enters from the top of the storage, transfers energy to the storage and exits from the bottom. During the storage discharging cycle, cold air enters from the bottom of the storage at temperature $T_c$, collects heat from the rocks and the PCM, and exits from the top.

Consider air flowing through an infinitesimal section of the storage, with length $dx$, as shown in Figure 2.1. The following energy flows and interactions taking place during an infinitesimal time $dt$ in the control volume are considered in our model:

- Energy of the air entering the control volume = $\dot{m} \cdot h_f(x, t) \cdot dt$
- Energy of the air exiting the control volume = $\dot{m} \cdot h_f(x + dx, t) \cdot dt$
- Energy transfer from the air to the storage = $\alpha \cdot a \cdot A \cdot dx \cdot [T_f(x, t) - T_s(x, t)] \cdot dt$
- Energy losses to the walls = $U_{wall} \cdot P \cdot dx \cdot [T_f(x, t) - T_\infty] \cdot dt$
Figure 2.1: Schematic showing the different heat transfer processes occurring within a combined storage during the storage charging cycle.
• Net energy conduction out of the control volume\(^1\) = \(\kappa_{\text{eff}} \cdot A \cdot \epsilon \cdot \left( \frac{\partial T_f}{\partial x} \bigg|_{x} - \frac{\partial T_f}{\partial x} \bigg|_{x+dx} \right) \cdot dt\)

• Energy increase of air in the control volume = \(\rho_f \cdot A \cdot dx \cdot \epsilon \cdot \left[ h_f(x, t + dt) - h_f(x, t) \right]\)

• Energy increase of storage material in the control volume = \(\rho_s \cdot A \cdot (1 - \epsilon) \cdot \left[ h_s(x, t + dt) - h_s(x, t) \right]\)

Applying the principle of energy conservation we get, for the fluid phase,

\[\rho_f Adx[\frac{\partial h_f}{\partial t} + \frac{\hat{m}}{\rho_f A \epsilon} \frac{\partial h_f}{\partial x} = \frac{\alpha a}{\rho_f \epsilon} (T_s - T_f) + \kappa_{\text{eff}} \frac{\partial^2 T_f}{\partial x^2} + \frac{U_{\text{wall}} P}{\rho A \epsilon} (T_\infty - T_f) \] (2.3a)

\[\rho_s (1 - \epsilon) \frac{\partial h_s}{\partial t} = \alpha a (T_f - T_s) \] (2.3b)

where the ‘+’ before the second term in equation 2.6a is valid during the storage charging cycle and the ‘−’ sign is valid during the storage discharging cycle. The specific enthalpy, \(h\), may be expressed as a function of temperature as

\[h(T) = \int_{T_{\text{ref}}}^{T} C_P(\chi) d\chi \] (2.4)

defining an average specific heat capacity such that

\[h(T) = C_{P,\text{avg}}(T - T_{\text{ref}}) = \int_{T_{\text{ref}}}^{T} C_P(\chi) d\chi \] (2.5)

allows us to express the governing equations (2.3) with the solid and fluid temperatures as the only unknowns,

\[\frac{\partial T_f}{\partial t} + \frac{\hat{m}}{\rho_f A \epsilon} \frac{\partial T_f}{\partial x} = \frac{\alpha a}{\rho_f C_f \epsilon} (T_s - T_f) + \kappa_{\text{eff}} \frac{\partial^2 T_f}{\partial x^2} + \frac{U_{\text{wall}} P}{\rho_f C_f \epsilon} (T_\infty - T_f) \] (2.6a)

\[\rho_s C_s (1 - \epsilon) \frac{\partial T_s}{\partial t} = \alpha a (T_f - T_s) \] (2.6b)

\(^1\)Although the conduction terms should be considered separately in both the solid and fluid phase equations, in the current model, an effective value of the storage conductivity, \(\kappa_{\text{eff}}\) is used and included in the fluid phase equation in order to simplify the numerical solution of the coupled system of partial differential equations that follow. \(\kappa_{\text{eff}}\) is calculated based on the correlation developed by Yagi and Kunii (1957) which considers the solid and fluid conductivity as well as the radiative heat transfer.
where $C_s$ and $C_f$ represent the average specific heat capacities of the solid at temperature $T_s$ and the fluid at temperature $T_f$ respectively. Equations 2.6 represent a coupled system of partial differential equations which are first order in time, $t$, and second order in the spatial coordinate, $x$. The solution further requires the definition of one initial condition and two boundary conditions for $T_f$ and one initial condition for $T_s$. Furthermore, physical correlations for the heat transfer coefficients, $\alpha$ and $U_{wall}$, the effective thermal conductivity, $\kappa_{eff}$, the average heat capacities, $C_f$ and $C_s$, the interfacial area per unit volume, $a$, and the void fraction, $\epsilon$ for each of the sensible and latent sections of the storage must be described along with the material densities, $\rho_f$ and $\rho_s$ for completing the model definition.

**Initial and Boundary Conditions**

The initial conditions need only be defined for the very first cycle, after which, the initial values of the solid and fluid temperatures for each subsequent cycle will be equal to those at the end of the previous cycle. The boundary conditions are defined for the fluid at the inlet, where the temperature is continuously measured and at the outlet, where no further heat losses are assumed.

\[
\begin{align*}
T_f(x,0) &= T_\infty, \quad (2.7a) \\
T_s(x,0) &= T_\infty \quad (2.7b)
\end{align*}
\]

**Charging Cycle**

\[
\begin{align*}
T_f(0,t) &= T_{in}(t), \quad (2.8a) \\
\left.\frac{\partial T_f}{\partial x}\right|_{L,t} &= 0 \quad (2.8b)
\end{align*}
\]

**Discharging Cycle**

\[
\begin{align*}
T_f(L,t) &= T_{in}(t), \quad (2.9a) \\
\left.\frac{\partial T_f}{\partial x}\right|_{0,t} &= 0 \quad (2.9b)
\end{align*}
\]

**Physical Correlations**

The thermo-physical property values for the working fluid, air, are taken from literature (Incropera and Dewitt, 2007). The properties of the rocks are averaged for the different types of rocks based on previous measurements by Zanganeh et al. (2012). The phase change material heat capacity is based on a phase transition model (Figure 2.2), where the PCM is assumed to melt uniformly between temperatures $T_{m_1}$ and $T_{m_2}$ as defined below,

\[
C_{PCM} = \begin{cases} 
C_{solid} & \text{if } T_s \leq T_{m_1} \\
\frac{\Delta h_{fus}}{T_{m_2}-T_{m_1}} & \text{if } T_{m_1} < T_s < T_{m_1} \\
C_{liquid} & \text{if } T_s \geq T_{m_2}
\end{cases} \quad (2.10)
\]

The corresponding values for AlSi$_{12}$, the phase change material used in this study are listed in Table 2.1. This phase transition model is predominantly based on the work by Beasley et al. (1989) who used it to simulate the phase change behaviour of paraffin wax and validated their heat transfer model using experimental data.
Energy input (J/Kg)

Temperature (°C)

$T_{m_1}$

$T_{m_2}$

$\Delta h_{fus}$

$C_{p,\text{solid}}$

$C_{p,\text{liquid}}$

$C_p = \frac{\Delta h_{fus}}{(T_{m_2} - T_{m_1})}$

Figure 2.2: Phase transition model used to simulate the melting behaviour of the phase change material inside the encapsulation.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\Delta h_{fus}$</td>
<td>466</td>
<td>KJ/Kg</td>
</tr>
<tr>
<td>$T_{m_1}$</td>
<td>573</td>
<td>°C</td>
</tr>
<tr>
<td>$T_{m_2}$</td>
<td>577</td>
<td>°C</td>
</tr>
<tr>
<td>$C_{\text{solid}}$</td>
<td>1038</td>
<td>J/KgK</td>
</tr>
<tr>
<td>$C_{\text{liquid}}$</td>
<td>1741</td>
<td>J/KgK</td>
</tr>
</tbody>
</table>

Table 2.1: Phase transition data for AlSi$_{12}$. Phase transition temperatures are taken from experimental observations during this study and the heat of fusion is taken from DSC measurements by Walser (2013). Solid and liquid heat capacities are taken from Wang et al. (2006).
Table 2.2: Coefficients for calculating the air to solid heat transfer coefficient in the latent section based on a staggered bank of tubes (Incropera and Dewitt, 2007).

<table>
<thead>
<tr>
<th>$Re_{max}$</th>
<th>$C$</th>
<th>$m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>&lt; 4</td>
<td>0.989</td>
<td>0.330</td>
</tr>
<tr>
<td>&lt; 40</td>
<td>0.911</td>
<td>0.385</td>
</tr>
<tr>
<td>&lt; 4000</td>
<td>0.683</td>
<td>0.466</td>
</tr>
<tr>
<td>&gt; 4000</td>
<td>0.193</td>
<td>0.618</td>
</tr>
</tbody>
</table>

The air to solid heat transfer coefficient, $\alpha$, for the sensible section is calculated using the following correlation proposed by Pfeffer (1964),

$$
\alpha_S = 1.26 \left[\frac{1 - (1 - \epsilon_S)^{5/3}}{W}\right]^{1/3} (C_p G)^{1/3} (\kappa_{air}/d_{rocks})^{2/3}
$$

(2.11)

where $W = 2 - 3\gamma + 3\gamma^5 - 2\gamma^6$ with $\gamma = (1 - \epsilon_S)^{1/3}$.

For the latent section, the air to solid heat transfer coefficient, is based on the correlation for a sequence of staggered tubes (Incropera and Dewitt, 2007)$^2$

$$
\alpha_L = C Re_{max}^m Pr_{air}^{1/3} \kappa_{air}/d_{tubes}
$$

(2.12)

where $C$ and $m$ are determined using the maximum Reynolds number from Table 2.2.

The overall heat transfer coefficient for the wall is calculated, from the insulation conductivities and thickness, using (Zanganeh et al., 2012),

$$
\frac{1}{U_{wall}} = \frac{1}{U_{inside}} + r_{inside} \sum_{j=1}^{n} \frac{1}{\kappa_j} \log\frac{r_{j+1}}{r_j}
$$

(2.13)

where $U_{inside}$ is based on the convective and radiative heat transfer coefficients taken from Beek (1962) and Ofuchi and Kunii (1965). The correlations are omitted here for brevity, for details the reader is referred to Zanganeh et al. (2012) or the original papers. The model was used to establish the required tank diameter and number of rows in the latent section for which stabilisation was predicted for a nominal mass flow rate of 0.0125 $Kg/s$, as shown in Figure 2.3.

The detailed design of the combined storage is described in chapter 3.

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$^2$In a staggered arrangement, the axes of tubes in adjacent rows is parallel to each other. In the combined storage design that follows the axes of tubes in adjacent rows are perpendicular to each other, hence a correction factor should be applied for the increased heat transfer coefficient during model validation.
Figure 2.3: Stabilisation of air outlet temperature during discharge predicted using the heat transfer model.
Chapter 3

Storage Design and Experimental Setup

This chapter outlines the major design decisions involved in the development of the combined storage prototype. The first section explains the selection criteria for the phase change material used in the latent section of the combined storage. An overview of the combined storage design is then given, along with the important design aspects which govern the fluid flow and heat transfer in the system, followed by a description of the major functional components of the experimental setup.

3.1 Phase Change Material Selection

The phase change material used in the latent section has a significant impact on the behaviour and performance of the combined storage (Commerford, 2012). A suitable phase change material should exhibit the following properties (Kenisarin, 2010)

- A melting temperature within the desired temperature range. The CSP collectors developed by Airlight energy SA are designed to deliver hot air at unto 650 °C. Accounting for variation in the value of direct normal incidence, the nominal inlet (hot) air temperature for the combined storage operation was 600 °C. Since the desired outlet air discharge temperature should be as close to the nominal hot air temperature, the desired phase transition temperature is between 560 and 590 °C.

- A high latent heat of fusion. This decreases the amount of phase change material required and enables more energy to be stored near the desired outlet air temperature.

- A high thermal conductivity, for rapid heat transfer within the encapsulation, which also reduces the amount of encapsulation surface required.

- Low undercooling and superheating, i.e. a small phase transition temperature range, providing all the latent heat at a nearly constant temperature.

- Small volume expansion during melting, to minimise void spaces within the encapsulation, which decrease the rate of heat transfer to and from the phase change material.

- Chemically stable, non toxic and compatible with an inexpensive encapsulation material, such as Stainless Steel, Copper or Aluminium, for a robust system capable of withstanding thousands of charging/discharging cycles.
3.1.1 Molten Salts and Metal Alloys

Most of the previous work on high temperature latent heat thermal storage systems involves the use of molten salts (Gomez et al., 2011) as phase change materials, since they are most suitable for the hot air temperatures of 300 - 400 °C by a wide range of parabolic trough and solar tower power plants. There are, however, a number of issues with the use of molten salts, such as their low thermal conductivity, large phase transition temperature range and large volume expansion during melting, which make their incorporation difficult and expensive from a thermal design point of view. The Airlight collector is, in contrast, capable of producing much higher air hot air temperatures and hence opens up the possibility of using metal alloys as phase change materials. Since metals have a thermal conductivity unto two orders of magnitude higher than molten salts, and an very small phase transition range, one may use them to develop simplified and hence, more cost effective, thermal storage designs.

3.1.2 AlSi$_{12}$

The eutectic alloy, containing 88% Aluminium and 12% Silicon by mass has a melting point of 576 ± 1 °C, along with a heat of fusion higher than other metal alloys with similar melting temperatures, as seen in Figure 3.1. It is also commercially available due to its widespread use in welding Aluminium. The thermo-physical properties of the alloy are tabulated in Table 3.1.
### Table 3.1: Thermophysical properties of AlSi₁₂ compared to molten salts and other metal alloys.

<table>
<thead>
<tr>
<th>Property</th>
<th>Molten Salts</th>
<th>Metal Alloys</th>
<th>AlSi₁₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>Melting point (°C)</td>
<td>100 – 900</td>
<td>&gt; 250</td>
<td>576</td>
</tr>
<tr>
<td>Heat of fusion (KJ/Kg)</td>
<td>high</td>
<td>high</td>
<td>560</td>
</tr>
<tr>
<td>Congruent melting</td>
<td>x</td>
<td>✓</td>
<td>574-577 °C</td>
</tr>
<tr>
<td>Volume expansion</td>
<td>high</td>
<td>low</td>
<td>5 %</td>
</tr>
<tr>
<td>Thermal Conductivity</td>
<td>very low</td>
<td>high</td>
<td>160 W/mK</td>
</tr>
<tr>
<td>Chemical stability</td>
<td>x</td>
<td>✓</td>
<td>under Argon atmosphere</td>
</tr>
<tr>
<td>Compatibility</td>
<td>x</td>
<td>✓</td>
<td>with stainless steel</td>
</tr>
<tr>
<td>Toxicity</td>
<td>high</td>
<td>low</td>
<td>low</td>
</tr>
<tr>
<td>Corrosion resistance</td>
<td>low</td>
<td>inert atmosphere</td>
<td>dry atmosphere</td>
</tr>
</tbody>
</table>

As seen from comparison in Table 3.1, metal alloys outperform molten salts at temperatures above 500 °C. Thus, after some initial testing (A), where the alloy showed integrity over multiple melting and solidification cycles while encapsulated in stainless steel, AlSi₁₂ was chosen as the phase change material for latent section of the combined storage.

### 3.2 Storage Design

The storage structure consists of a stainless steel tube (AISI 304) of length 1700 mm, outer diameter 400 mm and 3 mm thickness. Perforated plates are welded at a distance of 200 mm from the bottom of the tube and 100 mm from the top of the tube. The rocks and encapsulated PCM, i.e., the energy storing material, is placed in between the perforated plates, making the active storage length 1375 mm. The perforated plates homogenise the air flow, allowing the energy to be uniformly distributed, across the storage cross section. The presence of uniform flow was verified using CFD analysis, the results of which are summarised in Figure 3.2. Steel sections with double cones are used to connect the top and bottom of the storage to the smaller delivery tubes which are 200 mm in diameter. The double cone design was developed to ensure homogeneously distributed mass flow across the storage cross section, even when there is very little pressure drop across the active storage length.

#### 3.2.1 Sensible Heat Storage Section

The combined storage consists broadly of two sections. The bottom section is designed to store energy as sensible heat in rocks. The rocks have been excavated from the Rafzerfeld area in Zurich, Switzerland and consist of five different types - Helvetic Siliceous Limestone, Quartzite, Limestone, Calcareous Sandstone and Gabbro (Zanganeh et al., 2012). The rocks have an average diameter of 25 mm, giving a $\frac{D}{d}$ ratio of 15.76. The bulk void fraction of the sensible section was measured by constructing another steel tube with the same diameter, thickness and height as the sensible section of the combined storage, filling it with the storage rocks and measuring the volume of water which filled up the storage. A value of $\epsilon_s = 0.40 \pm 0.01$ was determined for the sensible part of the combined storage. It is important to note that this value depends on the $\frac{D}{d}$ ratio as well as the height of the rocks, since a higher column of rocks exerts more force on the lower rocks, compacting the system and decreasing the void fraction, and hence should be measured for systems of different sizes.
Figure 3.2: CFD results verifying flow homogeneity across tank cross section. The results were obtained using a steady state solver on a regular block mesh using ANSYS CFX®. Figure 3.2b shows that the maximum deviation in axial air velocity entering the storage is less than 10% of the average velocity at the charging inlet cross section. The double cones, along with the perforated plates ensure uniform distribution of air across the tank cross section, even when there is no pressure drop due to the storage material and there is a bend in the inlet delivery pipe.
Figure 3.3: Top and side views showing the arrangement of rows of encapsulated phase change material (AlSi$_{12}$) tubes stacked in the form of a mesh to effectively cover the storage cross section, providing a high interfacial surface area for heat transfer and heat transfer coefficient.

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tank Inner Diameter</td>
<td>394</td>
<td>mm</td>
</tr>
<tr>
<td>Tank Outer Diameter</td>
<td>400</td>
<td>mm</td>
</tr>
<tr>
<td>$L_{sensible}$</td>
<td>1270</td>
<td>mm</td>
</tr>
<tr>
<td>$L_{latent}$</td>
<td>90</td>
<td>mm</td>
</tr>
<tr>
<td>$a_{sensible}$</td>
<td>156</td>
<td>m$^2$/m$^3$</td>
</tr>
<tr>
<td>$a_{latent}$</td>
<td>103.45</td>
<td>m$^2$/m$^3$</td>
</tr>
<tr>
<td>Mass of AlSi$_{12}$</td>
<td>9.63</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of SS 316</td>
<td>13.14</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of Rocks</td>
<td>243.22</td>
<td>kg</td>
</tr>
<tr>
<td>$\epsilon_{sensible}$</td>
<td>0.4</td>
<td></td>
</tr>
<tr>
<td>$\epsilon_{latent}$</td>
<td>0.549</td>
<td></td>
</tr>
<tr>
<td>Thermal Capacity</td>
<td>40.29</td>
<td>KW$hr$</td>
</tr>
</tbody>
</table>

Table 3.2: Combined storage design parameters

3.2.2 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 phase change material, encapsulated in stainless steel (316) tubes with 16mm inner diameter and 1 mm thickness. The tubes are arranged in the form of rows, with each row containing 17 tubes of different lengths, in order to effectively pack the storage cross section while using a reasonable number of tubes, and hence keeping the filling and encapsulation work feasible, as shown in Figure 3.3a. Four such rows are then stacked on top of each other with the tube axes perpendicular to each other for adjacent rows, in the form of a mesh, depicted in Figure 3.3b. The mesh arrangement is an effort to increase the flow turbulence leading to a higher air to tube heat transfer coefficient than that obtained with parallel tube axes. Such an arrangement gives a total latent section length of 90 mm with an average void fraction of 0.549.
Figure 3.4: Combined storage design overview
3.2.3 Insulation

Thermal storage performance depends significantly on the quality and quantity of insulation. Significant thermal losses are associated with small, lab-scale thermal storages such as the one designed during this study, due to their intrinsically higher surface area to volume ratios. Due to thermal stratification, the combined storage was insulated with three different insulation materials (Table 3.3), with greater insulation thickness at the top, depending on the temperatures involved. The insulation materials and thicknesses are shown schematically in Figure 3.5. The insulation thicknesses were changed during the experimental campaign and the corresponding data for each experimental run were recorded.

Table 3.3: Insulation materials used for the combined storage prototype.

<table>
<thead>
<tr>
<th>Material</th>
<th>Conductivity (W/mK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Microtherm®</td>
<td>0.021-0.04</td>
</tr>
<tr>
<td>Rockwool®</td>
<td>0.05-0.07</td>
</tr>
<tr>
<td>Felt (Promaglaf G®)</td>
<td>0.04-0.09</td>
</tr>
</tbody>
</table>

3.3 Experimental Setup

Figure 3.7 shows a schematic of the experimental setup. The major components of the setup are listed below along with their functions:

- Air flow is established by means of an air blower. The air mass flow may be regulated by adjusting the blower speed through an inverter connected to the SCADA system.

Figure 3.5: Schematic of the insulation technique. Higher thickness of lower conductivity material (Microtherm®) is used for the upper part of the storage, which is subject to higher temperatures. The temperature of the outside surface of the insulation was monitored during the experimental campaign to ensure low thermal losses. Dimensions are given in mm.
A Bronkhorst® meter is used to measure the volume flow rate of air flowing through the system, which can then be converted into a mass flow rate. In order to obtain accurate values of the mass flow rate, long straight tubes of length 1 m are placed both before and after the measurement unit, ensuring fully developed flow through the meter.

Two valves (# 4, 5), placed after the mass flow meter, at two opposite branches of a T section, direct the air to the heater during the charging cycle and towards the bottom of the storage during the discharge cycle.

The heater, developed in house at Airlight Energy, uses resistance coils to heat up the air to the desired charging temperature. A valve (# 2), placed just before the heater inlet, prevents any hot air from reaching the plastic section of tubing between valves 2 and 3 during the discharge cycle.

Valves (# 3, 6) direct the air flow to the storage or to the surroundings, depending on the type of cycle (charge/ discharge) in progress.

Thermocouples embedded within the storage at regular axial intervals measure the axial temperature distribution evolution over time. The data is converted from analog to digital form by the SCADA board and then recorded at 10 second intervals through a software interface onto a laptop.

Valve # 1 was intended to direct the air out of the system during the discharging cycle, however, since it was advisable to maintain a flow through the heater while the resistances were hot, Valve # 3 was used as the exit to the surroundings during discharging.
Figure 3.7: Experimental Setup Schematic
Chapter 4

Observations and Results

4.1 Experimental Campaign

The major objective of this thesis was to design and construct a combined storage and experimentally obtain a stabilised air outlet temperature during storage discharging. However, for the purpose of validating mathematical models of such a system to enable the design of a scale-up, it was necessary to obtain detailed experimental data about the performance of the latent section without the presence of the rocks. Additionally, to observe the relative stabilisation as compared to a sensible only storage of the same dimensions and design, experimental data relating to the behaviour of a rocks only setup was also essential. Hence the experimental campaign was performed in three phases, for each of the following configurations, which are depicted schematically in Figure 4.1,

1. PCM Only, this configuration consists of only the four rows of encapsulated $AlSi_{12}$ as the active storage material. The heat is stored as sensible heat of the steel and PCM and as latent heat of the PCM.

2. Combined, this configuration consists of the encapsulated PCM section on top of the rocks.

3. Rocks Only, this configuration consists of only rocks, filled up to the total length of the combined storage. It is worth noting, that since the energy density of the encapsulated PCM is much more than the rocks, the total heat storage capacity of this configuration is less than that of the “combined” configuration.

4.2 Observations

Table 4.1 lists the experiments performed during the campaign along with the associated parameters. Experiments 5, 6, and 10 were performed with similar heater power ramps and inverter frequencies during charging and discharging to facilitate comparison between the three configurations. Also, using the data from experiments 9 and 11, one can compare the Combined to the Rocks Only configurations. A typical dataset generated during an experimental run is shown in Figure 4.2. The heater power, volume flow rate of air as measured by the volume flow meter, which could be connected to the inlet or outlet sides, and thermocouple temperatures are available for 10 second intervals.
Table 4.1: Summary of experimental campaign. The inverter frequency may be converted into the air mass flow rate using calibration curves available in the results dataset. Experiments 5, 6 and 10 were conducted under similar conditions to facilitate comparison across the three configurations. Experiments 9 and 11 were also conducted with similar test protocols, and may be used to compare the performance of the Combined and Rocks only configurations. The following discussion focuses mainly on these comparative experiments. The other experiments provide valuable data for validating mathematical models.
Figure 4.2: Typical result dataset obtained during an experimental run for the combined configuration (Experiment 9). The data includes air temperature for various axial locations inside the combined storage, solid (PCM) temperatures inside the middle tube for each of the four rows, the volume flow rate of air and the heater power measured at 10 sec intervals. The experimental parameters are given in Table 4.3. Axial position is measured as the height of the thermocouple from the top of the bottom perforated plate, i.e., the distance from the beginning of the sensible section.
Figure 4.3: Axial air temperature profiles at various times inside the combined storage. The experimental parameters are given in Table 4.3.

Inspite of our best efforts in making the setup air tight, differences were observed in between the air mass flow rates at the inlet and outlet, indicating air leakage. Since the air mass flow rate for a given inverter frequency is a function of the pressure drop across the system, the air mass flow rates at both the inlet and outlet were measured over the complete range of inverter frequencies, to obtain calibration curves between inverter frequency and mass flow rate, for each of the three configurations. These calibration curves may be used to obtain an initial estimate of the mass flow rates for an experimental run, which can then be more accurately determined using the measured temperature data and storage model, through energy balances.

4.2.1 Axial Temperature Profile

Figure 4.3 shows the axial temperature profile inside the combined storage during charging. The ‘S’ shaped curve, typical of thermally stratified heat storages, is observed, indicating the addition of a thin latent section preserves the thermocline and hence would have thermodynamic efficiency values similar to that of sensible heat storages. The data can be interpolated and used to estimate the amount of heat storage at a given time inside the storage in order to calculate charging and discharging efficiencies.

4.2.2 Radial temperature distribution and effect of the wall

Thermocouples were placed at different radial positions across the tank cross section, as depicted in Figure 4.4, for the PCM Only configuration. Figure 4.5 shows the temperature evolution for one row of PCM tubes. Temperatures very close to each other indicate flow homogeneity across the tank cross section. The tube outer surface temperatures are also observed to be very close to the temperatures near the tube centre, which is expected, due to the high thermal conductivity of AlSi_{12}.

The radial temperature distribution across the tank cross section in the sensible section is shown in Figure 4.6. The variation in temperature is observed to be higher here than the latent section, an effect which can be attributed to incoming radiation from the steel tank wall, which
heats up and cools down faster than the active storage material, due to its lower thermal inertia and higher thermal conductivity in comparison to the packed bed. Higher air velocities near the tank walls, which have been known to occur in packed bed storages (Hänchen et al., 2011), may also be partly responsible for non uniform radial temperature distribution in the sensible section.

4.2.3 Air outlet temperature during the discharge cycle

Significant stabilisation of the air outlet temperature during the discharge cycle is observed for all experiments with the Combined configuration. Figure 4.7 shows the inlet and outlet air temperatures during the discharge cycle during one such experimental run. The air outlet temperature initially decreases rapidly, stabilises near the phase transition range of $AlSi_{12}$, and then again starts falling rapidly when all the phase change material is solidified. An interesting observation is the change in slope of the temperature vs time curve, observed whenever a row phase change material tubes solidifies. These ‘kinks’ have also been predicted using the numerical heat transfer model (Commerford, 2012).

4.3 Qualitative comparison of the three configurations

Figure 4.8 shows the air temperature at the top of the storage for the three configurations with experimental parameters listed in 4.2. This temperature represents the inlet air temperature during the charging cycle and the outlet air temperature during the discharging cycle. The similar charging inlet temperature profiles enable one to qualitatively compare the performance of the three configurations during the discharge cycle.
Figure 4.5: Radial temperature distribution across tank cross section for the third PCM row during Experiment 5. Thermocouple placement is shown in Figure 4.4. The curves show homogenous temperature distribution across the tank for the latent section, indicating uniform flow across the tank cross section. The close proximity of the Air and PCM temperature profiles indicate excellent heat transfer between the two.
Figure 4.6: Radial temperature distribution across tank cross section for 3 axial positions within the sensible section during Experiment 6. The difference in temperatures for thermocouples at the same axial location but different radial positions may be attributed to the radiation heat transfer from the tank wall. Since the wall heats up and cools down faster than the rocks due to its lower thermal inertia and higher thermal conductivity, it increases the temperature of the radially offset thermocouples during charging and decreases it during discharging. The effect of the wall is small at the bottom, due to insignificant radiative heat transfer at lower temperatures and is maximum in the middle of the tank. Near the top of the storage tank, the effect is not observed, which might be due to additional radiative exchange with the latent section.
Figure 4.7: Inlet and Outlet Air Temperature during the discharge cycle for Experiment 8. Stabilisation of the outlet temperature is observed close to the phase transition range of AlSi$_{12}$, implying excellent heat transfer between the air and the phase change material. As expected, the air temperature starts dropping off rapidly when the latent section is completely solidified after about 1.5 hrs. The experimental parameters used are given in Table 4.2.
Figure 4.8: Comparison of the three storage configurations. The test protocols followed are given in Table 4.2. The solid lines show the temperature at the top of the storage (charging inlet/discharging outlet) and the dotted lines show the temperature at the bottom of the storage (charging outlet/discharging inlet). The higher charging outlet temperatures for the PCM only configuration imply that the input energy is not being completely stored. The PCM only configuration was charged for a longer period of time, to melt all the PCM in the four rows, indicating lower heat transfer in this configuration due to the lack of a back pressure created by the rock column.

Figure 4.9 is a zoomed version of Figure 4.8 showing the discharging cycle. The PCM only configuration is seen to perform the worst of the three configurations, since the air temperature drops quickly during the discharge cycle. With a thermal capacity of 3.97 KWhr, the latent section cannot store all the heat put in during charging, which continues till all the AlSi$_{12}$ is melted. The energy not captured by the storage manifests in terms of a charging outlet temperature higher than the ambient temperature. The Rocks only configuration, with a storage capacity of 38.9 KWhr, is able to store most of the energy input during the charging cycle, and provides a steadily decreasing outlet air temperature, typical of sensible heat storage designs (Commerford, 2012). The combined configuration provides a steady outlet air temperature, which stabilises near the phase transition temperature of AlSi$_{12}$, for a long duration during the discharge cycle, thus providing a proof of concept for the combined storage design. Although, the three experimental runs shown here were performed with similar charging and discharging inverter frequencies, the actual mass flow rates of air flowing through the storage may be different for the three configurations. An increase in the charging outlet air temperature for the combined configuration at the end of the charging period indicates a higher actual charging mass flow rate for the Combined configuration experiment as compared to the Rocks Only configuration.

Figure 4.10 shows a comparison of the air temperatures at the top and bottom of the storage for another set of experimental parameters, listed in Table 4.3. Similar charging inlet and outlet
Figure 4.9: Outlet air temperature at the top of the storage for the three configurations. The plots clearly show the superiority of the combined configuration in obtaining a stabilised air discharge temperature. The experimental parameters are the same as Figure 4.8 (Table 4.2).

<table>
<thead>
<tr>
<th>Experiment Number</th>
<th>Type</th>
<th>Charging Time (hrs)</th>
<th>$\dot{m}_{ch}$ (Kg/s)</th>
<th>$\dot{m}_{dis}$ (Kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>PCM Only</td>
<td>4:34</td>
<td>0.02</td>
<td>0.005</td>
</tr>
<tr>
<td>6</td>
<td>Combined</td>
<td>3:55</td>
<td>0.02</td>
<td>0.005</td>
</tr>
<tr>
<td>8</td>
<td>Combined</td>
<td>4:16</td>
<td>0.02</td>
<td>0.02</td>
</tr>
<tr>
<td>10</td>
<td>Rocks Only</td>
<td>3:55</td>
<td>0.02</td>
<td>0.005</td>
</tr>
</tbody>
</table>

Table 4.2: Experimental protocols for comparative tests between the three configurations. Best estimates of the mass flow rates are given based on calibration with the inverter frequency, but may not be accurate due to leakages. Hence, the comparison made here is qualitative.
Figure 4.10: Comparison of the Combined and Rocks Only configurations at a higher mass flow rate. The test protocols followed are given in Table 4.3. The solid lines show the temperature at the top of the storage (charging inlet/discharging outlet) and the dotted lines show the temperature at the bottom of the storage (charging outlet/discharging inlet).

Air temperature profiles for both configurations indicate similar mass flow rates during the charging cycle, implying similar input energy during charging. This set of experiments hence enable a more quantitative comparison between the two configurations from a thermodynamic point of view. The Combined configuration provides a stabilised outlet air discharge temperature, as mentioned in the preceding paragraphs. The Rocks Only configuration provides a steadily decreasing outlet discharge air temperature. It is worth noting that the outlet air temperature for the Rocks Only configuration is higher than that for Combined configuration, for most of the discharging cycle. This observation is in agreement with heat transfer model simulations, which predict that the Rocks Only configuration provides an outlet air temperature higher that for the Combined configuration, for the same input energy, during the first discharging cycle.

<table>
<thead>
<tr>
<th>Experiment Number</th>
<th>Type</th>
<th>Charging Time (hrs)</th>
<th>$\dot{m}_{ch}$ (Kg/s)</th>
<th>$\dot{m}_{dis}$ (Kg/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>9</td>
<td>Combined</td>
<td>3:09</td>
<td>0.028</td>
<td>0.0207</td>
</tr>
<tr>
<td>11</td>
<td>Rocks Only</td>
<td>3:08</td>
<td>0.028</td>
<td>0.0204</td>
</tr>
</tbody>
</table>

Table 4.3: Experimental protocols for comparative tests between the combined and rocks only configurations at higher mass flow rates. Best estimates of the mass flow rates are given based on calibration with the inverter frequency, but may not be accurate due to leakages. Based on comparison with simulations, this set of experiments are a better representation of the comparative performance of the two configurations.
Figure 4.11: Air temperature at the storage outlet for the two storage configurations during the discharge cycle for higher air mass flow rates. Stabilisation is observed for the combined storage, however, the rocks only storage provides higher temperatures. The experimental parameters are the same as Figure 4.10 (Table 4.3).
Chapter 5

Conclusions and Outlook

5.1 Conclusions

The combined storage concept has been experimentally shown to provide stabilised outlet air temperatures during the storage discharging cycle. The combined design has the ability to provide the benefits of high thermodynamic storage efficiency, similar to sensible only storage systems, while maintaining a nearly constant outflow air temperature, previously predicted for heat storage within phase change materials (Beasley et al., 1989), which typically have low thermodynamic efficiencies, since no thermal stratification is present (Bjurström and Carlsson, 1985). Since only a small amount of phase change material is needed for the thin latent section at the very top of the storage, in the combined design, the concept has the potential to be extremely cost effective, once an automated encapsulation process is developed.

The eutectic alloy of Aluminium and Silicon, AlSi\textsubscript{12}, is used as the phase change material in this study. The metallic alloy has many advantages over more commonly used molten salts, such as a higher thermal conductivity, low volume expansion during melting and a small phase transition temperature range. Encapsulation of this metal alloy within stainless steel tubes followed by sealing under an Argon purge is shown to be stable for up to 10 melting-solidification cycles.

The latent section of the combined storage was designed using rows of tubes containing the alloy, placed on top of each other in a mesh configuration. This design, with a high interfacial surface area and heat transfer coefficient, enables excellent heat transfer between the air and the alloy evidenced by the small difference between the air and solid temperatures (Figure 5.1), while providing a simple encapsulation technique for the phase change material from the perspective of manufacturability.

The mass flow rates of air during charging and discharging have a significant impact on the performance of the combined storage, hence mathematical modelling plays an extremely important role in the design of these systems, which have a range of mass flow rates, depending on their size, for optimal performance.

5.2 Outlook

The large amount of data generated during the experimental campaign may now be used to understand the behaviour of the storage better. The heat transfer models developed during this
Figure 5.1: Air and solid temperatures within the latent section of the combined storage during Experiment 6. A small difference in the air and solid temperature at the same axial position implies excellent heat transfer between the two.

Thesis and previous works may now be validated to develop a reliable mathematical model describing the behaviour of the combined storage. The data available for radially offset locations may be useful for extending the current model in two dimensions, incorporating the effects of radiation from the tank walls and radially varying void fraction, which cause non homogenous temperature distribution across the storage cross section. The effect of the PCM melting temperature on the thermodynamic efficiency of the storage may also be investigated using the heat transfer model.

Further experiments may also be performed to quantitatively compare the thermodynamic efficiencies of the combined storage to a sensible only storage. Special care needs to be taken to ensure that the system is free of any air leakages, to be able to measure the air mass flow rate accurately, which is essential to make such a quantitative comparison.

Metal alloys, such as the one used in this study, are very promising as latent heat storage materials for energy storage. However, limited data is available, on their use in such systems. The long term response of $\text{AlSi}_{12}$ to thermal cycling is largely unknown, and rapid thermal cycling tests need to be conducted with the alloy in stainless steel, and other encapsulations, such as copper, to determine any possible degradation. A safe, automated encapsulation process for such materials also needs to be developed to dramatically reduce time required and drive down the associated cost. An industrial scale combined storage may then be designed, using a validated heat transfer model, constructed and tested.
I would like to express my gratitude to Professor Aldo Steinfeld and Dr. Andreas Haselbacher for giving me the opportunity to work on this exciting project. I am especially grateful to Giw Zanganeh, for introducing me to this novel concept, and for his constant support, guidance and friendly supervision throughout the course of the thesis, which made it an enjoyable learning experience. I would also like to thank Dr. Gianluca Ambrosetti and Andrea Pedretti for helpful discussions during the course of this thesis and the excellent team at Airlight Energy SA, Vittorio, Pietro, Fabio, Fernando and others, without whom this thesis would not have been possible. I am always grateful to my family and friends for their constant encouragement and support.
Appendix A

$AlSi_{12}$ Testing and Encapsulation

A small amount of $AlSi_{12}$ was first tested by simulating the charging and discharging cycles by heating a stainless steel tube filled with the material in an oven. Temperatures for different regions inside the tube were measured for 3 cycles of heating and cooling through thermocouples inserted as shown in Figure A.2. The material exhibited no signs of phase segregation, interaction with the encapsulation or oxidation during the three cycles, and consistent heating and cooling curves for the different cycles. The phase transition temperature range was found to be 573 to 577 °C. The small value of temperature differences between the different regions inside the tube, as shown in Figure A.2, implies small Biot numbers, and hence the whole tube may be assumed to be at the same temperature for modelling purposes.

Based on the results of this initial testing, an encapsulation technique for all the 68 tubes to be used in the combined storage was developed, involving filling approximately 95% of the 16 mm internal diameter stainless steel tubes with molten $AlSi_{12}$ and sealing them with steel caps under Argon purge to prevent Aluminium oxidation during thermal cycling, as depicted in Figure A.3.
Figure A.1: Thermocouple readings over time for AlSi_{12} encapsulated in a stainless steel (316) tube. The temperatures are observed to be very close to each other due to the high thermal conductivity of the alloy, indicating a small Biot number. No change in phase transition temperature was observed over 3 cycles, indicating no significant phase segregation or oxidation inside the tube.

Figure A.2: Schematic for a single tube filled with AlSi_{12} showing the positions of the thermocouples relative to the alloy and the encapsulation.
Figure A.3: Encapsulation technique. Stainless steel tube were filled with molten $AlSi_{12}$, and sealed with caps under Argon purge.
Appendix B

Tube Placement

Figure B.1: Tubes of different lengths arranged inside a circular tank. Such an arrangement is used in the Latent section of the combined storage leading to a high specific surface area for heat transfer with the air.

The phase change material, AlSi, is encapsulated in cylindrical tubes. The tubes are arranged inside the tank in the form of a wire mesh, with identical rows of tubes on top of each other. The axes of adjacent rows are kept perpendicular to each other in an effort to maximise the heat transfer coefficient between the air and the tubes. The tube positions, for a particular row, inside the tank are determined based on the inner tank diameter, $D$, outer tube diameter, $d$, and spacing between the centrelines of adjacent tubes, $S_t$, as follows:

Let there be $n_r$ tubes in a row, the maximum number of tubes in one row is given by $\frac{D}{S_t}$, however
since $n_r$ must be a whole number, we take

$$n_r = \left\lceil \frac{D}{S_t} \right\rceil$$  \hspace{1cm} (B.1)

where $\lceil x \rceil$ denotes the greatest integer less than or equal to $x$. With $n_r$ defined, the gap between the outermost tube edge and the tank circumference, $g_e$ may be expressed as

$$g_e = \frac{D - d_l - (n_r - 1)S_t}{2}$$  \hspace{1cm} (B.2)

Using this edge gap, the position of the centerline of the $i^{th}$ tube from the tank circumference is given by

$$x_i = g_e + \frac{d_l}{2} + (i - 1)S_t$$  \hspace{1cm} (B.3)

and its maximum length is

$$L_i = 2 \sqrt{\left( R^2 - \left( |R - x_i| + \frac{d_l}{2} \right)^2 \right)}$$  \hspace{1cm} (B.4)

The above procedure may be used to calculate the lengths and positions of the tubes given a tube and tank size. The lengths hence calculated are the maximum lengths of the tubes for each position. However, allowances must me made in practice for fitting the tubes on placeholders and variabilities in tank diameter. Such allowances usually result in a decrease in the tube lengths compared to the values calculated using the procedure described above.
## Appendix C

### Accompanying Data Files

<table>
<thead>
<tr>
<th>Folder</th>
<th>File/Folder name</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design</td>
<td>Overview.xls</td>
<td>Overview of combined storage design, including parameters calculated for the perforated plates and tubes.</td>
</tr>
<tr>
<td>Design</td>
<td>PCM Tubes.xls</td>
<td>Measured weights, lengths and position of tubes containing the PCM inside the combined storage.</td>
</tr>
<tr>
<td>Design</td>
<td>Insulation Details.xls</td>
<td>Heights and thicknesses of various insulation materials used in the combined storage for different experiments.</td>
</tr>
<tr>
<td>Results</td>
<td>Experiment &lt;N&gt;, &lt;Configuration&gt; - .xls</td>
<td>All experimental data relating to experiment number &lt;N&gt;, folder containing all log files for the corresponding experiment along with additional tabulated data in .xlsx files.</td>
</tr>
<tr>
<td>Matlab</td>
<td>model.m</td>
<td>Main MATLAB file containing the mathematical model to be run.</td>
</tr>
<tr>
<td>Matlab</td>
<td>C_p_air.m</td>
<td>Function for calculating specific heat capacity of air for a given temperature.</td>
</tr>
<tr>
<td>Matlab</td>
<td>C_p_rocks.m</td>
<td>Function for calculating specific heat capacity of the rocks for a given temperature.</td>
</tr>
<tr>
<td>Matlab</td>
<td>C_PCM.m</td>
<td>Function for calculating specific heat capacity of $\text{AlSi}_{12}$ for a given temperature.</td>
</tr>
<tr>
<td>Matlab</td>
<td>Enthalpy_PCM.m</td>
<td>Function for calculating the specific enthalpy of $\text{AlSi}_{12}$ at a given temperature.</td>
</tr>
<tr>
<td>Matlab</td>
<td>H_air.m</td>
<td>Function for calculating specific enthalpy of air at a given temperature.</td>
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<tr>
<td>Matlab</td>
<td>h_PCM.m</td>
<td>Function for calculating the heat transfer coefficient for the latent section.</td>
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<tr>
<td>Matlab</td>
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<td>Function for calculating specific enthalpy of the rocks at a given temperature.</td>
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<tr>
<td>Matlab</td>
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<td>Function for calculating thermal conductivity of air at a given temperature.</td>
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<tr>
<td>Matlab</td>
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<td>Matlab</td>
<td>k_SS.m</td>
<td>Function for calculating thermal conductivity of steel at a given temperature.</td>
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<td>Function for calculating dynamic viscosity of the air at a given temperature.</td>
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<td>Function for calculating Prandtl number for air at a given temperature.</td>
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<tr>
<td>Matlab</td>
<td>rho_air.m</td>
<td>Function for calculating the density of air at a given temperature.</td>
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<tr>
<td>Autodesk Inventor</td>
<td>Inventor</td>
<td>All design and drawing files.</td>
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</tbody>
</table>

Figure C.1: Description of accompanying data files.
Figure C.2: Combined storage design sketch.
Bibliography


Kotzé, Johannes P; von Backström, Theodor W, and Erens, Paul J. Evaluation of a latent heat thermal energy storage system using alsi12 as a phase change material.


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